



All-Electric Central Plant Design Guide



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Abbreviations and Acronyms

Acronym	Meaning
AIM	American Innovation and Manufacturing
AVERT	Avoided Emissions and Generation Tool
AWHP	Air to water heat pump
BAS	Building automation system
BPS	Building performance standard
CARB	California Air Resources Board
CBE	Center for the Built Environment
CDF	Cumulative distribution function
CEDA	California Energy Design Assistance Program
COP	Coefficient of performance
COP _c	Coefficient of cooling performance
COP _h	Coefficient of heating performance
EPA	Environmental Protection Agency
GWP	Global warming potential
HFC	Hydrofluorocarbon
HRC	Heat recovery chiller
HVAC	Heating, Ventilation, and Air Conditioning
kBtu/h	Thousand British thermal units per hour
LRMER	Long run marginal emissions rates
LSC	Long-term system cost
NIST	National Institute of Standards and Technology

Acronym	Meaning
NLR	National Laboratory of the Rockies
NMEC	Normalized Metered Energy Consumption
OAT	Outside air temperature
PCE	Peninsula Clean Energy
PG&E	Pacific Gas & Electric
PLR	Part load ratio
ROM	Rough order of magnitude
SCE	Southern California Edison
SCR	Silicon controlled rectifiers
SDG&E	San Diego Gas & Electric
SNAP	Significant New Alternatives Policy
SHMER	Short run marginal emissions rates
SVP	Silicon Valley Power
TES	Thermal energy storage
TIER	Time independent energy storage
Title 24	California Building Energy Efficiency Standards
TOU	Time-of-use
TTR	Technology Transitions Rule
WSEC	Washington State Energy Code
WWHP	Water-to-water heat pump

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Introduction

As interest and demand grows for electrifying and decarbonizing buildings, the HVAC industry faces a wide range of 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 energy efficient. Many early all-electric installations have also suffered from design and installation issues due to equipment design failures, misapplication of new equipment, and lack of experience with new equipment constraints.

Standard HVAC designs for large commercial buildings rely on separate gas-fired hydronic heating and chilled water loops. This technology has slowly been refined with incremental energy efficiency improvements over several decades and there are existing resources for design and control best practices (Taylor S. , 2017; ASHRAE, 2024; Cheng, Wendler, & Raftery, 2024). Nevertheless, for hot water 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, Geronazzo, Cheng, & Paliaga, 2018; Raftery, Singla, Cheng, & Paliaga, 2024). Some of these challenges relate to all-electric plants as well. For all-electric plants, equipment offerings and design considerations are also new, and the HVAC industry has not yet established how to serve large commercial building heating loads effectively and energy efficiently. Many engineers lack the expertise, experience, or time to keep up with new and rapidly evolving equipment needed in order 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 to rapidly become familiarized with the new challenges and design considerations, abilities and limitations of new equipment options, and learn from the successes and challenges faced by early adopter projects.

This design guide aims to help meet some of these needs.

The guide focuses on all-electric central plant equipment that are primarily applicable to large commercial buildings. All-electric equipment typical of small to medium buildings are fairly well-established and cost competitive, and are not covered in this guide (e.g., variable refrigerant flow, water-to-air heat pumps, and packaged rooftop heat pumps).

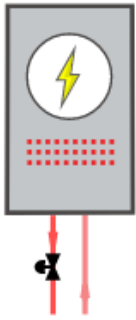
All-Electric Central Plant Equipment

Electric Boilers

Description

Electric boilers generate hot water or steam using electric resistance heating elements rather than combustion, operating at near 100% conversion efficiency or a coefficient of performance (COP) of 1. Electric boilers are simple, low maintenance, quiet, and compact, require no flue or gas piping, and can support efforts to eliminate on-site carbon emissions. In jurisdictions that require all-electric systems, electric boilers are one possible solution for space heating. Units are available in a wide

ELECTRIC BOILER



range of sizes that accommodate a broad range of applications. For steam applications, electric boilers are effectively the only option for all-electric building systems.

Conceptually similar, electric duct heaters use electric heating elements directly in the airstream. Electric duct heaters are very cost competitive, compared to hot water systems, because they eliminate the need for expensive pipe distribution. Standard duct heaters are single stage or multi-stage, but modulating capacity control is available with silicon controlled rectifiers (SCR), which can improve temperature control capability.

Challenges

Electric boilers can impose large electrical loads. For large systems, the electrical infrastructure requirements can be very expensive, impacting service and transformer sizing. Compared to typical gas-fired heating systems, the operating cost for electric boilers can also be very high due to the relatively high cost of electricity relative to natural gas and lack of efficiency multipliers seen in heat pumps.

Electric resistance heating is prescriptively banned for space heating by the California Building Energy Efficiency Standards (Title 24) Section 140.4(g) and the Washington State Energy Code (WSEC) Section C403.1.4, except in specific circumstances, such as for backup to the heat pump heating system or where only used as a very small portion of the overall building heating capacity (California Energy Commission, 2025) (Washington State Building Code Council, 2021).

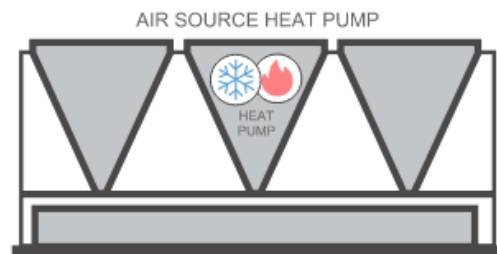
For these reasons, in California, electric boilers are often only applied as trim or backup heat sources and electric duct heaters only in limited applications.

Air-to-Water Heat Pumps

Description

Air-to-water heat pumps (AWHP) extract heat from outdoor air and transfer it into a hydronic loop that delivers hot water to coils, fan coils, radiant systems, or other hydronic heating terminals. AWHPs can operate in heating and cooling modes by reversing the refrigeration cycle through a reversing valve.

Compared to electric boilers, the refrigeration cycle in AWHPs offers higher efficiencies, sometimes greater than 2 - 3 COP, depending on conditions. Commercial AWHPs are increasingly used for electrification of central plants and can be paired with water-to-water heat pumps or thermal energy storage to increase capacity or enable heat recovery. Basic 2-pipe AWHPs have a reversing valve that allow them to operate in either heating or cooling mode using the ambient air as a heat source or sink, respectively. 4-pipe AWHPs have separate connections and provide the ability to simultaneously generate hot and chilled water simultaneously. A common misunderstanding is that 4-pipe AWHPs provide heat recovery capability. In some 4-pipe units, the simultaneous heating and cooling is achieved by operating separate circuits in the different modes using the ambient air as the sink and



source, rather than recovering from one loop to another. In some 4-pipe units, heat recovered from one loop to the other.

Many AWHPs utilize fixed speed scroll compressors but there are now newer modular, variable speed options that provide greater turndown capability and efficiency at low part loads.

Challenges

SIZE

AWHPs face two unavoidable spatial constraints compared to gas-fired boilers: they require outdoor installation and require roughly a full order of magnitude greater equipment area for each unit of heat output. Both constraints result from the fact that AWHPs rely on large finned tubed coils for heat transfer from ambient air to refrigerant to support the vapor-compression refrigeration cycle whereas natural gas boilers utilize the heat of combustion. Figure 1 provides a visual comparison of the spatial requirements for a 3 MBH boiler and two arrays of 4-module AWHPs with equivalent heating capacity.

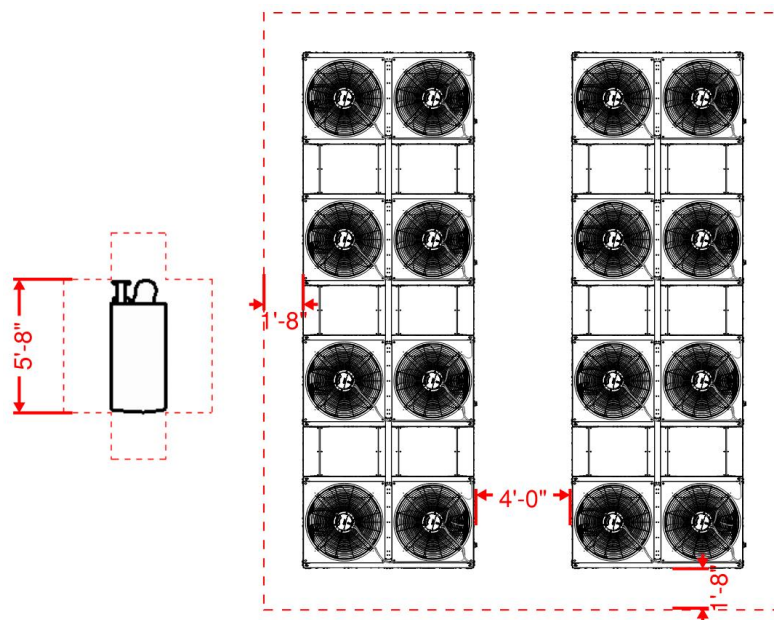


Figure 1. Size Comparison Between Boilers and Air-to-Water Heat Pumps

STRUCTURAL IMPACT

AWHPs can also impose a structural challenge because of their weight as well as the weight of their associated buffer tank(s). For instance, one manufacturer's four-module array weighs 10,700 lbs. The 1,000 gallon hot and chilled water buffer tanks that might be paired with it each weigh roughly 10,000 lbs and carry a pounds per square foot loading of 400 or more. By contrast, a water tube condensing boiler with an equivalent 1.8 MMbtu/h output might weigh only 1,800 lbs.

Since most new construction commercial applications require cooling in addition to heating, these spatial and weight penalties can be mitigated if not avoided entirely when AWHPs displace dedicated cooling-only equipment such as air-cooled chillers or water-cooled chiller plants with cooling towers. In retrofits and high rises with limited roof area to building square footage ratios, spatial constraints can, however, be a nearly insurmountable challenge. Consider an occupied 6-story multi-tenant building served by an existing rooftop boiler plant and direct expansion (DX) variable air volume (VAV) box cars. In such a scenario, the rooftop may already be crowded and AWHPs would require much more space than the boilers they are replacing. Both the spatial and structural constraints might therefore limit the viability of a full AWHP retrofit. In such a scenario, partial electrification might be

the path that proves most practical from a first cost, energy cost, and carbon emissions reduction perspective.

CLIMATE

Climate imposes important design constraints on AWHP solutions that do not exist for natural gas boilers. Perhaps most critically, there are ambient temperatures below which AWHPs cannot function. For one product, this operational limit is -13°F. Additionally, as the design ambient temperature decreases, the maximum achievable leaving water temperature also decreases. In the case of this product, the maximum design hot water supply temperature decreases from 140 to 122°F as design ambient temperature decreases from 32 to -13°F. Design capacity and efficiency also decrease as ambient design temperature decreases (Figure 2). As an

example, design capacity for one 2-pipe AWHP operating at 125°F hot water supply temperature decreases from 354 to 206 MBH and efficiency decreases from 2.06 to 1.84 COPh as ambient temperature decreases from 35 to 20°F. In simple terms, the more extreme the climate, the less attractive AWHP solutions become from a first cost and design efficiency perspective. In cold climates, AWHPs must be coupled with backup options like gas-fired or electric boilers.

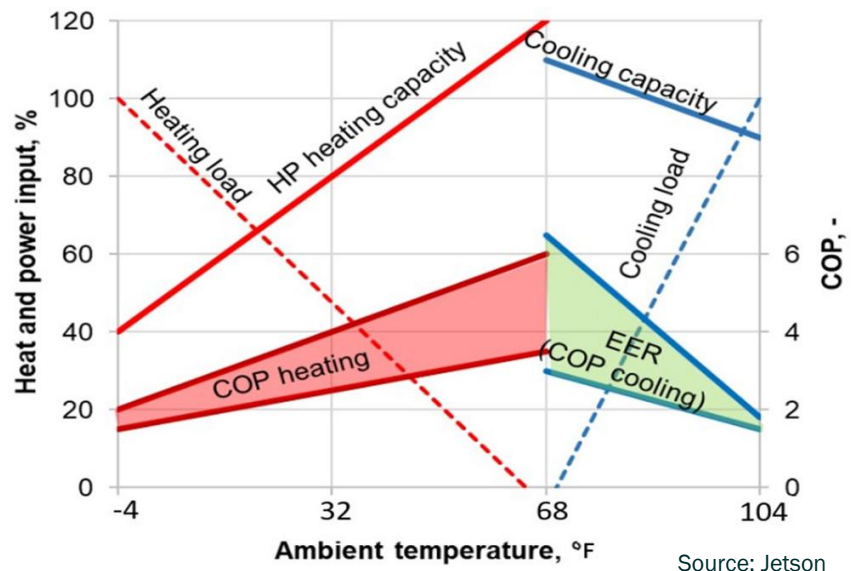


Figure 2. Dependence of Capacity and COP on Ambient Temperature

ELECTRICAL

In heating dominated climates, AWHPs pose a unique challenge since they can make buildings winter peaking instead of summer peaking. In a new construction context, this usually is not a major challenge since AWHP demand load can be designed for from the start. In retrofit contexts, it can occasionally be a major issue if the facility's service is already operating near maximum capacity. The reality, however, is that many if not most facilities have oversized electrical infrastructure relative to their demand loads because of inherently conservative prescriptive demand factor requirements imposed on electrical engineers during design by the National Electrical Code. Owners and engineers can employ a 30 day load study, as allowed under NEC 220.87, to validate that existing infrastructure can support the increase in demand load imposed by the proposed air-to-water heat pump solution. When evaluating the potential impacts of AWHPs, whether in a new or retrofit context, electrical engineers should also employ the non-coincident load language from NEC 220.60 to ensure that demand loads are not calculated based on an unrealistic scenario with both heating HVAC equipment and cooling HVAC equipment peaking concurrently.

COST

AWHPs require large air coils to effectively extract heat from cold ambient air. Considering equipment costs alone, AWHPs are significantly more expensive per unit 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. When space, structural, and electrical costs are factored, there can be an even greater cost add.

For AWHPs with heat recovery capability, there is a wide range in cost premium compared to the non-heat recovery options. For standard AWHPs, heat recovery adds nearly 80% in additional cost, so designers need to carefully consider whether the capability is needed and justified. For newer modular AWHPs, the cost add can be much smaller.

TEMPERATURE LIMITATIONS

Conventional AWHPs can generate hot water temperatures up to 120 to 140°F, though efficiency decreases at higher temperatures and the upper limit decreases in colder weather. In new construction contexts, hot water supply temperature is a design variable adjusted to optimize for first costs and ongoing energy costs. In retrofit contexts, determining whether the existing distribution systems and coils are suitable for the lower supply temperatures afforded by AWHPs is a critical evaluation criterion. In most cases, AWHPs replace existing natural gas boilers that have historically been designed at supply temperatures ranging from 140 to 180°F. Where the pre-existing design supply temperature is not much greater than an AWHP can stably provide (e.g., 150°F) and most load coils are 2-row, the likelihood that an AWHP system delivering 135°F will be problematic is low. At worst, pumps may need to be upsized and a few terminal units that are short on capacity may need to be replaced. If, however, the pre-existing system were designed from 180°F supply water and uses 1-row coils, then it is highly likely that the pre-existing distribution system, pumps, and coils, would not support a direct AWHP conversion due to the significantly lower coil capacities at lower supply water temperature.

Where real world system manipulations cannot be implemented, theoretical analysis is the backstop. Information about installed coils can be collected and their performance can be re-run through selection software using the new target hot water supply temperature. Where model information regarding the coils cannot be gathered, or selection software specific to the installed coils is no longer available, the next best option is to gather operational data from the existing coils and calibrate a surrogate coil in selection software to roughly match installed performance. For instance, if coil airflow, entering air temperature, leaving air temperature, entering water temperature, and leaving water temperature are known for an installed coil, a coil of equivalent dimensions and rows can be mocked up in software to match installed performance by manipulating fin density and feeds. While such a procedure may not be practical at building scale with 10s or 100s of coils, it can nonetheless provide a reasonable indication of whether pump, coil, or distribution issues are likely to arise and an opportunity to account for them in design.

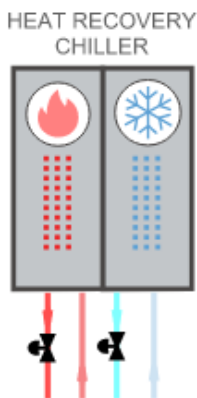
ACOUSTICS

Relative to natural gas boilers, large arrays of AWHPs can be quite loud with sound power levels potentially exceeding 90 dBA. Care must be taken when siting AWHPs relative to property lines where noise ordinances may exist, or placing them near or above acoustically sensitive spaces, e.g., conference rooms, without appropriate treatment. These constraints in and of themselves are rarely insurmountable. Sound barriers, proper vibration/isolation, and other noise mitigation measures are

often sufficient to address noise challenges, but their costs and complications should be included in any comprehensive analysis nonetheless.

Heat Recovery Chillers

Description



Heat recovery chillers (HRC) are water-to-water equipment that generate chilled water and are also able to use the rejected heat to simultaneously provide useful heating hot water. HRCs operate at higher lift conditions than typical heat rejection chillers that reject heat to condenser water systems. As a result, for a given cooling load, a HRC would generally draw more power than a comparable heat rejection chiller, but would be able to achieve a much higher combined heating and cooling COP because of the “free” recovered heat. The high combined COPs make HRCs an excellent option for decarbonization for the right applications.

HRCs with scroll compressors are available in small-to-mid tonnage ranges (<200 tons), are often modular in nature, and typically provide 120-140°F hot water temperatures. With modular units and variable speed modulation, scroll HRCs can provide excellent turndown and high efficiency under part load conditions. HRCs with screw compressors are common in mid-to-large tonnage ranges, provide good performance for variable loads, and can often generate 140-160°F hot water temperatures, or higher. Centrifugal HRCs are common in large tonnages and can provide excellent full load efficiencies but limited lift capability limits them to 120-140°F hot water temperatures.

Challenges

A primary limitation with heat recovery chillers is the need for simultaneous and balanced heating and cooling loads. Application is best suited where there are significant concurrent loads for a large portion of the year, such as in hospitals, laboratories, and campuses, or when coupled with thermal energy storage. Load imbalance may be addressed by staging on other equipment dedicated to cooling or heating, or through the use of dual condensers which can either recover or reject heat. Control is more complex for HRCs due to the need to maintain both chilled water and hot water supply temperatures.

New Design Considerations

Codes and Regulations

Building energy codes in an increasing number of states are driving a shift toward all-electric HVAC systems in large commercial applications. Examples of states where codes are pushing owners and designers in the direction of heat pumps include Washington, New York, California, and Colorado.

Section C403.1.4 of the WSEC prohibits the use of electric resistance and fossil fuel-fired heating equipment, effectively requiring heat pump heating in commercial buildings with a limited number of exceptions (Washington State Building Code Council, 2021). A separate fossil fuel compliance path

was carved out in the WSEC in response to legal challenges but includes additional requirements for added efficiency including purchasing energy efficiency credits and electrification-readiness.

New York's All-Electric Buildings Act was to require most new buildings seven stories or less to be all-electric effective January 1, 2026, and the expanding to all new buildings with limited exceptions starting January 1, 2029. The implementation date, however, has been delayed as of the time of this writing pending the outcome of legal challenges.

California's Title 24 includes all-electric HVAC systems as the primary means for prescriptive compliance in many building types; uses all-electric systems as the performance compliance path baseline for most building types; and includes an hourly life cycle cost metric known as Long-Term System Cost (LSC) that is biasing further in favor of all-electric solutions in each successive code cycle (California Energy Commission, 2025).

California's air quality agencies have also enacted regulations that will ban (Bay Area Air Quality Management District, 2023) or proposed regulations that will limit the sale (California Air Resources Board, n.d.) of gas-fired water heaters, boilers, and furnaces over a range of dates. These regulations will effectively expand the applicability of electrification requirements to existing buildings in need of equipment replacements, as well as further limit flexibility for new construction. Colorado and Washington also both have laws that drive electrification of existing buildings stock through Building Performance Standards (BPS). These laws establish building energy use budgets unique to each building based on ASHRAE Standard 100. The BPSs generally require the use of building energy use benchmarking and then auditing to reduce site energy use or emissions within a code mandated timeframe. When these standards are not met, recurring financial penalties can be levied for non-compliance. Many other cities and localities have also established similar BPSs.

While the above list of codes and regulations is not comprehensive, it does illustrate that jurisdictions are increasingly adopting measures to motivate or require heating electrification. Designers should be cognizant of new and evolving codes and regulations that relate not just to construction but also to ongoing building operations, whether for BPSs or for end of life equipment replacements.

Utility rates

Electricity and gas rates can vary markedly from one region to the next. As a result, the life cycle energy costs associated with converting from natural gas heating to all-electric AWHP heating can favor AWHPs in one locale, while strongly favoring gas boilers in another. A good example of this dichotomy is the difference in commercial electricity and gas rates between Boston, MA and Houston, TX, and the resulting difference in effective costs per MMBtu of heating output from AWHPs and boilers. In Houston, the combined transmission and generation electricity rates are low relative to the gas rates, while in Boston, the electricity rates are high relative to the gas rates. Assuming a heat pump COP_h of 2.5, and a boiler thermal efficiency of 85%, the net result is that AWHP heating reduces energy costs in Houston but increases energy costs in Boston, compared to boilers. This analysis is of course simplified and does not account for dynamically varying heat pump COP_h, nor does it account for utility demand charges or the positive impact of heat recovery for 4-pipe AWHP solutions. Unfortunately, in the case of this analysis, varying COP_h values and demand charges would only bias the results more in favor of boilers in Boston. Ambient temperatures are on average cooler in Boston than Houston—and thus AWHP COP_h lower—and the representative utility in Boston

from which the utility rates in Table 1 were derived has high maximum demand charges that penalize heat pump heating in the winter months when AWHPs are likely to impact maximum demand.

Table 1. Simplified Comparison of AWHP and Boiler Heating Costs per Unit Heat Output in Houston and Boston

City	Electricity Rate (\$/kW)	Natural Gas Rate (\$/therm)	Cost Per Heat Output (\$/MMBtu)	
			All-Electric Plant	Boiler Plant
Houston	\$0.08	\$1.00	\$9.38	\$11.76
Boston	\$0.19	\$1.71	\$22.27	\$20.12

Similar to emissions analyses, a rigorous cost analysis would require hourly heating and cooling load profiles, dynamic equipment efficiencies, and detailed utility rates schedules. Nonetheless, the sort of simplified analysis shown above can yield a ballpark indication for whether electrification is likely to appreciably increase or decrease utility costs early in the project scoping phase.

Even within a narrow geographic region, there can be wide variability in the structure of electric rate tariffs. Figure 3 shows the range of electricity demand charges for three Bay Area utilities: Pacific Gas & Electric (PG&E), Peninsula Clean Energy (PCE), and Silicon Valley Power (SVP). SVP has optional time-of-use (TOU) and fixed rate schedules. Summer on-peak demand charges vary by nearly a factor of 4X between some tariffs, and nearly 8X if the maximum demand occurs during the peak period for PG&E. PG&E has a strong winter demand charges whereas PCE and SVP have small to no charges. Where there are large demand rates, demand charges can often be responsible for the overwhelming majority of the monthly electricity bill. Electricity energy charges for the three utilities are shown in Figure 4 but do not vary as widely. PG&E is the primary investor-owned utility in Northern California, PCE is a community choice aggregator serving San Mateo County, and SVP is a small municipal utility district serving the city of Santa Clara. The variability across these utilities demonstrates that large differences in rates may occur within a local scale, not just regionally, and that in some cases building owners may have a choice of providers that offer very different rate structures, and which may have a significant impact on the economics of all-electric central plants.

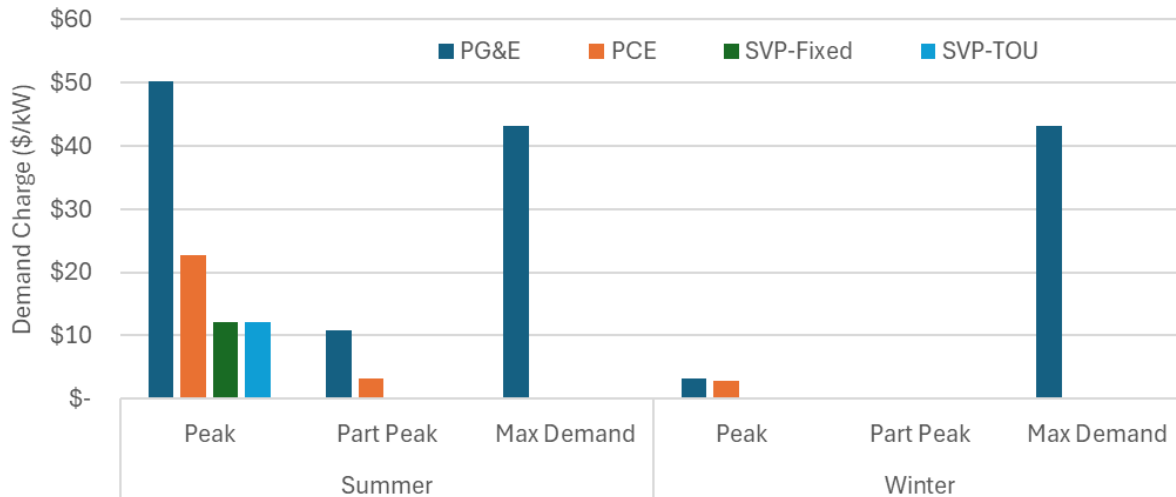


Figure 3. Electricity Demand Charges for Select Bay Area Utilities

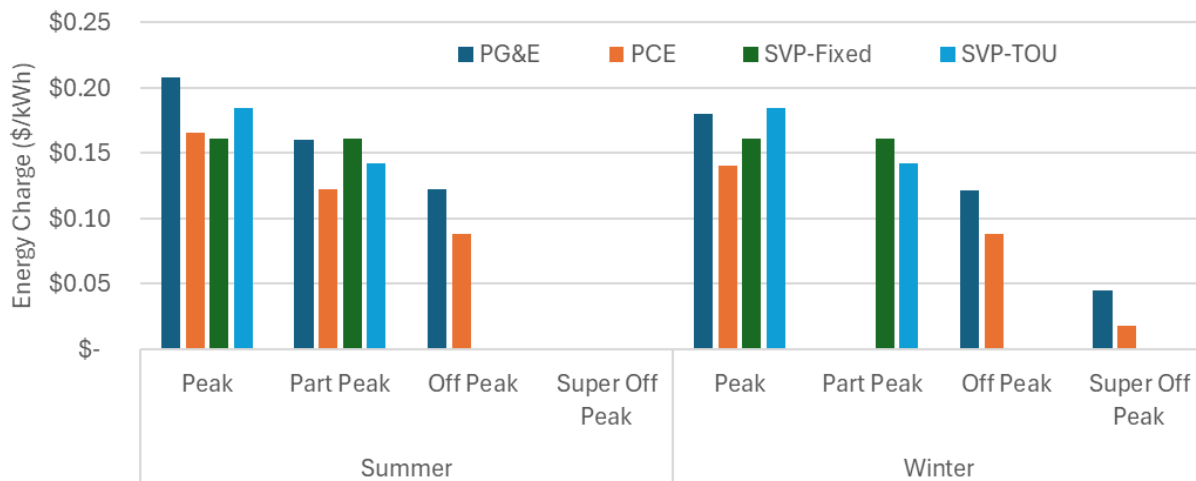


Figure 4. Electricity Energy Charges for Select Bay Area Utilities

The so-called “spark gap” remains an issue where electricity costs continue to remain high and natural gas costs continue to remain relatively low, such that electrification efforts cannot guarantee on-bill savings. The net effect is a conflict for many decision-makers who choose or are forced to electrify buildings despite higher first costs and higher operating costs. Projections of future changes to tariffs by PG&E through 2030 only suggest incremental increases in rates (Pacific Gas & Electric, 2025).

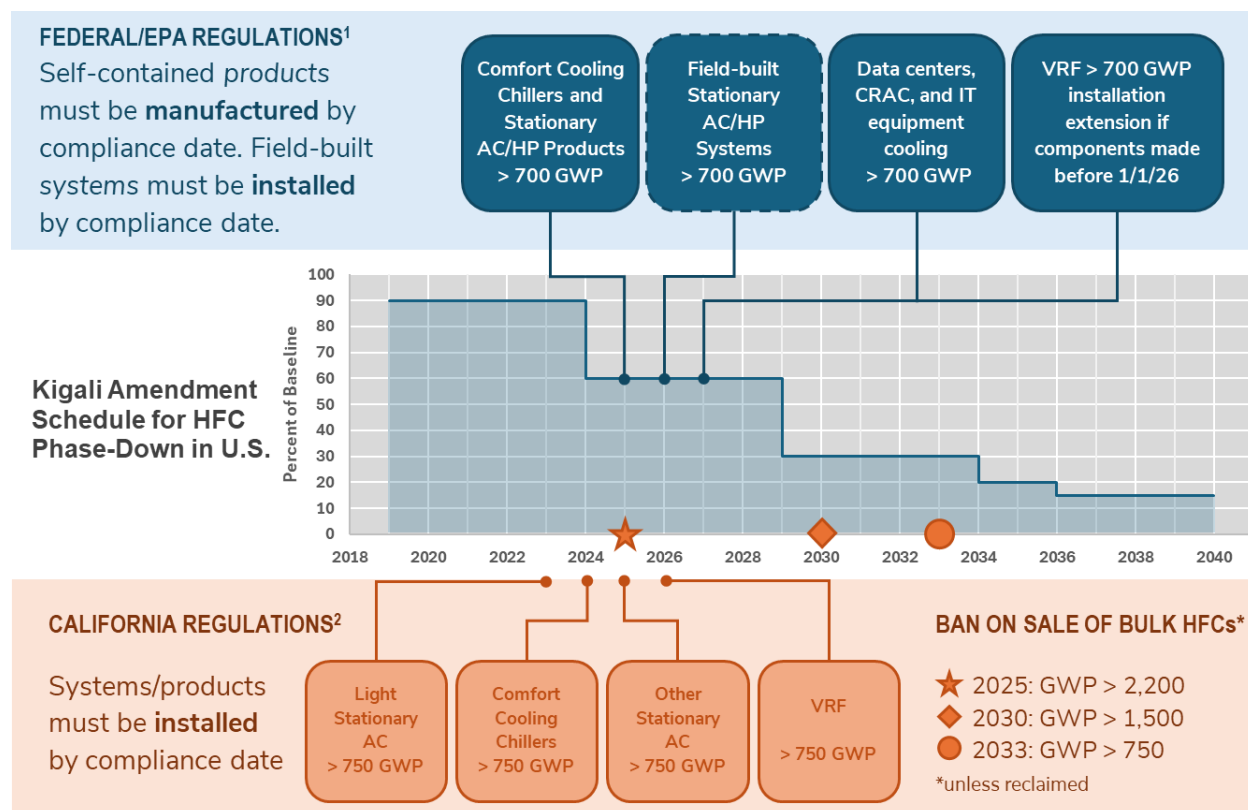
Refrigerant Restrictions

Recent restrictions on refrigerant global warming potential (GWP) at the federal and state levels have forced HVAC equipment manufacturers to adapt refrigeration equipment product lines.

The 2016 Kigali Amendment is an international agreement to the Montreal Protocol that phases down global production and consumption of hydrofluorocarbons (HFCs) due to their high GWP. The

United States formally ratified the Kigali Amendment in 2022, committing to reduce HFC consumption by 85% below baseline levels by 2036.

The EPA's [Significant New Alternatives Policy](#) (SNAP) Program, established under Section 612 of the Clean Air Act, manages substitutes for ozone-depleting substances and regulates refrigerant acceptability by end use. However, a 2017 D.C. Circuit Court ruling limited the EPA's authority under SNAP, determining that it could not require the replacement of non-ozone-depleting substances (such as HFCs).



Source: Taylor Engineers

Figure 5. Federal and California Refrigerant Regulations

To restore regulatory authority, Congress passed the [American Innovation and Manufacturing](#) (AIM) Act of 2020, granting the EPA explicit authority to phase down HFCs in alignment with Kigali. Under this authority, the EPA issued the [Technology Transitions Rule](#) (TTR) in 2023, which sets timelines for transitioning away from high-GWP refrigerants in new equipment across multiple sectors. Some subsequent rulemakings have proposed or adopted modifications to portions of the initial TTR requirements. The AIM Act mandates an 85% reduction in HFC production and consumption by 2036, consistent with Kigali, and provides the EPA with authority to issue additional regulations beyond the current TTR to achieve future phase-down steps.

Several states have enacted their own refrigerant regulations. California's Cooling Act (SB 1013) adopted HFC restrictions similar to those in prior federal SNAP rules and was implemented through [California Air Resources Board \(CARB\) regulations](#). [California SB 1206](#) further restricts the sale and distribution of high-GWP HFCs.

Figure 5 presents an overview of federal and California regulations for refrigerants used in stationary HVAC applications. Though the limits and effective dates differ slightly, each enacted phased requirements for different equipment classes over a range of years. A critical consideration is that the federal and state regulations were just the first step in addressing the U.S. commitment to the Kigali Amendment. The HFC phase down schedule for the U.S. includes another significant step in 2029, but the associated regulations for that have not yet been proposed. The implication is HVAC manufacturers have had to decide whether to develop new product lines that meet the current requirements or the anticipated future targets. The risk with the former is that new product lines may only have a short product life, making it difficult for manufacturers to recoup research and development costs. The risk with the latter is uncertainty about what the future restrictions will entail and when they will be enacted, but this approach offers the potential for longer product lives.

The transition away from high-GWP HFCs is driving a shift in the commercial HVAC industry toward alternative refrigerant classes.

Whereas most commercial HVAC systems have historically used A1 safety group refrigerants (as defined by ASHRAE Standard 34: low toxicity, non-flammable), new regulations increasingly favor A2L refrigerants (low toxicity, mildly flammable). A summary of common HVAC refrigerants is provided in Table 2.

The mild flammability of A2L refrigerants has important design and code implications, particularly regarding machinery room requirements and safety controls in ASHRAE Standard 15 (Table 3). Where A2L refrigerants are used in machinery rooms, Std 15 requires significantly higher emergency ventilation rates compared to A1 refrigerants, where the primary safety concern is asphyxiation from the displacement of oxygen during a leak. An additional requirement for A2L refrigerants is the automatic de-energization of refrigerant equipment and unclassified electrical devices larger than 1 kVA.

Table 2. Common HVAC Refrigerants

Refrigerant	Pressure Conditions	GWP (kg CO2 eq)	ASHRAE Class
R-123	Low	77	B1
R-514A	Low	1.7	B1
R-1233zd(E)	Low	1	A1
R-134a	Medium	1430	A1
R-513A	Medium	630	A1
R-515B	Medium	298	A1
R-1234yf	Medium	6	A2L
R-1234ze(E)	Medium	4	A2L
R-410A	High	2088	A1
R-454B	High	467	A2L
R-32	High	675	A2L

Low-pressure applications: large centrifugal compressors.
Medium-pressure applications: screw and smaller centrifugal compressors.
High-pressure applications: scroll compressors, unitary and packaged equipment.

Table 3. ASHRAE Standard 15 Machinery Room Requirements

Focus	A1 Refrigerants	A2L Refrigerants
Construction	Self-closing doors (8.9.2)	Non-combustible, 1-hr rated construction (8.11.3) Self-closing fire doors (8.11.2) Pipe penetrations tightly sealed (8.11.5)
Emergency shutdown button	None	Located outside door (8.11.7)
Emergency ventilation	Mechanical ventilation with detections above limit (8.9.8)	Mechanical ventilation with detections above limit, ventilation rate ~5X to 20X higher than A1 rate, additional inlet requirements
Additional Measures	None	Upon detection above limit, automatically de-energize refrigerant equipment and unclassified electrical devices > 1 kVA (8.11.6.2)

Note: This is a simplified summary comparing key machinery room requirements. See Standard 15 or applicable codes for more information or for differences in requirements for high probability systems or for other refrigerant classifications

Equipment Sizing and Selection

In conventional central plant design, the chilled water and hot water plants are separate systems and design calculations are performed independently, primarily based on peak design conditions. Consideration of performance under part load conditions is usually given for chilled water systems, such as through variable speed control, multiple staged units, and pony chillers. Traditionally, part load performance for hot water plants has not drawn as much attention, with many plants containing only a single boiler, limited stages for capacity control, and limited turndown capability. Recent research, however, has shown that gas-fired boilers are commonly oversized by a factor of two heating hot water systems, yet often operate at very low part load ratios with performance significantly degraded under these conditions (Raftery, Singla, Cheng, & Paliaga, 2024).

Oversizing and Turndown Capability

The design of efficient all-electric central plants demands closer evaluation of expected loads. Peak loads should be evaluated carefully to avoid oversizing. The conventional wisdom for gas-fired hot water plants is that there is little downside to oversizing. Higher capacity boilers are only incrementally larger and more expensive, whereas conservatism protects the designer and reduces the risk of capacity shortfalls. The existing paradigm of oversizing is not appropriate for AWHPs because they are often 3 to 5 times as expensive per unit heating output as equivalent natural gas boilers and require an order of magnitude more space. Cost and spatial impacts can be mitigated in new construction applications where AWHPs provide both cooling and heating and therefore displace air-cooled chillers, however retrofit applications do not benefit from this synergy where AWHPs solely replace existing heating systems. Given the significantly higher first costs of all-electric heating

equipment compared to gas-fired boilers, plus the electrical, structural, and space constraints discussed above, there is a much larger motivation to avoid oversizing all-electric heating equipment.

Aside from eliminating unnecessary safety factors in load calculations, some creative solutions to minimize heating loads exist. For instance, extending the morning warm up period from 1.5 hours to 3 hours and slowly ramping up zonal setpoints during warmup as opposed to step-changing setpoints from unoccupied setback to occupied setpoints was demonstrated in an example community college classroom lab building to nearly halve peak heating loads (Cheng, Raftery, & Wendler, 2024B). While this strategy may not impact sizing in a new construction context where warmup does not dictate sizing, it can impact sizing in a retrofit context where new equipment can be sized based on historical trends of maximum loads, which often occur during warmup.

Required equipment turndown is a critical component to consider in design. Turndown of chillers and AWHPs must be low enough to ensure that compressors are not cycling often. Frequent compressor cycling results in poor supply water temperature control, inefficient operation, and can also damage oil-lubricated compressors. Where conventional equipment sizing practices generally focus on peak conditions, multiple recent studies have reported frequent system operation at very low part loads and the negative impact of short cycling on plant efficiency (Raftery, Singla, Cheng, & Paliaga, 2024; Weitze, 2024; Cheng, et al., 2026). A common design pitfall is not considering all of the startup phases of the building(s). Often the “day 1” loads may be much lower when compared to the final full design loads. The plant turndown capability should consider the lowest possible operating state and loads to avoid frequent compressor cycling. This may require the use of a modular AWHPs with scroll compressors instead of a screw or centrifugal compressor type chillers, or unequal sized units.

Where central plant equipment are expected to serve both heating and cooling loads, designs should also consider the annualized load profiles and periods of concurrent heating and cooling loads. Where there are relatively few staged equipment, careful consideration should be given to the capability of providing adequate capacity to both heating and cooling under different loads and equipment stages. For example, consider a plant with two 2-pipe changeover AWHPs. During times when both hot water and chilled water are required, one AWP must have sufficient capacity to serve the entire heating and cooling load. Heat recovery, discussed in the next section, should also be evaluated carefully to understand how much simultaneous heating and cooling may be expected on an annual basis, and whether and how much heat recovery capacity is appropriate.

New and Evolving Product Lines

An additional challenge with equipment selection is that all-electric central plant equipment product lines are generally fairly new and are continuing to rapidly evolve. Conventional chillers and boilers have long since been established and mature, with relatively minor incremental changes to equipment capabilities over time. By contrast, AWHPs are relatively new and still actively evolving. In many cases, manufacturers, contractors, and designers are still learning the boundaries of appropriate applications, and the limitations and challenges for installation and operation. Manufacturers have taken different approaches in their solutions offered. Equipment capabilities can be very different from one product line to another and actively changing.

A research project funded by the California Energy Commission compiled a [summary of all-electric HVAC equipment](#) for large commercial buildings, to compare and contrast some of the differences in

equipment capabilities (Taylor Engineers, 2025). The summary includes definitions of different heat recovery approaches, highlighting differences in equipment capabilities to address imbalanced loads in heat recovery mode. Though the summaries of available AWHP and water-to-water heat pump (WWHP) equipment is incomplete and will fall out of date as product lines evolve and new ones are offered, it serves as a useful index to understand general similarities and differences in equipment offerings.

Heat Recovery

Heat recovery potential is a critical design consideration that can meaningfully impact the lifecycle energy cost and carbon emissions for any project evaluating all-electric central plants. Heat recovery offers the opportunity to significantly improve system efficiency for the right applications but the anticipated heat recovery potential must be carefully considered. Common building types that offer good heat recovery potential include: hospitals, laboratories, hotels, offices with moderate dry labs or server rooms. Many commercial applications only have limited heat recovery potential on an annual basis because the heating and cooling loads are asynchronous or only have a minor degree of overlap, often below the turndown capability of some equipment. Typical load calculations used for sizing conventional equipment only consider the separate peak design conditions. Evaluating annualized load profiles requires the use of building energy simulations and reasonable assumptions of internal load profiles, where energy models commonly assume unrealistically high internal gains schedules that don't account for diversity, partial occupancy, lighting controls, etc. and may exaggerate cooling loads.

Consider a cold-climate application that has a significant cooling baseload from manufacturing, dry labs, or IDF/MDF rooms. In such a scenario, heat recovery machines can provide “free” cooling while heating with a heating coefficient of performance (COP_h) greater than 3.5. Contrast this to the alternative without heat recovery. Cooling might still be “free” via an air or waterside economizer, but 2-pipe AWHP heating would be performed at a COP_h of less than 2.0 at below freezing ambient conditions. The difference in heating efficiency is dramatic and can mean the difference between a project that makes financial sense due to ongoing energy cost reductions and one that might not have a reasonable payback, if any, for the owner.

When hourly heating and cooling load profiles are available from either an energy model or historical trend data, estimating the percentage of annual heating that could theoretically be accomplished using heat recovery is a useful exercise. To do so, engineers can apply the following procedure:

1. For each hour, multiply the cooling load by $\frac{COP_h}{COP_h - 1}$, where COP_h is the system's estimated average heating efficiency in heat recovery mode. This is the heating output available from the cooling load for that hour.
2. For each hour, record the lesser of the above value and the actual heating load for the hour. This represents the system's maximum possible heating output for the hour.
3. Sum up the values from the preceding step for each hour of the year and divide by the annualized heating load. This is the percentage of annualized heating load that could theoretically be satisfied using system's heat recovery.

If the load data used in the above analysis come from an energy model, any economizers included in the model should be disabled prior to running the model. This should be done to maximize the heat available for recovery from the chilled water loop during each hour. In the real world, most buildings should be operated to maximize heat recovery up to that needed to satisfy the heating load for the hour, then economize to meet the remainder of the cooling load. Heat recovery is usually preferable to economizing since cooling is “free” either way, but heating is generally more efficient when recovering energy from chilled water as opposed to absorbing energy from ambient air.

The above analysis breaks down when storage is introduced into the design since energy storage decouples the time dependence of the heating and cooling loads.

Economizer throttling

Heat recovery is typically preferable to airside economizing in all-electric plant designs when ambient temperatures are less than 60 to 65 °F. An optimal sequence would therefore incrementally throttle airside economizers whenever the hot water loop was enabled, ambient temperatures less than 62 °F (adj.), and any heat recovery units are in heating only mode. This logic would reset maximum allowable economizer position in 5% to 10% increments, every 5 to 10 minutes, until no modules were operating in heating only mode, or the economizer were shut, i.e., air handlers operating at minimum ventilation. Once no modules were in heating only mode, the logic would progressively back off in small increments to only throttle economizing as much as necessary to meet the heating load with heat recovery. The logic described would be best implemented with trim and respond logic consistent with ASHRAE Guideline 36 (ASHRAE, 2024).

Pumping Configuration

There are two main methods for CHW and HHW plant pumping designs: variable-primary, and variable-primary, variable-secondary. Variable-primary plant designs have become the most common design for conventional chilled water and boiler plants, and with good reason. They are less expensive and reduce spatial requirements due to one fewer set of pumps, and reduce pumping energy relative to constant-primary, variable-secondary plants when chiller or boiler minimum flow rates are much less than design (Taylor S. , 2011). For modern all-electric plants that use modular AWHPs, the pumping energy benefit is often diminished for primary-only systems because minimum flowrates are typically a much larger fraction of design flowrate than for conventional centrifugal chillers. Minimum flow bypass control is also more challenging for modular AWHP plants. Accounting for all these considerations means that variable-primary, variable-secondary pumping option is preferable for plants with modular AWHPs.

Hot Water Coil Selection

Because heat pump systems use lower hot supply water temperatures than most conventional natural gas hot water boiler systems, proper coil selection is critical to maximize delta-T. Maximizing delta-T minimizes distribution and pump sizes, both of which reduce first costs and ongoing energy costs. A common misconception is that design primary flow—which is in some cases constrained by module minimum flow requirements—and design secondary flow must match. In practice, there is no issue with design secondary flow being less than design primary flow because of high delta-T coil selections.

In conventional VAV reheat designs, engineers should specify two-row coils instead of one-row coils as a minimum standard of care. Figure 6 illustrates the impact of the number of coil rows on delta-T and hot water return temperature over a range of supply water temperatures for a standard capacity 8-inch VAV reheat terminal at 250 cfm, with 63°F entering air temperature and 90°F leaving air temperature. Whereas a 2-row coil can achieve a reasonable 30°F delta-T with 135°F supply water temperatures, the performance of a 1-row coil degrades at lower supply water temperature to a degree that 1-row coils are effectively incompatible with AHP systems. While many vendors also offer custom three-row and four-row coils, these options come with a roughly linear increase in airside pressure drop relative to their two-row counterparts.

Furthermore, the market has expanded in recent years to include other terminal unit coil options that offer increases in waterside delta-T at lower cost and airside pressure drop penalty. The first option is an “oversized casing” that pairs a standard size inlet and flow cross with a casing and coil for a terminal that is one size larger—e.g., an 8-inch terminal unit with an “oversized casing” pairs an 8-inch inlet and flow cross with a 10-inch casing and coil. This approach effectively increases coil surface area and reduces airside coil velocity for a given airflow, yielding improved heat transfer performance and a greater waterside delta-T. Multiple vendors also offer a “high capacity coil” option that has a greater fin density than conventional coils. Higher fin density also improves waterside delta-T, albeit with a minor airside pressure drop penalty. Finally, at least one vendor offers a single circuit coil (“low water temperature”) option that has only a single hot water feed instead of the two or more feeds more common to standard reheat coils based on research findings from a CEC-funded study (Wendler, Raftery, & Cheng, 2024). These options consistently allow for 25°F delta-Ts, and in some cases 30°F delta-Ts, in applications with hot water supply temperatures as low as 120°F. Figure 7 shows performance of different terminal casing options (standard vs. oversized) and coil capacities (standard vs. high capacity) for an 8-inch VAV reheat terminal at 250 cfm, with 63°F entering air temperature and 90°F leaving air temperature.

For other designs that do not leverage reheat terminal units, even greater flexibility exists. For instance, in dual fan dual duct systems that rely exclusively on central heating coils, air handler hot water coils can be selected for 10°F to 15°F air-to-water approaches, allowing for both low hot water supply temperatures and high delta-Ts. In DOAS plus 4-pipe fan coil designs, a single coil with changeover valving can be used for both chilled and hot water duty at each fan coil. These coils are invariably sized for cooling duty, meaning that they deliver a high delta-T in heating operation. The changeover coil design therefore saves not only on fan coil hardware first costs, but also hot water distribution and pumping first costs and should be considered wherever possible in 4-pipe fan coil applications. Refer to the treatment by Steve Taylor included in the references for additional details on best practice changeover coil design (Taylor S. , 2017).

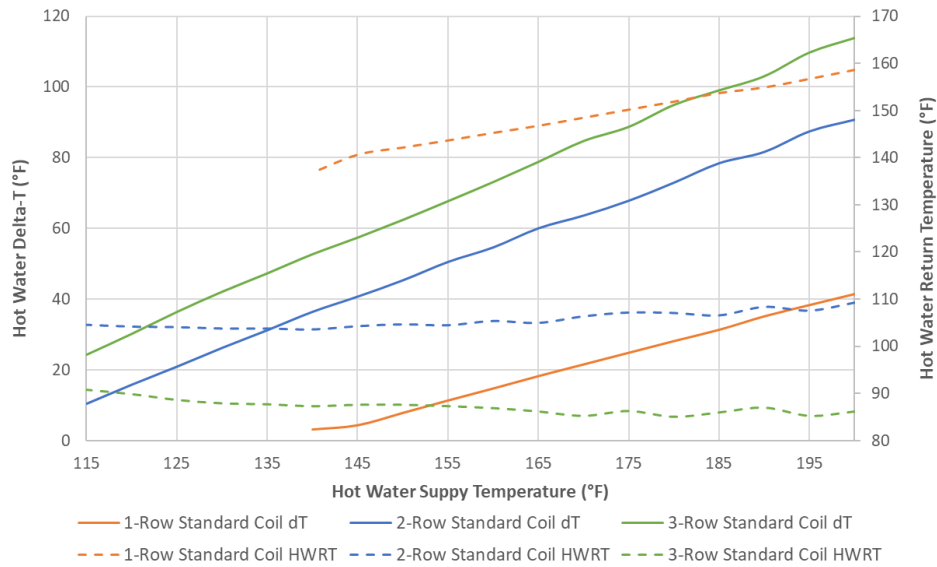


Figure 6. VAV reheat coil performance by number of coil rows

Coil performance based on a standard capacity 8-inch VAV reheat terminal at 250 cfm, with 63°F entering air temperature and 90°F leaving air temperature

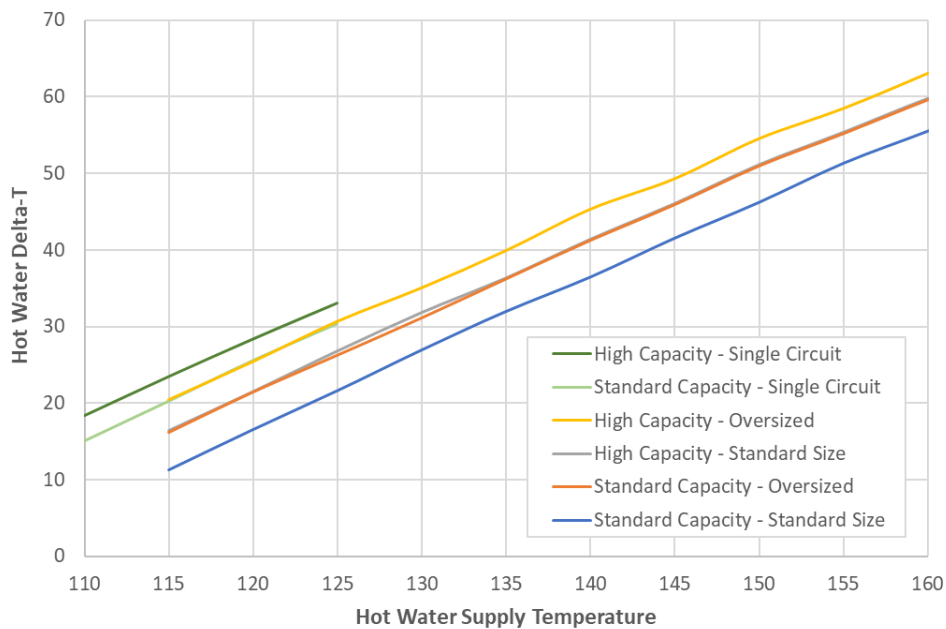


Figure 7. VAV reheat coil performance for different coil types

Coil performance based on an 8-inch VAV reheat terminal at 250 cfm, with 63°F entering air temperature and 90°F leaving air temperature

Defrost

Air-to-water heat pumps, when in heating mode, often require a defrost cycle that melts frost and ice buildup on the outdoor refrigerant coils when in heating mode. This frost buildup occurs on the outdoor AWHPs under certain outdoor conditions. Defrost loads are the highest in a dry bulb band of 20 °F to 40 °F, and when absolute humidity is high. In heating mode, the refrigerant enters the outdoor evaporator coil much lower than the outside air temperature so that the refrigerant can extract heat out of the atmosphere and provide heat to the HW loop. This cold refrigerant is often well below the outdoor dew point. Therefore, condensation occurs and freezes because the refrigerant and the coils are below 32 °F.

During defrost, heat is sourced from the hot water loop and rejected through the air coil to melt ice build-up. One defrost cycle typically lasts 3 to 5 minutes under most operating conditions. This use of heat from the HW to melt the ice means that the sum of the heat pumps' heating capacity becomes de-rated due to the defrost cycle. This heating de-rate must be considered in the design and sizing of heat pumps, especially for climates with winter outdoor design temperatures between 20 °F and 40 °F.

Another side effect of defrost is that the melted ice becomes condensate that then drains down out the bottom of the heat pump units. This condensate is a constant trickle at all times when in heating mode and at its highest rate when defrost occurs. Design considerations should be made on where to send this condensate drainage. If the heat pump units are on the roof of a building, the consistent condensate draining down onto the roof can cause biological growth and slipping hazards. Some roofing materials may have their warranty voided if there is constant water pooling. Roofs are meant to handle periodic rain events and then dry up, not constant draining throughout an entire season of the year. If it is determined by the design team that this condensate is to be drained, most jurisdictions require that the condensate for rooftop locations go to domestic drain pipes and not to storm drain.

Buffer Tanks

Buffer tanks are provided to increase loop volume which can reduce compressor cycling at low loads and minimize the risk of loop temperature changes during defrost of AWHPs. Buffer tanks are meant to be well-mixed, and not stratified, to help avoid loop temperature fluctuations. The most important consideration for locating a buffer tank, whether hot or chilled, is to place the tank where its mass will always be engaged. This means that the buffer tank should be within the primary circuit or decoupler of primary-secondary systems, and on the plant side of the minimum flow bypass in primary-only systems.

In hot water loops, the preferred location for the buffer tank is in the primary loop at the discharge from heat pump arrays. This location mitigates the impact of defrost on supply water temperature control since the cooler supply water that results from defrost mixes with the warmer water in the tank before being distributed to building loads. By contrast, locating the buffer tank on the return side or in the decoupler results in the loads immediately seeing a significant decrease in temperature during defrost, which is the scenario the tank is partially designed to avoid.

Chilled water loops with AWHPs do not have a defrost cycle. It is OK to locate a chilled water buffer tank within the primary supply, primary return, or the decoupler. All of these options yield equivalent reductions in cycling under the assumption that the buffer tank is well mixed.

Emissions Impacts

Reducing greenhouse gas emissions—whether driven by an owner directive, local codes, or otherwise—is often a central reason for electrifying building systems. Commonly, owners are interested in reducing *site* greenhouse emissions, which fall within the category of Scope 1 emissions under the Greenhouse Gas Protocol (GHG Protocol). In practice, a more holistic analysis should also consider *source* greenhouse emissions at the power plant(s). Source emissions fall within the category of Scope 2 emissions under the GHG protocol. Consider that electric boilers are a means of reducing *site* (Scope 1) greenhouse gas emissions from heating to zero, but ultimately result in a net increase in overall greenhouse gas emissions when including *source* (Scope 2) emissions relative to natural gas boilers on an annualized basis in most United States markets, based on current grid marginal generation resources.

Scope 2 emissions reductions from all-electric plant designs are best estimated with an hourly analysis that includes the following key inputs:

- Hourly annual heating and cooling load profiles.
- Estimates of all-electric plant and baseline option (e.g., natural gas boiler and air cooled chiller) efficiencies.
- Hourly grid marginal emissions factors for the project location.

The most common sources for hourly load profiles are annual energy simulations using tools like EnergyPlus and/or historical trend data for retrofit projects. For early design phase analyses where such profiles may not be readily available, simplified profiles assuming simple correlations between occupancy schedule, load, and ambient temperature often prove of value for rough order of magnitude (ROM) analyses.

Static equipment efficiencies are also often acceptable for early high-level analyses but efficiency models leveraging ambient air temperature and load independent variables are recommended for more exhaustive analyses. If ambient temperature is included as an equipment efficiency model independent variable, then hourly weather data are also required for the analysis; these data are readily available from the [National Laboratory of the Rockies](#) (NLR).

Grid marginal emissions factors account for the emissions rates resulting from the marginal (last in) generation equipment brought online to serve an increase in load. These factors therefore better reflect the impact of increasing electric load on the grid relative to average emissions factors, which account for the full mix of generation assets online during any hour in a given location. The unfortunate reality is that the marginal generation equipment is often far from the cleanest. Refer to William Ryan’s “Understanding Marginal Carbon Factors and Why They Matter” ASHRAE Journal article for a detailed treatment of this topic (Ryan, 2024).

Sources of grid marginal emissions data include the Environmental Protection Agency’s (EPA) Avoided Emissions and Generation Tool ([AVERT](#)) and NLR’s [Cambium](#) Tool. AVERT provides a source of present day short run marginal emissions rates (SRMER) for a variety of United States Regions, whereas Cambium offers both short run and long run marginal emissions rates (LRMER) for a variety of United States Regions at present and in future years under multiple future generation and emissions scenarios. Long run marginal emissions rates (LRMER) are distinguished from short run marginal emissions rates (SRMER) because they account for structural changes to the grid that

would result from a persistent, multi-year change in marginal demand. Since HVAC central plant projects typically persist for 10 or more years, LRMER rates are recommended for lifecycle analyses. Figure 8 presents a Cambium LRMER projection for California's ISO in 2025 using central (neither high nor low) estimates for generation technology costs, fuel prices, and demand growth.

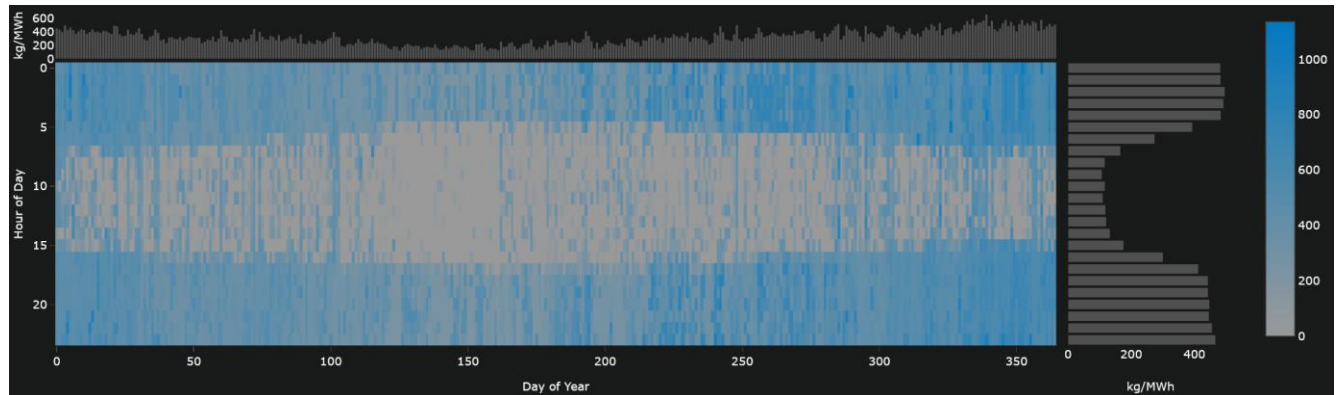


Figure 8. Projected 2025 CAISO LRMER Heat Map from Cambium

When an hourly analysis is not practical due to data input or budgetary constraints early in design, engineers can instead execute a high-level, rough order of magnitude (ROM) analysis using (a) estimated annual full load heating and cooling hours, (b) estimated annual average proposed and baseline equipment efficiencies, and (c) a best guess at load-weighted average marginal emissions for cooling and heating intuited from a graph such as Figure 8. It is important to emphasize that this sort of analysis is a ROM at best since both equipment efficiencies and grid marginal emissions are highly variable and therefore interact in difficult to estimate ways.

To bridge the gap between these two options, UC Berkeley's Center for the Built Environment (CBE) has recently published a [simplified hourly web tool](#) that supports early project phase emissions analysis using the same core inputs listed above. Simplified equipment performance maps are included in the tool, as well as real and modeled load profiles for many "typical" building applications across a variety of climates, to support carbon emissions savings projections.

All-Electric Plant Approaches

Basic AWHP Plant

AWHP plants offer a relatively simple solution for all-electric central plants. AWHPs can be arranged to provide heating-only, heating + cooling, or heat + cooling + heat recovery. Newer modular product lines provide flexibility to accommodate a range of plant sizes, while offering good turndown capability and excellent part load performance, particularly with variable speed models.

Nevertheless, there are limitations to AWHPs approaches:

- AWHPs are inherently inefficient during cold weather, and cannot operate in very cold conditions. While a design COPh above 2 is possible in mild West Coast climates, efficiency and capacity both drop rapidly as ambient temperature falls.

- AWHPs generally have limited operating ranges, with most units only able to produce 130-140°F hot water supply temperatures, which is often incompatible with existing hot water distribution and equipment in buildings. Furthermore, COPh decreases as AWHPs generate higher temperature hot water.
- AWHPs are also very expensive per unit capacity (roughly \$150 to 200 per kBtu/h of capacity, compared to \$15 to 30 per kBtu/h for condensing gas boilers), and have a large footprint (making that approach infeasible for existing buildings with limited space).
- Heat recovery is limited to applications where there are significant simultaneous heating and cooling loads.

See sections above on Air-to-Water Heat Pumps and Heat Recovery for more extensive discussions.

Limited turndown capability and extended operation at low part loads for non-modular units has been found to significantly impact operational COP in some applications. See Case Study #1 and (Weitze, 2024).

Plants with Heat Recovery, Storage, and Trim (TIER)

For larger central plants with significant heat recovery opportunities and design day heating load profiles with a large range between peak load and minimum load, designers should consider the suitability of plants that integrate heat recovery, TES, and a trim heat source.

Time independent energy recovery (TIER) is an innovative all-electric heating and cooling plant concept that integrates heat recovery chillers, thermal energy storage (TES), and AWHPs to overcome the shortcomings of alternative all-electric plant configurations. Traditional TES is used to shift or reduce peak cooling loads, whereas TIER leverages TES for heat recovery. The result is an all-electric system that maximizes heat recovery and smartly deploys the 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 (Gill, 2021; Stein & Gill, 2024).

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.

TIER with Condenser Water Storage

Figure 9 illustrates one TIER plant approach using condenser water as the TES medium. The building's cooling loads can be met by either a cooling-only chiller or HRC with the rejected heat stored in the TES tank as condenser water. When the TES tank is fully charged, heat is rejected instead to the atmosphere through cooling towers. The building's heating loads are served by the HRC using the condenser water TES as the heat source. When there is insufficient heat recovered, AWHPs can be used as a trim heat source to charge the TES tank. The TES allows for heat recovery across asynchronous heating and cooling loads. By maximizing heat recovery and storing recovered heat in the TES, the required AWHP capacity can be significantly reduced. The AWHPs can also be run at off peak times to charge the TES, at full capacity (avoiding short cycling inefficiencies), and to an intermediate temperature where they can achieve higher efficiencies (rather than at design hot water supply temperatures). Overall, the TIER design saves space, improves EE, supports grid-

interactive efficient building initiatives, and reduces first costs and operating costs compared to alternative all-electric plant approaches.

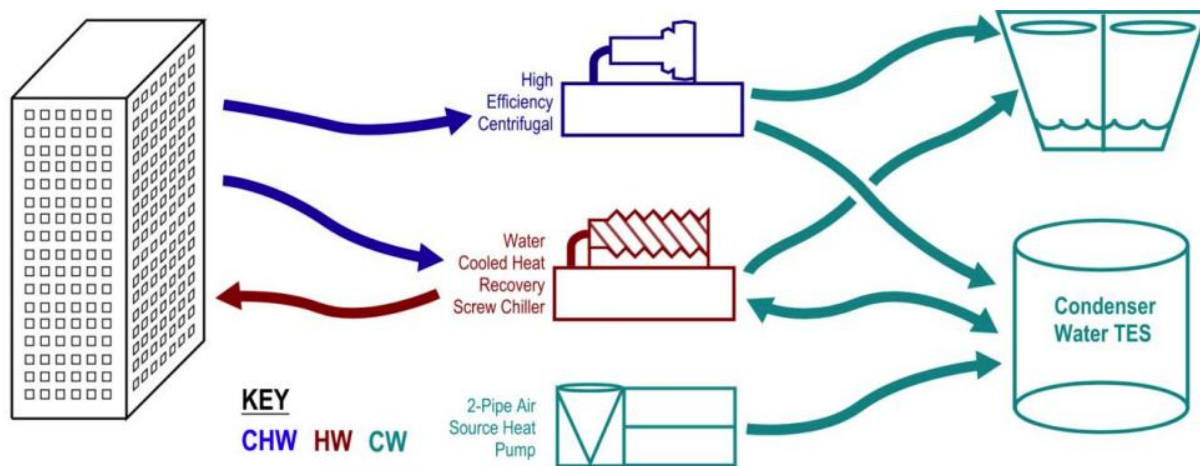


Figure 9. Condenser Water TIER System Diagram

Cheng et al. (2026) documented the improved heating and cooling COPs of the first operational TIER plant, compared to a similar AWHP reference plant. Table 4 shows the superior energy efficiency of an operational TIER plant relative to a comparable plant relying on AWHP heating. Both serve similar building loads, are located in the same climate, and monitored over the same period from April through August, 2025. The average heating COP for the TIER plant was 4.20, compared to 3.20 for the baseline plant, representing approximately a 31 percent improvement. In winter conditions, the difference in COP_h 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. Similarly, the TIER plant achieved an average cooling COP of 5.96, compared to 4.39 for the baseline system (which was effectively a water-cooled chiller plant), which is an improvement of roughly 36 percent. Average COPs are also reported for different seasonal periods in Table 4.

Evaluating COPs for heat recovery equipment is not straightforward because the work performed can simultaneously provide heating and cooling. For the comparisons shown in Table 4, 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.

Table 4. Average COPs for the TIER and Baseline Plants

Averaging Period	Heating COP		Cooling COP		Combined COP	
	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 AHP 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

Both buildings shown in Table 4 include small process cooling loads that continuously require chilled water. 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 chilled water 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 above.

TIER with Hot Water Storage

Figure 10 illustrates an alternative TIER plant configuration using hot water as the TES medium.

The main differences in the HW TIER concept, compared to the CW TIER concept, are:

1. The chillers are not interconnected in HW TIER like they are in CW TIER
2. All heat recovery must be through a screw HRC in HW TIER rather than a more efficient centrifugal chiller in CW TIER

- Hot water is created using a double lift envelope in CW TIER which is slightly less efficient than the single lift envelope in HW TIER
- AHPs must produce 130°F supply water in HW TIER which is less efficient than producing tepid water in CW TIER (though perhaps not with the double lift envelope)
- Larger tank from lower delta-T
- Higher tank jacket losses
- HWST setpoint reset is not possible

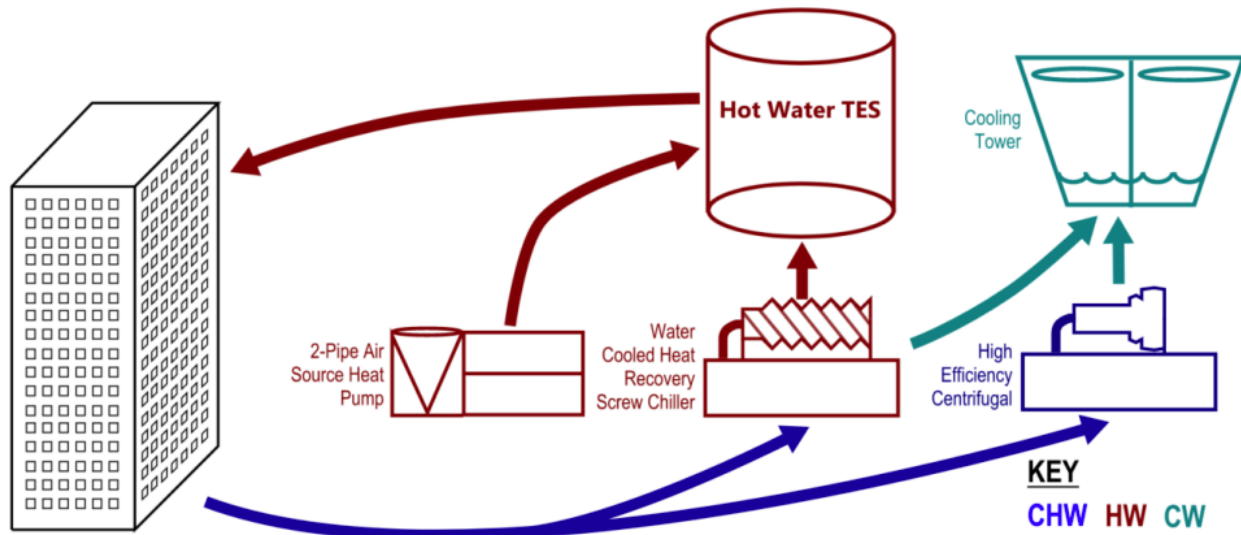


Figure 10. Hot Water TIER System Diagram

Simplified TIER Modeling Tools and Resources

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 its 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.).

Spreadsheet models of two TIER plant configurations provide a practical way for designers to explore and analyze the energy performance of TIER plants (Cheng, et al., 2026). 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 based on hourly energy analysis. 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. The CW TIER plant spreadsheet model is available online [here](#). The HW TIER plant spreadsheet model is available online [here](#).

Example system schematics and descriptions of the modes of operation for the CW and HW TIER plants are included in Appendix A. Detailed example sequences of operation for the CW TIER approach are included in Appendix B, and annotated control schematics for different modes for the CW TIER plant are provided in Appendix C.

Energy and Life Cycle Cost Evaluations

A parametric energy and life cycle cost analysis was conducted to develop broader understanding of TIER plant performance and to help generate guidance for future designs. Spreadsheet modeling tools were used to assess 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 – Pacific Gas & Electric
 - [CZ09] Los Angeles – Southern California Edison
 - [CZ12] Sacramento – Sacramento Municipal Utility District
- 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 is referenced by 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.”

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 11). This is a significant finding where all-electric central plants are broadly expected to increase energy costs compared to conventional chiller and boiler plants because of the high cost of electricity. 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.

THERMAL ENERGY STORAGE MEDIA

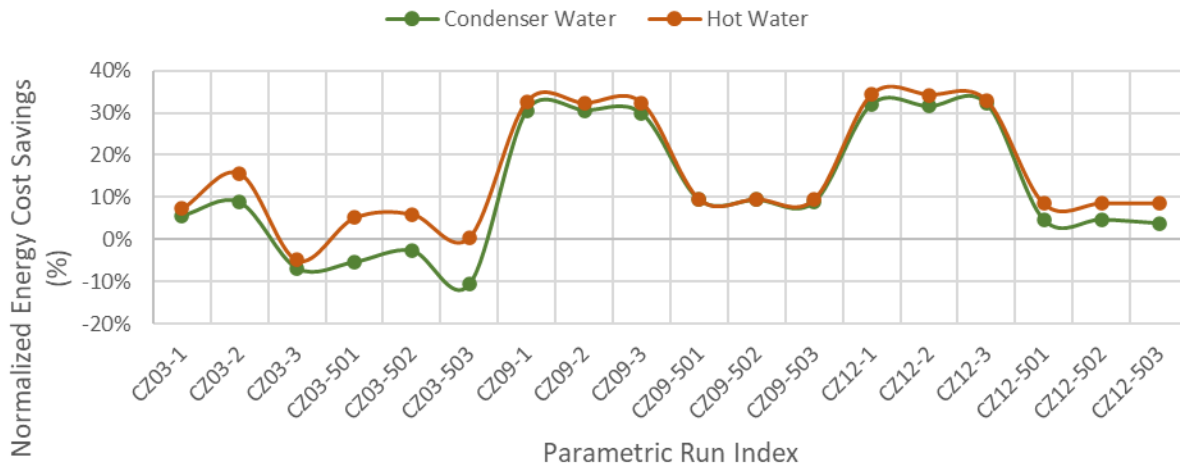


Figure 11. Parametric Energy Cost Savings vs. Baseline Plant

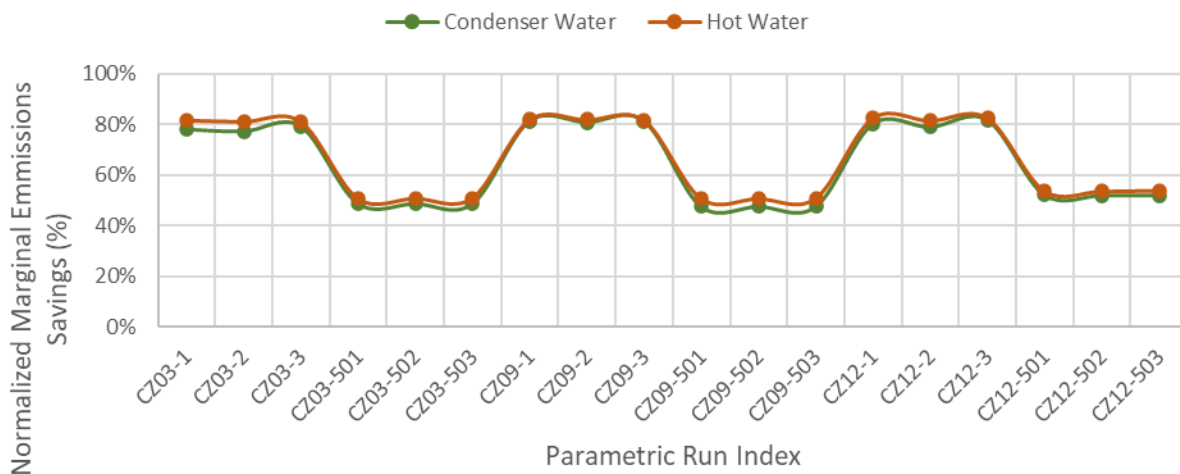


Figure 12. Parametric Marginal Emissions Savings vs. Baseline

Both TIER plant configurations were effective at reducing total energy consumption and therefore marginal emissions (Figure 12). 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 AWP sizing ratio is 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.

While both TIER plant configurations are clearly more efficient than the baseline system, the COPs for heating and cooling differ substantially between the plant types (Figure 13 and Figure 14).

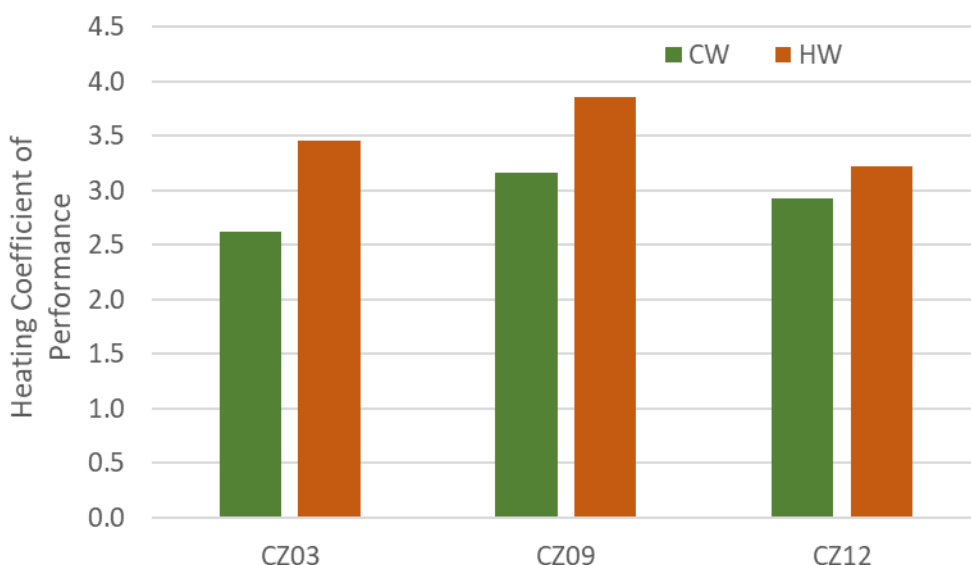


Figure 13. Heating COPh Condenser Water vs. Hot Water – Intermediate Tank with 0-ton Baseload

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. 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 14. 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 14). 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.

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 hot water preheat. For the modeled commercial building application, there is negligible domestic hot water load, so plant performance was assessed as a function of cooling baseload to represent buildings with a steady source of heat recovery.

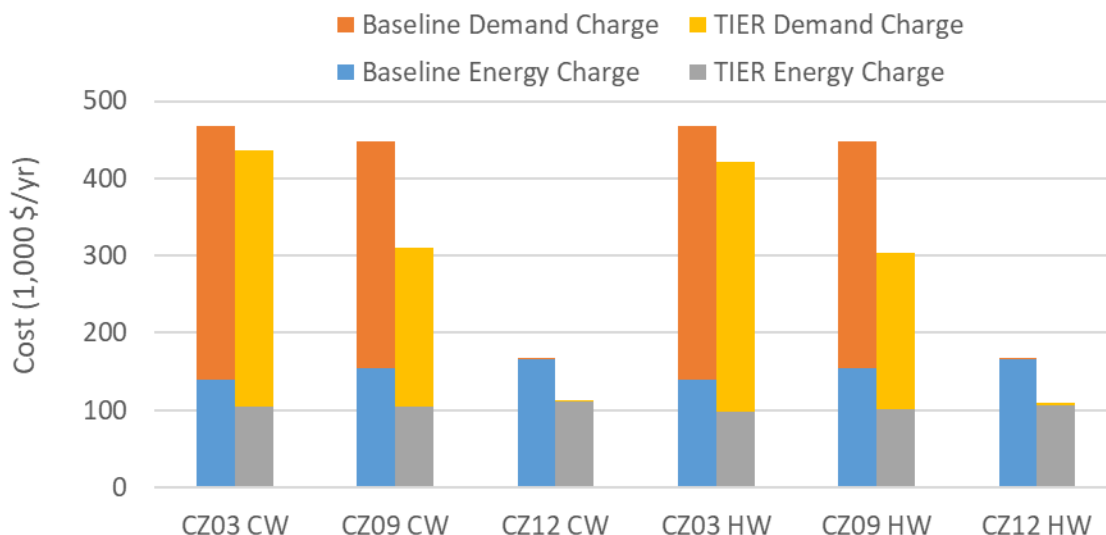


Figure 15. Energy Cost Breakdown by Region and Storage Media 0-ton baseload

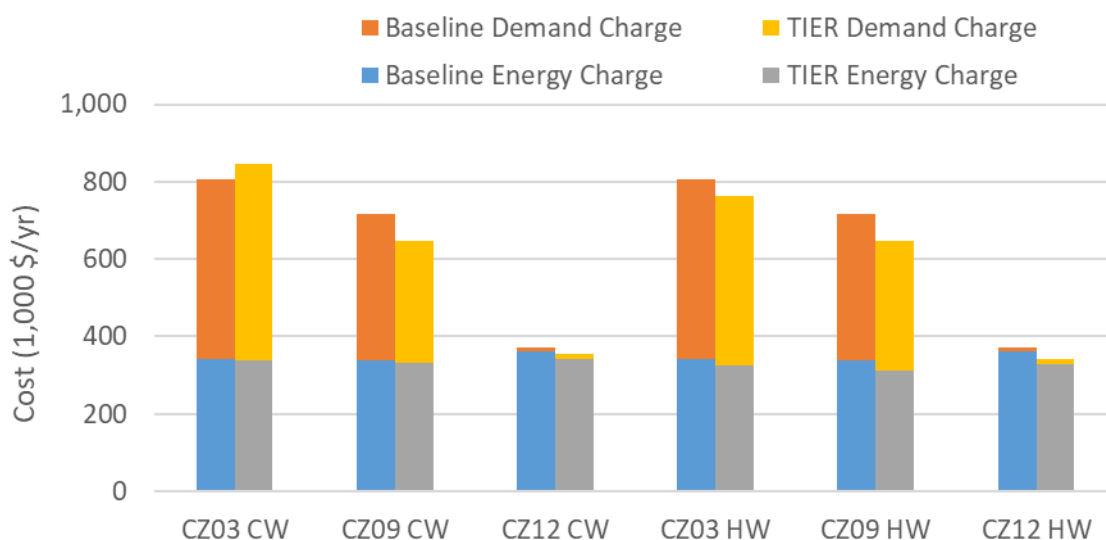


Figure 16. Energy Cost Breakdown by Region and Storage Media 50-ton baseload

With all other variables held constant, Figure 15 and Figure 16 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 is for an intermediately sized storage tank. 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 condenser water plant in CZ 3, 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 needs to be rejected to the cooling towers. CZs 9 and 12 already have high cooling loads relative to CZ 3 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 is now cooling load dominated across all 3 climates, causing the cooling towers to run during many more hours of the year at higher part loads where cooling equipment is less efficient. As cooling loads rise and heating loads stay constant, the TIER plant begins behaving much more like a conventional water-cooled chilled water 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 since the large cooling loads are served by high efficiency centrifugal chillers operating more at part load where they are more efficient while the heat recovery screw chiller operates at full load where it is as efficient as it can be. By contrast, the condenser water storage plant does a significant amount of cooling at high loads with the centrifugal chillers at higher part load fractions where they are less efficient and sometimes screw chillers when needed at part load where they are least efficient. Higher demand charges and more strict TOU penalties in CZ 3 exacerbate the issue, tipping the operating costs in favor of the baseline plant configuration which also leverages high efficiency centrifugal chillers.

LIFECYCLE COST STUDY

The Project Team conducted a lifecycle cost analysis across the 36 parametric runs defined in the previous section.

Instead of keeping the building design constant and resizing the plant entirely for each climate zone, the design cooling capacity of the plant 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.

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 17. 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 is less impactful and therefore outweighed by the higher tank costs.

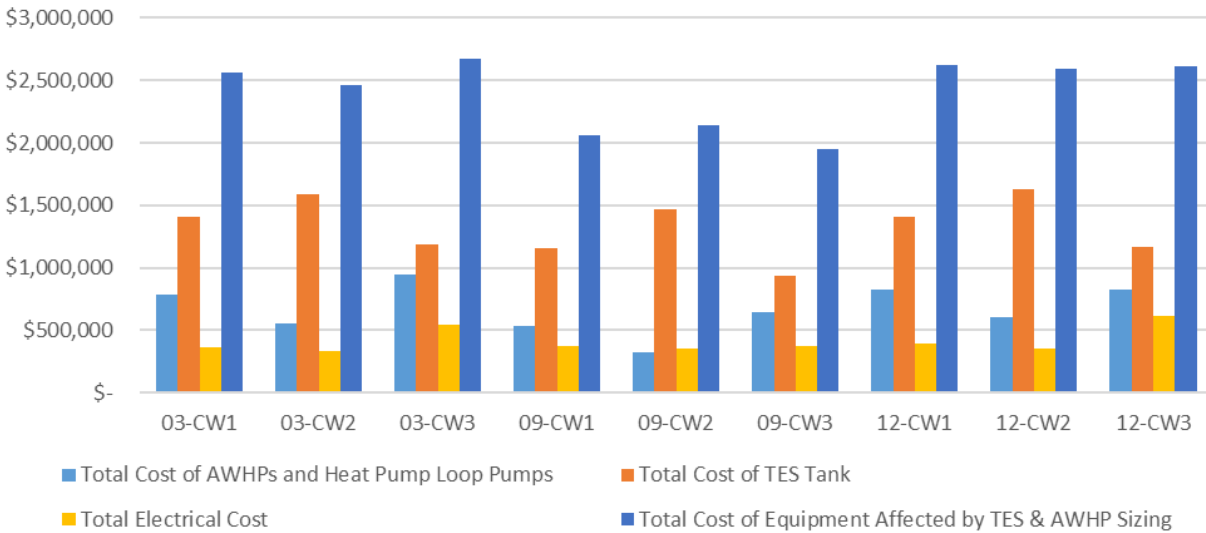


Figure 17. First Cost Comparison of Equipment Affected by TES Tank and AWHP Sizing by Climate Zone

First Cost Comparison: Thermal Energy Storage Media

Thermal energy storage media has negligible impact on the TES versus AWHP first cost tradeoffs across climate zones and cooling baseload quantity. As shown in Figure 18, each equipment cost category has similar magnitude and behavior between the condenser water and hot water storage plants.

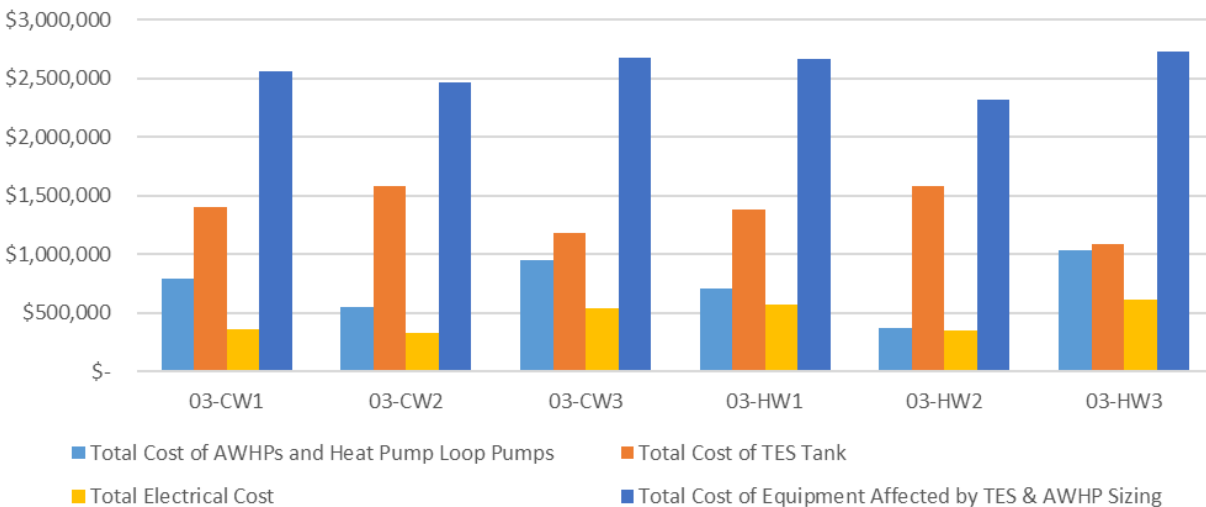


Figure 18. First Cost Comparison of Equipment Affected by TES Tank and AWHP Sizing by Storage Media

While the total cost of equipment affected by TES tank and AWHP sizing including electrical infrastructure is marginally higher in hot water storage plants compared to condenser water storage plants, the hot water 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 hot water vs. condenser water 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 hot water 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 condenser water 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 hot water loop directly. The hot water 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 condenser water 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 hot water 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.

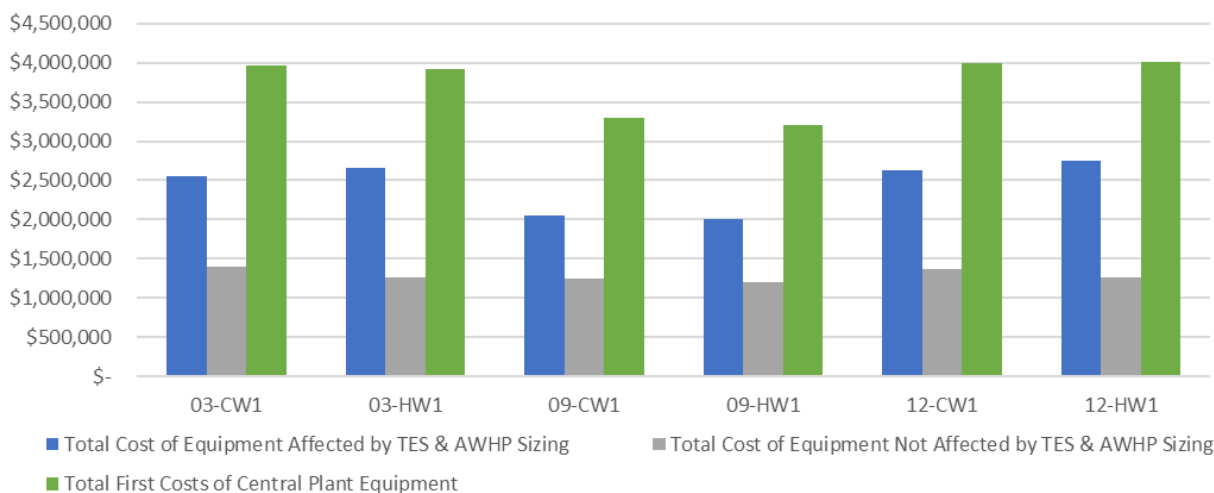


Figure 19. First Cost Comparison by Storage Media and Climate Zone

Net Present Energy Cost

The 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 NIST/DOE), federal and state tax rates, inflation (2.38 percent), discount rates (9.34 percent), and loan interest rates (7.0 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.

Lifecycle Cost Comparison

Incremental LCCs were calculated by adding incremental first costs to incremental net present energy costs. Figure 20 through Figure 22 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 hot water storage over condenser water storage primarily due to the high demand costs associated with higher AWHP usage given the utility rate structure assumed in CZ 3. Hot water storage runs with smaller AWHPs show 7 to 19 percent savings relative to most of the other plant configurations evaluated.

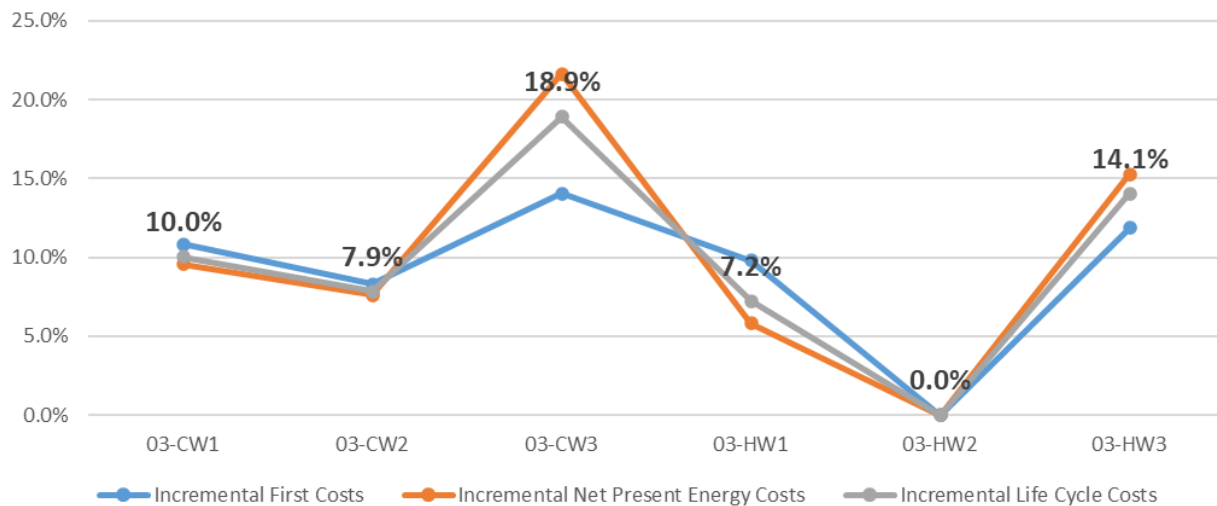


Figure 20. Incremental Life Cycle Cost Comparison – Climate Zone 3 with 0-ton Baseload

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 hot water storage TIER plant.

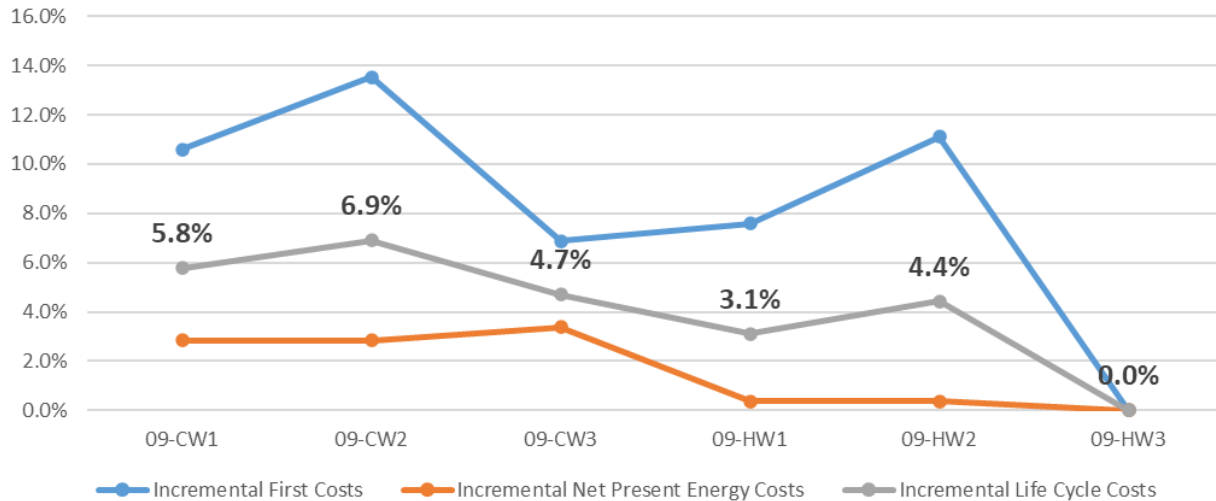


Figure 21. Incremental Life Cycle Cost Comparison – Climate Zone 9 with 0-ton Baseload

CZ 12 favors smaller AWHP plants and hot water 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.

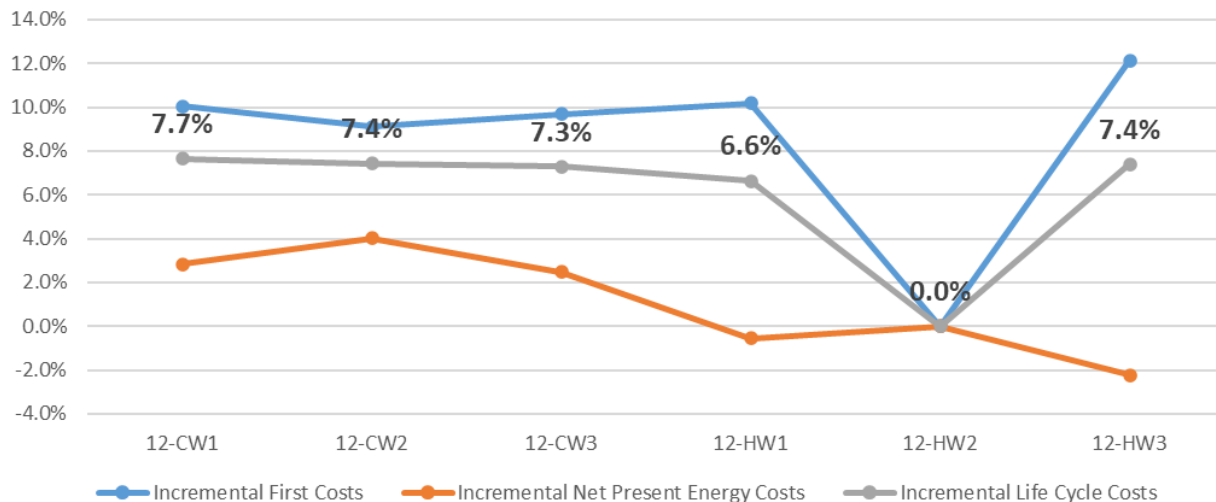


Figure 22. Incremental Life Cycle Cost Comparison – Climate Zone 12 with 0-ton Baseload

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.

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 outweigh 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.

Hybrid/partial electrification

In many cases, one or more of the considerations in the preceding sections may dissuade an owner from pursuing 100% heating electrification. This is especially true in retrofit situations where challenges related to weight, space, and electrical service capacity cannot be accounted for upfront as in new construction design. Fortunately, recent research (Raftery, Singla, Cheng, & Paliaga, 2024) demonstrates that most of the electrification benefit afforded by AWHPs can be achieved with partial electrification. Raftery's research analyzed hot water time series trend data from 259 commercial buildings in office, laboratory, medical office, library, and other building types across multiple U.S. climate zones from 59 organizations. The average building in the data set includes 2.2 years of data. Raftery's analysis of a subset of 152 buildings in the data set found that heat pumps sized for 10%, 20%, 30%, 40%, and 50% of maximum measured load across the trend period could serve an average of 41%, 69%, 84%, 93% and 97% of the total measured load, respectively. Furthermore, the research found that in a 21 building subset for which the requisite data were readily available, installed capacity exceeded the maximum observed demand by an average factor of approximately 2. In other words, a new AWHP array with only 10% of preexisting installed boiler capacity could very

well serve 20% of the actual peak heating load and meet 69% of the building's annualized heating load per these metrics.

It is also critical to consider that in most applications, heating loads are greatest during cold, winter early morning hours when (a) the grid is often dirtiest due to a dearth of renewable generation assets and (b) AWHP efficiencies are lowest. In other words, high loads hours are likely those in which AWHPs provide the smallest marginal benefit in emissions and potentially the largest marginal penalty in costs (utility rates depending) relative to natural gas boilers.

Taken together, these facts indicate that AWHP systems sized for a small fraction of the design load—as might be feasible due to spatial, structural, electrical, or budgetary constraints—should still yield most of the reductions in both emissions and energy cost savings (where afforded by utility rate structures) available from full electrification in typical applications.

When partially electrifying, the owner's goals dictate the approach. If the owner's directive is to maximally electrify the system given spatial, cost, structural, and electrical constraints, then the approach is straightforward: identify the limiting constraint, and size the system based on it. If on the other hand the owner's goal is to optimize for life cycle cost effectiveness, minimize first costs, or to only decarbonize up to the point where there are significantly diminishing marginal reductions in source emissions, then a deeper analysis is justified.

Many retrofit projects have the luxury of either time series hot water load data or time series natural gas metering data. Where these data sources are not available, modeled heating hourly loads can be used as a weak surrogate but should be calibrated to monthly natural gas billing data as best as possible. Using hourly load data, a cumulative distribution function (CDF) graph should be constructed with plant heating load (kBtu/h) on the x-axis and cumulative annual heating load, represented as a percentage, on the y-axis. For the example site shown in Figure 23, an array capable of outputting only 1000 kBtu/h (less than 50% of peak measured load) could satisfy nearly 90% of annual heating load. It is unlikely that a larger array would be either life cycle cost effective or justifiable based on emissions reductions alone.

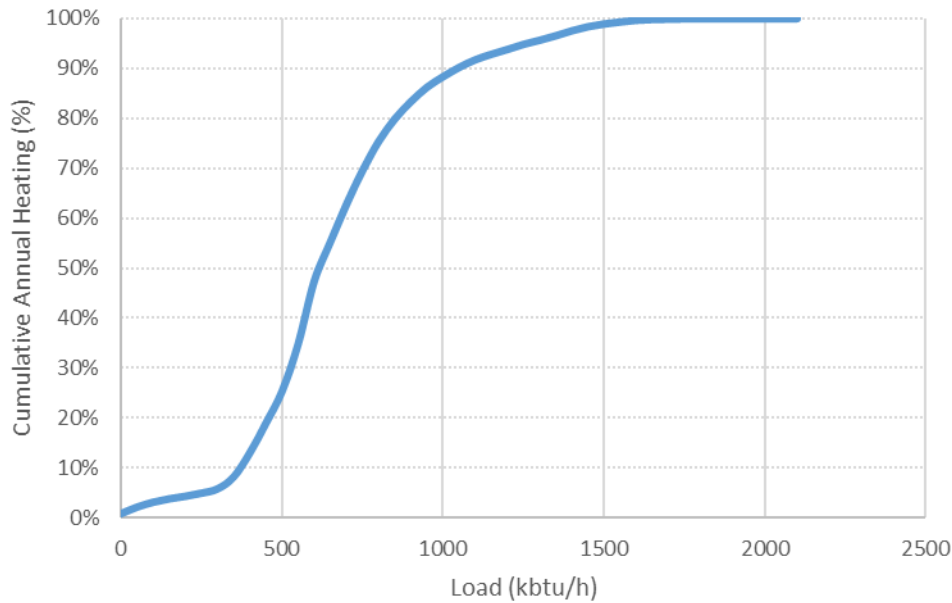


Figure 23. Representative Project Annual Heating CDF by Load

Two partial electrification approaches to consider:

Addition of a HRC to a central plant

One practical approach to partial electrification involves adding a HRC to a central plant with large cooling base loads and significant simultaneous heating and cooling needs (Figure 24). Where loads are sufficiently large to support this approach, screw HRCs can offer higher efficiencies and higher hot water temperatures than can generally be achieved by scroll compressors in AWHs. In this situation, the HRC can be prioritized to be staged on to meet cooling loads while also opportunistically recovering heat from the cooling load any time there is a need for heating. Additional cooling loads would be met by the remaining cooling-only equipment and additional heating loads would be met by the remaining heating-only equipment. Care must be taken to ensure that typical hot water loads exceed the minimum turndown capability of the HRC. An airside economizer can be throttled in cool weather to create a false cooling load, effectively using the mixed air stream as a heat source to increase the degree of simultaneous heating and cooling. For buildings with significant exhaust airflows, exhaust air heat recovery (e.g. a water coil in the exhaust air stream) can also provide an additional heat source in cool weather and as a heat sink in warm weather.

In retrofit applications, the addition of an HRC can potentially be coordinated with an end-of-life cooling-only chiller replacement, instead of a like-for-like replacement. This scenario offers the opportunity to leverage the equipment replacement costs toward energy efficiency while limiting the overall plant footprint where space is constrained. Important caveats with this approach:

1. If the HRC cooling capacity is needed to satisfy peak cooling loads, the HRC must have sufficient coincident heating loads to reject heat to, or otherwise have an alternative heat rejection option (e.g., a connection to a condenser water loop, an exhaust air coil, or a dry cooler).

2. Ensure that the HRC can deliver hot water supply temperatures compatible with the existing heating coils
3. As HRCs operate at much higher lift than cooling-only chillers, the current draw can be significantly higher than for the comparable cooling-only equipment. Ensure that electrical infrastructure can support these increased needs.

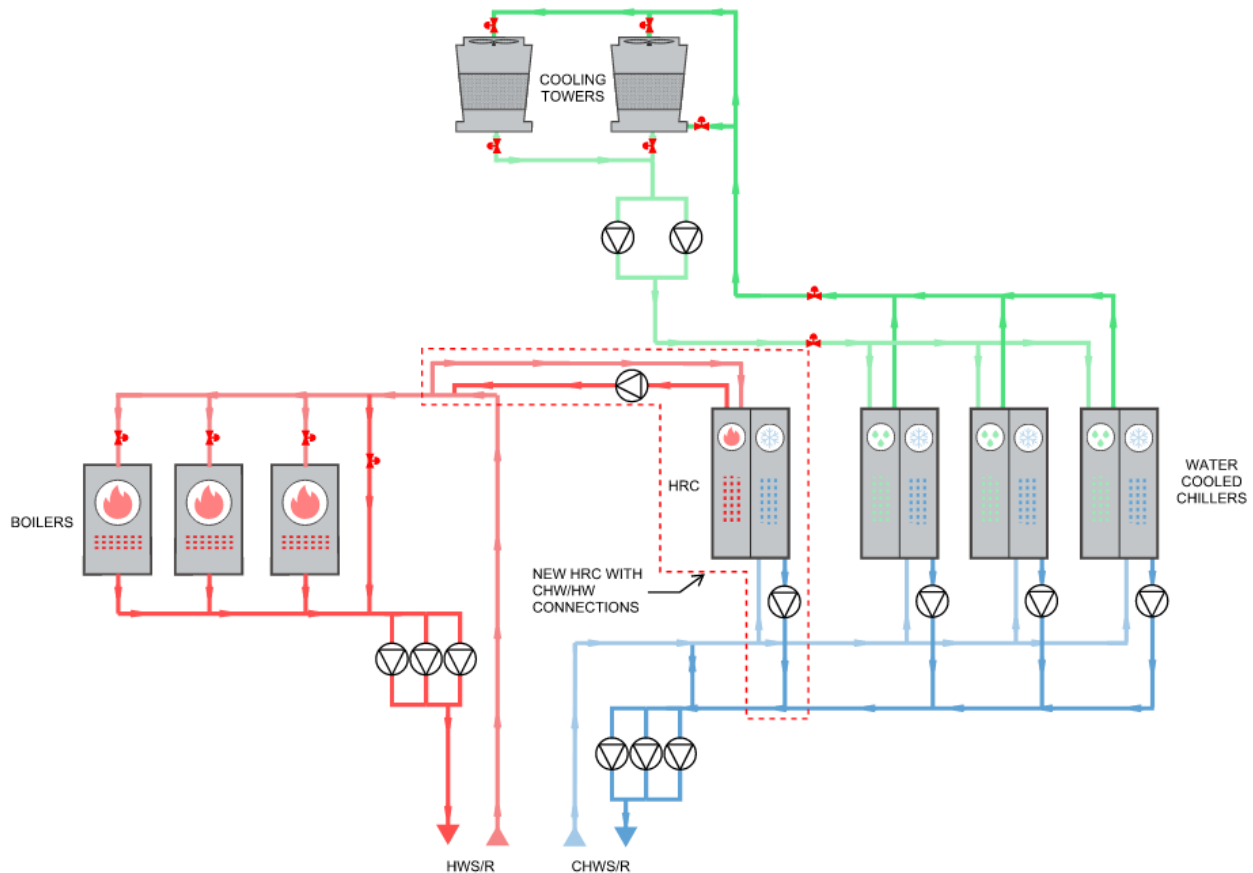


Figure 24. Partial electrification of a central plant with addition of a HRC

Addition of modular AWHPs

Where loads do not justify the use of a HRC, modular AWHPs can be used to serve a portion of a building's heating load. Modular AWHPs offer the flexibility of accommodating a wide range of capacities and variable speed units can provide good turndown and efficiency.

In a hybrid heating plant, the AWHPs can potentially be operated in an integrated or non-integrated fashion. In the former, the AWHPs and boilers operate together in series with AWHPs leading and the boilers boosting the supply temperatures when needed. One critical limitation with this scenario is ensuring that hot water return temperatures are compatible with the operating envelope of the AWHPs, particularly in retrofit applications. If the boilers run to provide higher supply water temperatures needed to achieve design capacities, the hot water coils must provide sufficient delta-Ts for the AWHPs to operate safely. Alternatively, a hybrid plant can operate in a non-integrated fashion where the AWHPs alone serve the off-peak heating loads to the extent that their capacity or maximum supply temperatures can handle. When that limit is reached, the AWHPs are staged off

and the plant is served solely by boilers. Non-integrated operation avoids the risk of loop temperatures being incompatible with the AWHP operating envelope, but requires that the boiler plant be sized to handle the full design heating load.

Case Studies

This chapter presents brief case studies of all-electric building systems that are in operation and that have energy monitoring data available for evaluation.

Case Study #1

An academic science building located in California Climate Zone 3 and constructed in 2019 is served by two 2-pipe changeover AWHPs to serve the building heating and cooling loads. The airside system operates 24/7 and consists of 100% outdoor air, variable air volume (VAV) air handling units with a run-around exhaust air heat recovery. All of the heating and cooling is done with 4-pipe terminal units so that there are no reheat loads. Table 5 provides a basic description of the building and its HVAC systems.

Table 5. Case Study 1 Building Description

Attribute	Description
Building Type	Academic Laboratory
Building Size	38,000 ft ²
Climate Zone	ASHRAE 3B CA CZ 3
Plant	Two 2-pipe changeover AWHPs. Each AHP: 85 tons, 736 KBH, 4 fixed speed compressors, design COP _c = 3.59, design COP _h = 2.35
HVAC System	VAV air handling units with 100% outside air, 24/7 operation, 4-pipe VAV terminals

One continuous year of operating data was evaluated to examine the performance of the AWHP heat pump plant. Figure 25 shows the average heating and cooling COPs of the units as a function of outside air temperature (OAT), along with the design heating and cooling COP values. Though there are other factors that influence COP (e.g., part load ratio [PLR] and lift), the measured COPs are fairly close to the design values at design ambient conditions.

The nominal expectation is for the cooling COP to decrease at higher OATs and the heating COP to decrease at lower OATs. While this can be seen over part of the ranges shown in Figure 25, the predominant result is the opposite effect: cooling COP decreasing at lower OATs and heating COP decreasing at higher OATs.

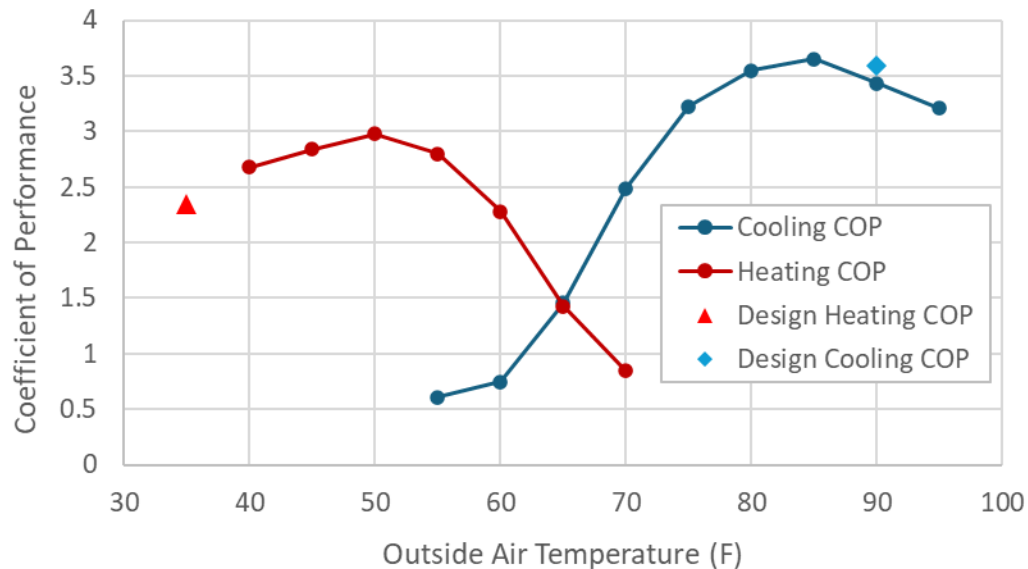


Figure 25. Case Study #1: Coefficient of Performance vs. Outside Air Temperature

Figure 26 shows the average plant PLR and the frequency distribution in heating and cooling modes as a function of OAT. Full load is evaluated here as the nominal capacity of one AWHP. The cooling PLR decreases as the OAT decreases and the heating PLR decreases as the OAT increases as expected. Figure 27 shows the frequency distribution as a function of PLR. The frequency distribution shows that the plant mostly operates in heating mode at moderate loads between PLRs of 20 to 60 percent where the AWHPs can operate stably to achieve reasonable COPh. Even with the airside heat recovery, heating is needed up to about 65 °F because of fume hood- and ventilation-dominated lab spaces. Still, the load was below the nominal 25 percent turndown limit for nearly half of the time when operating in heating mode. The overall load-weighted average COPh over the monitored year was 2.72.

In cooling mode the plant predominantly operates at extremely low PLRs: below the nominal 25 percent turndown limit for 91 percent of the time, and below 10 percent for 77 percent of the time. The excessive short cycling of the AWHPs in cooling mode is evidenced by the low load-weighted average COPc of 1.69. Overall, it appears that the decreases in COPc and COPh shown in Figure 25 are largely explained by very low part loads and poor part load performance of the AWHPs.

Exacerbating the low cooling loads was an issue observed with the AWHP staging in cooling mode that had both units often running even though the loads could have easily been handled by a single unit. The control logic would stage on the lag unit if the associated cooling coils were still demanding additional cooling, regardless of the measured load. Unfortunately this occurred frequently even though supply temperatures were at design and pumps were generating high differential pressures.

The 4-pipe VAV system, coupled with the airside heat recovery means that there is minimal opportunity for heat recovery at the plant. The design decision to omit 4-pipe heat recovery units in this plant seems to have been judicious, particularly given the significant cost to add heat recovery capability for these particular AWHPs. With two 2-pipe AWHPs and some need to provide both hot and chilled water simultaneously, one AWHP would need to be able to meet the full cooling load by

itself for OATs as high as 70°F, which they would easily be able to do. Given that the average heating load approaches the nominal heating capacity of one AWHP at the design temperature, but that both AWHPs could operate at those conditions, smaller AWHPs could have met the heating loads and provided better energy performance, particularly on the cooling side which was extremely oversized. The use of three smaller AWHPs, though more expensive to install, might also potentially be justifiable to avoid the excessive short cycling in cooling mode. This application would also be a good fit for modular AWHPs (though they were not available at the time that this project was in design) which provide more steps of unloading, particularly with the mismatch in peak heating and cooling loads.

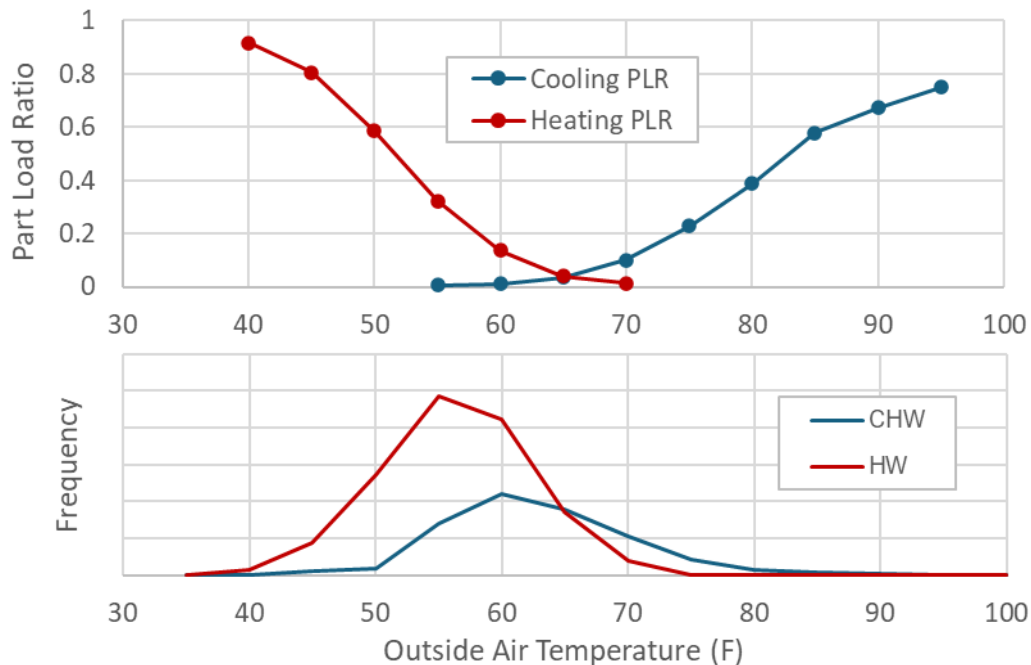


Figure 26. Case Study #1: Part Load Ratio and Frequency as a function of Outside Air Temperature

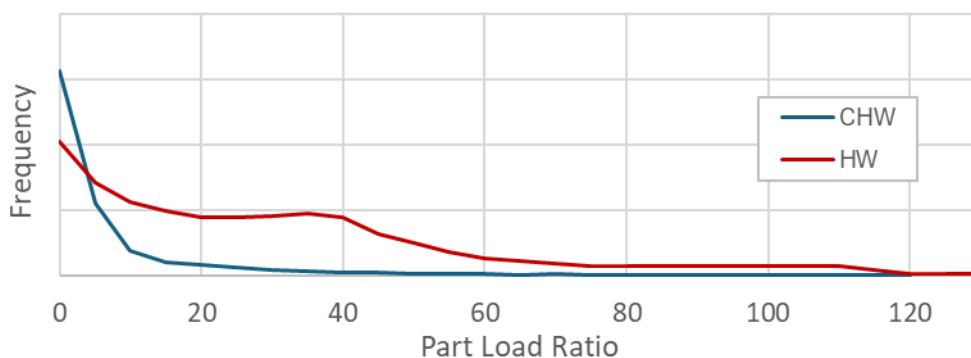


Figure 27. Case Study #1: Frequency chart of plant operation as a function of Part Load Ratio

Case Study #2

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 (Figure 9). One building completed construction with an operational TIER plant for most of 2025. 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. As both buildings serve similar functions and have similar equipment configurations, 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.

Table 6. Case Study 2 Building Descriptions

	TIER Plant	Baseline
Location	CA climate zone 4 (Sunnyvale, CA)	CA climate zone 4 (Santa Clara, CA)
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)

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 (HRCs, cooling chillers, AWHPs, and cooling towers). Figure 28 demonstrates how the tank is

discharged throughout the day by the HRCs, while simultaneously recharged by the cooling chillers, and then 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 chilled water loop runs out. The profile demonstrates how each piece of mechanical equipment interacts with the storage system asynchronously throughout the day to maximize system efficiency.

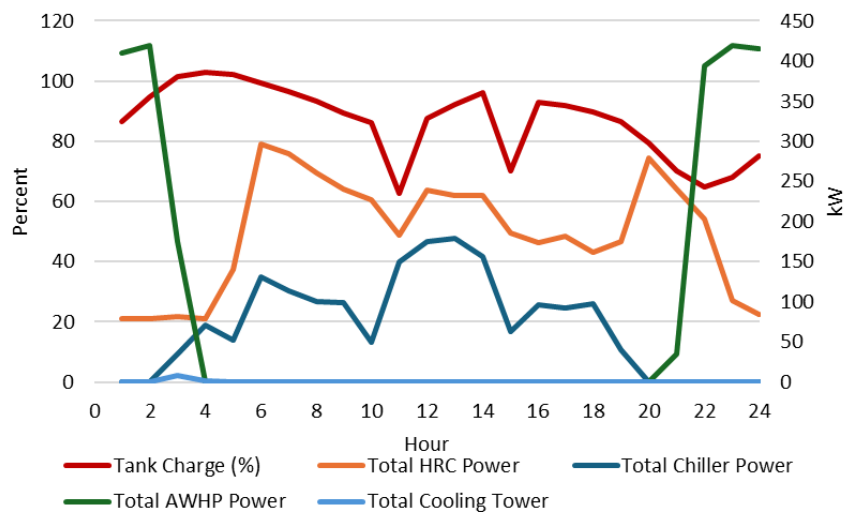


Figure 28. Hourly profile of the TIER plant for one representative day in February. Y-axis shows tank state of charge and power for AWHPs, HRCs, chillers, and cooling towers.

The TIER plant demonstrated superior performance over the analysis period (April through August), achieving higher overall average COPs in both heating and cooling modes (Table 7). 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 COP_h 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. Note that evaluating individual COP_c and COP_h values is not straightforward with a system that utilizes heat recovery and storage. The convention used in this analysis charges HRC and chiller power to heating if the heat is rejected to the TES, and HRC and chiller power to cooling only when they are rejecting heat to the cooling towers.

Table 7. Average COPs for the TIER and Baseline Plants

Averaging Period	Heating COP		Cooling COP		Combined COP	
	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 29 presents the average daily heating COP of both plants for different average daily OAT bins. As expected, the heating COP of both systems generally improves as average daily OAT increases, reflecting reduced compressor cycling and more favorable heat pump operating conditions.

The baseline plant is less sensitive to OAT, 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 OAT falls. This behavior is consistent with the characteristics of the TIER system and COP calculation conventions used here. There is also less available heat recovery from the chilled water loop forcing the less efficient AWHPs to operate more frequently to recharge the energy stored in the TES tank. Conversely at higher OATs, higher heat recovery availability and reduced cooling chiller cycling allow the TIER plant to produce heating hot water more efficiently, leading to an upward trend in COP relative to the baseline system.

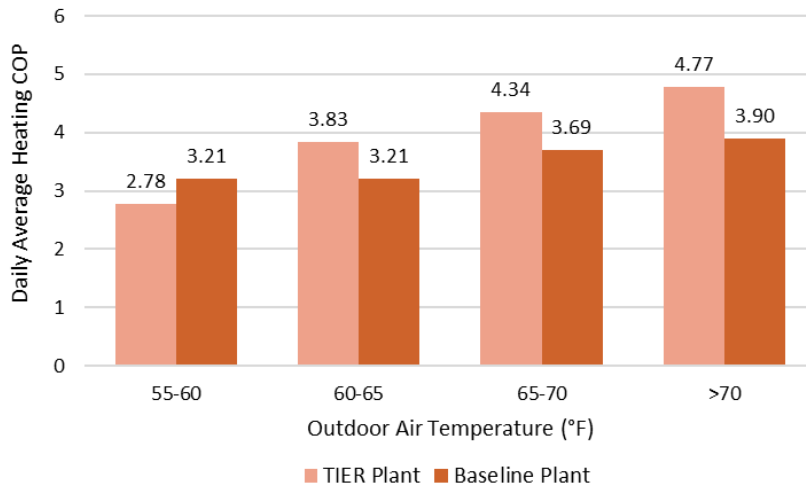


Figure 29. Daily average heating COP of the TIER plant and the baseline plant by average daily outdoor air temperature bins.

Figure 30 shows that across all average daily OAT 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 OAT is observed in both plants (see Figure 30). As OAT increases, cooling demands in both plants become higher and more sustained, allowing equipment to operate closer to steady-state. At lower OATs, 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 OATs. 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.

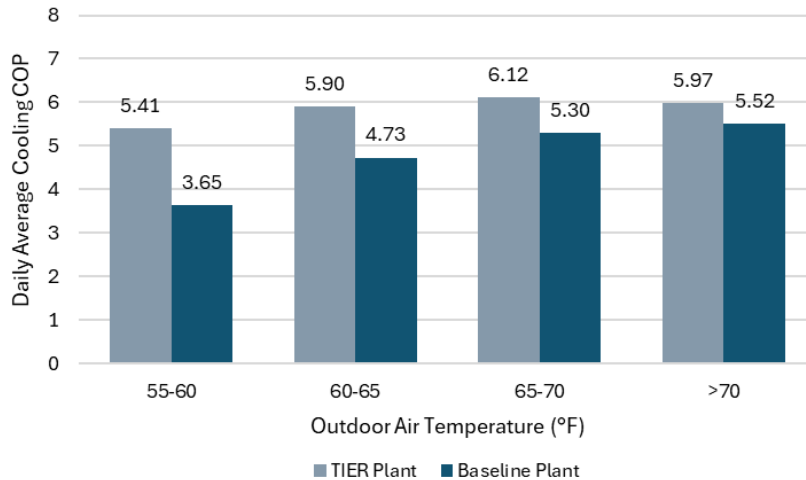


Figure 30. Daily average cooling COP of the TIER plant and the baseline plant by average daily outdoor air temperature bins.

Figure 31 shows the daily average heating COP of the TIER plant over a period slightly exceeding one year (January 2025 through January 2026) to evaluate its seasonal performance. The trend shows lower daily average heating COP values during the colder winter months 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 because of testing work conducted during this period.

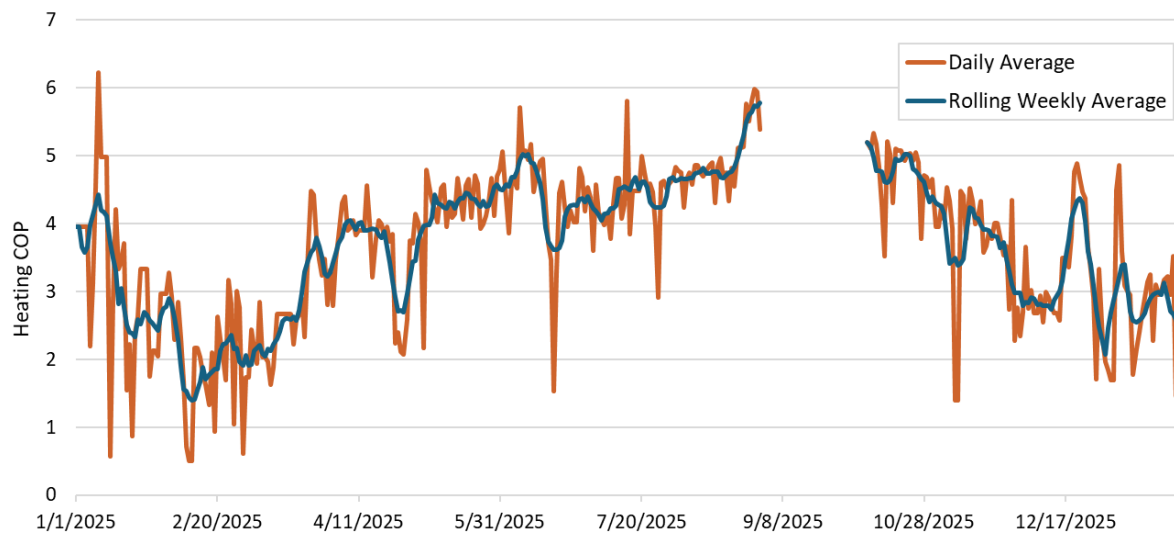


Figure 31. Daily average heating COP of the TIER plant from January 2025 to January 2026.

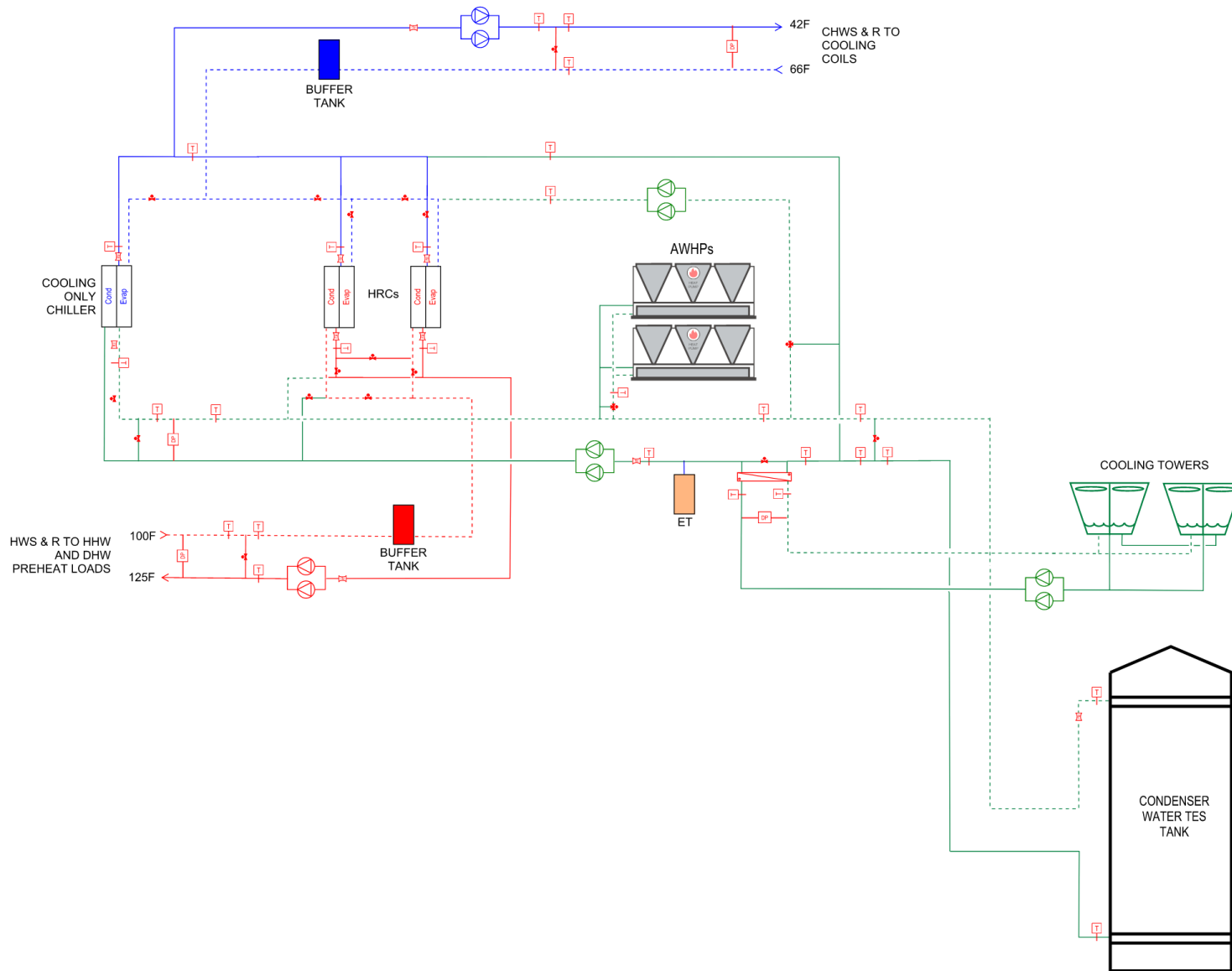
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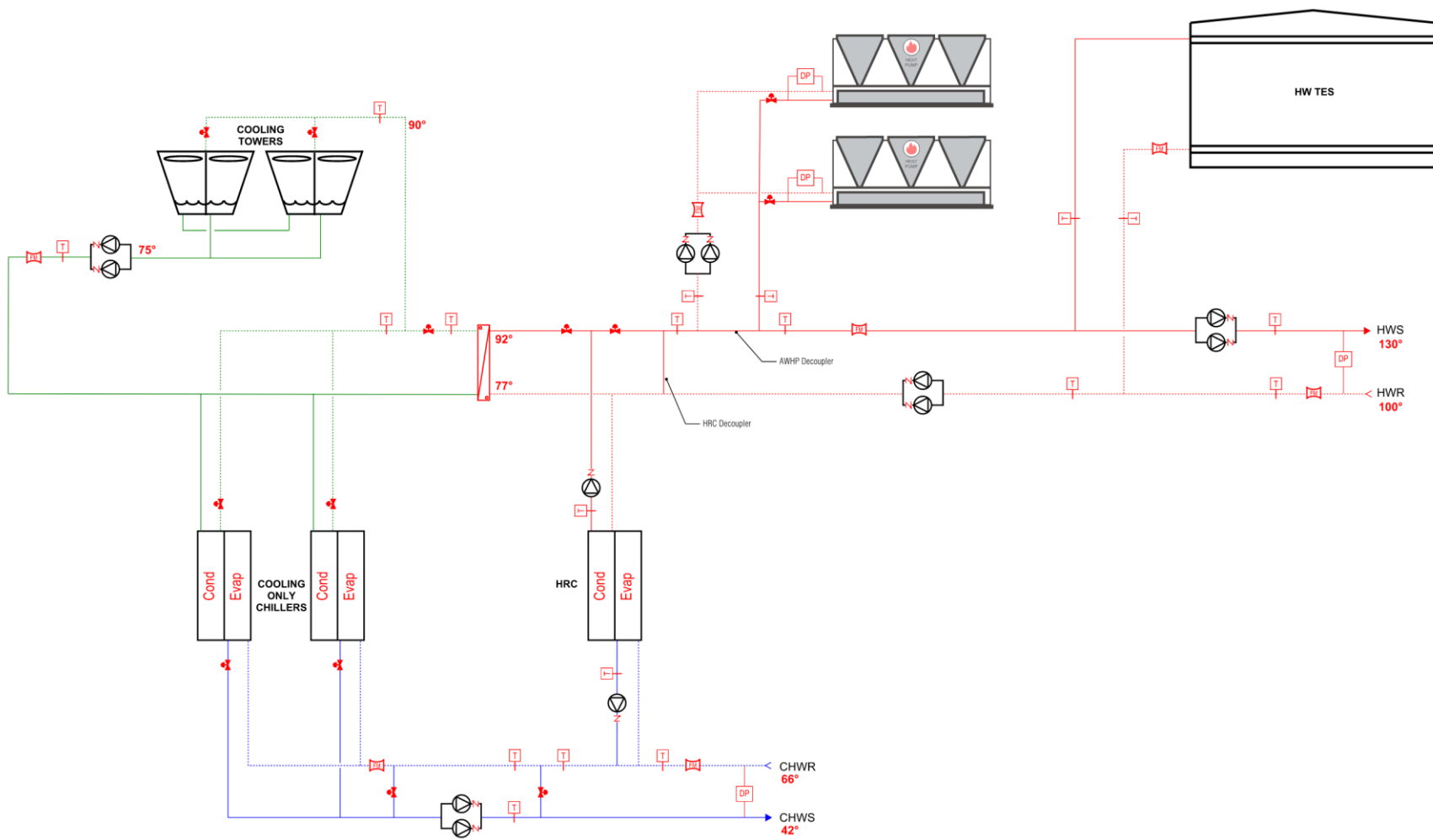
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Appendix A

Condenser Water and Hot Water TIER Plant System Schematics



Condenser Water TIER System Schematic



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 Sequences of Operation

SEQUENCES OF OPERATION

PART 1 - GENERAL

1.1 SUMMARY

- A. These control sequences include references to ASHRAE Guideline 36 and approved addenda. Where sequences are verbatim from Guideline 36, they are shown in **green text**. Not all informative text has been included. Sequences have been customized to include only Title 24 options where they take precedence over ASHRAE 90.1 and 62.1 requirements.

1.2 INFORMATION PROVIDED BY DESIGNER

- A. See equipment schedules on drawings for all setpoints unless otherwise noted below.

B. Chilled Water Plant

1. Chilled Water Loop Parameters
 - a. QDesignCool, the design cooling load of the plant (tons)
 - b. CapCoolChX, nominal cooling capacity of Chiller X at design conditions (tons)
 - c. CHWSTmin, the design plant chilled water supply temperature, 42°F.
 - d. **CHWSTmax**, 60°F
 - e. MinEvapFlowX, minimum evaporator flowrate for Chiller X per record submittal.
 - f. DesEvapFlowX, design cooling mode and heat recovery mode (applies to HRCs only) evaporator flowrates for Chiller X per record submittal (GPM)
 - g. MinCondFlowX, minimum condenser flowrate for Chiller X per record submittal.
 - h. DesCondFlowX, design cooling mode condenser flowrate for Chiller X per record submittal (GPM)
 - i. **PCHWFdesign**, design chilled water loop flowrate (GPM)
 - j. **CHW-DPmin**, 1 psi
 - k. LocalCHW-DPmin, 5 psi
 - l. DemandLimitMinCool shall be 10% of design load, or the lowest load % the machine will allow if greater, 40% (note: some chillers only allow an amperage, as opposed to a load, demand limit, in which case the amperage limit should be set at the amperage corresponding roughly to minimum load/speed operation at 42°F/80°F CHWST/CWRT lift conditions) (%)
2. Hot Water Loop Parameters
 - a. CapHeatChX, nominal heating capacity of Heat Recovery Chiller X at design conditions (kBtu/h)
 - b. **HWSTmax**, the design plant hot water supply temperature, 125°F
 - c. HWSTmin, 100°F
 - d. HW-DPmin, 1 psi
 - e. LocalHW-DPmin, 5 psi
 - f. **PHWFdesign**, design hot water loop flowrate (GPM)
 - g. DemandLimitMinHeat shall be 10% of design load, or the lowest load % the machine will allow if greater, 30% (note: some chillers only allow an amperage, as opposed to load, demand limit, in which case the amperage limit should be set at the amperage corresponding roughly to minimum load/speed operation at 60°F/125°F CHWST/HWST lift conditions) (%)
 - h. HW-LOT, the temperature above which the hot water loop is prohibited from operating (Lock out Temperature), 80°F
3. Condenser Water Loop Parameters

- a. HxDesFlow, design flowrate through the closed loop side of the cooling tower heat exchanger (GPM)
- b. TankMaxFlow, the maximum flowrate in/out of the TES tank above which stratification is disturbed (GPM) from the tank vendor
- c. TankIdealMaxCharge, the *ideal* maximum capacity of the tank, assuming all water volume including the thermocline and heel heights is usable (kBtu)
- d. NolMin, normal operating level of the tank at minimum temperature (ft)
- e. NolMax, normal operating level of the tank at maximum temperature (ft)
- f. CWP1Fdesign, design evaporator barrel closed condenser water loop flowrate (GPM)
- g. CWP2Fdesign, design condenser barrel closed condenser water loop flowrate (GPM)
- h. LiftMin, minimum lift, as estimated by the differential between CWRT and CHWST, for the cooling only chillers per the vendor (°F),
- i. LiftDesign, the design differential between CWRT and CHWST for the plant, 51°F
- j. HxApp, design approach across the tower heat exchanger (°F)
- k. CtApp, design cooling tower approach (°F)
- l. DT_{WB}, design wetbulb temperature (°F)
- m. Prcw-min, minimum city reclaimed water pressure, 15 psi
- n. PcwHxOpenLoopDp, **heat exchanger open condenser water loop differential pressure at design open condenser water loop flowrate**

1.3 INFORMATION PROVIDED BY (OR IN CONJUNCTION WITH) THE TESTING, ADJUSTING, AND BALANCING CONTRACTOR

- A. Coordinate with Section 230593 Testing, Adjusting and Balancing for setpoint determination. Any work not specifically listed in Section 230593 shall be provided under this Section.
- B. **Chilled Water Plant**
 - 1. Chilled Water Loop Parameters
 - a. **CHW-DPmax**, DP determined per TAB to provide design flow through devices downstream of the DP sensor
 - b. **LocalCHW-DPmax**, DP determined per TAB to simultaneously satisfy all remote CHW-DPmax setpoints.
 - 2. Hot Water Loop Parameters
 - a. **HW-DPmax**, DP determined per TAB to provide design flow through devices downstream of the DP sensor
 - b. **LocalHW-DPmax**, DP determined per TAB to simultaneously satisfy all remote HW-DPmax setpoints.
 - 3. Condenser Water Loop Parameters
 - a. DP1Max, Heat recovery chiller evaporator header DP determined per TAB to provide design flow through all chiller evaporator barrels simultaneously.
 - b. DP2Max, Chiller condenser header DP determined per TAB to provide design flow through all chiller condenser barrels simultaneously.
 - 4. Tower Water Loop Parameters
 - a. MinTWPSpeedX, Speed required to deliver minimum cooling tower flow across all tower cells where "X" represents the quantity of tower water pumps running.
 - b. T-level-min-fill, Tower basin fill "on" level (in.)
 - c. T-level-max-fill, Tower basin fill "off" level (in.)
 - d. **T-level-low-alarm**, Tower basin level that generates a low alarm (in.)
 - e. **T-level-high-alarm**, Tower basin level that generates a high alarm (in.)

PART 2 - PRODUCTS**2.1 NOT USED****PART 3 - EXECUTION****3.1 GENERAL**

- A. Contractor shall review sequences prior to programming and suggest modifications where required to achieve the design intent. Contractor may also suggest modifications to improve performance and stability or to simplify or reorganize logic in a manner that provides equal or better performance.
- B. Include costs for minor program modifications if required to provide proper performance of the system.
- C. Unless otherwise indicated, control loops shall be enabled and disabled based on the status of the system being controlled to prevent windup.
- D. When a control loop is enabled or reenabled, it and all its constituents (such as the proportional and integral terms) shall be set initially to a neutral value.
- E. A control loop in neutral shall correspond to a condition that applies the minimum control effect, i.e., valves/dampers closed, VFDs at minimum speed, etc.
- F. When there are multiple outdoor air temperature sensors, the system shall use the valid sensor that most accurately represents the outdoor air conditions at the equipment being controlled.
 - 1. Outdoor air temperature sensors at air-handler outdoor air intakes shall be considered valid only when the supply fan is proven on and the unit is in Occupied Mode or in any other mode with the economizer enabled.
 - 2. The outdoor air temperature used for optimum start, plant lockout, and other global sequences shall be the average of all valid sensor readings. If there are four or more valid outdoor air temperature sensors, discard the highest and lowest temperature readings.
- G. The term “proven” (i.e., “proven on”/“proven off”) shall mean that the equipment’s DI status point (where provided, e.g., current switch, DP switch, or VFD status) matches the state set by the equipment’s DO command point.
- H. The term “software point” shall mean an analog variable, and “software switch” shall mean a digital (binary) variable, that are not associated with real I/O points. They shall be read/write capable (e.g., BACnet analog variable and binary variable).
- I. The term “control loop” or “loop” is used generically for all control loops. These will typically be PID loops, but proportional plus integral plus derivative gains are not required on all loops. Unless specifically indicated otherwise, the guidelines in the following subsections shall be followed.
 - 1. Use proportional only (P-only) loops for limiting loops (such as zone CO2 control loops, etc.).
 - 2. Do not use the derivative term on any loops unless field tuning is not possible without it.

- J. To avoid abrupt changes in equipment operation, the output of every control loop shall be capable of being limited by a user adjustable maximum rate of change, with a default of 25% per minute.
- K. All setpoints, timers, deadbands, PID gains, etc. listed in sequences shall be adjustable by the user with appropriate access level whether indicated as adjustable in sequences or not. Software points shall be used for these variables. Fixed scalar numbers shall not be embedded in programs except for physical constants and conversion factors.
- L. Values for all points, including real (hardware) points used in control sequences shall be capable of being overridden by the user with appropriate access level (e.g., for testing and commissioning). If hardware design prevents this for hardware points, they shall be equated to a software point, and the software point shall be used in all sequences. Exceptions shall be made for machine or life safety.
- M. Alarms
 - 1. There shall be 4 levels of alarm
 - a. Level 1: Life-safety message
 - b. Level 2: Critical equipment message
 - c. Level 3: Urgent message
 - d. Level 4: Normal message
 - 2. Maintenance Mode. Operators shall have the ability to put any device (e.g., AHU) in/out of maintenance mode.
 - a. All alarms associated with a device in maintenance mode will be suppressed.
Exception: Life safety alarms shall not be suppressed.
 - b. If a device is in maintenance mode, issue a daily Level 3 alarm at a scheduled time indicating that the device is still in maintenance mode.
 - 3. Exit Hysteresis
 - a. Each alarm shall have an adjustable time-based hysteresis (default: 5 seconds) to exit the alarm. Once set, the alarm does not return to normal until the alarm conditions have ceased for the duration of the hysteresis.
 - b. Each analog alarm shall have an adjustable percent-of-limit-based hysteresis (default: 0% of the alarm threshold, i.e., no hysteresis; alarm exits at the same value as the alarm threshold) the alarmed variable required to exit the alarm. Alarm conditions have ceased when the alarmed variable is below the triggering threshold by the amount of the hysteresis.
 - 4. Latching. A latching alarm requires acknowledgment from the operators before it can return to normal, even if the exit deadband has been met. A nonlatching alarm does not require acknowledgment. Default latching status is as follows:
 - a. Level 1 alarms: latching
 - b. Level 2 alarms: latching
 - c. Level 3 alarms: nonlatching
 - d. Level 4 alarms: nonlatching
 - 5. Post-exit Suppression Period. To limit alarms, any alarm may have an adjustable suppression period such that once the alarm is exited, its post-exit suppression timer is triggered and the alarm may not trigger again until the post-exit suppression timer has expired. Default suppression periods are as follows:
 - a. Level 1 alarms: 0 minutes
 - b. Level 2 alarms: 5 minutes
 - c. Level 3 alarms: 24 hours
 - d. Level 4 alarms: 7 days

N. VFD Speed Points

To avoid operator confusion, the speed command point (and speed feedback point, if used) for VFDs should be configured so that a speed of 0% corresponds to 0 Hz, and 100% corresponds to maximum speed set in the VFD, not necessarily 60 Hz. The maximum speed may be limited below 60 Hz to protect equipment, or it may be above 60 Hz for direct drive equipment. Drives are often configured such that a 0% speed signal corresponds to the minimum speed programmed into the VFD, but that causes the speed AO value and the actual speed to deviate from one another.

1. The speed AO sent to VFDs shall be configured such that 0% speed corresponds to 0 Hz, and 100% speed corresponds to maximum speed configured in the VFD.

It is desirable that the minimum speed reside in the VFD to avoid problems when the VFD is manually controlled at the drive. But minimums can also be adjusted inadvertently in the VFD to a setpoint that is not equal to the minimum used in software. The following prevents separate, potentially conflicting minimum speed setpoints from existing in the BAS software and the drive firmware.

2. For each piece of equipment, the minimum speed shall be stored in a single software point; in the case of a hard-wired VFD interface, the minimum speed shall be the lowest speed command sent to the drive by the BAS. See Section 1.3 for minimum speed setpoints. The active minimum speed parameter shall be read every 60 minutes via the drive's network interface. When a mismatch between the drive's active minimum speed and the minimum speed stored in the software point is detected, the minimum speed stored in the software point shall be written to the VFD via the network interface to restore the active minimum speed parameter to its default value, and generate a Level 4 alarm.

The minimum speed parameter is read via the network interface to detect any changes in the minimum speed parameter. Upon detecting a change in the minimum speed setting, the correct minimum speed stored in a BAS software point is written back to the drive via the network interface to override any changes that are made locally to the minimum speed parameter at the VFD.

O. Trim & Respond Set-Point Reset Logic

1. T&R set-point reset logic and zone/system reset requests, where referenced in sequences, shall be implemented as described below.
2. A "request" is a call to reset a static pressure or temperature setpoint generated by downstream zones or air-handling systems. These requests are sent upstream to the plant or system that serves the zone or air handler that generated the request.
 - a. For each downstream zone or system, and for each type of set-point reset request listed for the zone/system, provide the following software points:
 - 1) Importance-Multiplier (default = 1)

Importance-Multiplier is used to scale the number of requests the zone/system is generating. A value of zero causes the requests from that zone or system to be ignored. A value greater than one can be used to effectively increase the number of requests from the zone/system based on the critical nature of the spaces served.

- 2) Request-Hours Accumulator. Provided SystemOK (see Section 3.1R) is true for the zone/system, every x minutes (default 5 minutes), add x divided by 60 times the current number of requests to this request-hours accumulator point.
- 3) System Run-Hours Total. This is the number of hours the zone/system has been operating in any mode other than Unoccupied Mode.

Request-Hours accumulates the integral of requests (prior to adjustment of Importance-Multiplier) to help identify zones/systems that are driving the reset logic. Rogue zone identification is particularly critical in this context, because a single rogue zone can keep the T&R loop at maximum and prevent it from saving any energy.

- 4) Cumulative%-Request-Hours. This is the zone/system Request-Hours divided by the zone/system run-hours (the hours in any mode other than Unoccupied Mode) since the last reset, expressed as a percentage.
 - 5) The Request-Hours Accumulator and System Run-Hours Total are reset to zero as follows:
 - a) Reset automatically for an individual zone/system when the System Run-Hours Total exceeds 400 hours.
 - b) Reset manually by a global operator command. This command will simultaneously reset the Request-Hours point for all zones served by the system.
 - 6) A Level 4 alarm is generated if the zone Importance-Multiplier is greater than zero, the zone/system Cumulative% Request Hours exceeds 70%, and the total number of zone/system run hours exceeds 40.
- b. See zone and air-handling system control sequences for logic to generate requests.
 - c. Multiply the number of requests determined from zone/system logic times the Importance-Multiplier and send to the system/plant that serves the zone/system. See system/plant logic to see how requests are used in T&R logic.
3. For each upstream system or plant setpoint being controlled by a T&R loop, define the following variables. Initial values are defined in system/plant sequences below. Values for trim, respond, time step, etc. shall be tuned to provide stable control. See Table 5.1.14.3.

Table 5.1.14.3 Trim & Respond Variables

Variable	Definition
Device	Associated device (e.g., fan, pump)
SP0	Initial setpoint
Spmin	Minimum setpoint
Spmax	Maximum setpoint
Td	Delay timer
T	Time step
I	Number of ignored requests
R	Number of requests from zones/systems
Sptrim	Trim amount
Spres	Respond amount (must be opposite in sign to Sptrim)
Spres-max	Maximum response per time interval (must be same sign as Spres)

Informative Note: The number of ignored requests (I) should be set to zero for critical zones or air handlers.

4. Trim & Respond logic shall reset the setpoint within the range Sp_{min} to Sp_{max} . When the associated device is off, the setpoint shall be SP_0 . The reset logic shall be active while the associated device is proven on, starting T_d after initial device start command. When active, every time step T , if $R \leq I$, trim the setpoint by Sp_{trim} . If there are more than I requests, respond by changing the setpoint by $Spres \cdot (R - I)$, (i.e., the number of requests minus the number of ignored requests) but no more than $Spres_{max}$. In other words, every time step T .

If $R \leq I$, change Setpoint by Sp_{trim}

If $R > I$, change setpoint by $(R - I) \cdot Spres$ but no larger than $Spres_{max}$

P. Equipment Staging and Rotation

1. Parallel equipment shall be lead/lag or lead/standby rotated to maintain even wear.
2. Two runtime points shall be defined for each equipment:
 - a. Lifetime Runtime: The cumulative runtime of the equipment since equipment start-up. This point shall not be readily resettable by operators.

Lifetime Runtime should be stored to a software point on the control system server so the recorded value is not lost due to controller reset, loss of power, programming file update, etc.

- b. Staging Runtime: An operator resettable runtime point that stores cumulative runtime since the last operator reset.

Staging Runtime provides a resettable runtime counter, which allows for reset of the staging runtime hours used for lead/lag or lead/standby rotation between maintenance intervals or equipment replacement while maintaining a separate log of the Lifetime Runtime. If runtime were not resettable, and logic relied only on Lifetime Runtime for determining staging lead/lag position, newly added equipment could run for years as the lead equipment before swapping rotation positions with older equipment per the logic below.

3. Lead/lag equipment: Unless otherwise noted, identical parallel staged equipment (such as CHW pumps and cooling towers) shall be lead/lag alternated when more than one is off or more than one is on so that the equipment with the most operating hours as determined by Staging Runtime is made the last stage equipment and the one with the least number of hours is made the lead stage equipment.

This strategy effectively makes it such that equipment are not "hot swapped", e.g., a pump would not be started and another stopped during operation just for runtime equalization. For example, assume there are two equipment and only one is on, but the operating equipment has exceeded the run hours of the disabled equipment. The equipment will not rotate positions until either a stage up or down occurs. If the plant stages up, then both equipment will be on and lead/lag position will switch; when the plant next stages down, the former lead equipment with more run hours will then turn off.

Expanding further, for a plant with three equipment, if all three are off or all are on, the staging order will simply be based on run hours from lowest to highest. If two equipment are on, the one with more hours will be set to be stage 2 while the other is set to stage 1; this may be the reverse of the operating order when the equipment were started. If two of the equipment are off, the one with the more hours will be set to be stage 3 while the other is set to stage 2; this may be the reverse of the operating order when the equipment were stopped.

Example with three pumps:

1. P-1 (1000 hours), 2 (950 hours), and 3 (900 hours) are all off. Staging logic makes lead/lag order: 3, 2, 1.
2. P-3 starts. Logic does not change its order since it is on by itself.
3. P-3 runs for 51 hours. Since it is on and others off, the lead/lag order does not change. It can run this way indefinitely and the order does not change.

4. *There is then a stage-up command. P-2 (the next in lead/lag order) is started. So, both P-2 and P-3 are on. P-3 now has more run hours than P-2. So, the Lead/lag order changes to: 2, 3, 1.*
5. *These two pumps run another 51 hours. Run times are P-1 (1000 hours), P-2 (1001), and P-3 (1002). No changes are made to lead/lag order because P-1 is off alone.*
6. *There is a stage down command. P-2 is now lead so it stays on. P-3 is shut off. The order for the two off pumps is now adjusted because P-1 has fewest run hours. Lead/lag order is now: 2, 1, 3.*
7. *P-2 runs for 100 more hours. It now has the longest runtime, but order does not change since it is on alone. Order is still 2, 1, 3.*
8. *There is a stage down or plant-off command. P-2 shuts off. Run times are P-1 (1000 hours), P-2 (1101), and P-3 (1002). Since all are off, order is switched to: 1, 3, 2.*

4. Lead/standby equipment:

- a. Unless equipment runs continuously, parallel equipment that are 100% redundant shall be lead/standby alternated when more than one of the equipment is off so that the equipment with the most operating hours as determined by Staging Runtime is made the last stage equipment and the one with the least number of hours is made the earlier stage equipment.

For example, assuming there are three equipment, if all three are off, the staging order will be based on run hours from lowest to highest.

- b. If equipment runs continuously, lead/standby positions shall switch at an adjustable day of the week and time (e.g., every Tuesday at 10:00 am) based on Staging Runtime; standby equipment shall first be started and proven on before former lead equipment is changed to standby and shut off.
 - 1) Variable speed fans and pumps shall have a deceleration rate of 1 Hz/second or slower set in BAS logic when disabled to prevent nuisance trips of operating equipment (e.g., chillers).

5. Exceptions to Lead/lag and Lead/standby rotation

- a. Operators with appropriate access level shall be able to manually command staging order via software points, but not overriding the In-Alarm or Hand-Operation logic in the following subsections.
 - 1) Staging order changes initiated via operator override shall be instituted as part of normal staging events.
 - 2) Staging order shall remain overridden until released by operators.
- b. Faulted Equipment:
 - 1) A faulted equipment is any equipment commanded to run that is either not running or unable to perform its required duty. If an operating equipment has any fault condition described subsequently, a Level 2 alarm shall be generated and a response shall be triggered as defined below.
 - a) Fans and Pumps
 1. Status point not matching its on/off point for 3 seconds after a time delay of 15 seconds while the equipment is commanded on.
 - b) Chillers
 1. Safety shutdown alarm condition either through network or hardwired alarm contact
 2. Chiller is manually shut off as indicated by the status of the Local/Auto switch from chiller gateway, or

3. Chiller status remains off 5 minutes after command to start (note: this condition only applies when a chiller first starts, i.e., once status is proven, then status is no longer used as a fault condition because status will come and go if chiller cycles on low load), or
 4. CHW isolation valve feedback indicates valve is not open 90 seconds after valve is commanded open, or
 5. CHW isolation valve feedback indicates valve is not closed 90 seconds after valve is commanded closed, or
 6. CW isolation valve feedback indicates valve is not open 90 seconds after valve is commanded open, or
 7. For 10 minutes, chilled water return temperature has been at least 5°F above the CHWST setpoint, and delta-T across the chiller, as determined based on the difference between chilled water return temperature and chilled water supply temperature measured at the chiller (i.e., not common CHWST), has been less than 3°F.
 8. If chiller condenser hot water supply temperature remains 15°F below setpoint for 15 minutes and delta-T across the chiller condenser, as determined based on the difference between hot water supply temperature and hot water return temperature measured at the chiller condenser (i.e., not common HWST), has been less than 10°F.
 9. Alarm per 3.2K.23 shall be user selectable as a fault condition; by default it shall not be a fault condition. Reset alarm Level from 3 to 2 when enabled as a fault condition.
- c) Air Source Heat Pumps
1. Safety shutdown alarm condition either through network or hardwired alarm contact, or
 2. If heat pump leaving water temperature remains 15°F below setpoint for 15 minutes and delta-T across the heat pump, as determined based on the difference between hot water supply temperature and hot water return temperature measured at the heat pump (i.e., not common HWST), has been less than 10°F.
- d) Cooling Towers
1. Tower fan has failed as defined above.
- 2) Upon identification of a fault condition:
- a) For fans, pumps, and cooling towers:
1. The next commanded off equipment in the staging order, Equipment "B", shall be commanded on while alarming Equipment "A" remains commanded on.
 2. If Equipment "B" fails to prove status (i.e., it also goes into alarm), it shall remain commanded on and the preceding step shall be repeated until the quantity of equipment called for by the current stage has proven on, or there are no more available equipment.
 3. Set alarming equipment to the last positions in the lead/lag or lead/standby staging order sequenced reverse chronologically (i.e., the equipment that alarmed most recently is sent to last position).
 4. Staging order of non-alarming equipment shall follow the even wear logic. Equipment in alarm can only automatically move up on the staging order if another equipment goes into alarm.
 5. Equipment in alarm shall run if so called for by the lead/lag or lead/standby staging order and present stage.
- b) For chillers and heat pumps:

1. The next commanded off equipment in the staging order, Equipment "B", shall be commanded on while alarming Equipment "A" is commanded off and set to the last position in the lead/lag staging order.
2. If Equipment B fails to prove status (i.e., it also goes into alarm), repeat the preceding step until the quantity of equipment called for by the lead/lag logic have proven on or until all equipment has been tried.
3. If all equipment has been tried and the quantity of non-alarming equipment is less than called for then the most recently alarmed equipment will remain commanded on.
4. Staging order of non-alarming equipment shall follow the even wear logic. Equipment in alarm can only automatically move up in the staging order if another equipment goes into alarm.
5. Equipment in alarm shall run if so called for by the lead/lag staging order and present stage.

The sequence for chillers and boilers differs from that used for pumps and cooling towers in that the alarming equipment does not remain commanded on until the next equipment proves status. The pump and tower logic mitigates the risk of lost loads and/or chain reaction trips of chillers and boilers by still taking advantage of any capacity the alarming equipment may provide until the lag equipment proves. This approach does not however typically work for chillers and boilers because bringing on the lag equipment while still commanding the alarming equipment to run may prevent a successful startup of the lag equipment. For example, in a parallel variable primary chilled water plant under low load conditions, starting a lag chiller while keeping the alarming chiller enabled may cause both chillers to trip on either low chilled water flow or low condenser water flow unless the minimum chilled water flow setpoint is changed to maintain minimum chilled water flow and condenser water pumps are staged to maintain minimum condenser flow through both chillers.

Example: For a set of (4) lead/lag equipment, the current staging order is Equipment A, B, C, then D. The current stage requires two equipment, so A and B are running. Then A goes into alarm. A is then commanded off at the same time as C is commanded on. If C then goes into alarm it is commanded off at the same time that D is commanded on. If D then goes into alarm it remains commanded on since all equipment has been tried. If B (the last equipment not in alarm) also goes into alarm then it remains commanded on (as the last alarming equipment with no non-alarming equipment available). At this point all equipment are in alarm and only B and D will remain commanded on until an equipment comes out of alarm. The staging order is B, D, A, C.

- c. Hand Operation. If a piece of equipment is on-in-hand (e.g., via an HOA switch or local control of VFD), the equipment shall be set to the lead device, and a Level 4 alarm shall be generated. The equipment will remain as lead until. Hand operation is determined by the following:

Any condition in which equipment appears to continue to run after being commanded off is considered a case of hand operation; in practice, this condition may arise due to other circumstances (e.g., a bad current transducer).

- 1) Fans and Pumps
 - a) Status point not matching its on/off point for 15 seconds after a time delay of 60 seconds when the equipment is commanded off.

Logic for hand operation of chillers, boilers, and cooling towers is not provided because sequences cannot stably respond to overrides by operators in all possible scenarios. For example, if a chiller is turned on in hand in a variable primary system with only one other chiller currently running, the control system would need to react by opening the isolation valves of the chiller placed in hand and either (1) immediately shutting down the former lead chiller or (2) changing the minimum chilled water flow setpoint, opening isolation valves, and possibly staging on condenser water pumps and cooling towers. Chillers, boilers, and cooling towers

should only be placed in hand by changing the staging sequence manually via the control system interface; they cannot be safely or stably operated in hand at the chiller/boiler/tower controllers.

Q. Air Economizer High Limits

1. Not Used.

R. Hierarchical Alarm Suppression

1. For each piece of equipment or space controlled by the BAS, define its relationship (if any) to other equipment in terms of “source,” “load,” or “system.”
 - a. A component is a “source” if it provides resources to a downstream component, such as a chiller providing chilled water (CHW) to an AHU.
 - b. A component is a “load” if it receives resources from an upstream component, such as an AHU that receives CHW from a chiller.
 - c. The same component may be both a load (receiving resources from an upstream source) and a source (providing resources to a downstream load).
 - d. A set of components is a “system” if they share a load in common (i.e., collectively act as a source to downstream equipment, such as a set of chillers in a lead/lag relationship serving air handlers).
 - 1) If a single component acts as a source for downstream loads (e.g., an AHU as a source for its VAV boxes), then that single-source component shall be defined as a “system” of one element.
 - 2) For equipment with associated pumps (chillers, boilers, cooling towers):
 - a) If the pumps are in a one-to-one relationship with equipment they serve, the pumps shall be treated as part of the system to which they are associated (i.e., they are not considered loads), as a pump failure will necessarily disable its associated equipment.
 - b) If the pumps are headered to the equipment they serve, then the pumps may be treated as a system, which is a load relative to the upstream equipment (e.g., chillers) and a source relative to downstream equipment (e.g., air handlers).
2. For each system as defined in Section 3.1R.1.d, there shall be a SystemOK flag, which is either true or false.
3. SystemOK shall be true when all of the following are true:
 - a. The system is proven on.
 - b. The system is achieving its temperature and/or pressure setpoint(s) for at least 5 minutes
 - c. The system is ready and able to serve its load
4. SystemOK shall be false while the system is starting up (i.e., before reaching setpoint) or when enough of the system’s components are unavailable (in alarm, disabled, or turned off) to disrupt the ability of the system to serve its load. This threshold shall be defined by the design engineer for each system.
 - a. By default, Level 1 through Level 3 component alarms (indicating equipment failure) shall inhibit SystemOK. Level 4 component alarms (maintenance and energy efficiency alarms) shall not affect SystemOK.
 - b. The operator shall have the ability to individually determine which component alarms may or may not inhibit SystemOK.
5. The BAS shall selectively suppress (i.e., fail to announce; alarms may still be logged to a database) alarms for load components if SystemOK is false for the source system that serves that load.

- a. If SystemOK is false for a cooling water system (i.e., chiller, cooling tower, or associated pump), then only high-temperature alarms from the loads shall be suppressed.
 - b. If SystemOK is false for a heating water system (i.e., boiler or associated pump), then only low temperature alarms from the loads shall be suppressed.
 - c. If SystemOK is false for an air-side system (air handler, fan coil, VAV box, etc.), then all alarms from the loads shall be suppressed.
6. This hierarchical suppression shall cascade through multiple levels of load-source relationship such that alarms at downstream loads shall also be suppressed.
7. The following types of alarms will never be suppressed by this logic:
 - a. Life/safety and Level 1 alarms
 - b. Failure-to-start alarms (i.e., equipment is commanded on, but status point shows equipment to be off)
 - c. Failure-to-stop/hand alarms (i.e., equipment is commanded off, but status point shows equipment to be on)

S. Time-Based Suppression

1. Calculate a time-delay period after any change in setpoint based on the difference between the controlled variable (e.g., zone temperature) at the time of the change and the new setpoint. The default time delay period shall be as follows:
 - a. For thermal zone temperature alarms: 10 minutes per °F of difference but no longer than 120 minutes
 - b. For thermal zone temperature cooling requests: 5 minutes per °F of difference but no longer than 30 minutes
 - c. For thermal zone temperature heating requests: 5 minutes per °F of difference but no longer than 30 minutes

3.2 CHILLED WATER PLANT

A. Chilled Water Loop Enable/Disable

1. The chilled water loop shall include an enabling schedule that allows operators to lock out the loop during off-hours, holidays, or any other scheduled event, e.g., to allow off-hour operation of HVAC systems except the chilled water loop. The default schedule shall be 24/7 (adjustable).
2. Enable the chilled water loop in the stage per 3.2C.5.a when the loop has been disabled for at least 15 minutes and:
 - a. Number of Chiller Plant Requests > I (I = Ignores shall default to 0, adjustable), and
 - b. The chilled water loop enable schedule is active.
3. Disable the chilled water loop when it has been enabled for at least 15 minutes and:
 - a. Number of Chiller Plant Requests ≤ I for 3 minutes, or
 - b. The chilled water loop enable schedule is inactive.

B. Hot Water Loop Enable/Disable

1. The hot water loop shall include an enabling schedule that allows operators to lock out the loop during off-hours, holidays, or any other scheduled event, e.g., to allow off-hour operation of HVAC systems except the hot water loop. The default schedule shall be 24/7 (adjustable).
2. Enable the hot water loop in the lowest hot water stage when the loop has been disabled for at least 15 minutes and:

- a. Number of Heating Hot-Water Plant Requests > I (I = Ignores shall default to 0, adjustable), and
 - b. OAT < HW-LOT, and
 - c. The hot water loop enable schedule is active.
3. Disable the hot water loop when it has been enabled for at least 15 minutes and:
 - a. Number of Heating Hot-Water Plant Requests ≤ I for 3 minutes, or
 - b. OAT > HW-LOT + 1°F, or
 - c. The hot water loop enable schedule is inactive.

C. Plant Staging

1. Stages

- a. Plant stages shall be defined per the table below, which indicates heat recovery chiller heating load by row and chiller cooling load by column. In each cell, the number preceding "C" designates the cooling stage and the number preceding "H" designates the heating stage.

		CHW Load (tons)					
		0	0 to 350	351 to 1060	1060 to 1710	1711 to 2020	2021+
HW Load (MBH)	0	0C0H	1C0H	2C0H	3C0H	4C0H	5C0H
	0 to 4100	0C1H	1C1H	2C1H	3C1H	4C1H	5C1H
	4101 to 8200	0C2H	1C2H	2C2H	3C2H	4C2H	5C2H
	8201 to 12300	0C3H	1C3H	2C3H	3C3H	4C3H	
	12301+	0C4H	1C4H	2C4H	3C4H		

b. Devices operating in each plant stage shall be per the table below.

		CHW Load (tons)					
		0	0 to 350	351 to 1060	1061 to 1710	1711 to 2020	2021+
HW Load (MBH)	0	NA: NA	CH1 CH2: NA	CH1&CH2: NA	CH1&CH2&HR3: NA	CH1&CH2&HR3&(HR1 HR2): NA	CH1&CH2&HR3&HR2&HR1: NA
	0 to 4100	NA: HR1 HR2	CH1 CH2: HR1 HR2	CH1&CH2: HR1 HR2	CH1&CH2&HR3: HR1 HR2	CH1&CH2&HR3&HR2: HR1	CH1&CH2&HR3&HR2&HR1*: NA
	4101 to 8200	NA: HR1&HR2	CH1 CH2: HR1&HR2	CH1&CH2: HR1&HR2	CH1&CH2&HR3: HR1&HR2	CH1&CH2&HR3&HR2*: HR1	CH1&CH2&HR3&HR2* & HR1*: NA
	8201 to 12300	NA: (HR1 HR2)&HR3	CH1 CH2: (HR1 HR2)&HR3	CH1&CH2: (HR1 HR2)&HR3	CH1&CH2&HR3*: (HR1 HR2)	CH1&CH2&HR3* & (HR1* HR2*): NA	
	12301+	NA: HR1&HR2&HR3	CH1 CH2: HR1&HR2&HR3	CH1&CH2: HR1&HR2&HR3	CH1&CH2&HR3*: HR1&HR2		

1)

Where:

- Chillers preceding the “:” have evaporators indexed to the chilled water loop and condensers indexed to the condenser water loop by default; chillers after the “:” have evaporators indexed to the condenser water loop and condensers indexed to the hot water loop.
- An “*” indicates a device with its evaporator indexed to the chilled water loop and its condenser indexed to the HW loop (i.e., direct CHW to HW heat recovery).
- “|” is an OR operator. Devices separated by an OR will be lead/lag rotated based on runtime per Paragraph 3.1P.
- Blue stages are those likely to occur in practice, grey stages are possible, though not probable, and white stages are improbable (excessive simultaneous heating and cooling) but are defined nonetheless out of conservatism.
- Waterside economizer may operate in any cooling stage, including 0, since loads are calculated as chiller loads.

2. Chillers shall be staged in part based on required cooling and heating capacity, $Q_{\text{required_cool}}$ and $Q_{\text{required_heat}}$.
 - a. $Q_{\text{required_cool}}$ is calculated based on chilled water return temperature (CHWRT, T4) entering the chillers, active chilled water supply temperature setpoint (CHWSTsp), and measured flow through the primary chilled water circuit flow meter (FLOWP, FM1), as shown in the equation below. $Q_{\text{required_cool}}$ used in logic shall be a 5-minute rolling average of instantaneous values sampled at a minimum of every 30 seconds.

$$Q_{\text{required_cool}} = \frac{\text{FLOWP}(\text{CHWRT} - \text{CHWST}_{\text{SP}})}{24} \text{ [tons]}$$
 - b. $Q_{\text{required_heat}}$ is calculated based on hot water return temperature (HWRT, T28) entering the heat recovery chillers, active hot water supply temperature setpoint (HWSTsp), and measured flow through the primary hot water circuit flow meter (FLOWP, FM6), as shown in the equation below. $Q_{\text{required_heat}}$ used in logic shall be a 5-minute rolling average of instantaneous values sampled at a minimum of every 30 seconds.

$$Q_{\text{required_heat}} = 0.490 * \text{FLOWP}(\text{HWST}_{\text{SP}} - \text{HWRT}) \text{ [kbtuh]}$$
 - c. When a stage up or stage down transition is initiated impacting enabled devices indexed to the CHW circuit, hold $Q_{\text{required_cool}}$ fixed at its last value until the longer of the successful completion of the stage change and 15 minutes. Successful completion shall be identified by all unfaulted chillers required of the stage proven on and the loop isolation valves (see 3.2C.7) proven in their commanded positions.
 - d. When a stage up or stage down transition is initiated impacting enabled devices indexed to the HW circuit, hold $Q_{\text{required_heat}}$ fixed at its last value until the longer of the successful completion of the stage change and 15 minutes. Successful completion shall be identified by all unfaulted chillers required of the stage proven on and the loop isolation valves (see 3.2C.7) proven in their commanded positions.
3. Chillers shall also be staged in part based on cooling and heating operative part load ratio, OPLR_cooling and OPLR_heating .
 - a. OPLR_cooling shall be calculated as $Q_{\text{required_cool}}$ divided by the sum of nominal chiller cooling capacities, CapCoolChX , in the current stage. OPLR_cooling shall be expressed as a percentage.
 - b. OPLR_heating shall be calculated as $Q_{\text{required_heat}}$ divided by the sum of nominal heat recovery chiller heating capacities, CapHeatChX , in the current stage. OPLR_heating shall be expressed as a percentage.
4. Staging shall be executed per the conditions below subject to the following requirements.
 - a. Each cooling/heat stage combination shall have a minimum runtime of 20 minutes.
 - b. Timers shall reset to zero at the completion of every stage change.
 - c. Supply temperature sensor used in staging logic shall be common sensor (T1) in the system header, not individual chiller sensors.
 - 1) Except: Use the average of sensors of enabled chiller's indexed to the CHW loop (T5, T7, T8, T9) if T1 has failed.
5. Cooling Staging
 - a. When the chilled water loop is enabled, the loop shall be initialized to Cooling Stage 0 (0CxH) if either of the following waterside economizer enable conditions is true and Stage 1 (1CxH) otherwise.
 - 1) Tank Cycle Flag is SixtyFortyTwo.
 - 2) Tank Cycle Flag is EightySixty; $\text{PHXLWT} \leq \text{CHWST}_{\text{sp}} + 8^{\circ}\text{F}$ for 5 minutes; and Condenser Loop Mode is Excess Heat Rejection. Assume a PLR_{HX} of 0.1 for the purposes of this calculation; refer to 3.2D.1.a.

- b. When only the Waterside Economizer is enabled (Cooling Stage 0 (0CxH), stage up to Cooling Stage 1 (1CxH) if any of the following is true:
 - 1) CHW supply temperature is $2^{\circ}\text{F} > \text{setpoint}$ for 20 minutes and $Q_{\text{required_cool}} > 100$; or
 - 2) CHW supply temperature is $4^{\circ}\text{F} > \text{setpoint}$ for 10 minutes and $Q_{\text{required_cool}} > 100$; or
 - 3) CHW supply temperature is $10^{\circ}\text{F} > \text{setpoint}$ for 20 minutes and $Q_{\text{required_cool}} \leq 100$; or
 - 4) CHW supply temperatures is $12^{\circ} > \text{setpoint}$ for 10 minutes and $Q_{\text{required_cool}} \leq 100$.
 - 5) The waterside economizer disable conditional in 3.2D.1.c.1) is true.
 - c. In all other stages, stage up if any of the following is true AND chilled water loop anti-cycling control is not active per 3.2D.6.b:
 - 1) Efficiency Condition: $Q_{\text{required_cool}}$ exceeds the upper limit for the current stage for 15 minutes and OPLR_cooling is not decreasing at a rate greater than 2.5% per minute averaged over 5 minutes; or
 - 2) Failsafe Conditions:
 - a) CHW supply temperature is $3^{\circ}\text{F} > \text{setpoint}$ for 15 minutes; or
 - b) CHW DP is $2 \text{ psi} < \text{setpoint}$ for 15 minutes, both chilled water pumps are proven on, and speed command $> 99\%$.
 - d. When only the Waterside Economizer is enabled in the next lower stage, stage down to Cooling Stage 0 (0CxH) if both of the following are true:
 - 1) WSE is enabled; and
 - 2) Either
 - a) CHWRT after the WSE feed at $T_3 < \text{CHWST setpoint}$ for 5 minutes.
 - b) Operating chiller cycles off under low load per 3.2D.6; CHWRT after the WSE feed at $T_3 < \text{CHWST setpoint} + 8^{\circ}\text{F}$ for 5 minutes; and $Q_{\text{required_cool}} < 100$.
 - e. In all other stages, stage down if both of the following are true:
 - 1) Efficiency Condition: $Q_{\text{required_cool}}$ is less than the upper limit for the next lower stage for 15 minutes and OPLR_cooling is not increasing at a rate greater than 2.5% per minute averaged over 5 minutes; and
 - 2) The failsafe stage up conditions are not true.
6. Heating Staging
- a. When the hot water loop is enabled, the loop shall be initialized to Heating Stage 1 (x1H).
 - b. Stage up if any of the following is true AND hot water loop anti-cycling control is not active per 3.2E.5.b:
 - 1) Efficiency Condition: $Q_{\text{required_heat}}$ exceeds the upper limit for the current stage for 15 minutes and OPLR_heating is not decreasing at a rate greater than 2.5% per minute averaged over 5 minutes; or
 - 2) Failsafe Conditions:
 - a) HW supply temperature is $5^{\circ}\text{F} < \text{setpoint}$ for 15 minutes; or
 - b) HW DP is $2 \text{ psi} < \text{setpoint}$ for 15 minutes, both hot water pumps are proven on, and speed command $> 99\%$.
 - c. Stage down if both of the following are true:
 - 1) Efficiency Condition: $Q_{\text{required_heat}}$ is less than the upper limit for the next lower stage for 15 minutes and OPLR_heating is not increasing at a rate greater than 2.5% per minute averaged over 5 minutes; and
 - 2) The failsafe stage up conditions are not true.

7. Loop Isolation Valve Indexing Matrix

- a. Isolation valves V13 through V18 isolate the CHW loop from the CW loop at the HRC evaporators at varying locations based on Cooling and Heating Stage.
- b. Isolation valves MV19 through MV21 isolate the HW loop from the CW loop at the HRC condensers at varying locations based on Cooling and Heating Stage.
- c. These valves are indexed and controlled as a function of plant stage. The matrix below defines the position of each valve by stage. See Section 3.2C.9 for valve actuation timing when staging events occur.

Stage	Chillers	V13	V14	V15	V16	V17	V18	MV19	MV20	MV21
0C0H	NA: NA	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
0C1H	NA: HR1 HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
0C2H	NA: HR1&HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
0C3H	NA: (HR1 HR2)&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
0C4H	NA: HR1&HR2&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
1C0H	CH1 CH2: NA	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
1C1H	CH1 CH2: HR1 HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
1C2H	CH1 CH2: HR1&HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
1C3H	CH1 CH2: (HR1 HR2)&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
1C4H	CH1 CH2: HR1&HR2&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
2C0H	CH1&CH2: NA	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
2C1H	CH1&CH2: HR1 HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
2C2H	CH1&CH2: HR1&HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
2C3H	CH1&CH2: (HR1 HR2)&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
2C4H	CH1&CH2: HR1&HR2&HR3	OPEN	OPEN	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	CLOSED
3C0H	CH1&CH2&HR3: NA	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
3C1H	CH1&CH2&HR3: HR1 HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
3C2H	CH1&CH2&HR3: HR1&HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	CLOSED	OPEN
3C3H	CH1&CH2&HR3*: (HR1 HR2)	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	OPEN	NOTE (2)

Stage	Chillers	V13	V14	V15	V16	V17	V18	MV19	MV20	MV21
3C4H	CH1&CH2&HR3*: HR1&HR2	OPEN	OPEN	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	OPEN	NOTE (2)
4C0H	CH1&CH2&HR3&(HR1 HR2): NA	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN
4C1H	CH1&CH2&HR3&HR2: HR1	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	OPEN	CLOSED	OPEN	OPEN
4C2H	CH1&CH2&HR3&HR2*: HR1	OPEN (1)	CLOSED	OPEN	OPEN	OPEN	OPEN	OPEN	NOTE (2)	OPEN
4C3H	CH1&CH2&HR3*&(HR1* HR2*): NA	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	NOTE (2)
5C0H	CH1&CH2&HR3&HR2&HR1: NA	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN
5C1H	CH1&CH2&HR3&HR2&HR1*: NA	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	NOTE (2)	OPEN	OPEN
5C2H	CH1&CH2&HR3&HR2*&HR1*: NA	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	OPEN	NOTE (2)	OPEN
(1) Exception: valve closes when either waterside economizer valve V25 or MV26 is not proven closed. See Section 3.2D.1 for WSE valve control.										
(2) Valve modulates to create extra evaporator load on chiller indexed to CHW loop (evaporator) and HW loop (condenser). See Section 3.2D.3.c.1)g).										

8. Chiller Isolation Valve Indexing Matrix

- a. Chiller bypass and isolation valves V1 through MV12 control flow through the evaporator and condenser barrels.
- b. These valves are indexed and controlled as a function of plant stage. The matrix below defines the position of each valve by stage. See section 3.2C.9 for valve actuation timing when staging events occur.

Stage	Chillers	V1	V2	V3	V4	MV5	MV6	MV7	MV8	MV9	MV10	MV11	MV12
0C0H	NA: NA	OPEN	OPEN	OPEN	OPEN	0% (5)	0%	0%	0%	0%	0%	0%	0%
0C1H	NA: HR1 HR2	OPEN	OPEN	OPEN	OPEN	0% (5)	0%	NOTE (3) 0%	100% 0%	0% NOTE (3)	0% 100%	0%	0%
0C2H	NA: HR1&HR2	OPEN	OPEN	OPEN	OPEN	0% (5)	0%	NOTE (3)	100%	NOTE (3)	100%	0%	0%
0C3H	NA: (HR1 HR2)&HR3	OPEN	OPEN	OPEN	OPEN	0% (5)	0%	NOTE (3) 0%	TBD% 0%	0% NOTE (3)	0% TBD%	NOTE (3)	TBD%
0C4H	NA: HR1&HR2&HR3	OPEN	OPEN	OPEN	OPEN	0% (5)	0%	NOTE (3)	TBD%	NOTE (3)	TBD%	NOTE (3)	TBD%
1C0H	CH1 CH2: NA	CLOSED OPEN	NOTE (6) OPEN	OPEN CLOSED	OPEN NOTE (6)	100%	NOTE (1)	0%	0%	0%	0%	0%	0%
1C1H	CH1 CH2: HR1 HR2	CLOSED OPEN	NOTE (6) OPEN	OPEN CLOSED	OPEN NOTE (6)	100%	NOTE (1)	NOTE (3) 0%	100% 0%	0% NOTE (3)	0% 100%	0%	0%
1C2H	CH1 CH2: HR1&HR2	CLOSED OPEN	NOTE (6) OPEN	OPEN CLOSED	OPEN NOTE (6)	100%	NOTE (1)	NOTE (3)	100%	NOTE (3)	100%	0%	0%
1C3H	CH1 CH2: (HR1 HR2)&HR3	CLOSED OPEN	NOTE (6) OPEN	OPEN CLOSED	OPEN NOTE (6)	100%	NOTE (1)	NOTE (3) 0%	TBD% 0%	0% NOTE (3)	0% TBD%	NOTE (3)	TBD%
1C4H	CH1 CH2: HR1&HR2&HR3	CLOSED OPEN	NOTE (6) OPEN	OPEN CLOSED	OPEN NOTE (6)	100%	NOTE (1)	NOTE (3)	TBD%	NOTE (3)	TBD%	NOTE (3)	TBD%
2C0H	CH1&CH2: NA	CLOSED	CLOSED	CLOSED	CLOSED	100%	NOTE (1)	0%	0%	0%	0%	0%	0%
2C1H	CH1&CH2: HR1 HR2	CLOSED	CLOSED	CLOSED	CLOSED	100%	NOTE (1)	NOTE (3) 0%	100% 0%	0% NOTE (3)	0% 100%	0%	0%
2C2H	CH1&CH2: HR1&HR2	CLOSED	CLOSED	CLOSED	CLOSED	100%	NOTE (1)	NOTE (3)	100%	NOTE (3)	100%	0%	0%
2C3H	CH1&CH2: (HR1 HR2)&HR3	CLOSED	CLOSED	CLOSED	CLOSED	100%	NOTE (1)	NOTE (3) 0%	TBD% 0%	0% NOTE (3)	0% TBD%	NOTE (3)	TBD%
2C4H	CH1&CH2: HR1&HR2&HR3	CLOSED	CLOSED	CLOSED	CLOSED	100%	NOTE (1)	NOTE (3)	TBD%	NOTE (3)	TBD%	NOTE (3)	TBD%
3C0H	CH1&CH2&HR3: NA	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	0%	0%	0%	0%	TBD%	NOTE (1)
3C1H	CH1&CH2&HR3: HR1 HR2	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	NOTE (3) 0%	100% 0%	0% NOTE (3)	0% 100%	TBD%	NOTE (1)
3C2H	CH1&CH2&HR3: HR1&HR2	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	NOTE (3)	100%	NOTE (3)	100%	TBD%	NOTE (1)
3C3H	CH1&CH2&HR3*: (HR1 HR2)	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (3) 0%	TBD% 0%	0% NOTE (3)	0% TBD%	NOTE (4)	TBD%
3C4H	CH1&CH2&HR3*: HR1&HR2	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (3)	TBD%	NOTE (3)	TBD%	NOTE (4)	TBD%

Stage	Chillers	V1	V2	V3	V4	MV5	MV6	MV7	MV8	MV9	MV10	MV11	MV12
4C0H	CH1&CH2&HR3&(HR1 HR2): NA	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	TBD% 0%	NOTE (1) 0%	0% TBD%	0% NOTE (1)	TBD%	NOTE (1)
4C1H	CH1&CH2&HR3&HR2: HR1	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	NOTE (3)	100%	TBD%	NOTE (1)	TBD%	NOTE (1)
4C2H	CH1&CH2&HR3&HR2*: HR1	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (3)	TBD%	NOTE (4)	TBD%	NOTE (2)	NOTE (1)
4C3H	CH1&CH2&HR3*&(HR1* HR2*): NA	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (4) 0%	TBD% 0%	0% NOTE (4)	0% TBD%	NOTE (4)	TBD%
5C0H	CH1&CH2&HR3&HR2&HR1: NA	CLOSED	CLOSED	CLOSED	CLOSED	TBD%	NOTE (1)	TBD%	NOTE (1)	TBD%	NOTE (1)	TBD%	NOTE (1)
5C1H	CH1&CH2&HR3&HR2&HR1*: NA	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (4)	100%	NOTE (2)	NOTE (1)	NOTE (2)	NOTE (1)
5C2H	CH1&CH2&HR3&HR2*&HR1*: NA	CLOSED	CLOSED	CLOSED	CLOSED	NOTE (2)	NOTE (1)	NOTE (4)	100%	NOTE (4)	100%	NOTE (2)	NOTE (1)

"TBD" means the valve will be greater than 0% open but may be less than 100% open. The position will be determined per TAB to yield flow balanced proportionally to design for all chillers operating in the stage.

(1) Valve modulates to maintain a target condenser barrel or loop flow. See Section 3.2F.3.a.

(2) Valve modulates to maintain a target evaporator flow for enabled chillers with evaporators indexed to the CHW loop and condensers indexed to the CW loop. See Section 3.2D.3.c.1)e).

(3) Valve modulates to maintain a target evaporator flow for enabled chillers with evaporators indexed to the CW loop and condensers indexed to the HW loop. See Section 3.2F.3.b.

(4) Valve modulates to maintain a target evaporator flow for enabled chillers with evaporators indexed to the CHW loop and condensers indexed to the HW loop. See Section 3.2D.3.c.1)f).

(5) Exception: valve opens to 100% when either waterside economizer valve V25 or MV26 is not proven closed. See Section 3.2D.1 for WSE valve control.

(6) Valve shall be commanded closed unless chiller has cycled off under low load conditions per 3.2D.6.b, in which case it shall be commanded open.

9. Staging Event Sequencing

- a. When a stage change is initiated, valves and setpoints must be sequenced to ensure plant stability. The subsequent logic defines the necessary order of operations for any sequencing event.
 - 1) Loop isolation valves referenced in the event sequencing below are those in the matrix in Section 3.2C.7.
 - 2) Chiller isolation and bypass valves referenced in the event sequencing below are those in the matrix in Section 3.2C.8.
- b. Step 1: Chiller Disabling
 - 1) Disable the following chillers:
 - a) Any chillers not operating in the new stage;
 - b) Any enabled HRCs with evaporators being re-indexed from the CW loop to the CHW loop or vice-versa; or

The CWP-1A/B evaporator loop unintentionally bypasses flow through CH-1 and CH-2 when all evaporator loop index valves are at least partially open during Step 2a when re-indexing HRC evaporators from the CHW loop to the CW loop or vice versa. The bypass path steals water from the enabled HRC evaporators and risks losing minimum flow and temperature control. The closed evaporator loop isolation valve must therefore remain closed while the next evaporator loop isolation valve fully closes per the matrix in Section 3.2C.7 to avoid flow disturbances. During this transition period, the HRC with an evaporator being re-indexed from the CW loop and the CHW loop or vice-versa has a fully isolated evaporator loop and must first be disabled during Step 1a and later re-enabled during Step 3.

- c) HRC-3 with a condenser being re-indexed from the CW loop and the HW loop or vice-versa.

When changing the source of condenser water from the CW loop to the HW loop or vice versa, HRC-3 can experience a large sudden change in entering condenser water temperature by up to 70°F. Large enough fluctuations in entering condenser water temperature can cause bearing pressure fault trips in the 19DV chillers. The 19DV chillers may also have issues changing active setpoint control from the condenser barrel leaving water temperature to the evaporator barrel leaving water temperature or vice versa while the chiller is running. Disabling HRC-3 during Step 1a before re-indexing the condenser barrel and re-enabling the chiller during Step 3 avoids these issues.

- 2) Once the chiller(s) is proven off, close the disabled chiller's condenser isolation valve (for HRCs) or open the chiller's condenser bypass valve (for series counterflow chillers, SCF) if applicable per the matrix in Section 3.2C.8. Note that this includes closing MV6 as applicable to the stage.
 - 3) When the controller of the chiller(s) being disabled indicates no request for chilled water flow or (3) minutes have elapsed, close the chiller's evaporator isolation valve (for HRCs) or open the chiller's evaporator bypass valve (for series counterflow chillers, SCF) if applicable per the matrix in Section 3.2C.8. Note that this includes closing MV5 as applicable to the stage.
- c. Step 2a: Loop Indexing
 - 1) Wait 60 seconds after the final commands in the proceeding step, then proceed to the following actions as applicable.
 - 2) When indexing an enabled HRC's evaporator from the CW loop to the CHW loop:
 - a) Not Used. See 3.2C.9.b.1)b).

The meanings of "CHW loop evaporator isolation valve" and "CW loop evaporator isolation valve" vary depending on the stage. If, for instance HRC-2 is indexing from the CW to the CHW loop, the "CHW loop evaporator isolation valves" are V15 and V16 and the "CW loop evaporator isolation valves" are V13 and V14. If HRC-1 on the other hand is doing the same, then the "CHW loop evaporator isolation valves" are V13 and V14 and there aren't any "CW loop evaporator isolation valves".

- 3) When indexing a disabled HRC's evaporator from the CW loop to the CHW loop:
 - a) Close CW loop evaporator isolation valves as required for the stage per Section 3.2C.7.
 - b) Once CW loop isolation valves are proven closed, open the CHW loop evaporator isolation valves as required for the stage.
- 4) When indexing an enabled HRC's condenser from the HW loop to the CW loop:
 - a) Set the HRC to CHWST control with a setpoint equal to the current loop CHWST setpoint.
 - b) Open CW loop condenser isolation valve as required for the stage per Section 3.2C.7.
 - c) Once CW loop isolation valve is proven open, close HW loop condenser isolation valves as required for the stage.

The meanings of "CW loop condenser isolation valve" and "HW loop condenser isolation valve" vary depending on the stage. If, for instance HRC-3 is indexing from the HW to the CW loop, the "CW loop condenser isolation valve" is MV21 and the "HW loop condenser isolation valve" is MV20. If HRC-1 on the other hand is doing the same, then the "CW loop condenser isolation valve" is MV19 and there isn't a "HW loop condenser isolation valve".

- 5) When indexing a disabled HRC's condenser from the HW loop to the CW loop:
 - a) Set the HRC to CHWST control with a setpoint equal to the current loop CHWST setpoint.
 - b) Concurrently open CW loop condenser isolation valve and close HW loop condenser isolation valve as required for the stage per Section 3.2C.7.
 - 6) When indexing an enabled HRC's evaporator from the CHW loop to the CW loop:
 - a) Not Used. See 3.2C.9.b.1)b).
 - 7) When indexing a disabled HRC's evaporator from the CHW loop to the CW loop:
 - a) Close CHW loop evaporator isolation valves as required for the stage per Section 3.2C.7.
 - b) Once CHW loop evaporator isolation valves are proven closed, open CW loop evaporator isolation valves as required for the stage.
 - 8) When indexing an enabled HRC's condenser from the CW loop to the HW loop:
 - a) Open HW loop condenser isolation valve as required for the stage per Section 3.2C.7.
 - b) Once the HW loop condenser isolation valve is proven open, close CW loop condenser isolation valve as required for the stage.
 - c) Set the HRC to HWST control with a setpoint equal to the current loop HWST setpoint.
 - 9) When indexing a disabled HRC's condenser from the CW loop to the HW loop:
 - a) Concurrently open HW loop condenser isolation valves and close CW loop condenser isolation valves as required for the stage per Section 3.2C.7.
 - b) Set the HRC to HWST control with a setpoint equal to the current loop HWST setpoint.
- d. Step 2b: Already Enabled Chiller Isolation Valve Indexing and Control Loop Enabling
- 1) For already enabled chiller(s), set evaporator isolation valve position or enable valve control loop per Section 3.2C.8 concurrent with the *first* loop

- evaporator isolation valve being commanded to its required positions for the new stage.
- 2) For already enabled chiller(s), set condenser isolation valve position or enable valve control loop per Section 3.2C.8 60 seconds after the *last* loop condenser isolation valve being commanded to its required positions for the new stage.
- e. Step 3: Chiller Enabling
- 1) After the final commands in the proceeding step, proceed to the following actions as applicable.
 - 2) Open/enable the evaporator isolation valve (HRC) or close the evaporator bypass valve (series counterflow chiller, SCF) of the chiller(s) to be enabled in the new stage if applicable per the matrix in Section 3.2C.8. Note that this includes opening/enabling MV5 as applicable to the stage.
 - 3) Concurrently, open/enable the condenser isolation valve (HRC) or close the condenser bypass valve (series counterflow chiller, SCF) of the chiller(s) to be enabled in the new stage if applicable per the matrix in Section 3.2C.8. Note that this includes opening/enabling MV6 as applicable to the stage.
 - 4) Enable the chiller(s) when all of the following are true:
 - a) The lead tower water pump, TWP-1A or B, has been proven on for 10 minutes if Excess Heat Rejection Mode is enabled.
 - b) Evaporator flow exceeds 95% of MinEvapFlowX for 3 minutes.
 - c) Condenser flow exceeds the following for 3 minutes:
 1. 95% of 0.10*DesCondFlowX in the case of HRC-1 or HRC-2.
 2. 95% of 0.40*DesCondFlowX in the case of HRC-3.
 3. 95% of 0.15*DesCondFlowX in the case of CH-1 or CH-2.

On startup, the 19DV chillers experience large fluctuations in leaving condenser water temperature when condenser flow is low. These fluctuations can cause a bearing pressure fault trip. The 19DV chiller condenser isolation valves therefore control to a sufficiently high minimum condenser flow rate on startup to prevent that fault from occurring without exceeding the design evaporator flow rate of one operating HRC so as to accidentally send potentially cold leaving condenser water to the HRC evaporators.

D. Chilled Water Loop Control

1. Waterside Economizer
 - a. Waterside economizer enabling shall be dictated in part based on predicted tower heat exchanger leaving water temperature (PHXLWT). PHXLWT shall be calculated as:

$$PHXLWT = T_{WB} + PA_{HX} + PA_{CT}$$

$$PA_{HX} = DA_{HX} * PLR_{HX}$$

$$PA_{CT} = m * (DT_{WB} - T_{WB}) + DA_{CT}$$

Where,

T_{WB} = current wetbulb temperature

PA_{HX} = predicted heat exchanger approach

PA_{CT} = predicted cooling tower approach

DA_{HX} = design heat exchanger approach, HxApp

PLR_{HX} = Predicted heat exchanger part load ratio (current chilled water flow rate, FM1, divided by design HX closed loop flow rate, HxDesFlow)

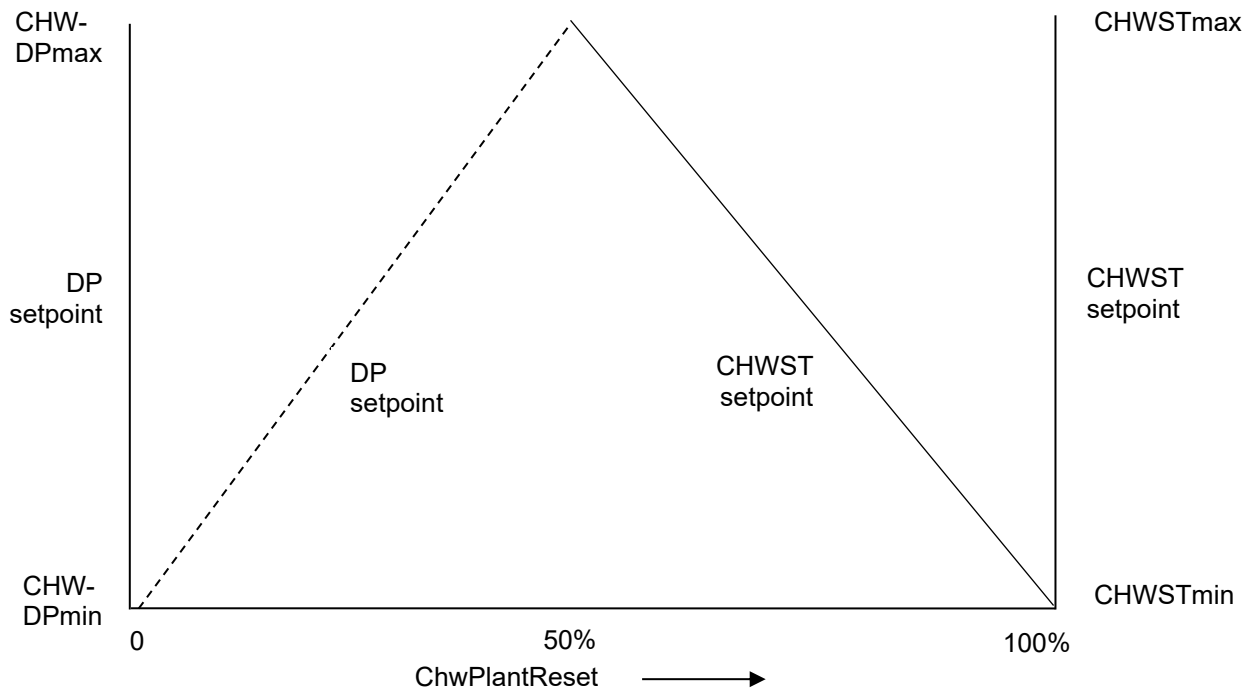
DT_{WB} = design wetbulb temperature

DA_{CT} = design cooling tower approach, CtApp

m = output of PHXLWT tuning logic below.

- b. PHXLWT Tuning
 - 1) Decrease "m" by 0.02 when the economizer is disabled if the economizer remained enabled for greater than 60 minutes.

- 2) Increase “m” by 0.02 when the economizer is disabled if the economizer remained enabled for less than 30 minutes and was disabled based on condition 3.2D.1.c.1)a).
 - 3) “m” shall be limited to the range of -0.2 to 0.5.
 - 4) “m” shall initialize at 0 upon first plant start up and shall not be reinitialized every time the chilled water loop is disabled/enabled. Rather, “m” holds its value when the chilled water loop is disabled and tuning resumes from that value when the chilled water loop is re-enabled. Store “m” in non-volatile memory.
- c. Enable/Disable
- 1) Disable waterside economizer whenever the chilled water loop is disabled, or any of the following is true:
 - a) CHWRT at T3 > CHWRT at T2 – 1°F for 5 minutes.
 - b) Tank Cycle Flag is SixtyFortyTwo (see 3.2F.1.a) and T2 > 70°F or T2 < 50°F for 10 minutes.
 - c) Tank Cycle Flag is EightySixty (see 3.2F.1.a) and Condenser Loop Mode is not Excess Heat Rejection (see 3.2F.2).
 - 2) When the chilled water loop is already enabled, enable the waterside economizer whenever waterside economizer has been disabled for 20 minutes, and either:
 - a) Tank Cycle Flag is SixtyFortyTwo and 50°F < T2 < 70°F for 5 minutes.
 - b) Tank Cycle Flag is EightySixty; PHXLWT < CHWRT at T2 – 3°F for 5 minutes; and Condenser Loop Mode is Excess Heat Rejection.
 - 3) When WSE is enabled:
 - a) Open V25.
 - b) Modulate MV26 to maintain CHWRT at T3 at a setpoint of Condenser Loop CCWST at T20 + 1°F using a direct acting control loop.
 - 4) When WSE is disabled:
 - a) Close V25 and MV26.
 - 5) Valve MV26 shall generate up to 1 Condenser Isolation Valve Request per 3.2F.3.a.4) based on valve position. It shall not generate requests based on flow.
2. Chilled Water Loop Reset
- a. Chilled water supply temperature setpoint CHWSTsp and pump differential pressure setpoint CHW-DPsp shall be reset based on the current value of the logic variable ChwPlantReset as shown below and described subsequently.



- 1) From 0% loop output to 50% loop output, reset DP setpoint from CHW-DPmin to CHW-DPmax.
- 2) From 50% loop output to 100% loop output, reset CHWST setpoint from CHWSTmax to CHWSTmin.
- 3) ChwPlantReset variable shall be reset using Trim & Respond logic with the following parameters:

Variable	Value
Device	Any CHW Pump Distribution Loop
SP0	100%
Spmin	0%
Spmax	100%
Td	15 minutes
T	5 minutes
I	2
R	Cooling CHWST Reset Requests
Sptrim	-2%
Spres	+3%
Spres-max	+7%

- 4) CHWST Plant Reset loop shall be enabled when the chilled water loop is enabled and disabled otherwise.
- 5) When a plant stage change is initiated, CHW Plant Reset logic shall be disabled and value fixed at its last value for the longer of 15 minutes and the time it takes for the plant to successfully stage.
- 6) Where more than one remote DP sensor serves a given set of primary pumps, remote DP setpoints for all remote sensors serving those pumps shall increase in unison. Note: if remote sensors have different CHW-

DPmax values, then the amount each DP setpoint changes per percent loop output will differ.

- b. The CHWST setpoint for each chiller shall equal the output of the plant reset loop output with the following exceptions:
 - 1) The minimum setpoint shall be inhibited from being less than the low limit logic variable CHWSTsafety.

Under low load/flow conditions, evaporator approach (CHWST – SST) increases to as much as 5°F. The 19DV chiller is configured to cycle off when CHWST < setpoint – 5°F. The machine also triggers a low suction temperature recycle at 34°F SST. The combination of these variables and a 42°F CHWST therefore mean that a low suction temperature trip can occur prior to a low load recycle (42°F – 5°F – 5°F = 32°F). Resetting the minimum CHWST setpoint under low load mitigates this risk.

- a) In Cooling Stage 0, CHWSTsafety shall be 42°F. Otherwise, CHWSTsafety shall reset linearly from 46°F when Qrequired_cool ≤ 100 to 42°F when Qrequired_cool ≥ 150.
- b) CHWSTsafety shall be locked at 46°F whenever in Stage 1CxH and the chiller with evaporator indexed to the CHW loop is proven off or has been proven on for < 15 minutes.
- 2) When both CH-1 and CH-2 are enabled, CH-1's setpoint shall equal the 5-minute rolling average of the following equation.

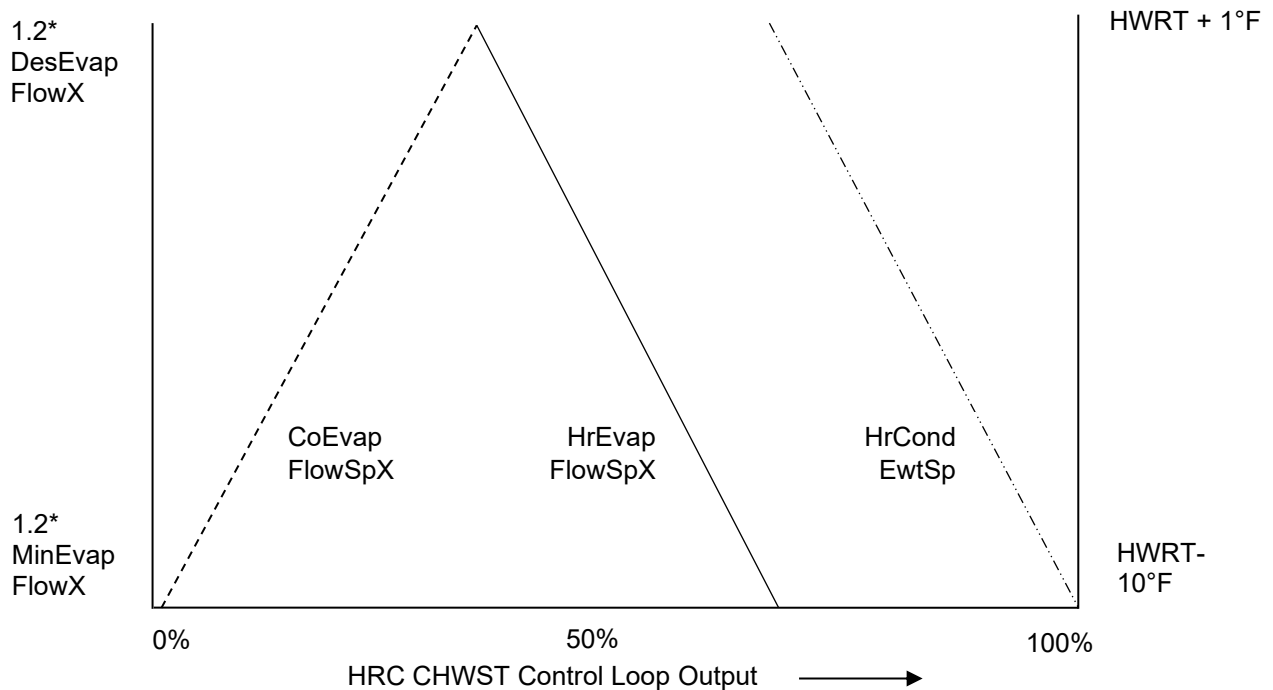
$$CWHST_{SP,CH-1} = T4 - (T4 - CHWST_{SP,Plant}) * \frac{CapCoolCh1}{CapCoolCh1 + CapCoolCh2}$$

Using a rolling average avoids sudden fluctuations in chiller setpoint that may induce plant instability. Weighting the setpoint by the design capacity ratio of the series chillers improves efficiency when the upstream chiller is selected to provide more of the load. The efficiency of even identical chillers in series can be optimized by shifting load to the upstream chiller which is more efficient due to the warmer CHWST. This is usually determined by iteratively varying this setpoint to minimize combined chiller power using chiller selection software.

3. CHWST Setpoint Control

- a. Chillers CH-1 and CH-2 directly control CHWST to setpoint based on a setpoint written to the chiller's onboard controller over BACnet.
- b. When HRC-1, HRC-2, or HRC-3's condenser is indexed to the condenser water loop, that chiller shall directly control CHWST to setpoint based on a setpoint written to the chiller's onboard controller over BACnet. Refer to Section 3.2C.9 for timing of indexing from HWST to CHWST setpoint control.
- c. When HRC-1, HRC-2, or HRC-3's condenser is indexed to the hot water loop while its evaporator is indexed to the chilled water loop, the chiller shall directly control HWST setpoint (see Section 3.2E.2) and CHWST shall be controlled to setpoint by modulating evaporator flow and entering condenser water temperature (i.e., varying heating load).
 - 1) For each enabled HRC with its condenser indexed to the hot water loop and evaporator indexed to the chilled water loop, a direct acting HRC CHWST control loop with an output ranging from 0% to 100% shall be used to maintain leaving evaporator water temperature (T7, T8, or T9) at setpoint.
 - a) The loop shall be enabled with a bias of 33% whenever the HRC is proven on in a stage where its condenser is indexed to the hot water loop and its evaporator is indexed to the chilled water loop immediately *after* staging completion. Loop shall be disabled with output set to 33% otherwise.
 - b) The loop shall utilize a +/-2.5°F deadband around the target CHWST setpoint to minimize hunting.

- c) Loop must be slow (stability is more critical than tight temperature control) to avoid chiller trips. By default, aim for P&I gains that prevent loop output from changing more than 33% in 15 minutes.
- d) Loop output shall be mapped as shown below and described subsequently.



1. From 0% to 33% output, reset the *target* evaporator flowrate setpoint of enabled chillers with evaporators indexed to the CHW loop and condensers indexed to the CW loop, CoEvapFlowSpX, from 1.2*MinEvapFlowX to 1.2*DesEvapFlowX, where "X" designates a chiller's tag.
 2. From 33% to 67% loop output, reset the evaporator flowrate setpoint of the enabled HRC with its evaporator indexed to the CHW loop and condenser indexed to the HW loop, HrEvapFlowSpX, from 1.2*DesEvapFlowX to 1.2*MinEvapFlowX, where "X" designates a given chiller's tag.
 3. From 67% to 100% output, reset the entering condenser water temperature setpoint of the enabled HRC with its evaporator indexed to the CHW loop and condenser indexed to the HW loop, HrCondEwtSp, from HWRT + 1°F to HWRT - 10°F. HWRT shall be the 5-minute rolling average of T28 sampled once every 30 seconds.
- e) A reverse acting control loop shall maintain the evaporator flow of each enabled chiller with its evaporator indexed to the CHW loop and condenser indexed to the CW loop at the *lowest* target value of CoEvapFlowSpX by modulating the chiller's evaporator isolation valve, MV5, MV9, or MV11. CoEvapFlowSpX setpoint(s) shall be determined from each enabled HRC CHWST control loop per Section 3.2D.3.c.1)d)1.
1. Map loop output from 10% valve position at 0% loop output to 100% valve position at 100% loop output.

2. Loop shall be enabled whenever NOTE (2) is listed for the stage (see Section 3.2C.8) at the time indicated in Section 3.2C.9 Step 2b or 3.

Note that the implication of this timing is that these valves can be controlled to a setpoint output from a disabled HRC CHWST control loop based on its output when disabled (33% by default per the above). This is so because the valve flow control loop of a newly enabled chiller will be enabled during Step 2b of the stage change (see 3.2C.9), but its HRC CHWST control loop will not start until the chiller is proven on in Step 3.

3. Loop shall be disabled whenever NOTE (2) is not listed for the stage (see Section 3.2C.8) at the time indicated in Section 3.2C.9 Step 1 or 2b (i.e., during chiller disabling or when the valve changes to a static position for the new stage).
 4. Loop shall be biased to launch from 100% (valve full open).
 5. Control loop shall be configured with a setpoint deadband equal to +/-2% of chiller DesEvapFlowX to minimize valve hunting.
- f) A reverse acting control loop shall maintain the evaporator flow of each enabled HRC with its evaporator indexed to the CHW loop and condenser indexed to the HW loop at HrEvapFlowSpX by modulating the chiller's evaporator isolation valve, MV7, MV9, or MV11.
1. Map loop output 10% valve position at 0% loop output to 100% valve position at 100% loop output.
 2. Loop shall be enabled whenever NOTE (4) is listed for the stage (see Section 3.2C.8) at the time indicated in Section 3.2C.9 Step 2b or 3.

Note that the implication of this timing is that these valves can be controlled to a setpoint output from a disabled HRC CHWST control loop based on its output when disabled (33% by default per the above). This is so because the valve flow control loop of a newly enabled chiller will be enabled during Step 2b of the stage change (see 3.2C.9), but its HRC CHWST control loop will not start until the chiller is proven on in Step 3.

3. Loop shall be disabled whenever NOTE (4) is not listed for the stage (see Section 3.2C.8) at the time indicated in Section 3.2C.9 Step 1 or 2b (i.e., during chiller disabling or when the valve changes to a static position for the new stage).
4. Loop shall be biased to launch from 50% (valve 55% open).
5. HrEvapFlowSpX shall be limited to no less than the value, HrEvapFlowLimitX:

$$\left(\frac{1.1 * FM1 - 1.2 * DesEvapFlow_{CO}}{MinEvapFlow_{HR}} \right) MinEvapFlowX$$
 - a. Where, DesEvapFlow_{CO} is the sum of DesEvapFlowX for enabled chillers with condensers indexed to the CW loop; and
 - b. MinEvapFlow_{HR} is the sum of MinEvapFlowX for enabled chillers with evaporators indexed to the CHW loop and condensers indexed to the HW loop.
 - c. HrEvapFlowLimitX used in logic shall be a 1-minute rolling average of instantaneous values sampled at minimum once every 10 seconds.
6. Control loop shall be configured with a setpoint deadband equal to +/-2% of chiller DesEvapFlowX to minimize valve hunting.

The above logic is necessary to avoid potentially starving the chilled water loop. This could happen if HRC evaporator flowrate were reset to a value below that required to meet CHW

loop flow with all CO chillers at 120% of design flowrate. Setting a minimum HRC evaporator loop flow that is reset off current chilled water loop flow prevents that from happening.

- g) A direct acting loop shall maintain HRC condenser entering water temperature at HrCondEwtSp by modulating MV19, MV20, or MV21 to bleed condenser water into the HW loop. Valve under loop control shall be that designated with NOTE (2) in the Loop Isolation Valve matrix in Section 3.2C.7.
 - 1. Map loop output from 0% valve position at 0% loop output to 100% valve position at 100% loop output.
 - 2. Loop shall be enabled whenever any HRC CHWST control loop is enabled.
 - 3. When all HRC CHWST control loops are disabled, this loop shall be disabled and output set to 0%.
 - 4. When more than one HRC has an enabled HRC CHWST control loop, HrCondEwtSp shall be the setpoint output from the HRC CHWST control loop of the HRC nearest to the CW loop interconnection (i.e., the HRC with the highest tag number that has an evaporator indexed to the CHW loop and condenser indexed to the HW loop) and the control point shall be the condenser entering water temperature of that HRC (T31, T32, or T33).
- 4. Chilled Water Pumps
 - a. CHW pumps shall be lead/lag controlled per Section 3.1P.
 - b. Enable lead CHW pump 24/7.
 - c. Chilled water pumps shall be staged as a function of CHWFR, the ratio of current chilled water flow, FLOWP, as measured by the loop flow meter to design primary pump flow, PCHWFdesign.
 - d. Enable the lag pump whenever either:
 - 1) CHWFR > 47% for 10 minutes.
 - 2) Pump speed command > 99% for 5 minutes.
 - e. Disable the lag pump whenever any of the follow is true:
 - 1) Pumps staged up on CHWFR condition and CHWFR < 47% for 10 minutes.
 - 2) Pump staged up on speed condition and pump speed command < 30% for 5 minutes.
 - 3) The lead pump is disabled.
 - f. Remote DP shall be maintained at a setpoint of CHW-DPsp determined by the reset scheme described herein. CHW-DPsp shall be maintained by a reverse acting PID loop running in the controller to which the remote sensor is wired; the loop output shall be a DP setpoint for the local primary loop DP sensor hardwired to the plant controller. Reset local DP from LocalCHW-DPmin psi at 0% loop output to LocalCHW-DPmax at 100% loop output. Loop shall be enabled whenever any pump is proven on and disabled otherwise.
 - g. When any pump is proven on, pump speed will be controlled by a reverse acting PID loop maintaining the local primary DP signal at the DP setpoint output from the remote sensor control loop. All pumps receive the same speed signal.
 - h. Where multiple remote DP sensors exist, a PID loop shall run for each sensor. The DP setpoint for the local DP sensor shall be the highest DP setpoint output from each of the remote loops.
- 5. Chilled Water Loop Minimum Flow Control
 - a. CH-1 and CH-2 shall share a common chilled water minimum flow control loop since their evaporators are in series when enabled.
 - b. Each HRC shall have its own chilled water minimum flow control loop.

- c. Loops shall be reverse acting with an output varying from 0% to 100%. Each loop shall act to maintain minimum flow as measured by the chiller's local evaporator flow meter at $1.2 * \text{MinEvapFlowX}$ for that chiller.
 - d. Enable loop whenever the chiller's evaporator is indexed to the chilled water loop, as determined by stage, and its evaporator chiller isolation valve is commanded >0% open (HRC) or at least one evaporator chiller bypass valve (SCF) is commanded shut.
 - e. Disable loop whenever either the chiller is proven off, or the chiller's evaporator is indexed to the condenser water loop, as determined by stage, and the evaporator loop isolation valves have proven to the positions required of the stage (HRC) or both chillers are disabled (SCF).
 - f. Map each enabled loop to requested minimum flow bypass valve position, MinFlowPosX , in proportion to loop output from 0% to 100%. "X" is a unique identifier for each control loop.
 - g. Command the minimum flow bypass valve, MV23, to the largest MinFlowPosX of enabled control loops.
6. Chilled Water Loop Low Load Anti-Cycling Control
- a. The subsequent anti-cycling logic shall apply to Cooling Stage 1 (1CxH) only.
 - b. Whenever either of the following occurs for the chiller indexed to the chilled water loop, disable the chiller and inhibit cooling stage up until demand limit equals 100%:
 - 1) Chiller cycles off under internal control with no faults active and CHWST as measured by the chiller's internal sensor < current CHWST setpoint (indicating low load cycling); or
- When inlet guide vane (IGV) position is nearly shut under low load, moderate lift conditions, insufficient refrigerant is sent to the condenser barrel to replenish the high side float chamber, in turn vapor locking the bearing lubricant pump and causing a bearing pressure fault trip. This condition aims to prevent that fault from occurring.*
- 2) Chiller inlet guide vane position < 1% for 4 minutes (adj.) and CHWST as measured by the chiller's internal sensor < current CHWST setpoint (indicating low load cycling) + 0°F offset (adj.). This second condition shall apply to CH-1 and CH-2 only.
 - c. Enable the chiller when chiller CHWST as measured by the chiller's internal sensor > current CHWST setpoint + 10°F.
 - d. When the chiller is enabled, demand limit the chiller per the following trim and respond logic.

Trim & Respond Variables

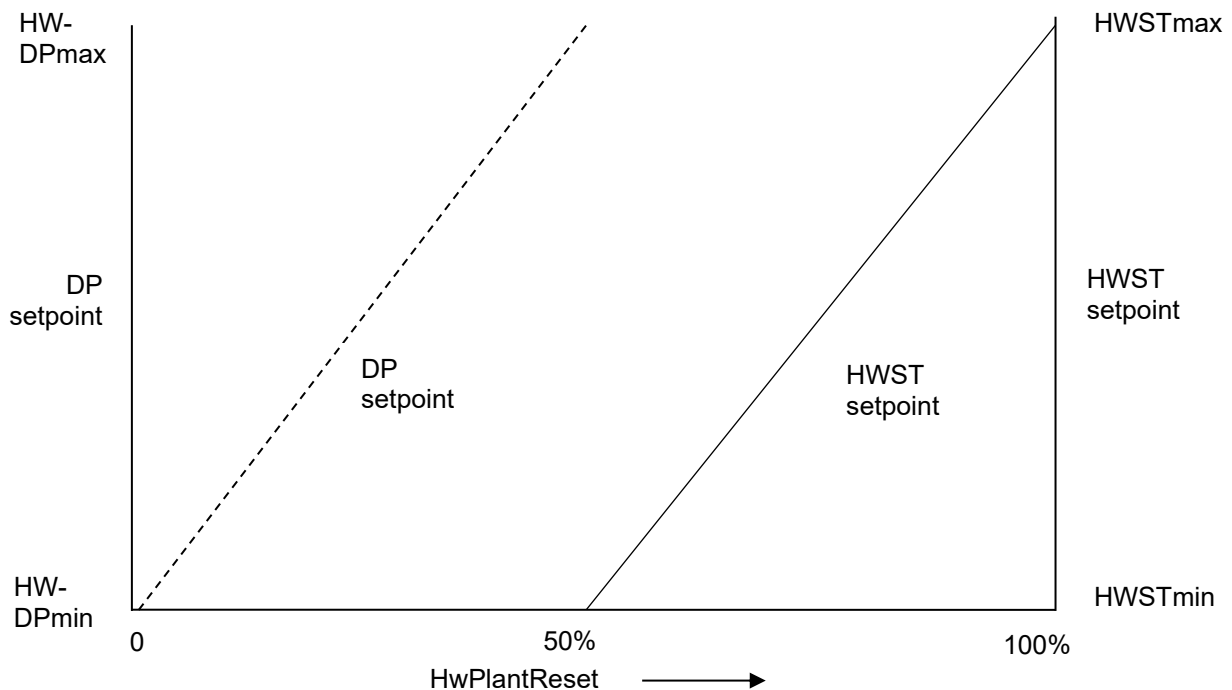
Variable	Value
Device	Chiller
SP0	DemandLimitMinCool
Spmin	DemandLimitMinCool
Spmax	100%
Td	10 min
T	2 min
I	0
R	Demand Limit Reset Requests

Sptrim	-0%
Spres	+5%
Spres-max	+15%

- 1) Demand Limit Reset Requests
 - a) If CHWST > CHWST from 2 minutes prior + 2°F, generate 3 requests.
 - b) Else, if CHWST > CHWST from 2 minutes prior + 1°F, generate 2 requests.
 - c) Else, if CHWST > CHWST from 2 minutes prior, generate 1 request.
 - d) Else, generate 0 requests.
 - e) Inhibit all requests when CHWST < CHWST setpoint.
- 2) Logic intentionally inhibits trimming. Once demand limit reaches 100% or 45 minutes has elapsed since the loop was enabled, disable loop and set demand limit to 100%. Demand limit shall remain 100% until the chiller cycles back off and the reset loop is re-enabled at SP0.

E. Hot Water Loop Control

1. Hot Water Loop Reset
 - a. HWSTmax in the following logic shall be 125°F.
 - b. Hot water supply temperature setpoint HWSTsp and pump differential pressure setpoint HW-DPsp shall be reset based on the current value of the logic variable called HwPlantReset as shown below and described subsequently.



- c. From 0% loop output to 50% loop output, reset DP setpoint from HW-DPmin to HW-DPmax.
- d. From 50% loop output to 100% loop output, reset HWST setpoint from HWSTmin to HWSTmax.
- e. HwPlantReset variable shall be reset using Trim & Respond logic with the following parameters:

Variable	Value
----------	-------

Device	Any HW Pump Distribution Loop
SP0	100%
Spmin	0%
Spmax	100%
Td	15 minutes
T	5 minutes
I	2
R	Heating HWST Reset Requests
Sptrim	-2%
Spres	+3%
Spres-max	+7%

- f. HWST Plant Reset loop shall be enabled when the hot water loop is enabled and disabled otherwise.
 - g. When a plant stage change is initiated, HW Plant Reset logic shall be disabled and value fixed at its last value for the longer of 15 minutes and the time it takes for the plant to successfully stage.
 - h. Where more than one remote DP sensor serves a given set of primary pumps, remote DP setpoints for all remote sensors serving those pumps shall increase in unison. Note: if remote sensors have different HW-DPmax values, then the amount each DP setpoint changes per percent loop output will differ.
 - i. The HWST setpoint for each HRC shall equal the output of the plant reset loop output.
2. HWST Setpoint Control
 - a. Chillers HRC-1, -2, and -3 directly control HWST to setpoint when their condensers are indexed to the HW loop based on a setpoint written to the chiller's onboard controller over BACnet. Refer to Section 3.2C.9 for timing of indexing from CHWST to HWST setpoint control if the condenser was previously indexed to the condenser water loop.
3. Hot Water Pumps
 - a. HW pumps shall be lead/lag controlled per Section 3.1P.
 - b. Enable lead HW pump whenever any HRC indexed to the hot water loop, as determined by stage, has a condenser isolation valve commanded >0% open. Disable the lead HW pump otherwise.
 - c. Hot water pumps shall be staged as a function of HWFR, the ratio of current hot water flow, FLOWP, as measured by the loop flow meter to design primary pump flow, PHWFdesign.
 - d. Enable the lag pump whenever either:
 - 1) HWFR > 47% for 10 minutes.
 - 2) Pump speed command > 99% for 5 minutes.
 - e. Disable the lag pump whenever any of the following is true:
 - 1) Pumps staged up on HWFR condition and HWFR < 47% for 10 minutes.
 - 2) Pumps staged up on speed condition and pump speed command < 30% for 5 minutes.
 - 3) The lead pump is disabled.
 - f. Remote DP shall be maintained at a setpoint of HW-DPsp determined by the reset scheme described herein. HW-DPsp shall be maintained by a reverse acting PID loop running in the controller to which the remote sensor is wired; the loop output shall be a DP setpoint for the local primary loop DP sensor hardwired to the plant controller. Reset local DP from LocalHW-DPmin psi at 0% loop output to LocalHW-

DPmax* at 100% loop output. Loop shall be enabled whenever any pump is proven on and disabled otherwise.

- 1) LocalHW-DPmax* variable shall be reset using Trim & Respond logic with the following parameters:

Variable	Value
Device	Any HW Pump Distribution Loop
SP0	LocalHW-DPmax0
Spmin	LocalHW-DPmin
Spmax	LocalHW-DPmax
Td	10 minutes
T	2 minutes
I	0
R	Heating Head Pressure Control Requests
Sptrim	+1.0 psi
Spres	-2.0 psi
Spres-max	-4.0 psi

- a) When HWRT entering the HRCs as measured by T28 < 70°F, LocalHW-DPmax0 shall equal LocalHW-DPmin. LocalHW-DPmax0 shall equal LocalHW-DPmax otherwise.
- b) Heating Head Pressure Control Requests
1. If HWST < HWST setpoint – 20°F, generate 1 request.
 2. Else, if HWST < HWST setpoint – 30°F, generate 2 requests.
 3. Else, generate 0 requests.
- g. When any pump is proven on, pump speed will be controlled by a reverse acting PID loop maintaining the local primary DP signal at the DP setpoint output from the remote sensor control loop. All pumps receive the same speed signal.
- h. Where multiple remote DP sensors exist, a PID loop shall run for each sensor. The DP setpoint for the local DP sensor shall be the highest DP setpoint output from each of the remote loops.
4. Hot Water Loop Minimum Flow Control
- a. Each HRC shall have its own hot water minimum flow control loop.
 - b. Loops shall be reverse acting with an output varying from 0% to 100%. Each loop shall act to maintain minimum flow as measured by the chiller's local condenser flow meter at 1.1*MinCondFlowX for that chiller.
 - c. Enable control loop whenever the chiller's condenser is indexed to the hot water loop, as determined by stage, and its condenser isolation valve is commanded >0% open.
 - d. Disable loop whenever the chiller is proven off, or the chiller's condenser is indexed to the condenser water loop, as determined by stage, and the condenser loop isolation valves have proven to the positions required of the stage.
 - e. Map each enabled loop to requested minimum flow bypass valve position, MinFlowPosX, in proportion to loop output from 0% to 100%. "X" is a unique identifier for each control loop.
 - f. Command the minimum flow bypass valve, MV24, to the largest MinFlowPosX of enabled control loops.
5. Hot Water Loop Low Load Anti-Cycling Control
- a. The subsequent anti-cycling logic shall apply if:
 - 1) Plant is in Heating Stage 1 (xC1H); and
 - 2) Enabled HRC has its evaporator indexed to the condenser water loop.

- b. When the HRC indexed to the hot water loop cycles off with no faults active AND HWST as measured by the HRC's internal sensor is > current HWST setpoint (indicating low load cycling), disable the chiller and inhibit heating stage up until demand limit equals 100%.
- c. Enable the chiller when HRC HWST as measured by the chiller's internal sensor < current HWST setpoint – 10°F.
- d. When the HRC is enabled, demand limit the chiller per the following trim and respond logic.

Trim & Respond Variables

Variable	Value
Device	Chiller
SP0	DemandLimitMinHeat
Spmin	DemandLimitMinHeat
Spmax	100%
Td	10 min
T	2 min
I	0
R	Demand Limit Reset Requests
Sptrim	–0%
Spres	+5%
Spres-max	+15%

- 1) Demand Limit Reset Requests
 - a) If HWST < HWST from 2 minutes prior – 2°F, generate 3 requests.
 - b) Else, if HWST < HWST from 2 minutes prior – 1°F, generate 2 requests.
 - c) Else, if HWST < HWST from 2 minutes prior, generate 1 request.
 - d) Else, generate 0 requests.
 - e) Inhibit all requests when HWST > HWST setpoint.
- e. Logic intentionally inhibits trimming. Once demand limit reaches 100% or 45 minutes has elapsed since the loop was enabled, disable loop and set demand limit to 100%. Demand limit shall remain 100% until the chiller cycles back off and the reset loop is re-enabled at SP0.

F. Closed Condenser Water Loop Control

1. Tank Charge Parameters

- a. Tank Cycle Flag
 - 1) Set the cycle parameter to EightySixty whenever both of the following are true:
 - a) All tank temperature sensors measure > 50°F for 10 minutes.
 - b) (FM13 – FM14 – FM_{CCW Makeup}) flow is greater than +0 GPM (flow out of lower port of tank/tank is charging with heat rejected from the condenser barrel loop) for 10 minutes.
 - 2) Set the cycle parameter to SixtyFortyTwo whenever both of the following are true:

- a) All tank temperature sensors measure $< 75^{\circ}\text{F}$ for 10 minutes.
- b) $(\text{FM13} - \text{FM14} - \text{FM}_{\text{CCW Makeup}})$ flow < 0 GPM (flow into lower port of tank/tank is discharging to provide a heat source to the evaporator barrel loop) for 10 minutes.
- 3) When the system is re-initialized (e.g., following a controller restart):
 - a) Set the Tank Cycle Flag based on the tank temperature condition only from the above two clauses without a time delay.
 - b) If neither condition is true, set the mode based on the flow condition only in the above two clauses without a time delay.
- b. Tank thermal energy charge, TankCharge (kBtu), shall be a 5-minute rolling average of the instantaneous tank charge sampled at minimum once every 30 seconds.
- c. Instantaneous tank charge shall be estimated as follows:

$$\text{TankIdealMaxCharge} * \left(\frac{\sum_{i=1}^n \frac{T_i - 42^{\circ}\text{F}}{80^{\circ}\text{F} - 42^{\circ}\text{F}}}{n} - \frac{(0.526 * \text{Thermocline} + \text{Heel})}{\text{NoMin}} \right)$$

1) Where:

- a) TankIdealMaxCharge is the *ideal* maximum capacity of the tank, assuming all water volume including the thermocline and heel heights is usable (kBtu).
- b) n is the number of tank temperature sensors along the vertical axis of the tank.
- c) Thermocline is the estimated thermocline height (5 ft, adj.) (Note: the thermocline is useful on the first tank pass, hence the scale factor).
- d) Heel includes the upper and lower heel heights above and below the diffusers (3 ft, adj.)
- e) NoMin is the normal operating level of the tank (ft.) at minimum temperature
- d. Calculate the % tank charge, PercentTankCharge, as follows:

$$\frac{\text{TankCharge}}{\text{TankIdealMaxCharge} * \left(1 - \frac{(0.526 * \text{Thermocline} + \text{Heel})}{\text{NoMin}} \right)} * 100$$

- e. Calculate current tank charge/discharge rate, TankChargeRate (kBtu/h), as follows:

$$(\text{TankCharge} - \text{TankCharge}_{5\text{minutesprior}}) * 12$$

- f. Calculate current % tank charge/discharge rate, PercentTankChargeRate (%/h), as follows:

$$\frac{\text{TankChargeRate}}{\text{TankIdealMaxCharge} * \left(1 - \frac{(0.526 * \text{Thermocline} + \text{Heel})}{\text{NoMin}} \right)} * 100$$

2. Condenser Loop Mode

- a. Closed condenser water loop shall be in one of three modes: AWHP Charge Assist, Excess Heat Rejection, and Tank Charge/Discharge.
- b. AWHP Charge Assist Mode
 - 1) Enable AWHP Charge Assist Mode whenever PercentTankCharge $< 95\%$ and any of the following is true for 5 minutes continuously:
 - a) $\text{PercentTankCharge}_t < \text{PercentTankCharge}_{t-10} - 5\% * f_1$ (30%/hour default)
 - b) $\text{PercentTankCharge}_t < \text{PercentTankCharge}_{t-30} - 10\% * f_2$ (20%/hour default)
 - c) $\text{PercentTankCharge}_t < \text{PercentTankCharge}_{t-120} - 30\% * f_3$ (15%/hour default)
 - d) $\text{PercentTankCharge}_t < \text{PercentTankCharge}_{t-240} - 40\% * f_4$ (10%/hour default)

- e) $\text{PercentTankCharge}_t < \text{PercentTankCharge}_{t-360} - 50\% * f_5$ (8.3%/hour default)
 - f) $\text{PercentTankCharge} + 8\%/hr * (T\text{toWarmup} - 2) < 95\%$.
 - g) Where:
 - 1. t is the current time in minutes
 - 2. f_x are tuning factors initialized to 1 on first plant startup.
 - 3. $T\text{toWarmup}$ is the time, in hours, from the present time until morning warmup begins (4 AM, default).
 - 2) Disable AWHP Charge Assist Mode whenever either of the following is true:
 - a) None of the AWHP Charge Assist Mode enable conditions is true for 15 minutes and $\text{HeatRecoveryRate} > 0$ for 15 minutes. See 3.2L.17 for HeatRecoveryRate definition.
 - b) $(\text{FM13} - \text{FM14} - \text{FM}_{\text{CCW Makeup}})$'s flowrate > 0 (flow out of the bottom of the tank) for 5 minutes and $T23 > 70^\circ\text{F}$ for 5 minutes.
 - 3) Tuning Factor Calibration
 - a) Each AWHP Charge Assist Enable tuning factor, f_1 to f_5 , shall be calibrated on an ongoing basis. Factors must be stored in non-volatile memory so that they are retained following plant controller shutdown.
 - b) If during a 24-hour period beginning at morning warmup (4 AM default) AWHP Charge Assist Mode is enabled based on a startup condition using a given factor, f_x , and PercentTankCharge does *not* decrease below 30% prior to the same warmup time the following day, increase f_x by 0.02.
 - c) If during a 24-hour period beginning at morning warmup (4 AM default) AWHP Charge Assist Mode is enabled based on a startup condition using a given factor, f_x , and PercentTankCharge decreases below 30% prior to the same warmup time the following day, decrease f_x by 0.02.
 - d) Tuning logic shall only be enabled on weekdays.
 - c. Excess Heat Rejection Mode
 - 1) Enable Excess Heat Rejection Mode whenever all of the following is true.
 - a) AWHP Charge Assist Mode is disabled for 5 minutes
 - b) $(\text{FM13} - \text{FM14} - \text{FM}_{\text{CCW Makeup}}) > 50$ (flow out of the bottom of the tank) for 5 minutes.
 - c) $\text{PercentTankCharge} > 95\%$ for 5 minutes or $T23 > 70^\circ\text{F}$ for 5 minutes.
 - 2) Disable Excess Heat Rejection Mode whenever $\text{FM13} < 0$ for 1 minute (indicating reverse flow through tower heat exchangers) after MV29 has been proven closed for 1 minute and MV30 has been proven open for 1 minute.
 - d. Tank Charge/Discharge Mode
 - 1) Tank Charge/Discharge Mode shall be enabled whenever neither AWHP Charge Assist Mode nor Excess Heat Rejection Mode is enabled.
3. Closed Condenser Water Loop Control Valves
- a. MV6, MV8, MV10, MV12
 - 1) Valve Control Loop Enable/Disable
 - a) For a given valve, control loops defined below shall be enabled in stages for which a given valve is designated "NOTE (1)" in Section 3.2C.8.
 - b) Timing of loop enabling shall correspond to Step 3 in Section 3.2C.9 during staging event sequencing.
 - c) Timing of loop disabling shall correspond to Step 1 in Section 3.2C.9 during staging event sequencing.
 - d) Exceptions:
 - 1. If no valves are designated with "NOTE (1)" in a stage (Stages 0CxH only) in Section 3.2C.8 and AWHP Charge Assist

- Condenser Water Loop Mode is enabled, enable the MV6 control loop. MV6 control loop shall be disabled when AWWP Charge Assist is disabled.
- 2) Each valve shall be controlled by an independent reverse acting control loop acting to maintain associated condenser barrel flow at setpoint.
 - a) Valve to flow meter mappings shall be as follows:
 1. MV6: FM7
 2. MV8: FM8
 3. MV10: FM9
 4. MV12: FM10
 - b) Control loop shall be configured with a setpoint deadband equal to +/- 2% of DesCondFlowX for the associated chiller to minimize valve hunting.
 - c) Whenever MV6's control loop is enabled but both V2 and V4 are commanded open (as might occur in Stage 0CxH or under low load cycling control per 3.2D.6.b), 500 GPM (adj.) shall be used for DesCondFlowX.
 - 3) Valve flow setpoint shall vary based on Tank Cycle Flag status and Condenser Loop Mode. Flow setpoint loops shall be enabled and disabled concurrently with associated Valve Control Loops.
 - a) When Condenser Loop Mode is AWWP Charge Assist, a direct acting control loop shall maintain T19 at target setpoints of 80°F and 60°F when the Tank Cycle Flag status is EightySixty and SixtyFortyTwo respectively.
 1. Loop output from 0% to 100% shall reset each chiller's condenser water flowrate setpoint from $0.10 * \text{DesCondFlowX}$ to DesCondFlowX.
 2. Bias loop output to 20% so it launches from roughly 30% of DesCondFlowX.
 3. Loop shall be common to all enabled valves (MV6, MV8, MV10, MV12)
 4. Provide ramp functions to prevent changes in target T19 setpoint of more than 1°F every 90 seconds to avoid large changes in chiller condenser flow rate when Tank Cycle Flag status changes.
 - b) When Condenser Loop Mode is Excess Heat Rejection, a reverse acting loop shall maintain the associated chiller's head pressure at minimum. This applies to CH-1, CH-2. And HRC-3 only.
 1. Loop output from 0% to 100% shall reset the associated chiller's condenser water flowrate setpoint from DesCondFlowX to $0.10 * \text{DesCondFlowX}$.
 2. Bias loop output to 50% so it launches from roughly 50% of DesCondFlowX.
 3. Each isolation valve designated with "NOTE (1)" for the stage in Section 3.2C.8 shall have its own loop.
 4. "Head pressure" signal shall be read through the chiller's network interface. Coordinate with chiller submittals.
 - c) When Condenser Loop Mode is Tank Charge/Discharge, a direct acting control loop shall maintain the associated chiller's CWRT sensor (T12, T13, T14, T15 respectively) at target setpoints of 80°F and 60°F when the Tank Cycle Flag status is EightySixty and SixtyFortyTwo respectively.
 1. Loop output from 0% to 100% shall reset the associated chiller's condenser flowrate setpoint from $0.10 * \text{DesCondFlowX}$ to DesCondFlowX.

2. Bias loop output to 50% so it launches from roughly 50% of DesCondFlowX.
3. Each isolation valve designated with "NOTE (1)" for the stage in Section 3.2C.8 shall have its own loop.
4. Provide ramp functions to prevent changes in target chiller CWRT setpoints of more than 1°F every 90 seconds to avoid large changes in chiller condenser flow rate when Tank Cycle Flag status changes.
- d) In any Condenser Loop Mode, a reverse acting loop shall maintain the associated chiller's head pressure at minimum. This applies to HRC-1 and HRC-2 only.
 1. The valve flow setpoint determined by Sections 3.2F.3.a.3)a) or 3.2F.3.a.3)c) shall not exceed the maximum condenser water flowrate setpoint determined by this loop.
 2. Loop output from 0% to 100% shall reset the associated chiller's maximum condenser water flowrate setpoint from DesCondFlowX to $0.10 * \text{DesCondFlowX}$.
 3. Bias loop output to 50% so it launches from roughly 50% of DesCWFlowX.
 4. Each isolation valve designated with "NOTE (1)" for the stage in Section 3.2C.8 shall have its own loop.
 5. "Head pressure" signal shall be read through the chiller's network interface. Coordinate with chiller submittals.

The 30XW chillers have minimum lift requirements that may require leaving condenser water temperatures that exceed the target setpoint especially when the Tank Cycle Flag status is SixtyFortyTwo. Always enabling head pressure control when modulating the condenser isolation valve prevents low lift issues.

- e) MV6's flow setpoint shall be no less than $0.15 * \text{DesCondFlowX}$ for 15 minutes (adj.) after MV6's control loop is enabled.
- f) MV12's flow setpoint shall be no less than $0.40 * \text{DesCondFlowX}$ for 10 minutes (adj.) after MV12's control loop is enabled.

On startup, the 19DV chillers experience large fluctuations in leaving condenser water temperature when condenser flow is low. These fluctuations can cause a bearing pressure fault trip. The 19DV chiller condenser isolation valves therefore control to a sufficiently high minimum condenser flow rate on startup to prevent that fault from occurring without exceeding the design evaporator flow rate of one operating HRC so as to accidentally send potentially cold leaving condenser water to the HRC evaporators.

- 4) Condenser Isolation Valve Requests
 - a) If the measured flow is less than 80% of setpoint while setpoint is greater than zero and the valve command is greater than 95% for 1 minute, send 3 requests.
 - b) Else, if the measured flow is less than 90% of setpoint while setpoint is greater than zero and the valve command is greater than 95% for 1 minute, send 2 requests.
 - c) Else, if the valve command is greater than 95%, send 1 request until the valve command is less than 85%.
 - d) Else, if the valve command is less than 95%, send 0 requests.
- b. MV7, MV9, MV11
 - 1) Valve Control Loop Enable/Disable
 - a) Control loops defined below shall be enabled in stages for which a given valve is designated "NOTE (3)" in Section 3.2C.8.
 - b) Timing of loop enabling shall correspond to Step 3 in Section 3.2C.9 during staging event sequencing.

- c) Timing of loop disabling shall correspond to Step 1 in Section 3.2C.9 during staging event sequencing.
- 2) Each evaporator isolation valve shall be controlled by an independent reverse acting control loop acting to maintain associated evaporator flow (FM3, FM4, FM5 respectively) at setpoint. Control loop shall be configured with a setpoint deadband equal to +/-2% of chiller DesEvapFlowX to minimize valve hunting.
- 3) Each HRC's evaporator supply temperature as measured by T7, T8, or T9 shall be maintained at setpoint by a direct acting loop that resets evaporator flow setpoint and *requested* MV27 and MV28 position.
 - a) Evaporator supply temperature setpoint shall be:
 - 1. TWST Setpoint + HxApp, limited to a minimum of 42°F and a maximum of 60°F, when Excess Heat Rejection Mode is enabled irrespective of Tank Cycle Flag.
 - 2. Otherwise, setpoint shall be 60°F when the Tank Cycle Flag is EightySixty and 42°F when the Tank Cycle Flag is SixtyFortyTwo.
 - b) Provide ramp functions to prevent changes in HRC evaporator supply temperature setpoints of more than 1°F per minute to allow adequate time for chiller internal controls to respond.
 - c) Each evaporator supply temperature control loop shall be enabled when the associated chiller is proven ON and the associated MV7, MV9, or MV11 control loop is enabled. Otherwise, the loop shall be disabled and output set to 25%.
 - d) Loop shall be biased to launch from 25%.
 - e) Loop output shall be mapped as follows:
 - 1. From 0% to 50% loop output, reset evaporator flow setpoint from DesEvapFlowX to MinEvapFlowX.
 - 2. From 50% to 100% loop output, concurrently reset *requested* MV27 position from 0% open to 100% open and *requested* MV28 position from 100% open to 0% open.
- 4) See Section 3.2F.3.c for control of MV27 and MV28.
- 5) Evaporator Isolation Valve Requests
 - a) If the measured flow is less than 80% of setpoint while setpoint is greater than zero and the valve command is greater than 95% for 1 minute, send 3 requests.
 - b) Else, if the measured flow is less than 90% of setpoint while setpoint is greater than zero and the valve command is greater than 95% for 1 minute, send 2 requests.
 - c) Else, if the valve command is greater than 95%, send 1 request until the valve command is less than 85%.
 - d) Else, if the valve command is less than 95%, send 0 requests.
- c. MV27 and MV28
 - 1) MV27 and MV28 positions shall be dictated by two different loop types: evaporator supply temperature control loops (see 3.2F.3.b.3)) and a loop maintaining chiller evaporator entering water temperatures below 80°F.
 - 2) Evaporator entering water temperature as measured by T11 shall be maintained at less than 80°F by a direct acting control loop whose output is mapped to sequence *requested* MV27 and MV28 positions.
 - a) Loop shall be enabled whenever any evaporator supply temperature control loop as defined in 3.2F.3.b.3) is enabled. Loop shall be disabled with output set to 0% otherwise.
 - b) Loop output shall be mapped as follows: from 0% to 100% output concurrently reset *requested* MV27 position from 0% to 100% open and *requested* MV28 position from 100% to 0% open.

- 3) MV27 shall be controlled to the high *requested* position of the active evaporator supply temperature control loops and the evaporator entering temperature control loop.
- 4) MV28 shall be controlled to the low *requested* position of the active evaporator supply temperature control loops and the evaporator entering temperature control loop.
- d. MV29 and MV30
 - 1) Whenever Excess Heat Rejection Mode is enabled for 10 minutes, MV29 shall be commanded shut and MV30 commanded open.
 - 2) Otherwise, a direct acting control loop shall modulate MV29 and MV30 to prevent tank flowrate, as measured by (FM13 – FM14 – FM_{CCW Makeup}) (flow out of the lower port is positive), from exceeding TankMaxFlow. From 0% to 100% loop output, concurrently reset MV30 from 0% open to 100% open and MV29 from 100% open to 0% open.
 - 3) Provide ramp functions to prevent changes in MV29 and MV30 commands of more than 5% per minute to avoid large changes in chiller entering condenser water temperature when Excess Heat Rejection Mode is enabled or disabled.
- e. V31 and V32
 - 1) V31 shall be open at all times by default for condenser water loop, chilled water loop, and hot water loop expansion/contraction. Operators shall be able to command valve shut from graphics.
 - a) When V31 is commanded closed by operators:
 1. Simultaneously close MV29 and V31 and open MV30 and V32. MV29 and MV30 command shall override control logic in Section 3.2F.3.d above.
 2. Display a large red banner on all plant graphics indicating “TES tank is out of service. Plant is in hand with limited heating availability.”
 3. Generate a Level 2 TES Tank Out of Service Alarm.
 4. Override Condenser Loop Mode to AWHF Charge Assist. (Note: this could potentially be wasteful—i.e., Excess Heat Rejection Mode may be preferable—but keeping sequences simple during this failsafe condition or maintenance condition is the more critical consideration.
 5. Limit available plant stages to 0C0H through 5C0H and 0C1H through 4C1H and trigger heating and cooling stage down events to the highest available cooling and heating stages immediately if required.
 - b) When V31 is commanded open by operators:
 1. Simultaneously open V31 and close V32 while releasing MV29 and MV30 to automatic control.
 2. Clear banners on all plant graphics.
 3. Release the override on Condenser Loop Mode.
 4. Allow normal staging to resume without limits on available stages.
 - 2) Configure V31 and V32 if possible to prevent changes in valve position of more than 10% per minute to avoid large changes in chiller entering condenser water temperature when the TES tank is put in and out of service.
- f. V33
 - 1) Open V33 when either of the following is true.
 - a) TES tank water level is 2” below minimum (lowest temperature) normal operating level, NoMin, and V31 is commanded open.
 - b) V31 is commanded closed

- 2) Close V33 when TES tank water level is above NolMin and V31 is commanded open.
- g. MV37 and MV38
 - 1) Whenever AWHP Charge Assist Mode is enabled and either AWHP is proven on, a direct acting control loop shall modulate to maintain T18 at current AWHP setpoint – 2°F. Loop shall be disabled and output set to 0% otherwise.
 - a) Loop output from 0% to 100% shall index the valves from full recirculation to fully open to the main loop.
 - b) Loop output shall be mapped as follows: from 0% to 100% output concurrently reset MV37 position from 100% to 0% open and MV38 position from 0% to 100% open.
 - c) Bias loop output to 50% so it launches from 50% recirculation.
 - d) Note: this control loop indirectly acts to maintain minimum AWHP entering water temperature and forces all enabled AWHPs to full load by preventing them from making setpoint.
4. Air-to-Water Heat Pumps
 - a. Enable both AWHPs whenever AWHP Charge Assist Mode is enabled and either CWP-2A or CWP-2B is proven on. AWHPs shall be disabled otherwise.
 - b. When Tank Cycle Flag is EightySixty, AWHP setpoint shall be 86°F.
 - c. When Tank Cycle Flag is SixtyFortyTwo, AWHP setpoint shall be 77°F (minimum allowed by manufacturer).
5. Closed Condenser Water Evaporator Loop Pumps (CWP-1A, -1B)
 - a. CWP-1 pumps shall be lead/lag controlled per Section 3.1P.
 - b. Lead Pump Enable/Disable
 - 1) Enable lead CWP-1 pump whenever the stage requires that an operating heat recovery chiller have its evaporator indexed to the condenser water loop.
 - 2) Disable the lead CWP-1 pump otherwise.
 - 3) Enable the pump when the stage change is initiated; disable only after the stage change is complete.
 - c. CWP-1 pumps shall be staged as a function of CWP1FR, the ratio of total evaporator barrel flow of chillers with evaporators indexed to the condenser water loop, FlowEvap, as measured by the evaporator flow meters (FM3, FM4, FM5) to design CWP1 flow, CWP1Fdesign.
 - d. Enable the lag pump whenever either:
 - 1) CWP1FR > 47% for 10 minutes.
 - 2) Pump speed command > 99% for 5 minutes.
 - e. Disable the lag pump whenever any of the following is true:
 - 1) Pumps staged up on CWP1FR condition and CWP1FR < 47% for 10 minutes.
 - 2) Pumps staged up on speed condition and pump speed command < 30% for 5 minutes.
 - 3) The lead pump is disabled.
 - f. When any pump is proven on, pump speed shall be controlled by a reverse acting control loop acting to maintain condenser loop DP, DP1, at setpoint.
 - g. DP1 setpoint shall be reset using Trim & Respond logic with the following parameters:

Variable	Value
Device	Any CWP1
SP0	DP1Max
Spmin	2 psi
Spmax	DP1Max

Td	5 minutes
T	5 minutes
I	0
R	Evaporator Isolation Valve Requests (See 3.2F.3.b.5))
Sptrim	-0.5 psi
Spres	+1.0 psi
Spres-max	+2.0 psi

- 1) DP1 setpoint reset loop shall be enabled whenever any CWP1 is enabled and disabled otherwise.
 - 2) When a plant stage change is initiated, reset logic shall be disabled and value fixed at its last value for the longer of 5 minutes and the time it takes for the plant to successfully stage.
6. Closed Condenser Water Condenser Loop Pumps (CWP-2A, -2B)
- a. CWP-2 pumps shall be lead/lag controlled per Section 3.1P.
 - b. Lead Pump Enable/Disable
 - 1) Enable lead CWP-2 pump whenever either:
 - a) The stage requires that an operating chiller have its condenser indexed to the condenser water loop, or
 - b) The MV6 control loop is enabled (see Section 3.2F.3.a.1d)), or
 - c) The waterside economizer valve V25 is commanded open.
 - 2) Disable the lead CWP-2 pump otherwise.
 - 3) When enabling based on plant stage, enable the pump when the stage change is initiated. When disabling based on plant stage, enable only after the stage change is complete.
 - c. CWP-2 pumps shall be staged as a function of CWP2FR, the ratio of total condenser barrel flow of chillers with condensers indexed to the condenser water loop, FlowCond, as measured by the loop flow meter FM11 to design CWP2 flow, CWP2Fdesign.
 - d. Enable the lag pump whenever either:
 - 1) CWP2FR > 47% for 10 minutes.
 - 2) Pump speed command > 99% for 5 minutes.
 - e. Disable the lag pump whenever any of the following is true:
 - 1) Pumps staged up on CWP2FR condition and CWP2FR < 47% for 10 minutes.
 - 2) Pumps staged up on speed condition and pump speed command < 30% for 5 minutes.
 - 3) The lead pump is disabled.
 - f. When any pump is proven on, pump speed shall be controlled by a reverse acting control loop acting to maintain condenser loop DP, DP2, at setpoint.
 - g. DP2 setpoint shall be reset using Trim & Respond logic with the following parameters:

Variable	Value
Device	Any CWP2
SP0	DP2Max
Spmin	2 psi
Spmax	DP2Max
Td	5 minutes
T	5 minutes
I	0
R	Condenser Isolation Valve

	Requests (See 3.2F.3.a.4))
Sptrim	-0.5 psi
Spres	+1.0 psi
Spres-max	+2.0 psi

- 1) DP2 setpoint reset loop shall be enabled whenever any CWP2 is enabled and disabled otherwise.
- 2) When a plant stage change is initiated, reset logic shall be disabled and value fixed at its last value for the longer of 5 minutes and the time it takes for the plant to successfully stage.

G. Tower Water Loop

1. Tower Water Supply Temperature (TWST) Setpoint
 - a. Setpoint shall be reset as a function of Tank Cycle Flag, Condenser Loop Mode, and Waterside Economizer Status.
 - 1) When Excess Heat Rejection Mode is enabled and the Waterside Economizer is commanded off:
 - a) Target chiller lift, LIFT, shall be reset from LiftMin when Qrequired_cool is 10% of QdesignCool to LiftDesign when Qrequired_cool is 100% of QdesignCool.
 - b) Chiller condenser delta-T, CwDT, shall be calculated as the 5-minute rolling average of T16 – T20 where samples are taken at minimum once every 30 seconds.
 - c) TWST setpoint shall equal current CHWST setpoint + LIFT – CwDT – HxApp. Setpoint shall be limited to a maximum of 77°F.
 - 2) When Excess Heat Rejection Mode is enabled and the Waterside Economizer is commanded on:
 - a) Setpoint shall be CHWST setpoint – HxApp – 1°F.
 - 3) In any other Mode with a Tank Cycle Flag of EightySixty:
 - a) Setpoint shall equal 60°F – HxApp.
 - 4) In any other Mode with a Tank Cycle Flag of SixtyFortyTwo:
 - a) Setpoint shall equal 42°F – HxApp.
 - b. Note that setpoints will not be achievable under certain wet bulb conditions. Subsequent sequences institute limits on tower fan and TWP speeds.
2. Tower Water Pumps
 - a. TWP pumps shall be lead/lag controlled per Section 3.1P.
 - b. Enable lead TWP pump whenever either:
 - 1) MV30 command > 0% open for 30 seconds and FM14 flow > 50 GPM for 30 seconds.
 - 2) Excess Heat Rejection Mode is enabled and the chilled water loop is enabled.

When changing the source of chiller condenser water from the TES tank to the tower water loop, the 19DV chillers can experience a large sudden change in entering condenser water temperature. Large enough fluctuations in entering condenser water temperature can cause bearing pressure fault trips in the 19DV chillers. Enabling the tower water loop and allowing 10 minutes for the TWST to stabilize before the TES tank bypass valves start changing the source of chiller condenser water lessens entering condenser water temperature fluctuations and mitigates the risk of bearing pressure fault trips.

- c. Disable lead TWP pump whenever either:
 - 1) MV30 command = 0% open for 1 minute.
 - 2) FM14 flow < 50 GPM for 5 minutes.
- d. TWP pumps shall be staged based on speed:
 - 1) Enable the lag pump whenever:

- a) Pump speed command > 90% for 5 minutes.
 - 2) Disable the lag pump whenever either of the following is true:
 - a) Pump speed command < 40% for 5 minutes.
 - b) The lead pump is disabled.
 - e. Pump speed command shall be the lower of that output by two loops, each of which shall be enabled whenever any TWP is proven on. Loops shall be disabled otherwise.
 - 1) A direct acting control loop shall maintain closed condenser water HX leaving temperature, T24, at setpoint. Setpoint shall equal TWST Setpoint (see 3.2G.1) + HxApp. Reset requested speed from MinTWPSpeedX at 0% loop output to 100% at 100% loop output.
 - 2) A direct acting loop shall maintain tower loop HX DT (T27 – T26) at a setpoint equal to closed condenser water loop HX DT (T19 – T24) – 2°F. Reset requested speed from MinTWPSpeedX at 0% loop output to 100% at 100% loop output. *(This loop prevents speed runaway when target setpoint cannot be achieved.)*
 - f. MinTWPSpeedX shall be the TAB determined pump speed needed to maintain minimum tower flow per 1.3B.4.a, where X corresponds to the pump stage (1 or 2), since different speeds will be required for each stage.
 - 1) MinTWPSpeedX shall be set to the greater of 75% (adj.) and its TAB determined value for 10 minutes when the loop is enabled under condition 3.2G.2.b.2) above.
 - g. Provide ramp functions to prevent changes in TWP speed of more than 10% per minute to avoid large changes in chiller entering condenser water temperature.
- 3. Cooling Tower Fans
 - a. Cooling tower fans shall be enabled whenever both of the following are true:
 - 1) Either TWP is proven on for 1 minute.
 - 2) TWST (T26) > TWST setpoint + 1°F for 1 minute.
 - b. Cooling tower fans shall be disabled whenever either of the following is true:
 - 1) Both TWPs are proven off.
 - 2) For 1 minute, tower fan speed loop output is 0% and TWST (T26) < TWST setpoint – 1°F.
 - c. When tower fans are commanded on, a direct acting control loop maintaining T26 at TWST setpoint shall reset tower fan speed from minimum to CtMaxSpeed in proportion to loop output. Loop shall be disabled when fans are commanded off.
 - d. CtMaxSpeed shall reset from 70% when Qrequired_cool = 0 to 100% when Qrequired_cool = 50% of QdesignCool.
- 4. Tower Loop Makeup Water
 - a. Make-up water valves, MV22 or MV50, shall cycle based on the two tower water fill level sensors.
 - 1) Convert the basin level pressure readings from psi to in. H2O (1 psi = 27.71 inches of water) and subtract the elevation difference from bottom of basin to the sensor level so that the readings correspond to basin elevation.
 - 2) Note: modulating duty is only provided as a water hammer protection mechanism so valve actuation can be slowed if required.
 - b. MV22 shall open fully when the city reclaimed water pressure exceeds Prcw-min plus 5 psig (adj.) for 30 minutes (adj.) and either the low signal of both tower water level sensors falls below T-level-min-fill or when one fill level signal is above T-level-max-fill and one is below T-level-min fill. MV22 shall close fully when the city reclaimed water pressure drops below Prcw-min (adj.) for 1 minute (adj.) or the high signal of both tower water level sensors goes above T-level-max-fill.
 - c. MV50 shall open fully when the city reclaimed water pressure drops below Prcw-min (adj.) for 1 minute (adj.) and either the low signal of both tower water level sensors falls below T-level-min-fill or when one fill level signal is above T-level-

max-fill and one is below T-level-min fill. MV50 shall close fully when the city reclaimed water pressure exceeds Prcw-min plus 5 psig (adj.) for 30 minutes (adj.) or the high signal of both tower water level sensors goes above T-level-max-fill.

5. Sidestream Filtration System
 - a. The cooling tower filtration pump shall run from 10 PM to 1 AM (adjustable) daily provided that either TWP-1A or TWP-1B has proven on during the same day.
 - b. While the pump is commanded on, a single tower basin sweep valve, V34 through V36, shall be commanded open at any time while the others remain closed. Rotate the open isolation valve sequentially every 60 minutes.
 - c. When the pump is ready to shut off after all cells have been filtered, open the purge valve, V41, for 25 seconds, then stop pump.

H. Water Treatment Override

1. Chilled Water Loop
 - a. Every night at 10:00pm:
 - 1) Command all air handler HW valves fully closed,
 - 2) Switch all air handler 2-position changeover valves from the HW loop to the CHW loop.
 - 3) Command all air handler CHW valves fully open after the longer of the nominal timing of the changeover valves and 5 minutes after the HW valves are commanded closed,
 - 4) Command all other building (and garage distribution) CHW valves fully open except the cleanroom 2-pipe fan coil units, and
 - 5) A reverse acting control loop maintaining FM1 at 1325 GPM (3 ft/s through 12" chilled water loop piping with 25% distribution variance factor) shall reset CHW pump speed command from minimum to 100% in proportion to loop output. Provide ramp functions to prevent changes in CHW pump speed of more than 5% per minute.
 - b. 55 minutes (adj.) after the air handler CHW valves are commanded open, release all air handler modulating and changeover valves, all building (and garage distribution) CHW valves, and the CHW pump speed command to normal control.
2. Hot Water Loop
 - a. Every night at 12:00am if HRC-1 or HRC-2 has no faults active and the plant stage is not 4C0H or 5C0H:
 - 1) If the plant stage is 0C0H, 1C0H, 2C0H, or 3C0H, command either HRC-1 or HRC-2's condenser isolation valve fully open depending on which HRC has fewer operating hours and no faults active,
 - 2) Command all air handler CHW valves fully closed,
 - 3) Switch all air handler 2-position changeover valves from the CHW loop to the HW loop.
 - 4) Command all air handler HW valves fully open after the longer of the nominal timing of the changeover valves and 5 minutes after the CHW valves are commanded closed,
 - 5) Command all other building (and garage distribution) HW valves fully open,
 - 6) Enable lead HW pump, and
 - 7) A reverse acting control loop maintaining FM6 at 585 GPM (3 ft/s through 8" hot water loop piping with 25% distribution variance factor) shall reset HW pump speed command from minimum to 100% in proportion to loop output. Provide ramp functions to prevent changes in HW pump speed of more than 5% per minute.
 - b. 55 minutes (adj.) after the air handler HW valves are commanded open, release MV8, MV10, all air handler modulating and changeover valves, all building (and

garage distribution) HW valves, lead HW pump enable, and the HW pump speed command to normal control.

3. Tower Water Loop
 - a. Every night at 2:00 am, if all tower water pumps are off and the tower water pumps have not accumulated at least 20 minutes of runtime in the last 24 hours, then start the lead tower water pump. After 20 minutes, release the lead tower water pump to normal control.

I. Freeze Protection

1. Low TES Tank Charge
 - a. When PercentTankCharge < 5%, disable all HRCs with evaporators indexed to the condenser water loop and inhibit heating stage up until PercentTankCharge > 15%.
 - b. Re-enable the disabled HRCs when PercentTankCharge > 15%.
2. Low Saturated Suction Temperature
 - a. When refrigerant evaporator temperature as polled through the chiller's network interface every 15 seconds is less than 34°F for 1 minute (adj.) generate a Level 2 chiller alarm, disable the chiller, and inhibit cooling and heating stage-up until 45 minutes (adj.) have elapsed and refrigerant evaporator temperature is greater than 40°F.
 - b. Re-enable the disabled chiller when 45 minutes (adj.) have elapsed and refrigerant evaporator temperature is greater than 40°F.

J. Airside Economizer Disabling

1. Set the logic variable, EconTesLockout, to TRUE whenever PercentTankCharge < 75%.
2. Set EconTesLockout to FALSE whenever any of the following is true:
 - a. PercentTankCharge > 90%; OAT > (OAT when EconTesLockout was set to TRUE + 10°F); and OAT > 63°F.
 - b. PercentTankCharge > 85%; OAT > (OAT when EconTesLockout was set to TRUE + 15°F); and OAT > 63°F.
 - c. PercentTankCharge > 95%.
3. Refer to air handler sequences for use of EconTesLockout to lock out airside economizers.

This sequence intentionally locks out airside economizers for heat recovery when TES tank charge is approaching the level at which the tank will need to cycle through from 80°F/60°F to 60°F /42°F. Note that from an energy perspective, heat recovery is always preferable to economizing in all-electric systems since the cooling can be conceptualized as "free" either way, and using recovered heat (rather than AWHPs) to charge the tank allows for more efficient heating.

K. Alarms

1. Maintenance interval alarm when pump has operated for more than 3000 hours as indicated by the Staging Runtime: Level 4. Reset the Staging Runtime interval counter when alarm is acknowledged.
2. Maintenance interval alarm when chiller has operated for more than 1000 hours as indicated by the Staging Runtime: Level 4. Reset the Staging Runtime interval counter when alarm is acknowledged.
3. Chiller alarm: level 2
4. Emergency off switch: Level 1

5. Tower level
 - a. If tower water level sensor indicates water level below T-level-low-alarm, generate a Level 2 alarm.
 - b. If tower water level sensor indicates water level above T-level-high-alarm, generate a Level 3 alarm.
 - c. If level sensors differ by 2", generate a Level 3 sensor disparity alarm.
6. Pump or tower fan alarm is indicated by the status input being different from the output command for 15 seconds.
 - a. Commanded on, status off: Level 2. Do not evaluate alarm until the equipment has been commanded on for 15 seconds.
 - b. Commanded off, status on: Level 4. Do not evaluate the alarm until the equipment has been commanded off for 60 seconds.
7. Valve alarm is indicated by the end switch status being different from the output command for 90 seconds.
 - a. Commanded open, status not open: Level 2. Do not evaluate alarm until the equipment has been commanded open for 90 seconds.
 - b. Commanded closed, status not closed: Level 4. Do not evaluate alarm until the equipment has been commanded closed for 90 seconds.
8. Valve alarm is indicated by the analog position feedback being different from the output command by more than 10% for 90 seconds: Level 2
9. Sensor Failure:
 - a. Sensor shall be deemed outside of its widest possible operating range if any of the following are true:
 - 1) Feedback less than 2 mA from any 4 to 20 mA transducer; or
 - 2) Temperature reading less than 0°F from any temperature sensor.
 - b. Any sensor that goes outside of its widest possible operating range.
 - 1) If the sensor is used for monitoring only: Level 3.
 - 2) If the sensor is used for control: Level 2.
10. Sidestream Filtration System Alarm: Level 2
11. Refrigerant Leak Alarms
 - a. Refrigerant detector indicates evacuate level alarm: Level 1
 - b. Refrigerant detector malfunction or warning level alarm: Level 2
 - c. Refrigerant caution level alarm: Level 3
12. Excessive CW approach indicating water side fouling: If leaving condenser water temperature is more than 4°F below refrigerant condensing temperature for 15 minutes at least 15 minutes after chiller start: Level 4
13. Excessive CHW approach indicating water side fouling: If leaving chilled water temperature is more than 4°F above refrigerant evaporator temperature for 15 minutes at least 15 minutes after chiller start: Level 4
14. High water conductivity (total dissolved solids) from TDS controller: Level 3
15. High evaporator leaving chilled water temperature (more than 5°F above setpoint) for more than 15 minutes: Level 3. Inhibit alarm if either:
 - a. Chiller is not internally controlling CHWST (i.e., a HRC controlling to condenser leaving temperature).
 - b. Chiller has been commanded on for less than 30 minutes.
16. Low HRC condenser leaving hot water temperature (more than 5°F below setpoint) for more than 15 minutes: Level 3. Inhibit alarm if either:
 - a. HRC is not internally controlling HWST (i.e., a HRC controlling to evaporator leaving temperature).

- b. HRC has been commanded on for less than 30 minutes.
- 17. High CHW loop temperature if chilled water supply temperature as measured by T1 > Setpoint + 10°F for 15 minutes: Level 2. Inhibit until loop has been enabled for 30 minutes.
- 18. Low HW loop temperature if hot water supply temperature as measured by T30 < Setpoint – 15°F for 15 minutes: Level 2. Inhibit until loop has been enabled for 30 minutes.
- 19. Low TES Charge
 - a. When PercentTankCharge < 25%: Level 3.
 - b. When PercentTankCharge < 10%: Level 2.
- 20. TES Fill Level
 - a. If TES water level sensor indicates water level 3" below fill valve open level, NoMin: Level 2 indicating potential leak.
 - b. If TES water level sensor indicates water level 2" above maximum (highest temperature) level, NoMax: Level 3 indicating overfill condition.
- 21. AWWP Alarm: Level 2. Alarms mapped through network interface shall indicate alarm cause.
- 22. High heat exchanger open loop DP alarm, Level 3, if heat exchanger pressure drop > 1.5*PcwHxOpenLoopDp for 5 minutes
- 23. Chiller High Cycling Frequency: Level 3, if a chiller has greater than 12 starts within a 12-hour period.

L. Performance Monitoring

- 1. All calculations listed below shall be performed at least once every 30 seconds. Time averaged values shall be recorded at least once every 5 minutes. The averaging period shall equal the trending interval. The most recent time averaged value shall be shown on graphics.
- 2. For each of the below parameters store and display on graphics:
 - a. Current 5-minute rolling average value
 - b. Year to date integrated value (e.g., year to date kWh of a given chiller's kW readings, or ton-hours for Plant Cooling Load, etc.)
 - c. Year to date average value
 - d. Current year peak value (does not apply to runtimes)
 - e. Previous year integrated value
 - f. Previous year average value
 - g. Previous year peak value (does not apply to runtimes)
- 3. Device Runtime. Runtime of all plant chillers, HRCs, AWWPs, pumps, and cooling towers.
- 4. Device Power. Power (kW) of all plant chillers, HRCs, AWWPs, pumps, and cooling towers as read through network interfaces.
- 5. Plant Cooling Power. Calculate Plant Cooling Power as the sum of:
 - a. Cooling Only Chiller kW
 - b. HRC kW when HRCs have their condensers indexed to the CW loop
 - c. CHWP-1A/B kW
 - d. CWP-1A/B kW
 - e. TWP-1A/B kW
 - f. CT-1 to -3 kW
- 6. Plant Heating Power. Calculate Plant Heating Power as the sum of:
 - a. HRC kW when HRCs have their condensers indexed to the HW loop
 - b. AWWP kW

- c. HHWP-1A/B kW
- d. CWP-2A/B kW

7. Plant Cooling Load. Calculate Plant Cooling Load as:

$$Q_{\text{Plant}} = \frac{FM1(T4 - T1)}{24} [\text{tons}]$$

8. Plant Heating Load. Calculate Plant Heating Load as:

$$Q_{\text{Plant}} = \frac{FM6(T30 - T28)}{24} [\text{tons}]$$

9. Individual Chiller Cooling Load. Calculate load for each operating chiller using flowrate through the equipment, $FLOW_D$; chilled water return temperature entering the equipment, $CHWRT_D$; and chilled water supply temperature leaving the equipment, $CHWST_D$. Flow and temperature shall be determined via hardwired temperature sensors, not values read through the chiller(s)' network interface(s).

$$Q_D = \frac{FLOW_D (CHWRT_D - CHWST_D)}{24} [\text{tons}]$$

10. Individual HRC Heating Load. Calculate load for each operating heat recovery chiller using flowrate through the equipment, $FLOW_D$; hot water return temperature entering the equipment, $HWRT_D$; and hot water supply temperature leaving the equipment, $HWST_D$.

$$Q_D = 0.490 * FLOW_D (HWST_D - HWRT_D) [\text{kbtuh}]$$

11. Waterside Economizer Cooling Load. Calculate as Plant Cooling Load minus the sum of Individual Chiller Cooling Loads.

12. Plant Cooling Efficiency. Calculate Plant Cooling Efficiency (COPc) as:

$$COPc = \frac{\text{Plant Cooling Load} [\text{tons}]}{\text{Plant Cooling Power} [\text{kW}]} * 3.517$$

13. Plant Heating Efficiency. Calculate Plant Heating Efficiency (COPh) as:

$$COPh = \frac{\text{Plant Heating Load} [\text{tons}]}{\text{Plant Heating Power} [\text{kW}]} * 3.517$$

14. Plant Total Efficiency. Calculate Plant Total Efficiency (COPh) as:

$$COPc = \frac{\text{Plant Heating Load} * 3.517 + \text{Plant Cooling Load} * 3.517}{\text{Plant Heating Power} + \text{Plant Cooling Power}}$$

15. Flow-Based Tank Charge Rate. Calculate flow-based tank charge/discharge rate, FlowBasedTankChargeRate (kBtu/h), as: $0.5 * (FM13 - FM14 - FM_{CCW \text{ Makeup}}) * (T22 - T23)$. $(FM13 - FM14 - FM_{CCW \text{ Makeup}})$ shall be positive when flow is out of the bottom of the tank and negative when flow is into the bottom of the tank.

16. AWHP Output. Calculate AWHP output, AwhpOutput (kBtu/h), as: $0.5 * FM11 * (T19 - T16)$. AwhpOutput shall be set to 0 whenever both AWHPs are proven off.

17. Heat Recovery Rate. Calculate heat recovery rate, HeatRecoveryRate (kBtu/h), as FlowBasedTankChargeRate – AwhpOutput.

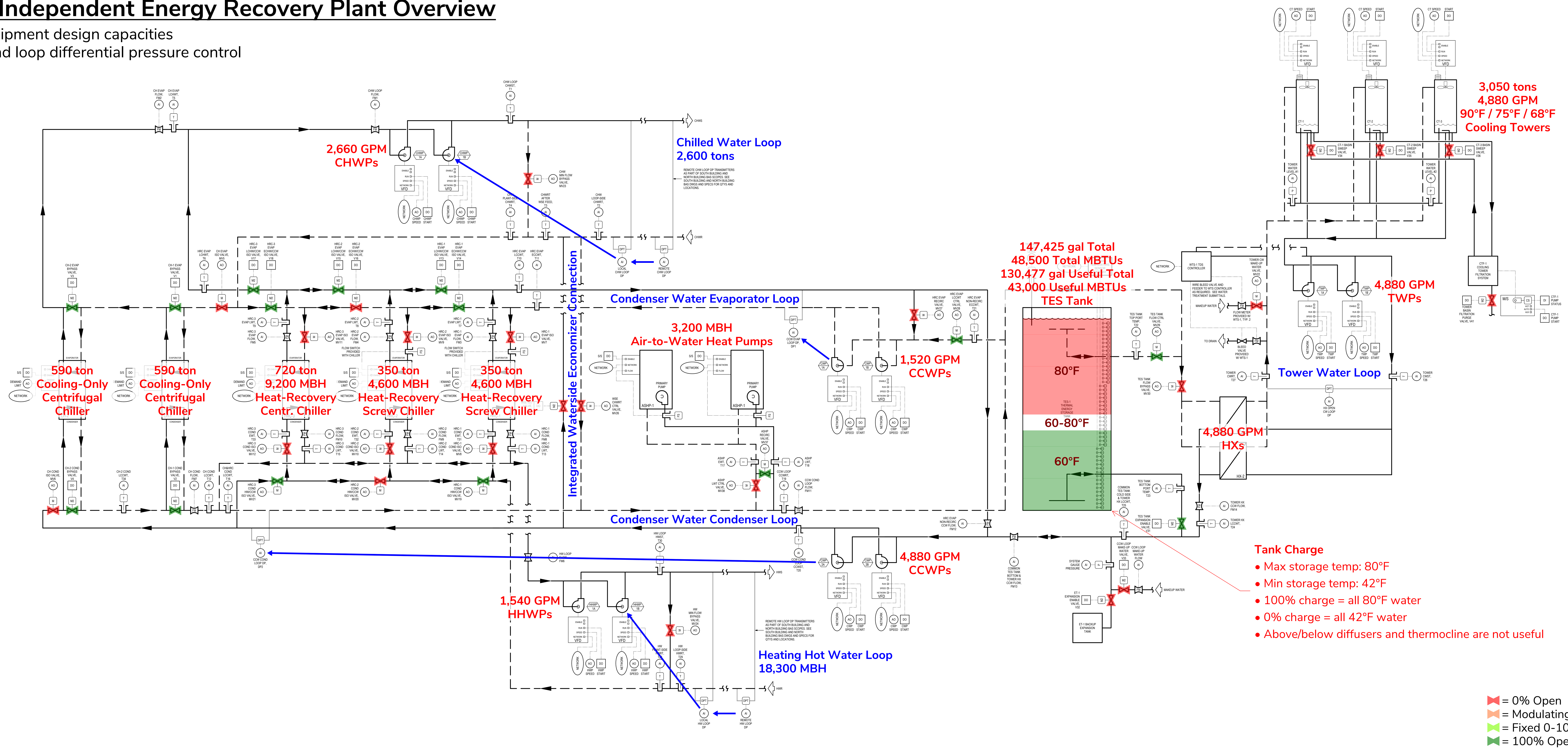
END OF SECTION 259000

Appendix C

Condenser Water TIER Control Schematics

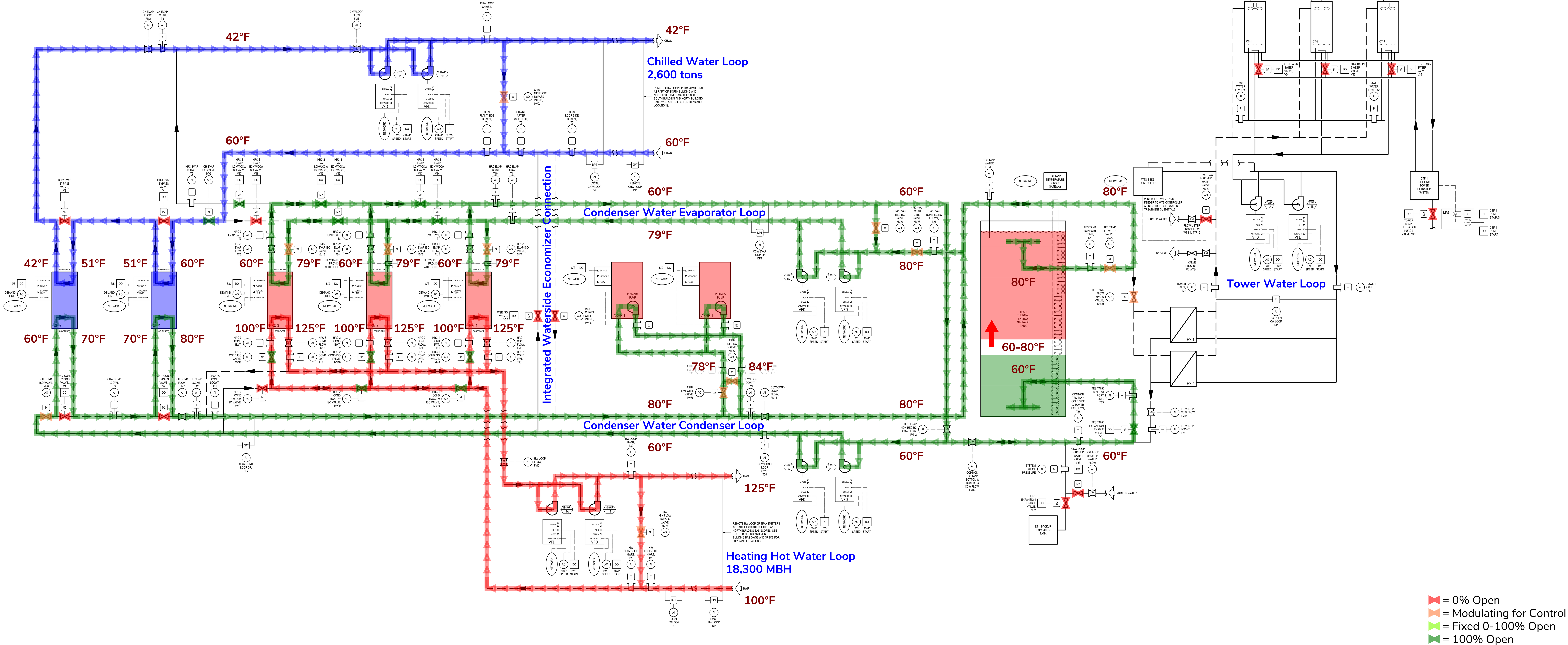
Time Independent Energy Recovery Plant Overview

CUP equipment design capacities
Loops and loop differential pressure control



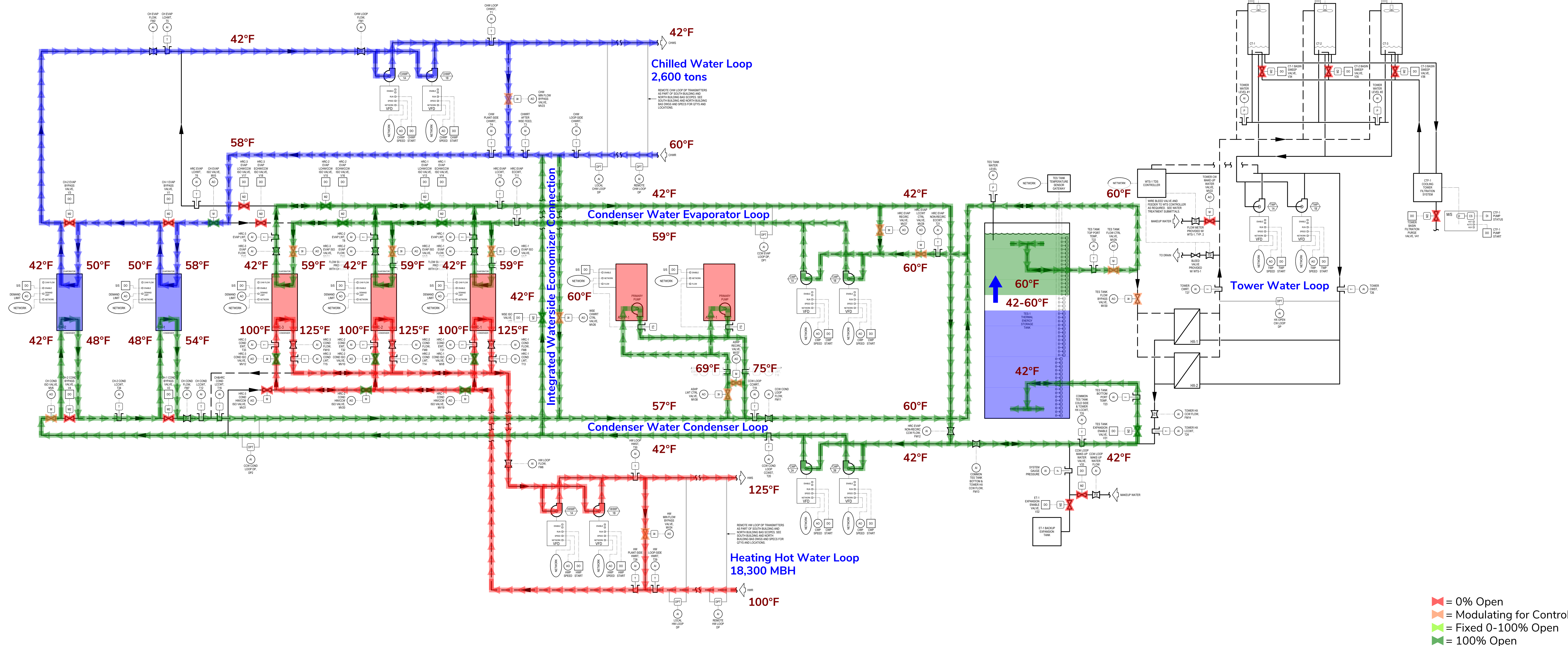
Time Independent Energy Recovery Plant Overview

Typical flow pathways when TES tank split is 80/60



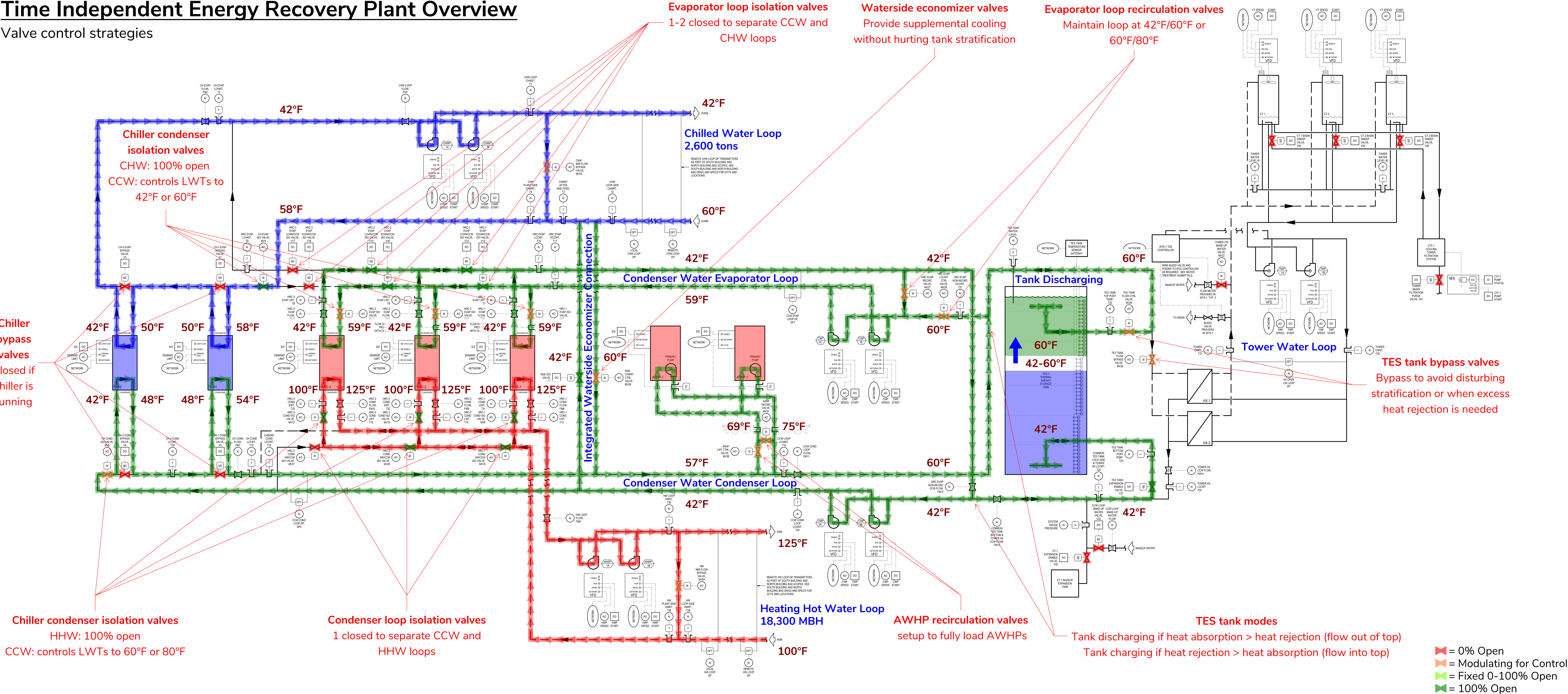
Time Independent Energy Recovery Plant Overview

Typical flow pathways when TES tank split is 60/42



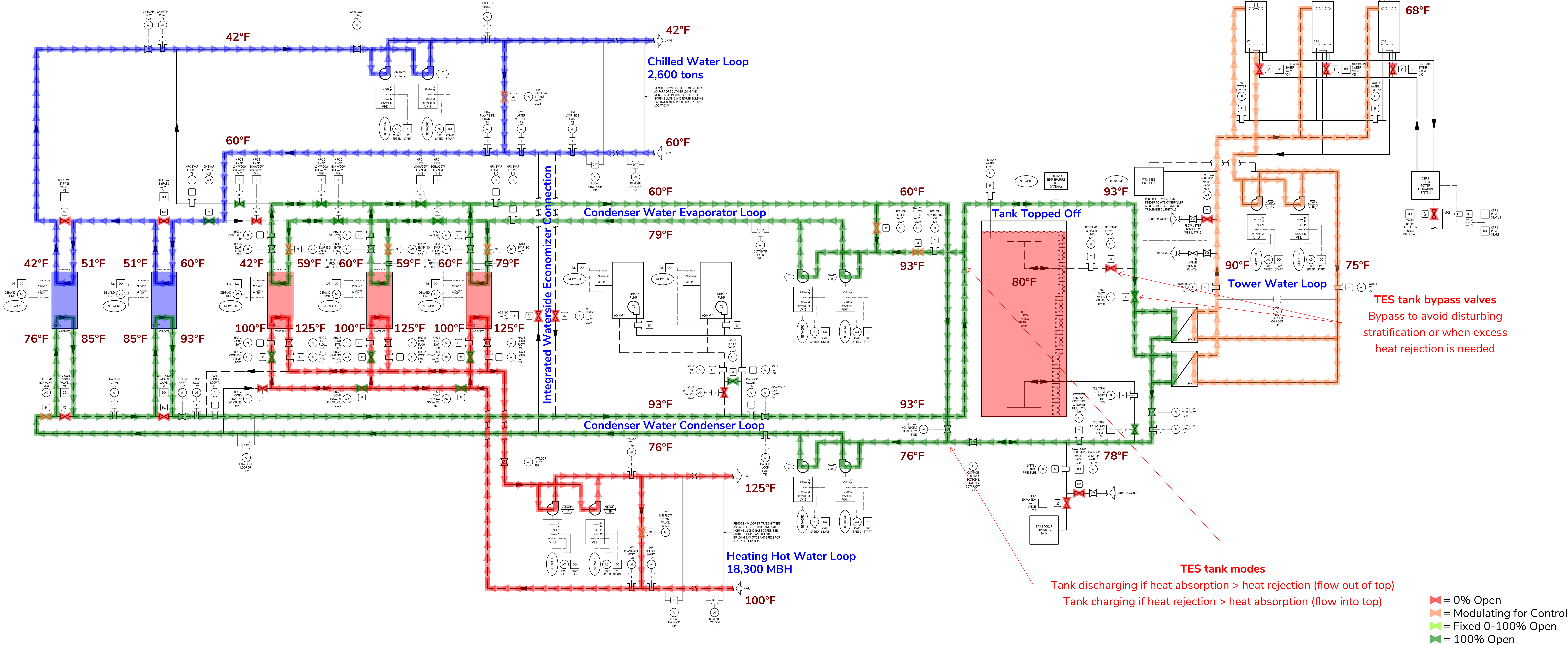
Time Independent Energy Recovery Plant Overview

Valve control strategies



Time Independent Energy Recovery Plant Overview

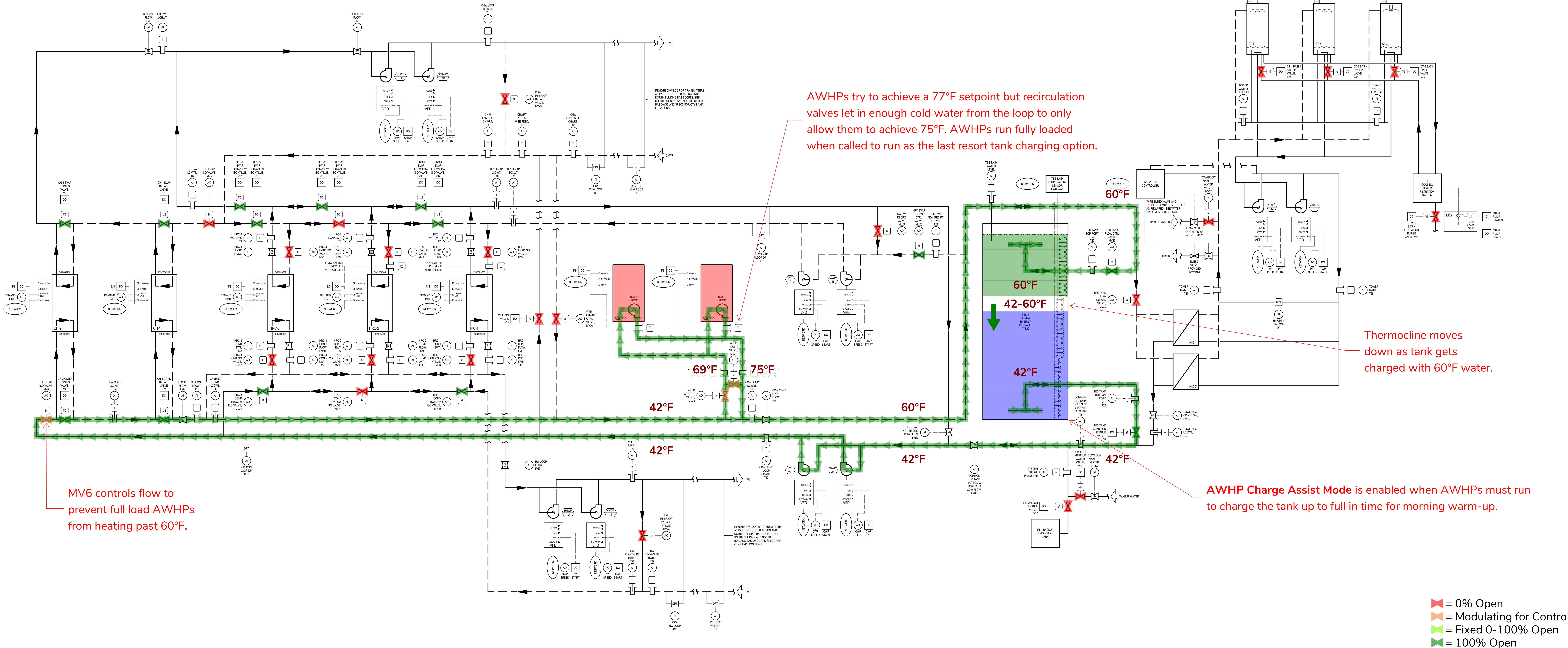
Valve control strategies



No Cooling, No Heating

Mild weekend night with no 24/7 load

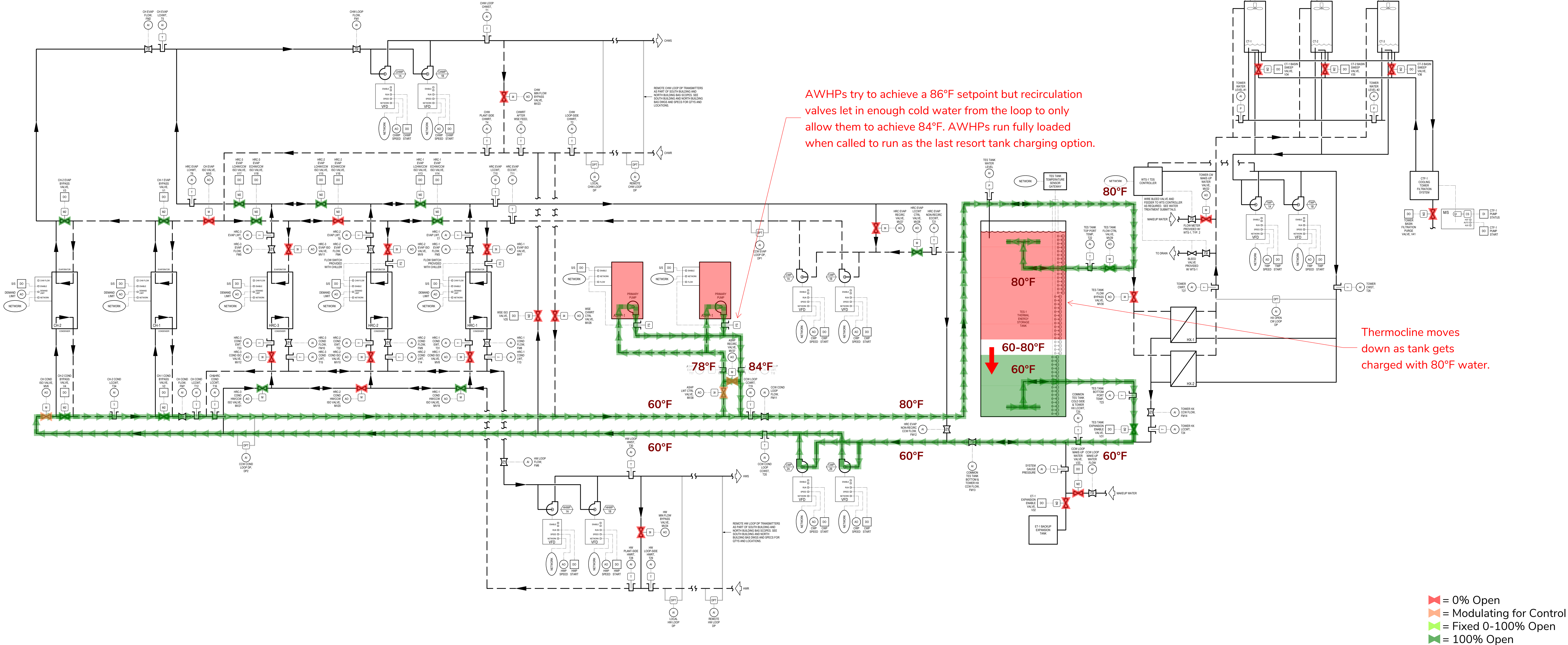
TES tank will not be fully charged with 80°F water by the next morning warm-up without AWHP assistance



No Cooling, No Heating

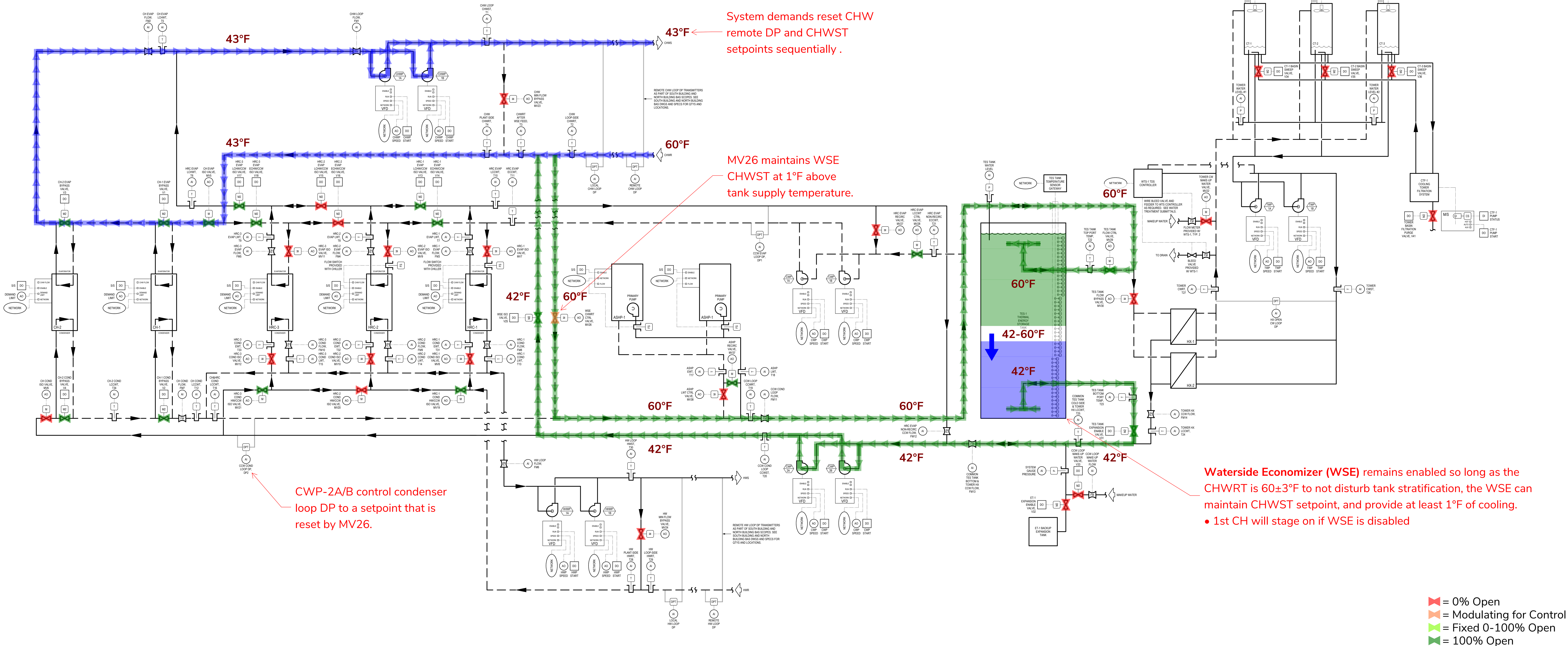
Mild weekend night with no 24/7 load

TES tank will not be fully charged with 80°F water by the next morning warm-up without AWHP assistance

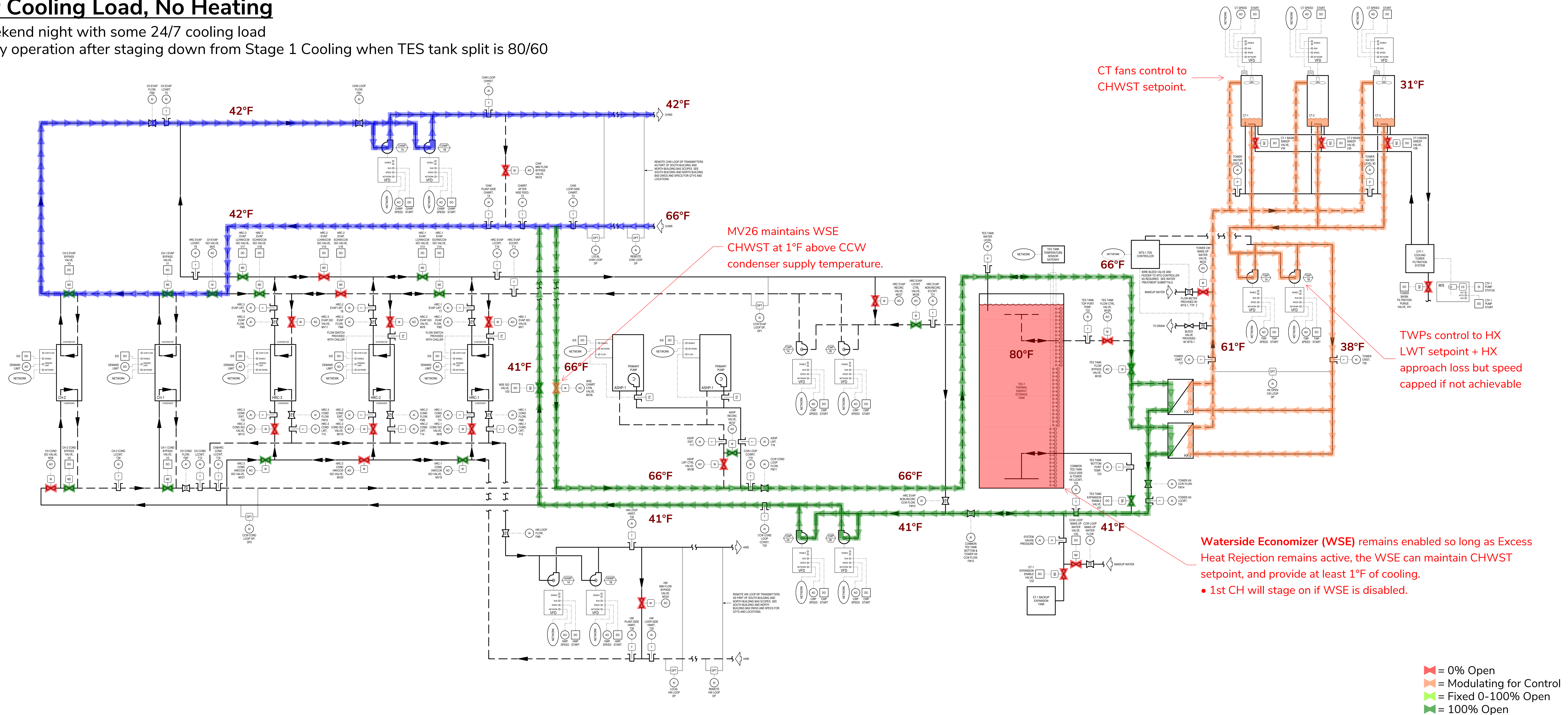


Minor Cooling Load, No Heating

Mild weekend night with some 24/7 cooling load
WSE only operation after staging down from Stage 1 Cooling when TES tank split is 60/42



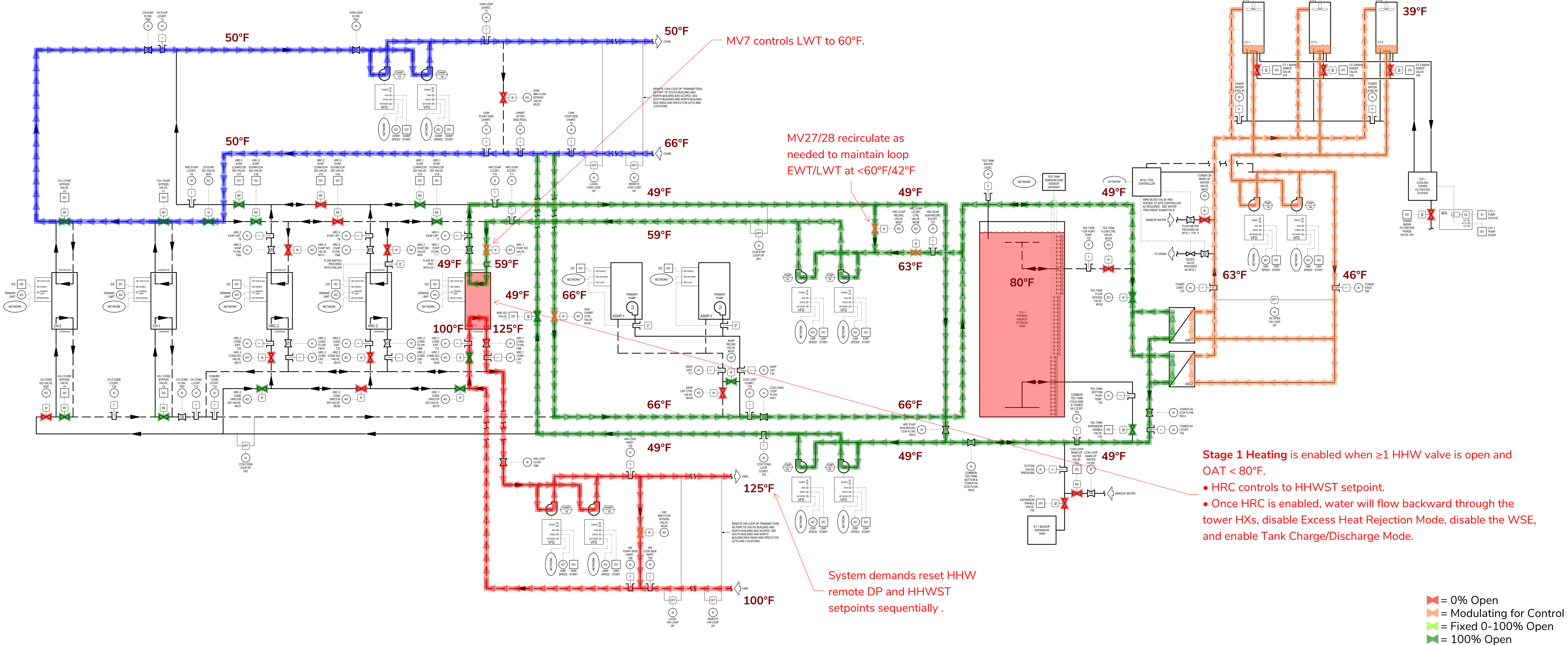
Mild weekend night with some 24/7 cooling load
WSE only operation after staging down from Stage 1 Cooling when TES tank split is 80/60



Minor Cooling Load, Minor Heating Load

Mild weekday morning with some 24/7 cooling load

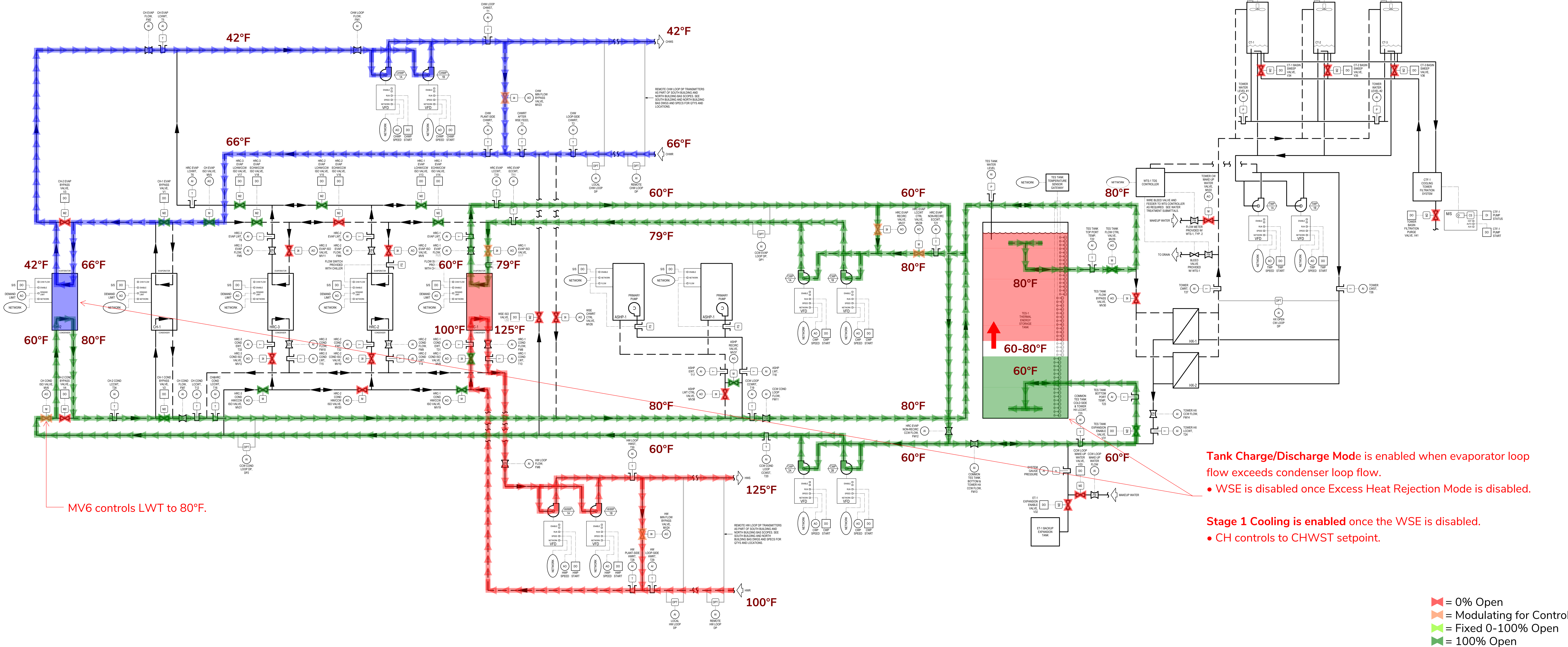
Start of Stage 1 Heating with WSE operation (**0C1H**) when Excess Heat Rejection Mode is still active



Minor Cooling Load, Minor Heating Load

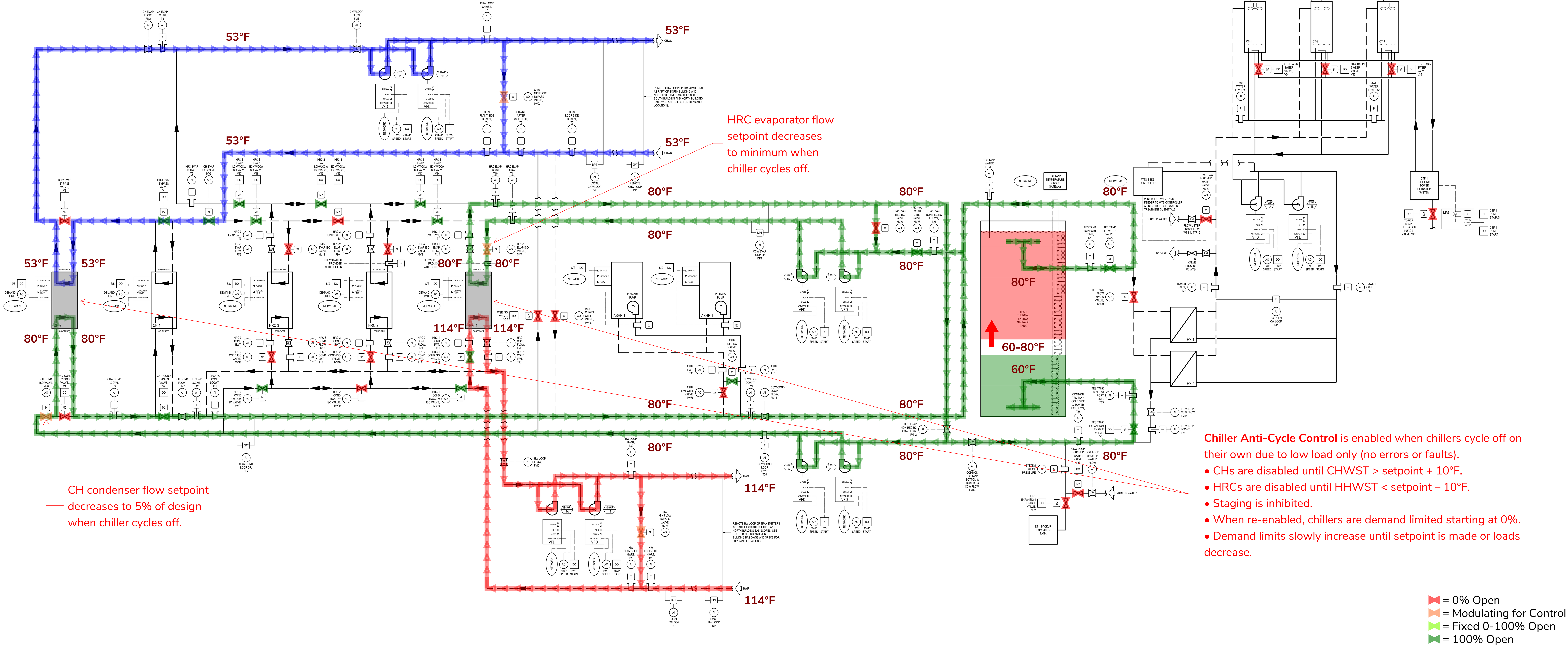
Mild weekday morning with some cooling load

Stage 1 Cooling with Stage 1 Heating (1C1H) after Excess Heat Rejection Mode is disabled, WSE is disabled, and Tank Discharge Mode is enabled



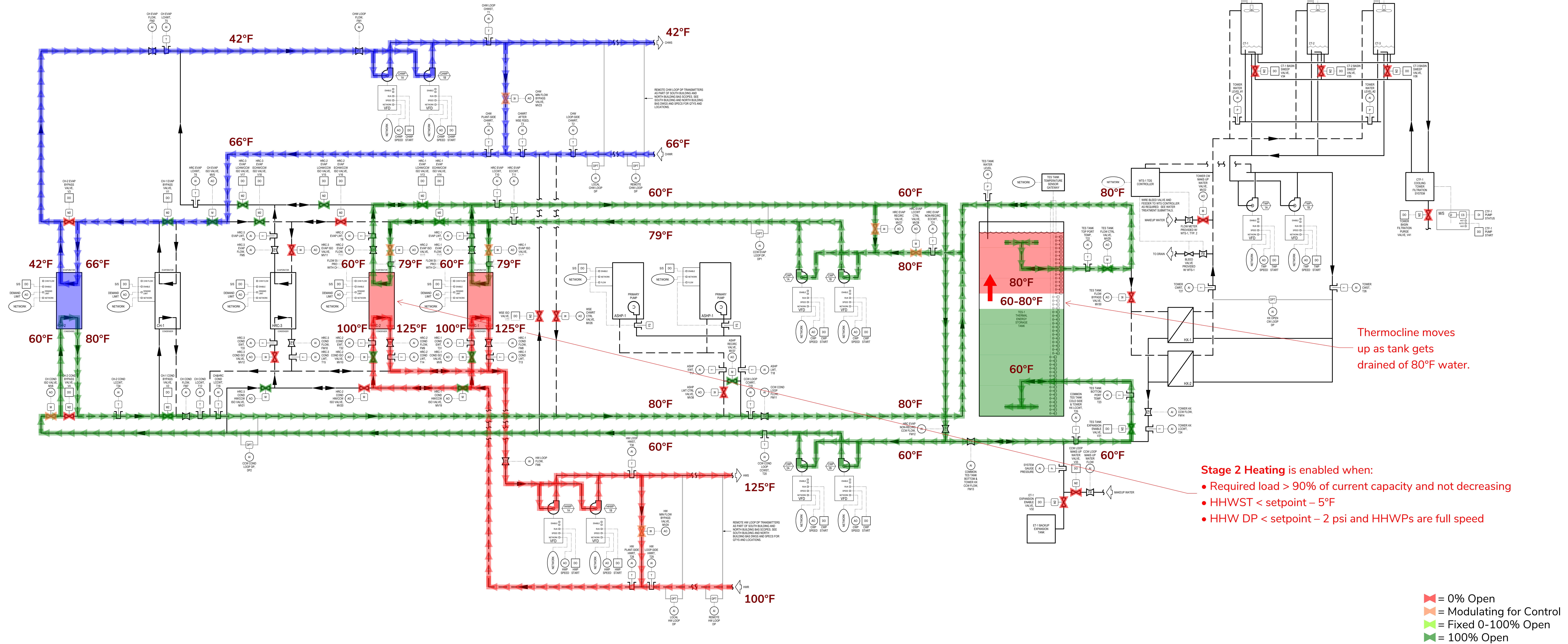
Minor Cooling Load, Minor Heating Load

Mild weekday morning with some cooling load
Stage 1 Cooling with Stage 1 Heating (1C1H) but loads are too low to support full-time operating chillers



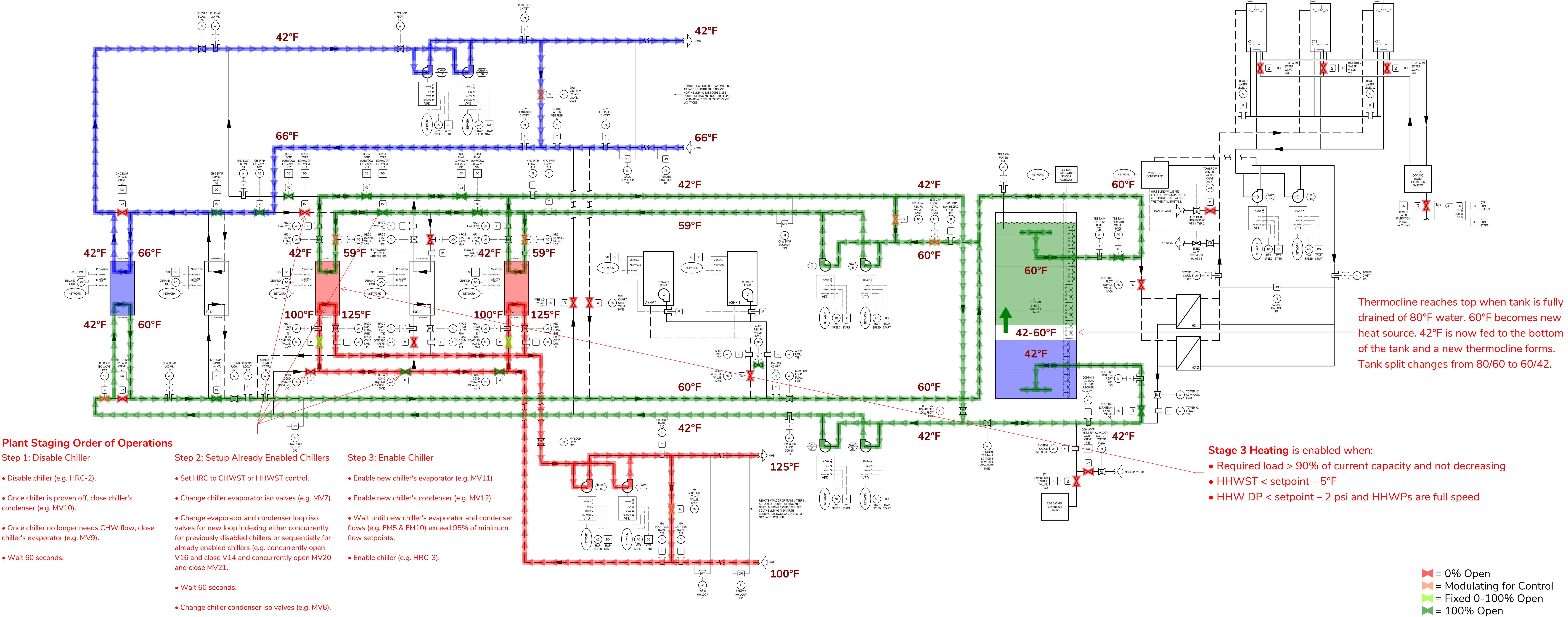
Minor Cooling Load, Moderate Heating Load

Colder weekday morning with some cooling load
Stage 1 Cooling with Stage 2 Heating (1C2H)



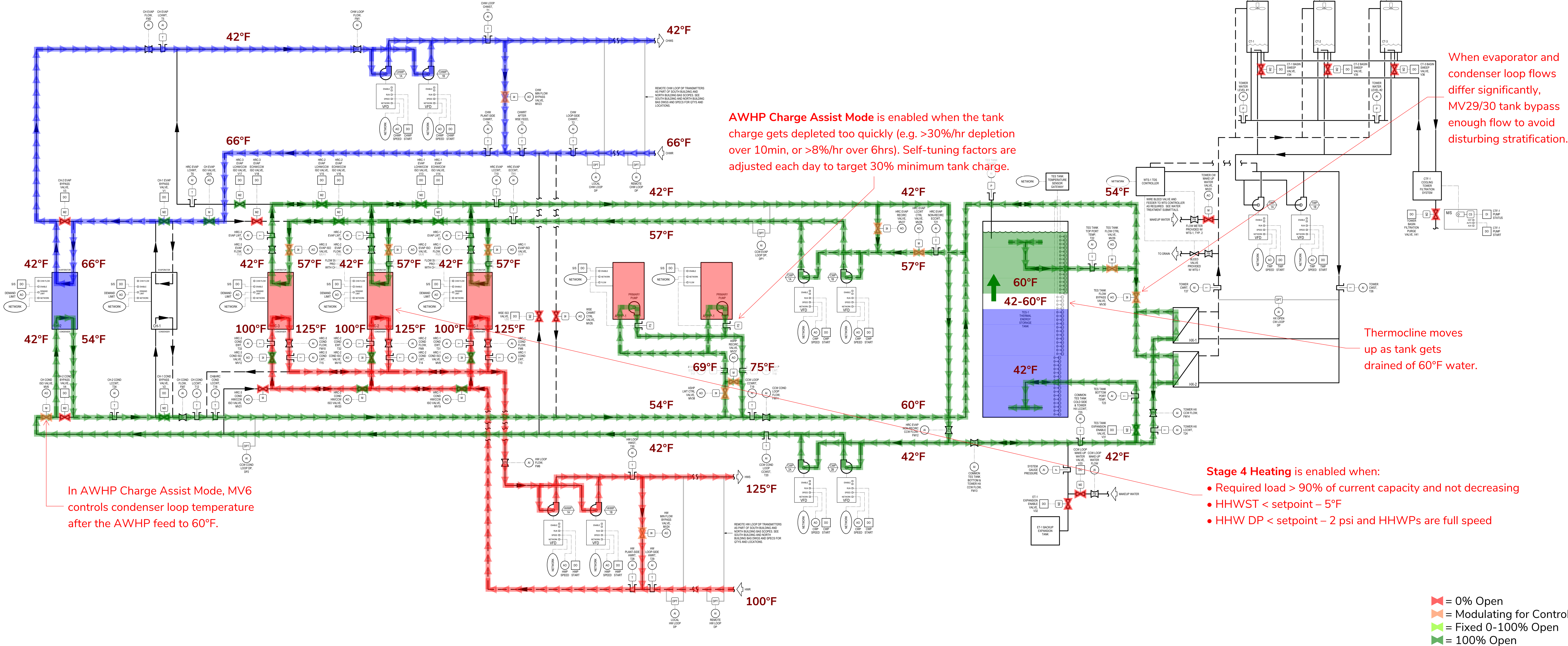
Minor Cooling Load, High Heating Load

Very cold weekday morning with some cooling load
Stage 1 Cooling with Stage 3 Heating (1C3H)



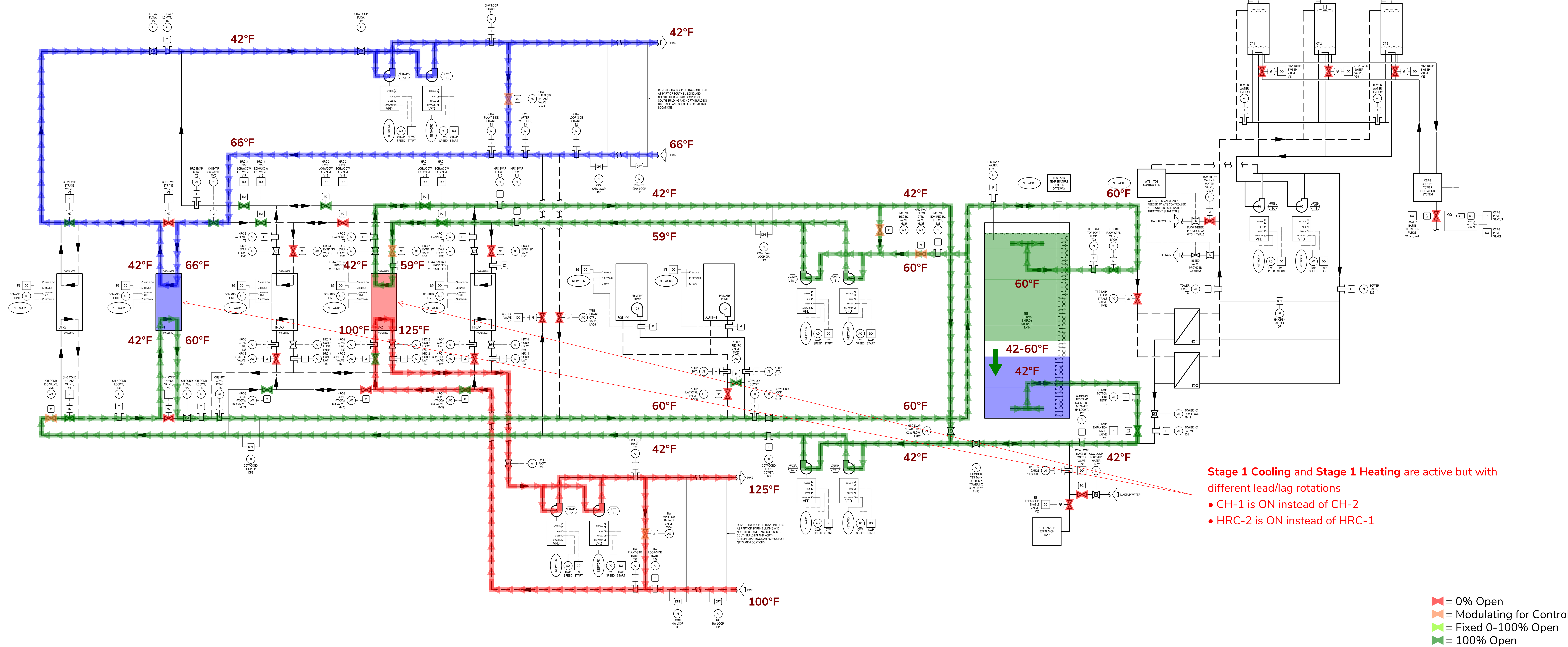
Minor Cooling Load, Design Day Heating Load

Coldest weekday morning with some cooling load
Stage 1 Cooling with Stage 4 Heating (1C4H)



Minor Cooling Load, Minor Heating Load

Mild weekday afternoon with some heating load
Stage 1 Cooling with Stage 1 Heating (1C1H)

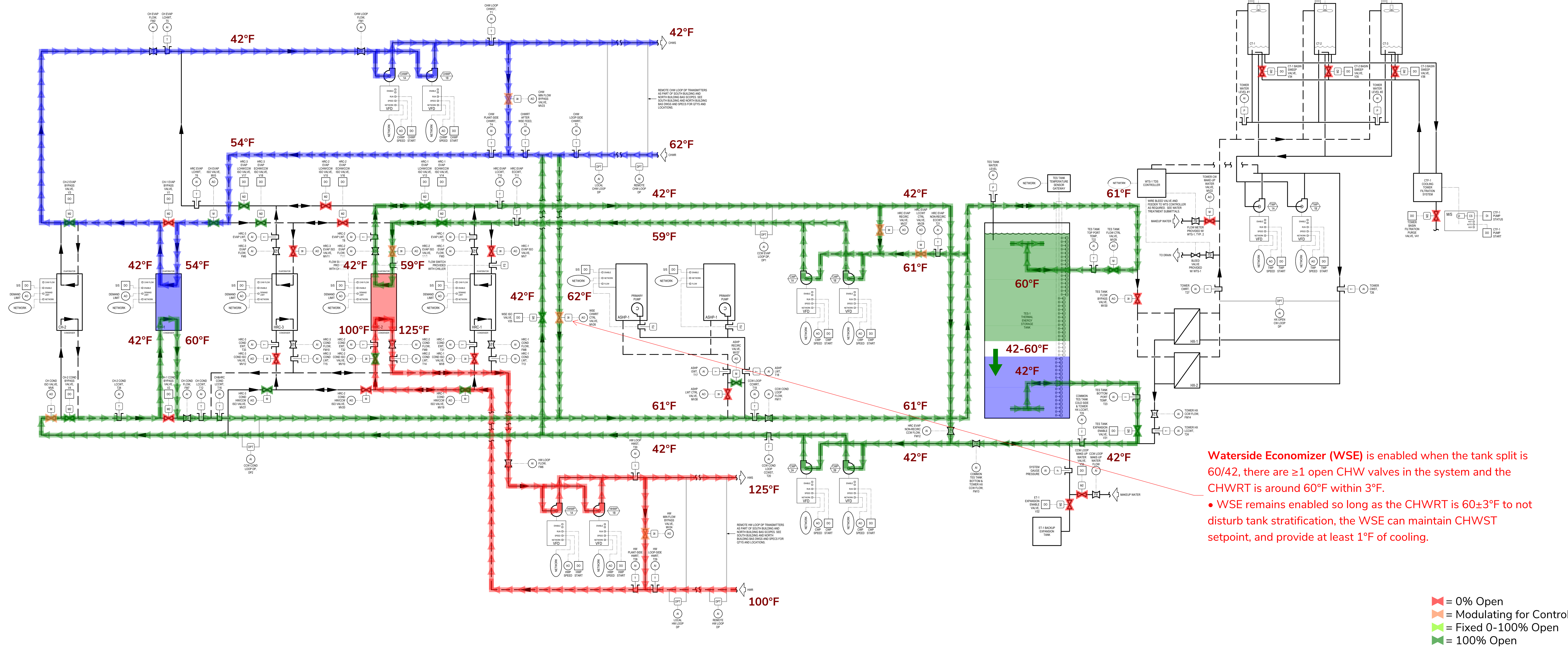


Stage 1 Cooling and Stage 1 Heating are active but with different lead/lag rotations

- CH-1 is ON instead of CH-2
- HRC-2 is ON instead of HRC-1

Minor Cooling Load, Minor Heating Load

Mild weekday afternoon with some heating load
Stage 1 Cooling with Stage 1 Heating with WSE operation (1C1H)

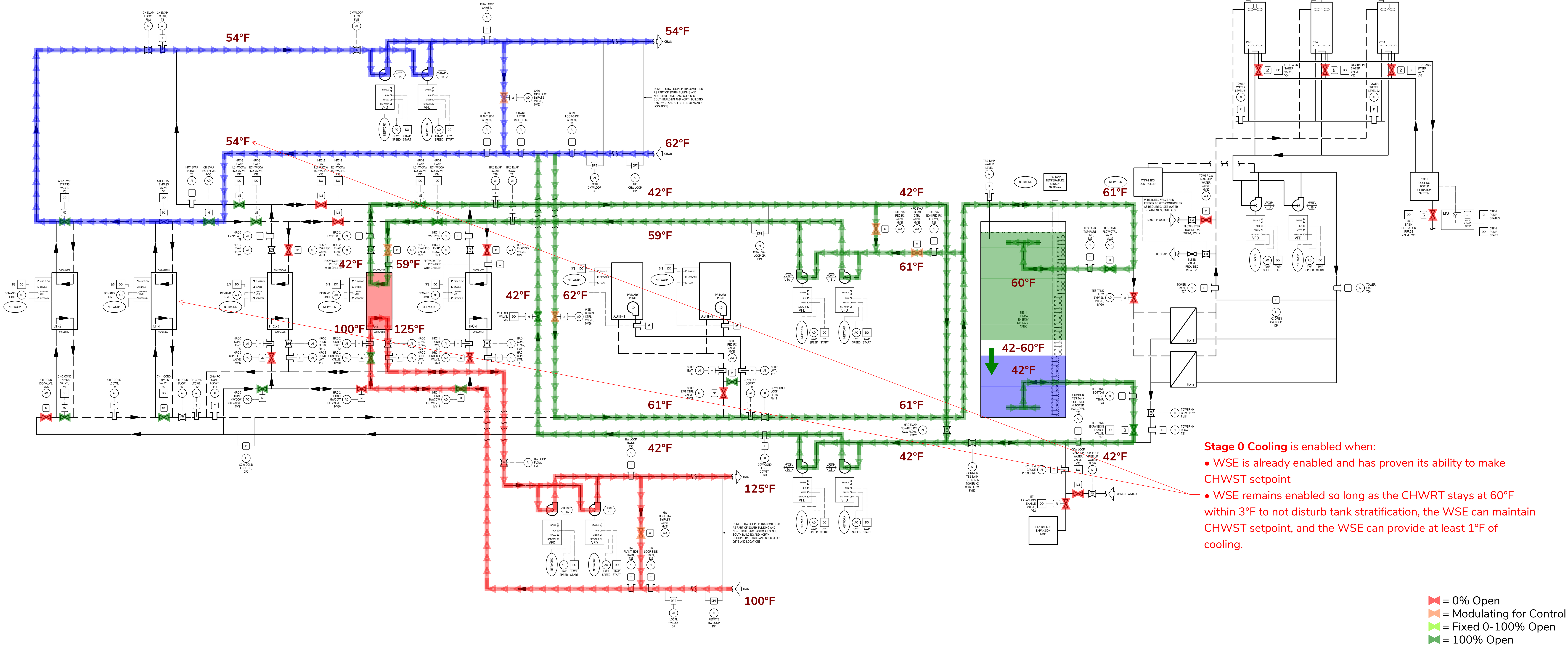


Waterside Economizer (WSE) is enabled when the tank split is 60/42, there are ≥ 1 open CHW valves in the system and the CHWRT is around 60°F within 3°F.

- WSE remains enabled so long as the CHWRT is $60 \pm 3^\circ\text{F}$ to not disturb tank stratification, the WSE can maintain CHWST setpoint, and provide at least 1°F of cooling.

Minor Cooling Load, Minor Heating Load

Mild weekday afternoon with some heating load
Stage 0 Cooling with Stage 1 Heating with WSE operation (0C1H)



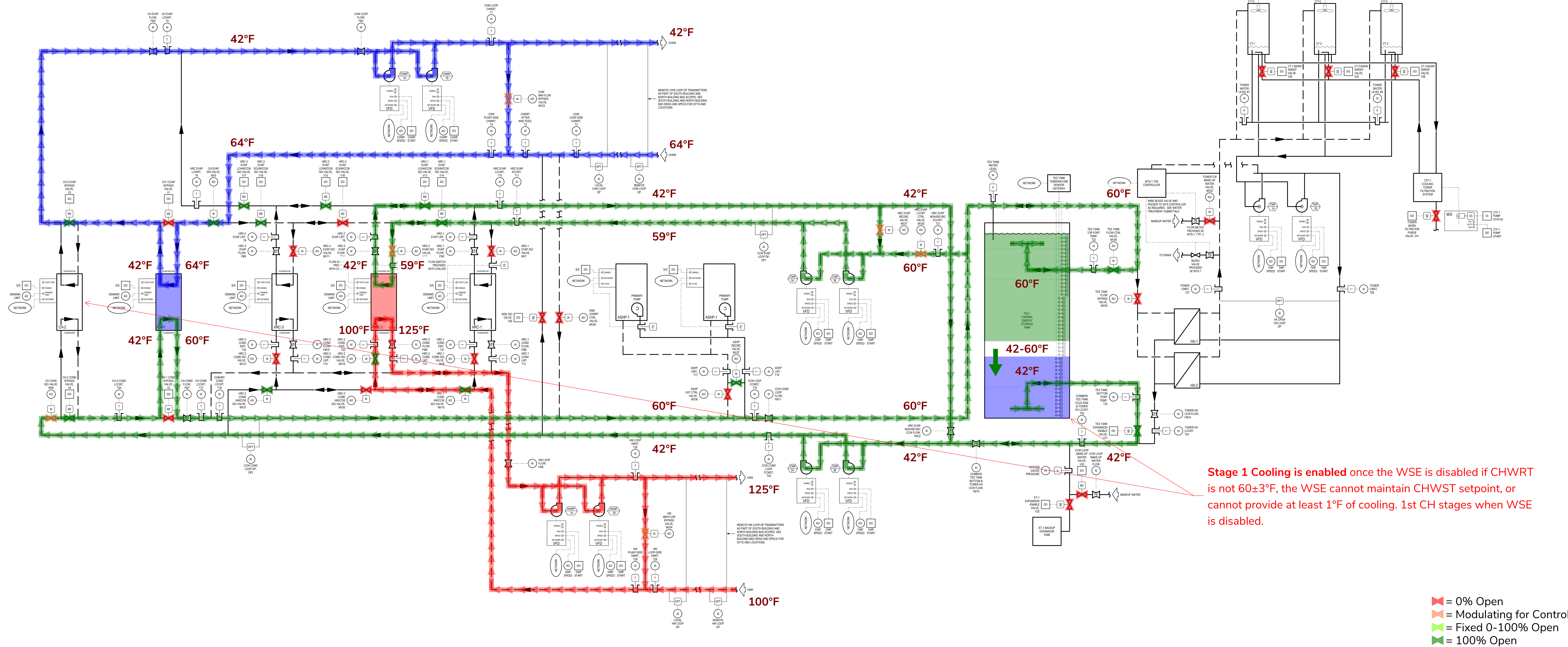
Stage 0 Cooling is enabled when:

- WSE is already enabled and has proven its ability to make CHWST setpoint
- WSE remains enabled so long as the CHWRT stays at 60°F within 3°F to not disturb tank stratification, the WSE can maintain CHWST setpoint, and the WSE can provide at least 1°F of cooling.

Red arrow = 0% Open
Orange arrow = Modulating for Control
Green arrow = Fixed 0-100% Open
Blue arrow = 100% Open

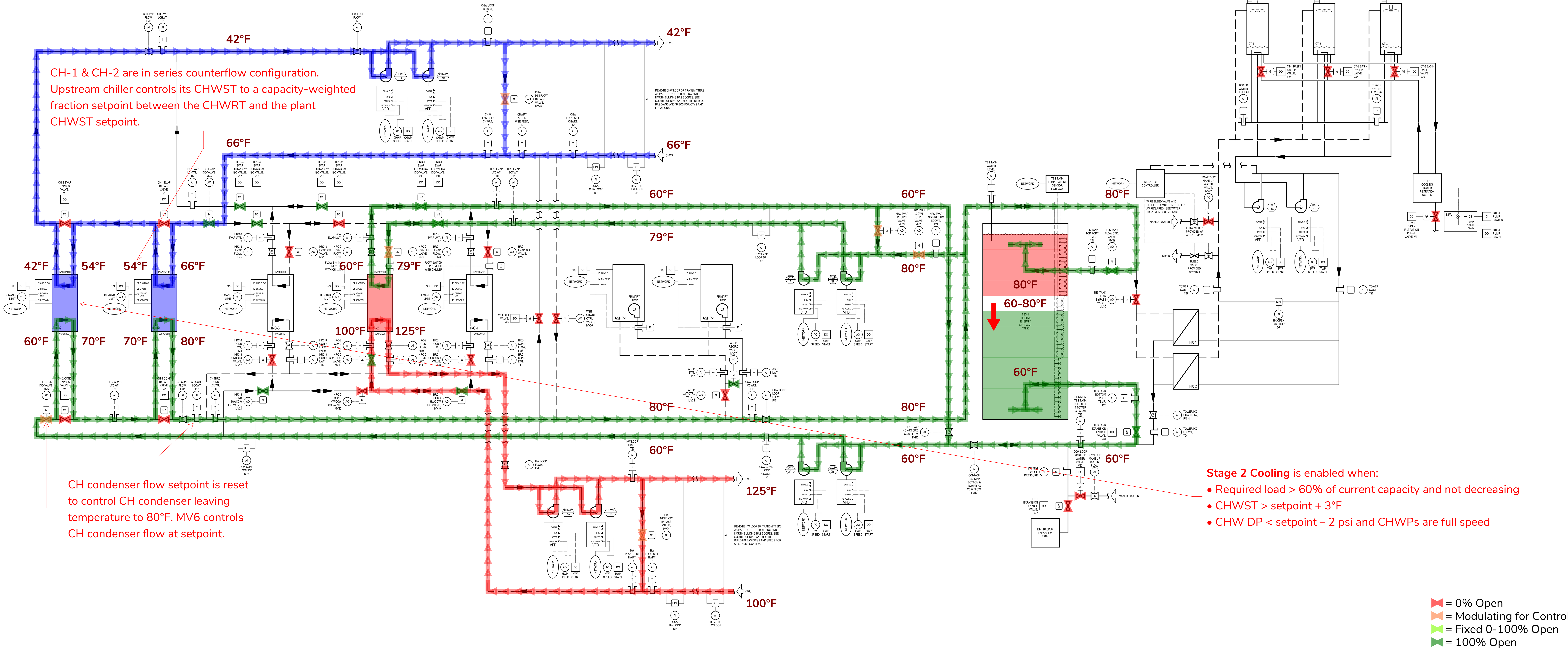
Minor Cooling Load, Minor Heating Load

Mild weekday afternoon with some heating load
Stage 1 Cooling with Stage 1 Heating (1C1H)



Moderate Cooling Load, Minor Heating Load

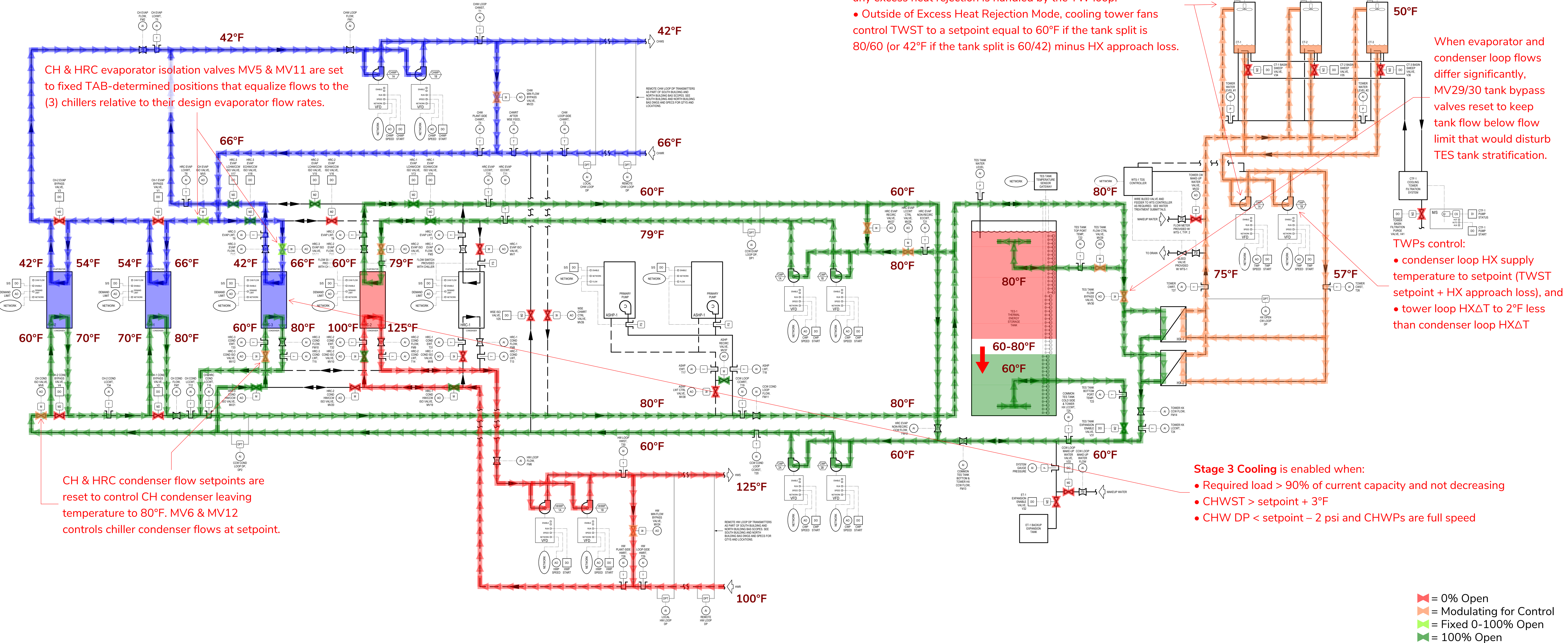
Slightly warmer weekday afternoon with some heating load
Stage 2 Cooling with Stage 1 Heating (2C1H)



Moderately High Cooling Load, Minor Heating Load

Warmer weekday afternoon with some heating load
Stage 3 Cooling with Stage 1 Heating (3C1H)

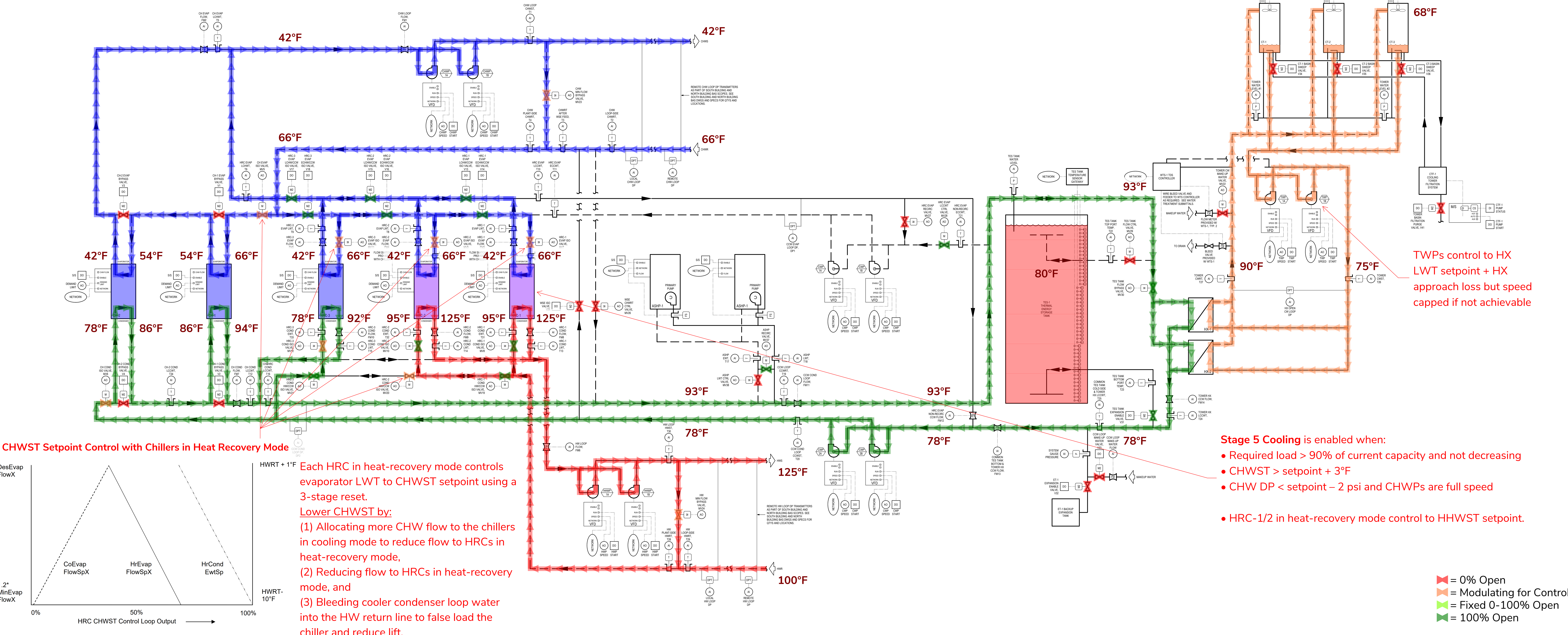
- TWP's are enabled when the tank bypass is active and condenser loop flow exceeds evaporator loop flow. The tank is not fully charged yet so Excess Heat Rejection is not active but any excess heat rejection is handled by the TW loop.
- Outside of Excess Heat Rejection Mode, cooling tower fans control TWST to a setpoint equal to 60°F if the tank split is 80/60 (or 42°F if the tank split is 60/42) minus HX approach loss.



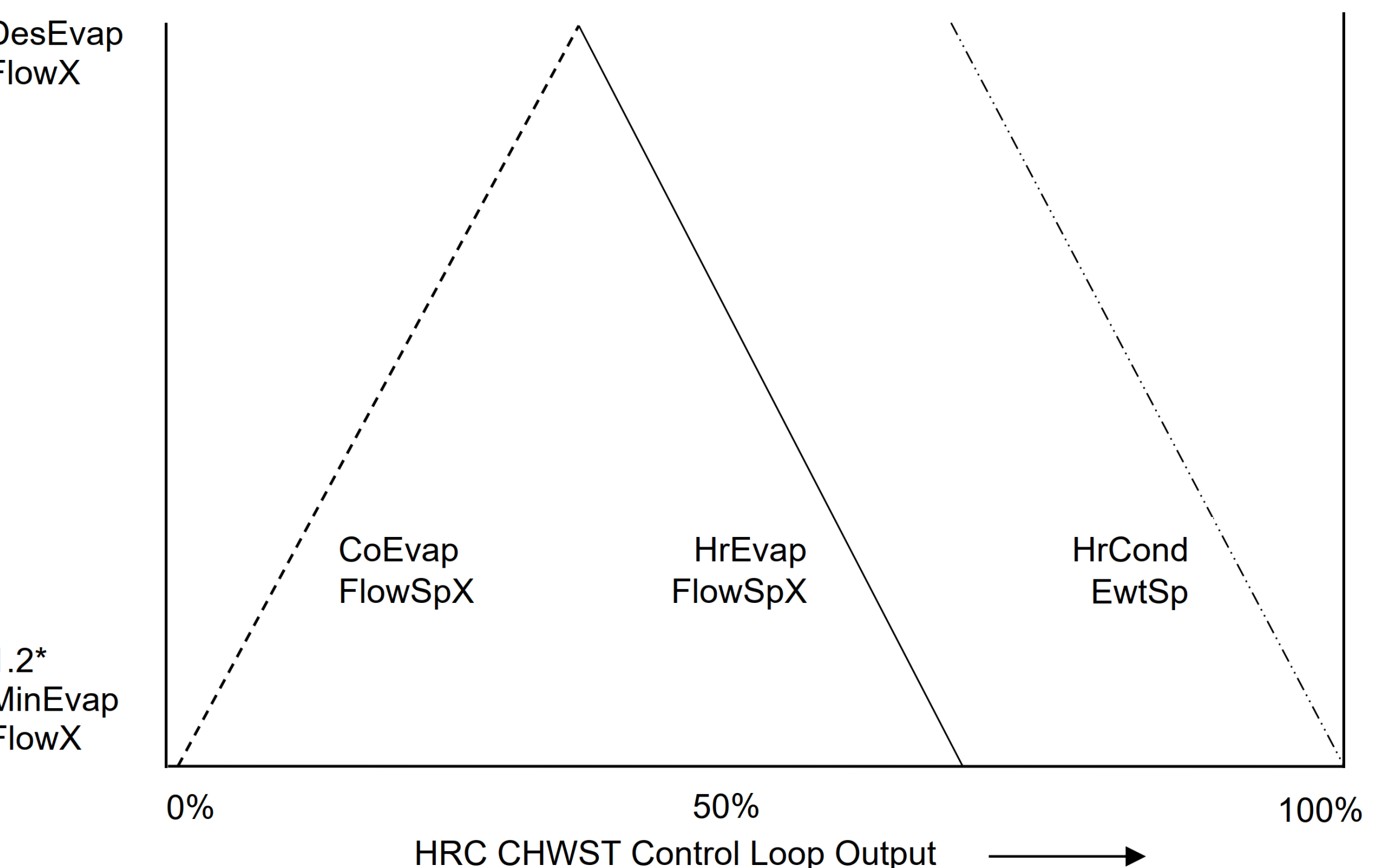
Hot weekday afternoon with some heating load
Stage 4 Cooling with Stage 1 Heating (**4C1H**)

Design Day Cooling Load, Moderate Heating Load

Hottest weekday afternoon with moderate heating load (highly unlikely)
Stage 5 Cooling with Stage 2 Heating (5C2H)



CHWST Setpoint Control with Chillers in Heat Recovery Mode



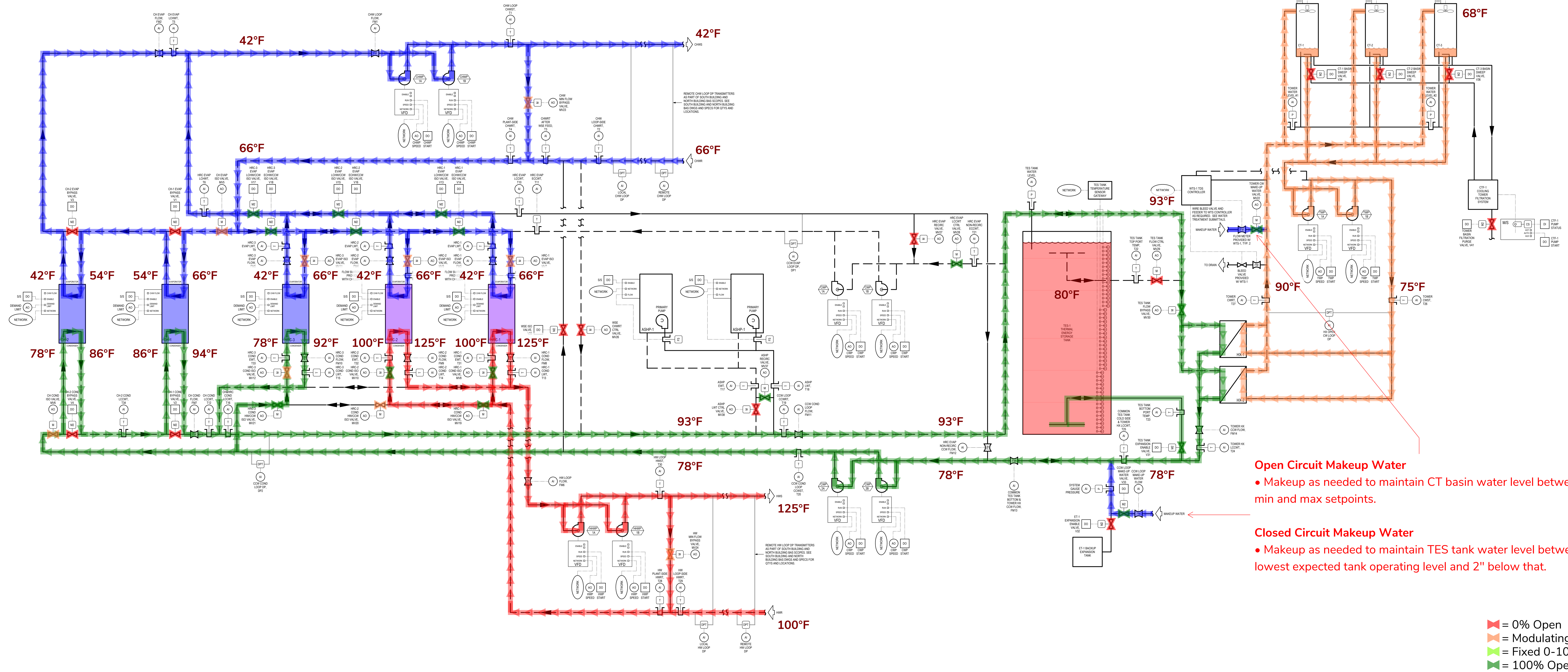
Each HRC in heat-recovery mode controls evaporator LWT to CHWST setpoint using a 3-stage reset.
Lower CHWST by:
(1) Allocating more CHW flow to the chillers in cooling mode to reduce flow to HRCs in heat-recovery mode,
(2) Reducing flow to HRCs in heat-recovery mode, and
(3) Bleeding cooler condenser loop water into the HW return line to false load the chiller and reduce lift.

- Stage 5 Cooling is enabled when:
- Required load > 90% of current capacity and not decreasing
 - CHWST > setpoint + 3°F
 - CHW DP < setpoint – 2 psi and CHWPs are full speed
 - HRC-1/2 in heat-recovery mode control to HHWST setpoint.

- ✖ = 0% Open
- ◡ = Modulating for Control
- ▬ = Fixed 0-100% Open
- ▬ = 100% Open

Makeup Water Control

Closed and open circuit makeup water

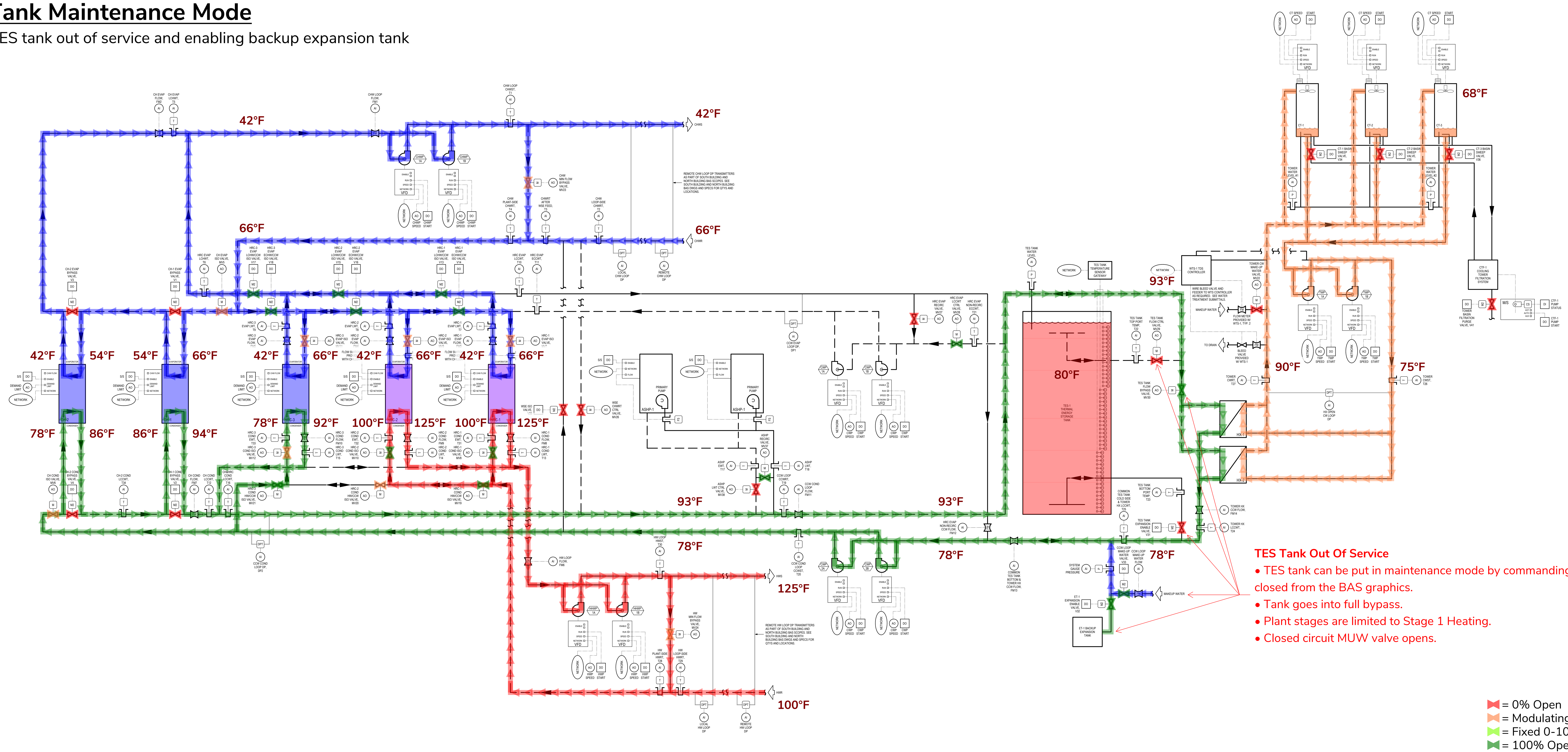


- Open Circuit Makeup Water**
- Makeup as needed to maintain CT basin water level between min and max setpoints.
- Closed Circuit Makeup Water**
- Makeup as needed to maintain TES tank water level between lowest expected tank operating level and 2" below that.

- 🔴 = 0% Open
- 🟠 = Modulating for Control
- 🟢 = Fixed 0-100% Open
- 🟩 = 100% Open

TES Tank Maintenance Mode

Taking TES tank out of service and enabling backup expansion tank

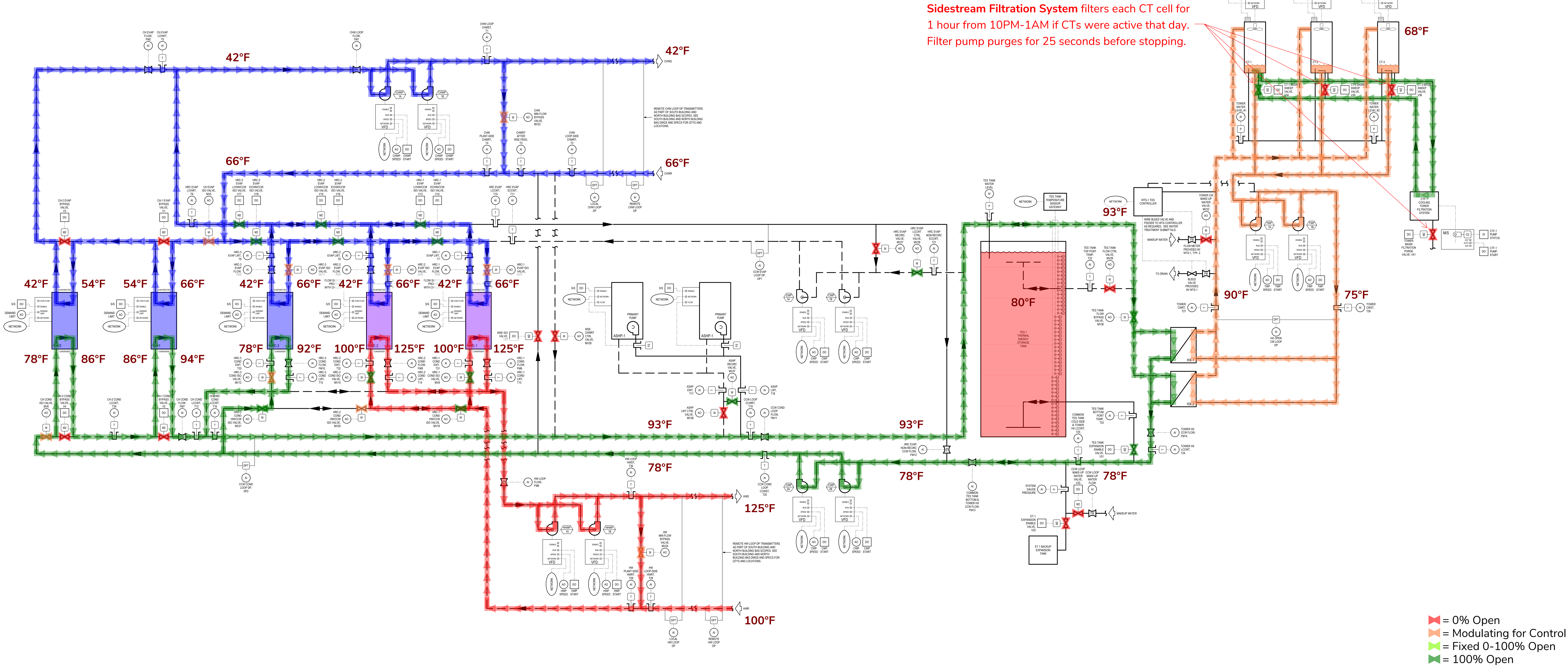


- TES Tank Out Of Service**
- TES tank can be put in maintenance mode by commanding V31 closed from the BAS graphics.
 - Tank goes into full bypass.
 - Plant stages are limited to Stage 1 Heating.
 - Closed circuit MUW valve opens.

- ✖ = 0% Open
- ◡ = Modulating for Control
- ▬ = Fixed 0-100% Open
- ▬ = 100% Open

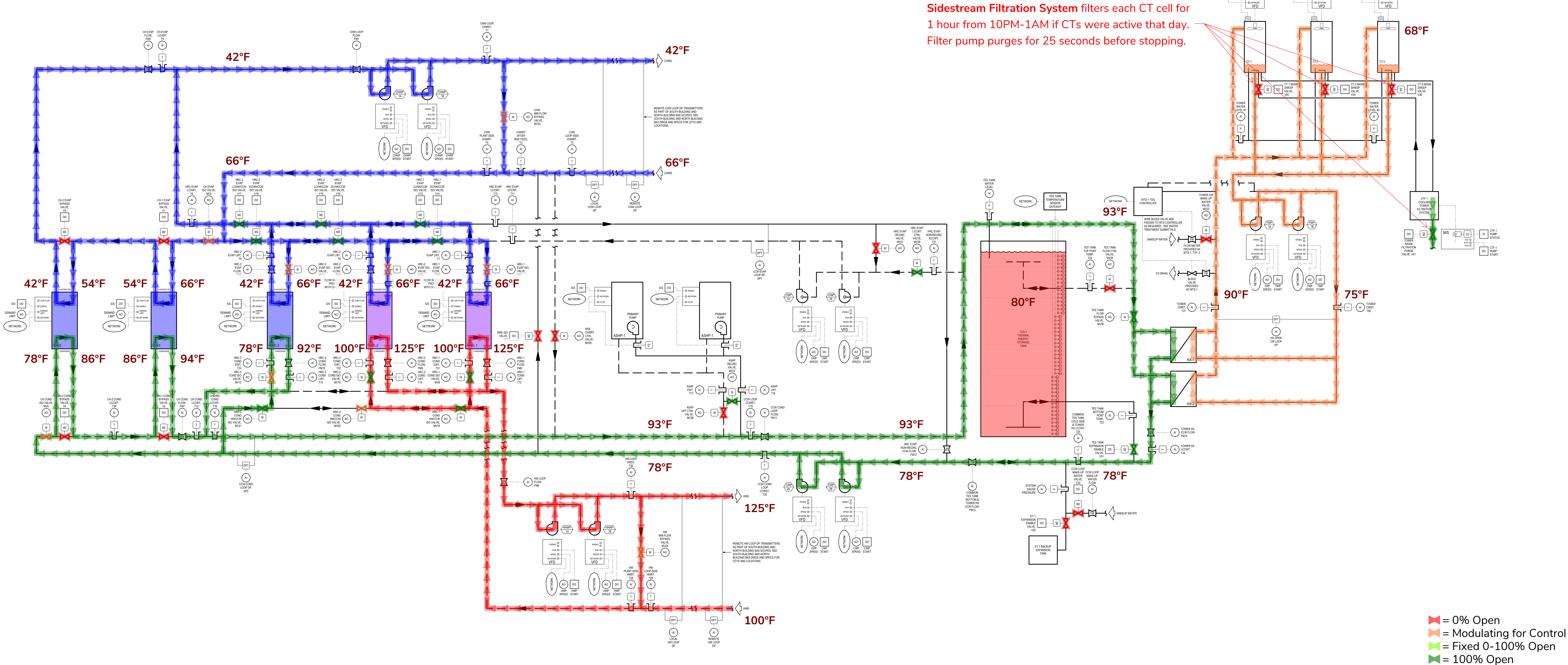
Cooling Tower Filtration System

Filtration schedule and purge control



Cooling Tower Filtration System

Filtration schedule and purge control



Sidestream Filtration System filters each CT cell for 1 hour from 10PM-1AM if CTs were active that day. Filter pump purges for 25 seconds before stopping.