Solving the Large Building All-Electric Heating Problem

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The push for building HVAC electrification[1] (i.e., eliminating on-site fossil fuel consumption) poses new challenges for heating large buildings and campuses in a practical and efficient way. Common small- and medium-building all-electric solutions such as air-to-air heat pumps and variable refrigerant flow systems do not scale well for large building applications, and most existing large-building solutions require compromises. One novel solution, time-independent energy recovery (TIER), is an all-electric central plant design that combines thermal energy storage and energy recovery to improve on existing alternatives for large commercial and mixed-use buildings with respect to energy efficiency, cost-effectiveness, equipment spatial requirements and support of grid-interactive efficient building initiatives.

State of the Market

Currently, four primary options exist in the market for generating heat using electricity for large buildings:

- Air-source (air-to-water) heat pumps, which generate hot water using heat extracted from ambient air via the vapor compression refrigeration cycle;
- Electric boilers, which rely on electric resistance heat to generate hot water;
- Wire-to-air electric resistance coils, which are typically used at the zone level in terminal units such as variable air volume (VAV) and fan-powered boxes; and
- Heat recovery chillers, which generate chilled water and hot water simultaneously, but require either simultaneous heating and cooling loads in the building or a separate heat source or sink.

Each of these options is fraught with one or more major challenges related to equipment and installation costs, spatial requirements, energy efficiency and carbon emissions.

Air-Source Heat Pumps

Air-source heat pumps (ASHPs) are probably the most carbon-friendly option on the market that does not rely

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on heat recovery. In Santa Clara, Calif., where the design heating temperature is 29°F (-2°C), one market leader's product yields a heating coefficient of performance (COPh) of approximately 2.1 when generating 120°F (49°C) water at design ambient dry bulb.

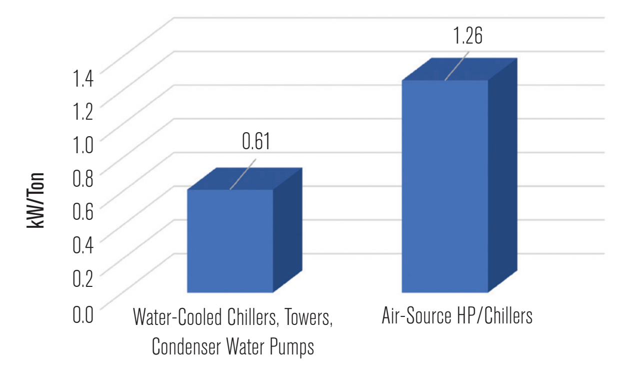
The efficiency of heat pumps is highly dependent on ambient air temperatures. While a design COPh above 2 is possible in mild west coast climates where the author practices, efficiency and capacity both drop rapidly as ambient temperature falls. Currently available air-to-water heat pumps cannot generally be used when ambient temperatures are below about 0°F (-18°C) while producing hot water of approximately 120°F (–49°C) (assuming R-410A refrigerant).

ASHPs are also very expensive per unit capacity (roughly $150/MBH to $200/MBH [$511 000/MW to $682 000/MW] vs. $15/MBH to $30/MBH [$51 000 MW to $102 000 MW] for high quality condensing gas boilers). And, because they use ambient air to extract heat, they require multiple units with large footprints to generate heat at scale. On large high-rise projects, it can be nearly impossible to find sufficient roof space for ASHPs.

The use of multiple units in large installations necessitates costly piping and controls for each unit. Most ASHPs on the market have very high minimum flow rates, which usually requires providing a primary pump dedicated to each unit, further adding to first costs. ASHP plants are also likely to experience higher ongoing maintenance costs than other plant options because of the quantity of devices involved and the complexity of the equipment itself; large ASHPs typically have four to six scroll compressors, at least two refrigeration circuits and multiple condenser fan motors, increasing the likelihood of some device failing or requiring service.

One benefit of ASHP designs is that almost all ASHPs inherently can provide cooling as well as heating; they require the ability to defrost the outdoor coils when operating in cool weather, which is usually accomplished by reversing the cycle, i.e., becoming a chiller. Thus, ASHPs can switch to cooling mode in the summer, reducing the size of the cooling plant serving the same building and offsetting some of the first cost from the ASHPs. Unfortunately, currently available ASHPs are not very efficient in cooling mode, commonly yielding efficiencies of about 9.5 EER to 10 EER (1.2 kW/ton to 1.3 kW/ton [0.34 kW/kW to 0.37 kW/kW] or 2.8 COP to 2.9 COP) at AHRI Standard 550/590 conditions.

FIGURE 1 Cooling full-load efficiency of typical water-cooled and ASHP chiller plants.



Contrast this with a well-designed water-cooled chiller plant that operates at about 0.60 kW/ton to 0.65 kW/ton (0.17 kW/kW to 0.18 kW/kW) or 5.4 COP to 5.9 COP at design conditions, including condenser water pumps (CWP) and cooling towers. This reality makes it almost impossible to comply with either ASHRAE Standard 90.1-2019[2] or California Title 24[3] using the performance approach when replacing water-cooled plant cooling capacity with ASHP capacity since the baseline cooling system for large buildings under both standards is a chiller plant with water-cooled plant variable speed centrifugal chillers.

Figure 1 shows a typical efficiency comparison. On one recent project where our company used AHSPs for heating, we were able to use part of the available ASHP cooling capacity to provide 30% of the design cooling plant capacity, with the rest provided by a high-efficiency all-variable speed water-cooled plant; using any more of the ASHPs in cooling mode resulted in not complying with code and increasing energy costs.

Electric Resistance

Electric resistance-based electric heating options such as electric boilers and wire-to-air coils do not present the same spatial or mechanical first-cost challenges as ASHPs. Relative to ASHP plants, which are typically limited to supply temperatures of around 120°F (49°C), electric boilers can generate 160°F to 180°F (71°C to 82°C) supply temperatures like conventional natural gas boiler plants, and thus can benefit from the higher hot water delta Ts (e.g., 40°F [22°C]) and smaller pipe and pumps sizes that result from supplying hotter water.

Another major benefit of zone-level electric resistance heating coils is that they eliminate parasitic pipe heat losses inherent to all water-based designs. Preliminary research[4] indicates these losses can be as large as the amount of heat needed for space conditioning.

Both electric resistance design strategies are, however, limited by thermodynamics to a peak COPh of 1. Even in states like California, which generates much of its electricity from zero-carbon wind, solar and hydro plants, the grid is not low-carbon in the early morning when heating systems peak. Resistance heating options are therefore likely to remain worse than natural gas boilers on a carbon basis in at least the near term in most parts of the country after accounting for generation, transmission and distribution losses.

Electric resistance options can additionally present new challenges to electrical engineers by making buildings winter-peaking instead of summer-peaking. This is particularly an issue in cold climates, but winter-peaking can also occur with electric resistance heating options in mild west coast climates. Not only will winter-peaking increase building electrical service sizes vs. current practice, but the entire utility distribution system would have to be up-sized at considerable expense.[5]

Code compliance can also be an issue with electric resistance heating systems. ASHRAE Standard 90.1-2019's Energy Cost Budget Method, for instance, allows electric resistance heat but puts the proposed design up against a fan-powered box system baseline with zero reheat. California Title 24 prescriptively prohibits electric resistance with few exceptions and does not include electric resistance heat in any of its performance method baseline system types.

Heat Recovery Chillers

Another alternative is to use heat recovery chillers that can provide high-efficiency simultaneous heating and cooling when concurrent heating and cooling loads exist. In most applications this condition does not occur when heating loads are at their highest. When heating loads are high (e.g., on a cold winter day or during morning warm-up), there is typically little or no cooling load because cold ventilation outdoor air provides all the cooling needed. The time-dependency issue with heat recovery chillers is sometimes addressed with geothermal heat exchange systems, wherein heat absorbed from the building in warm summer weather is rejected to the earth, and the heat needed to warm the building in cold winter weather is extracted from the earth. However, geothermal bore fields for large buildings are extremely expensive to install, especially when site limitations require deep bores, and are prone to performance degradation over time when the heating and cooling loads are not well balanced.

Summary

The current market presents owners with two mediocre options for all-electric heating and cooling systems: either accept the large space requirements and high first costs inherent to ASHPs, or select an electric resistance option that increases energy cost and may yield worse carbon performance than a natural gas boiler plant for the foreseeable future while electricity is still primarily generated from fossil fuels. Current applications with heat recovery chillers are limited or are prohibitively expensive when coupled with geothermal systems.

The Solution

The key to solving these issues, as indicated by MacCracken,[6] is coupling thermal energy storage (TES) with heat recovery. TES has long been used as an HVAC strategy for peak shifting, primarily as a cost-saving strategy through reduced demand and peak utility charges, but rarely as an energy recovery mechanism. Multiple versions of thermal energy storage systems exist, including:

- Condenser water (CW) storage (stratified and unstratified)
- Hot water (HW) storage;
- Chilled water (CHW) storage;
- Ice storage; and
- Phase-change material (PCM) storage;

Combining TES with energy recovery leads to the concept of time-independent energy recovery (TIER), an all-electric central plant design that improves on the existing alternatives for large commercial and mixed-use buildings with respect to energy efficiency, cost effectiveness, equipment spatial requirements and support of grid-interactive efficient building (GEB) initiatives.

All TIER plants have three components in common: a TES component, an energy recovery component (heat recovery chillers) and a trim heat source component (usually ASHPs, but these can be electric boilers in cold climates or where roof space is limited). When

FIGURE 2 Direction of heat transfer for a hot water TIER system.

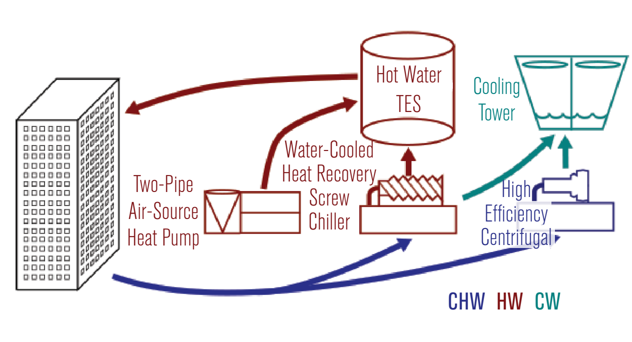
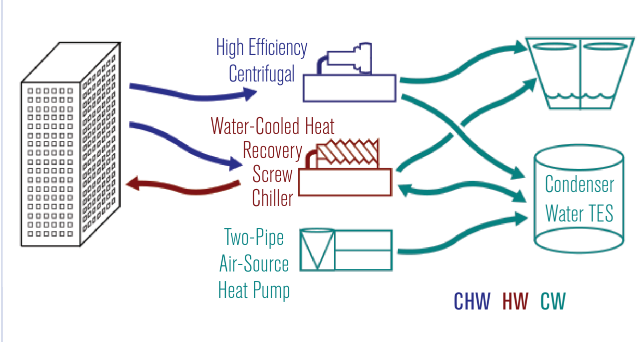


FIGURE 3 Direction of heat transfer for a condenser water TIER system.



combined, these elements allow efficient water-to-water chillers to perform heat recovery even when heating and cooling loads are not simultaneous, as is done with a geothermal system, while avoiding the high costs and temperature degradation inherent to geothermal designs.

Though perhaps initially nonintuitive, each TES approach can be used to store energy for heating irrespective of whether the medium is 130°F (54°C) hot water, 80°F (27°C) condenser water or 32°F (0°C) ice. The first two will be used to illustrate this concept. In a design with a hot water storage tank, trim ASHPs (which are only sized for a fraction of design heating load), charge the hot water tank throughout a heating design day. Heat recovery chillers also charge the tank by pulling any available heat from the chilled water loop and rejecting it to the tank. During winter mornings when the building is heating dominated, the tank discharges; in the afternoon when combined building heat recovery and trim ASHP capacity exceeds heating load, the tank charges. *Figure 2* illustrates the energy flow paths for the hot water storage system design.

In a condenser water storage design, trim air-source heat pumps, which are again only sized for a fraction of design heating load, charge the condenser water tank throughout a cold day with tepid 80°F (27°C) water. Heat rejection loads from the condenser side of chillers in "cooling mode" also charge the tank with 80°F (27°C) water. During winter mornings when the building is heating dominated, the tank discharges as heat recovery chillers extract more heat from the tank than the ASHPs, and any chillers in "cooling mode" reject to the tank; in the afternoon, when combined chilled water heat rejection load and trim ASHP capacity exceeds building heating load, the tank charges. In the summer, the heat recovery chillers can be indexed to the chilled water loop to provide cooling. *Figure 3* illustrates the energy flow paths for the condenser water storage system design.

Understanding Condenser Water TIER

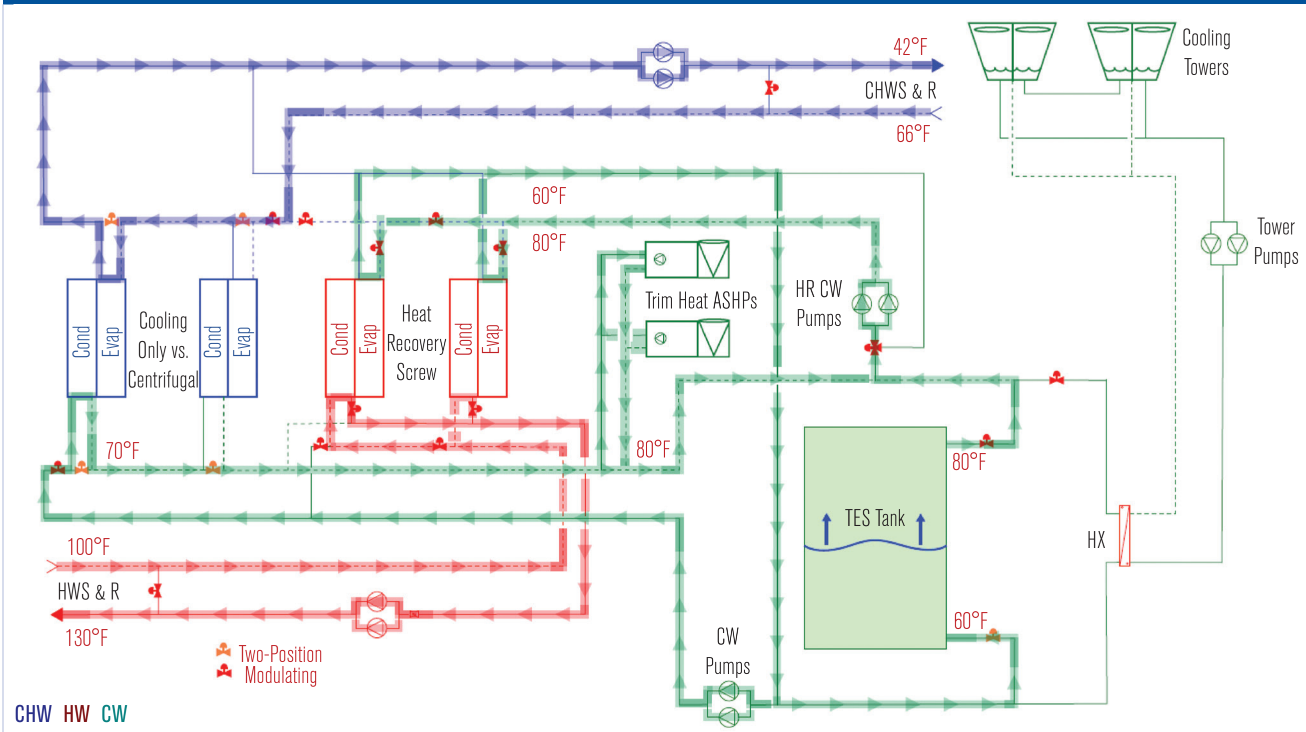
The remainder of this article focuses on condenser water as the storage medium of choice to illustrate the benefits of TIER since we believe condenser water is the best option for many applications. Many of the subsequent benefits also apply to other TIER TES schemes, but all approaches are not equal. Pros and cons of alternative TES strategies are discussed at the end of the article.

The condenser water TIER plants we have designed take heat rejected from cooling loads via high-efficiency, low-lift, centrifugal chillers and store it in a TES tank at tepid temperatures between 60°F (16°C) and 80°F (27°C). Tank temperature excursions down to 40°F (4.4°C) are allowed on peak heating days to minimize tank size.

When energy is needed for building heating, heat is extracted from the tank using water-to-water heat recovery chillers. In effect, the cooling chillers and heat recovery chillers are placed in a cascade configuration: the cooling chillers have a lift envelope of 40°F (4.4°C) chilled water supply temperature to 80°F (27°C) condenser water leaving temperature, while the heat recovery chillers have a lift envelope of 60°F (16°C) evaporator supply temperature to the active hot water supply temperature setpoint, typically 110°F (43°C) to 140°F (60°C) for all-electric designs.

During most days in California's mild climate zones where the author practices, the energy recovered from cooling loads alone can satisfy heating loads. During the small fraction of the year when heat recovery alone

FIGURE 4 Cool day morning operation of a condenser water TIER system.



cannot satisfy heating demand, trim ASHPs are used to charge the storage tank.

Figures 4, *5* and *6*, which are simplified and adapted from a project for which we used a condenser water TIER design, show an example plant in a few typical modes of operation to illustrate the design concept.* Flow paths for chilled water (CHW), condenser water (CW) and hot water (HW) are traced in each.

Figure 4 illustrates a typical cold morning operation condition during which the TES tank discharges. Both red heat recovery chillers are in operation, supplying hot water to the building at 130°F (54°C) on the condenser side while extracting heat from the TES tank on the evaporator side. Any cooling loads that the building might have—e.g., due to 24/7 IT spaces, data centers, lab equipment, etc.—are concurrently addressed by a blue variable speed "cooling-only" machine. The condenser water rejected from this machine, which is 70°F (21°C) in this example, is then passed through the trim ASHPs, which act to boost the condenser water charging the top of the tank to 80°F (27°C). The amount of heat the blue cooling-only chiller and the ASHPs are adding to the tank is less than the amount of heat the red heat recovery chillers are removing from the tank, so on balance the tank is discharging (decreasing in temperature).

Later during the same day, when heating loads decrease and cooling loads increase, the net result is that the tank charges (increases in average temperature). During the example condition in *Figure 5*, only one red heat recovery chiller is providing heating while drawing energy from the TES tank. Two-cooling only blue chillers are cooling the building in a series configuration while head pressure control on the condenser side is modulating flow through the cooling-only machines' condenser barrels to achieve the target condenser water leaving temperature of 80°F (27°C) needed to charge the tank. The air-source heat pumps are off because BAS logic has determined that heat rejection loads alone will be sufficient to charge the tank by the end of the business day, i.e., bring the tank up to an average temperature of about 80°F (27°C).

*Astute readers may notice that the TES tank, which is open to the atmosphere, is not hydraulically isolated from the closed-loop chilled water and hot water systems. This plant was designed for a low-rise campus of two buildings where the tank was readily located near the high point of the campus to avoid the energy waste that would have otherwise been incurred by introducing pressure-sustaining valves. When atmospheric TES tanks are used in high-rise applications, they must be installed low to be structurally feasible; hydraulically isolating them with heat exchangers avoids pressure-sustaining requirements.

FIGURE 5 Cool day afternoon operation of a condenser water TIER system.

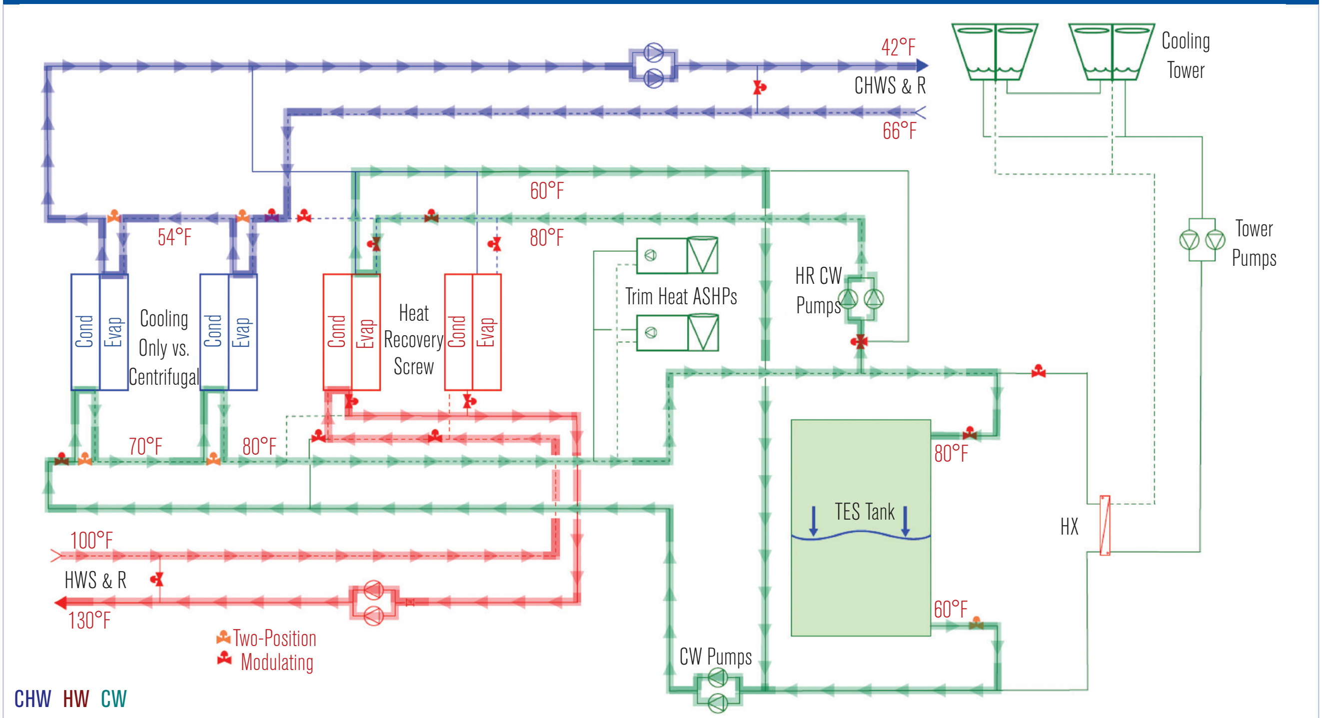


FIGURE 6 Warm day afternoon operation of a condenser water TIER system.

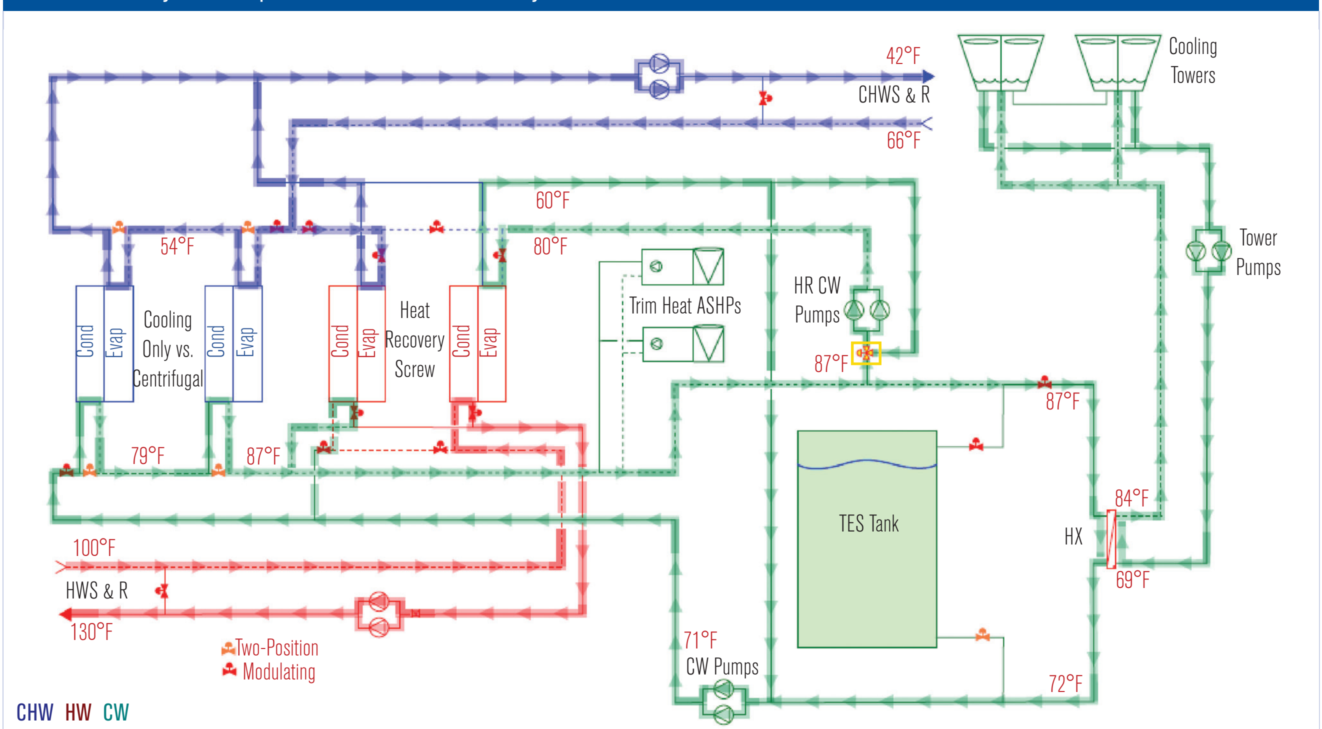
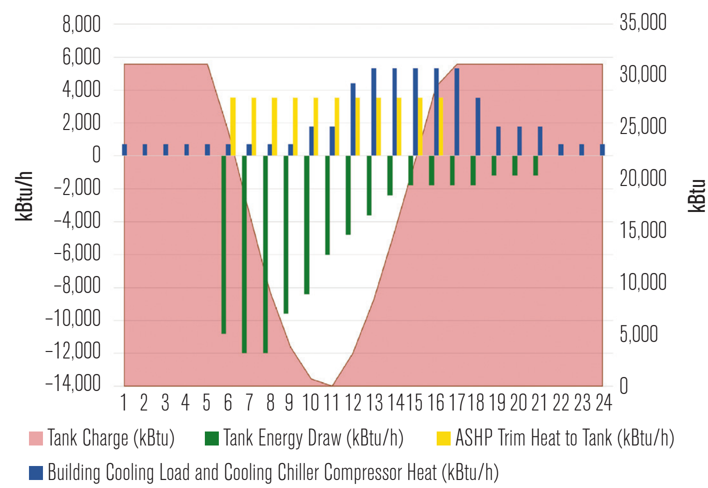


Figure 6 shows a high cooling load condition as might occur during the afternoon of a warm day. In this scenario, one of the red heat recovery chillers has been indexed into "cooling mode" and is connected on the evaporator side to the chilled water loop while rejecting heat at low lift to the condenser water loop. Any building heating loads are served by the one remaining heat recovery chiller indexed to the hot water loop. A mixing

valve upstream of the heat recovery chiller evaporator inlets (shown and boxed in yellow) prevents water warmer than 80°F (27°C) from entering the heating heat recovery chiller's evaporator barrel as is required by many chiller manufacturers for continuous operation. Since the day is warm, morning heating loads were small, meaning the tank is already fully charged by early afternoon. Therefore, all excess heat is rejected through the cooling towers, which are isolated with a heat exchanger to prevent dirty tower water from entering the tank or the chilled or hot water loops.

These three schematics illustrate three of many possible modes of operation for this plant. For instance, it is possible to index valves such that heat recovery chillers reject heat from the chilled water loop directly to the hot water loop. Such a configuration might be optimal when loads are fairly balanced between the chilled water and hot water loops; otherwise, it will result in serving the cooling load at higher lift than necessary (e.g., if there are 200 tons (703 kW) of cooling load but only 50 tons (176 kW) of heating load), in which case operating in one of the other aforementioned modes is preferable. Reviewing all operating modes is beyond the scope of this article, but this discussion hopefully conveys the flexibility this plant configuration provides to optimize operational efficiency under a variety of heating and cooling load conditions.

FIGURE 7 Tank discharge profile, TIER design with two 1,765 kBtu/h ASHPs.



Note that the peak tank energy draw is less than the design heating load because heat recovery chiller compressor energy makes up the difference.

The Benefits of TIER

Spatial Requirements

While TES designs are often thought of as space intensive, the TIER solution is a space saver relative to a conventional ASHP plant. This is because load shifting allows the TIER design to reduce ASHP capacity dramatically.

A traditional TES tank is used for cooling peak shifting, not for heat recovery, and is typically sized to either ride through the utility peak period without running chillers or trim some fraction of chiller capacity throughout that period. A TIER TES tank is sized to ensure that on a design heating day, heating loads can be met during all hours of the day using the available heat recovered from the building(s) and trim heat source energy added to the tank.

Designers can manipulate tank size by providing more or less trim heat source capacity. The more trim heat source capacity is available, the smaller the tank can be while riding through sustained heating peaks. But as the tank gets smaller, some of the opportunities for heat recovery are lost and plant efficiency gets worse. Since providing more tank capacity generally reduces overall project costs and improves plant efficiency, tank capacity should be maximized to the extent that spatial and project aesthetic constraints allow.

At a certain point, however, there is no value in increasing tank size further since doing so will not yield additional reductions in trim heat source size. This is so because the amount of heat required to warm a building over a 24-hour peak heating day does not change irrespective of the amount of load shifting the tank can provide—heat recovery and trim heat must meet that load over a 24-hour period.

In the real example below, two ASHPs totaling 3,530 kBtu/h (1 MW) of capacity at near design ambient dry bulb of 36°F (2.2°C) were proposed along with a condenser water tank providing 31,200 kBtu (9.1 MWh) of storage for a 1.1 million ft^2 (102 000 m^2) office/dry computer lab complex with a design heating load of approximately 16,000 kBtu/h (4.7 MW). A 110,000 gallon, 50 ft tall, 20 ft diameter (416 000 L, 15 m tall, 6 m diameter) tank was selected for the project. *Figure 7* shows a simplified charge and discharge profile for this scheme on a design day.

The TIER design allowed us to provide two ASHPs totaling 3,530 kBtu/h (1 MW) where 10 ASHPs totaling over 16,000 kBtu/h (4.7 MW) would have otherwise been required. The relative footprints of these two designs are shown in *Figure 8* and *Figure 9*.

FIGURE 8 1.1 million ft² building conventional ASHP farm.

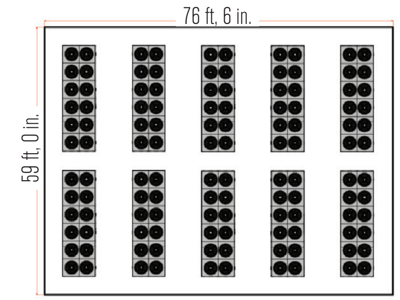
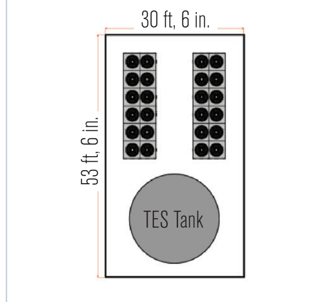


FIGURE 9 1.1 million ft² building TIER TES tank and ASHP alternative.



TIER TES tanks are invariably taller than the ASHPs they replace (height is desirable to improve tank storage efficiency since thermal stratification yields a thermocline of at least a few feet in height; the shorter the tank the more volume is trapped in the thermocline, and thus the greater the total tank volume must be)—40 ft (12 m) taller in this case study—so finding an optimal location for the tank can be a challenge.

EQUATION 1 Cascaded chiller heating efficiency.

$$\mathrm{COPh}_{\mathrm{Cascade}} = \frac{1}{\dfrac{1}{\mathrm{COPh}_{\text{heat recovery chiller}}} + \dfrac{1 - \dfrac{1}{\mathrm{COPh}_{\text{heat recovery chiller}}}}{\mathrm{COPh}_{\text{cooling chiller}}}}$$

We have thus far found success siting these tanks in parking garages. Typically, the TES tank is smaller than the fire water storage tank needed for a high-rise. On one high-rise we have been preliminarily approved to use the TES tank as the fire water tank, further reducing project costs and spatial requirements.

Spatial analyses also illustrate one of the unique benefits of condenser water TIER relative to other forms of TIER, including HW and CHW: while a HW or CHW TES tank's capacity is limited by the delta T of the loads it serves, a condenser water tank serves as a source for heat recovery chillers, so it can have a much higher delta T. For a CHW TES system, delta T is typically in the range of 18°F to 25°F (11°C to 14°C). A HW TES tank storing 120°F (49°C) water achievable by the ASHPs trim changing the tank might similarly be limited to a 25°F (14°C) delta T without multirow heating coil selections.

A condenser water TES tank by contrast can readily be sized for 40°F (22°C) delta T or more. While the tank is intended to operate with a 20°F (11°C) delta T between 60°F (16°C) and 80°F (27°C) on most days to minimize the lift overlap between cooling-only machines and heat recovery machines to maximize cascade efficiency, on design heating days, the tank can cycle through one more time down from 60°F (16°C) to 40°F (4.4°C). The overall delta T with the TIER design is therefore 40°F (22°C), allowing for a compact tank.

Efficiency

The condenser water TIER solution—and all TIER approaches for that matter—are significantly more energy efficient than a conventional ASHP plant.

Consider first that the IPLV of a typical variable speed centrifugal chiller that would be used to cool a large building and charge the TES tank in a TIER design is on the order of 0.35 kW/ton (0.10 kW/kW); this corresponds to a COPh of 11. The COPh of heat recovery chillers boosting water from 60°F to 125°F (16°C to 52°C) should be greater than 5. The cascaded COPh is therefore roughly 3.7 (*Equation 1*).

Contrast this to the COPh of one representative ASHP product, which varies from 2.1 at design ambient conditions (32°F [0°C]) to 3.1 under more mild ambient conditions (60°F [16°C]) when supplying 120°F (49°C) water. Perhaps most important, any energy extracted from the building and stored in the TES tank for later or concurrent heating use effectively provides "free" cooling—it is simply a by-product of the heating process.[†]

[†]The typical paradigm is to view the recovered heat from heat recovery chillers as a "free" by-product of the cooling process. With TIER designs we prefer to flip the paradigm since the objective is to recover as much energy from the building on cold days as possible to minimize the use of trim heat sources. This recovered heat makes the associated cooling in turn "free."

TABLE 1 Condenser water and hot water TIER design lift heating efficiency comparison.

	CONDENSER WATER HEAT RECOVERY			HOT WATER HEAT RECOVERY		
DEVICE	SOURCE (°F)	SINK (°F)	COPh	SOURCE (°F)	SINK (°F)	COPh
Cooling Only Chiller	40	80	12.72	–	–	–
Heat Recovery Chiller	60	140	4.2	40	140	3.5
NET HEATING EFFICIENCY			3.36			3.5

Note that on a design day, when the ASHPs are charging the TIER tank with tepid 80°F (27°C) water, their COPh will increase to approximately 3.75, yielding a cascaded COPh of 2.4. In other words, even on a design day when both the ASHPs and heat recovery chillers are operating, the TIER design will still yield superior energy efficiency to a conventional ASHP plant.

TIER also improves water efficiency in designs with water-cooled chillers since any heat recovered from the building for later or concurrent heating use is avoided cooling tower heat rejection and evaporation.

Cost

Not only are condenser water TIER designs significantly more efficient than ASHP designs, but they also cost less. Since water-cooled chillers (typically 250 ton to 400 ton [879 kW to 1407 kW] screw chillers or larger centrifugal machines) are used as the primary heating machines in a CW TIER plant, they can efficiently serve double-duty as cooling machines for the plant. For instance, in the example plant discussed previously with two cooling-only chillers and two heat recovery chillers, on a hot day one of the heat recovery chillers swings to cooling duty and operates in parallel with the cooling chillers as shown in *Figure 6*.

On a design cooling day, both heat recovery chillers can swing to cooling duty. Owners, therefore, avoid paying for nearly as much redundant tonnage as they do when using a separate ASHP plant for heating. In effect, a TIER design swaps out multiple ASHPs for a TES storage tank and converts cooling-only chiller capacity—which already needed to exist for cooling duty—to heat recovery chiller capacity.

Preliminary pricing from the 1.1 million ft^2 (102 000 m^2) project discussed previously indicated the conversion to TIER would yield mechanical equipment savings on the order of $900,000. These savings do not account for the electrical, controls, piping or opportunity cost savings from reclaimed space that will result as well. The TIER redesign replaces eight ASHPs, with a budget price of $1,840,000, with one TES tank with a budget price of $960,000. And as noted above, the tank is basically free if it can double as the fire water storage tank. Chiller cost per ton is lower for the screw heat recovery machines in this plant than the cooling-only variable speed centrifugal machines, showing that large heat recovery chillers are not necessarily more expensive than their cooling-optimized counterparts.

Condenser water TIER saves space, improves energy and water efficiency and reduces costs relative to a conventional ASHP plant, making it an all-around win for owners and the environment.

Alternative Storage Approaches

While the author believes that condenser water (CW) TES is the most promising TIER approach for most applications, hot water (HW), chilled water (CHW), ice and phase-change materials (PCM) all have their own benefits and shortcomings.

Hot Water Storage

Hot water (HW) storage is perhaps the most intuitive alternative for a system used to solve a heating problem. The energy flows were conceptually shown in *Figure 2*.

A supposed advantage of HW storage relative to condenser water (CW) storage is that it eliminates the cascade chiller configuration and allows for greater morning peak shifting in locales with higher utility rates in the morning warm-up hours where that matters. Eliminating the cascade configuration would seem to yield a significant energy benefit, but in practice the difference is relatively small as *Table 1* illustrates.

Consider an application requiring 40°F (4.4°C) chilled water and 140°F (60°C) hot water. COPh for one manufacturer's 300 ton (1055 kW) heat recovery screw chillers for such an application is approximately 3.5. In a condenser water storage design for the same application, a variable speed cooling-only centrifugal chiller would operate at 40°F (4.4°C) chilled water supply temperature (CHWST) and 80°F (27°C) condenser water return temperature (CWRT), and a heat recovery screw chiller would operate at 60°F (16°C) CWST and 140°F (60°C) hot water supply temperature (HWST). At the noted conditions, the efficiency of a centrifugal chiller might be ~0.30 kW/ton (~0.09 kW/kW) or 12.7 COPh; the COPh of a heat recovery machine might be 4.2. The net process heating COP, therefore, works out to 3.36 with condenser water storage. The condenser water storage approach also requires slightly more pumping energy, but the overall efficiency delta is minor.

Hot water (HW) storage allows nearly full heating peak load shifting because most of the heating peak period loads are met using energy stored in the tank, plus a small amount of trim ASHP capacity. In contrast, a CW storage design only shifts the portion of the energy required to charge the TES outside of the heating peak period. Heating peak load shifting is only of benefit in areas with higher utility rates during the morning peak heating period.

HW TES also eliminates the potential for low heating load heat recovery chiller cycling, which can be an issue with CW TES designs if a HW buffer tank is not provided.

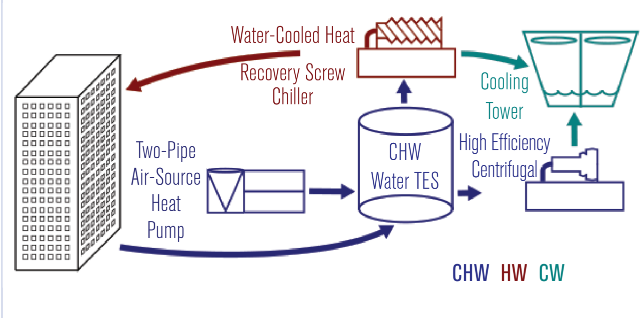
The benefits of HW storage are offset by several drawbacks that must be given close consideration. First, ASHPs must be able to generate hot water at the same temperature as the heat recovery chillers feeding the TES tank. This can be problematic since many ASHPs are limited to a maximum HWST of approximately 120°F (49°C). The ASHPs therefore dictate the maximum design hot water supply temperature, lowering the hot water delta T achievable by the plant. Lower hot water delta Ts require larger tanks, bigger HW pipes and larger pumps. Each of these factors contributes to higher first costs. This issue does not exist with CW storage because the ASHPs reject heat at tepid conditions to a CW storage tank.

Because ASHPs must generate design HW temperature in the middle of winter, the HW TES design may not be viable in very cold climates since many ASHPs cannot produce 120°F (49°C) water at extreme ambient conditions as noted previously. The cascade introduced by CW storage eliminates this issue.

Hot water storage does not allow for demand-based hot water temperature resets. Instead, the hot water supply temperature needs to be fixed at the tank charge temperature throughout the day to maintain proper stratification and ensure the worst-case temperature is always available as demand varies. In a CW storage solution, HWST can be reset based on demand, which should make up for the small full-load efficiency penalty discussed previously.

Last, hot water storage tanks are subject to significantly greater jacket losses than CW storage tanks, which spend many more hours close to neutral relative to ambient in all climates.

FIGURE 10 Direction of heat transfer for a chilled water TIER system.



Chilled Water Storage

Another interesting TIER alternative is chilled water (CHW) storage, for which the energy flows are illustrated in *Figure 10*.

The most compelling reason to consider chilled water TIER is that it integrates exceptionally well with conventional peak shifting schemes. It therefore represents a viable all-electric retrofit strategy for existing campus and district chilled water TES plants. CHW TES also eliminates low-load cooling chiller cycling concerns; CW TES requires a buffer tank to avoid this issue in systems with insufficient base load.

A downside of chilled water TIER is that the storage tank needs to be approximately twice as large as a condenser water tank because the design range is on the order of 20°F to 25°F (11°C to 14°C) instead of 40°F (22°C). This is not an issue if the tank is also designed for cooling peak shifting since that requirement will drive the tank size in many applications; but, it is an issue in non-campus designs where peak shifting is not a primary driver.

Chilled water TIER also prohibits chilled water supply temperature reset when charging the tank since operating at design chilled water range is required to maximize tank storage, maintain stratification and ensure the water stored in the tank is cold enough to serve loads irrespective of varying temperature requirements later in the day. This is in contrast to condenser storage, which allows for demand-based chilled water supply setpoint reset.

Finally, chilled water TES creates a less efficient cascade than condenser water TES any time trim heat is required. This is because the ASHPs end up doing a small fraction of the total required lift (the difference between the saturated suction temperature required to extract heat from ambient air up to the saturated condensing temperature required to reject heat to the heating hot water loop) or create excess "lift overlap." For instance, instead of an air-source heat pump absorbing heat at 32°F (0°C) and rejecting it as 80°F (27°C) condenser water, followed by a heat recovery chiller supplying 60°F (16°C) water from its evaporator barrel and rejecting heat as 140°F (60°C) hot water, an air-source heat pump ends up absorbing heat at 32°F (0°C) and rejecting it as 60°F (16°C) chilled water, followed by a heat recovery chiller supplying 40°F (4.4°C) chilled water and rejecting heat as 140°F (60°C) hot water.

The latter cascade will be less efficient with most equipment and is also problematic for some ASHPs on the market. One market leader's product, for instance, cannot supply water colder than 77°F (25°C). This in turn creates 37°F (21°C) of "lift overlap" (77°F [25°C] on the condenser leaving side of one machine, 40°F [4.4°C] on the evaporator leaving side of the next) where only 20°F (11°C) of overlap needs to exist.

Ice Storage

Ice storage has many of the same pros and cons as CHW storage and conceptually ties into a plant in the same way, so CHW storage serves as a useful point of reference. Ice tanks could replace the CHW tank in *Figure 10* since the energy flows are otherwise identical. The primary benefit of ice storage relative to CHW storage is energy density and therefore space. Because ice storage captures energy in the latent heat of fusion, only roughly 12% and 24% as much volume is required to store energy in ice as in chilled water (20°F [11°C] delta T) and condenser water (40°F [22°C] delta T), respectively. Total floor area savings are, however, not as dramatic as these figures would suggest because ice TES systems are typically broken up into many smaller vessels instead of one monolithic tank.

The primary downsides of ice storage are that it requires adding glycol to the cooling loop to prevent freezing, which reduces heat exchange efficiency and adds a maintenance complication; it requires that "chilled fluid" supply temperatures be below freezing whenever the storage is being charged, thus creating high lift conditions for cooling year-round; and it creates an even less efficient cascade than the chilled water design with more lift overlap.

Phase-Change Materials

Phase-change materials (PCMs) are like ice in that they store energy in the latent heat of fusion. Conceptually, they could replace HW storage, CW storage or CHW storage in any of the schematics in *Figure 2*, *Figure 3* or *Figure 10*. The main benefit of PCMs is that, like ice, they dramatically reduce the TES footprint. The key downside of PCM solutions is that they are typically significantly more expensive than any of the other TES options and therefore may not be life-cycle cost-effective.

Conclusions

Further research is required to investigate the applications and climates for which each of the above storage options is the most life-cycle cost effective. In the meantime, the author encourages designers to begin exploring TIER, especially the novel concept of condenser water TIER, as an option on their all-electric jobs. It is likely that, regardless of the approach taken, TIER will unlock the potential of all-electric solutions for big buildings by improving energy efficiency while reducing costs and spatial requirements relative to typical ASHP designs.

References

1. Jadun, P, C. McMillan, D. Steinberg, M. Muratori, et al. 2017. "Electrification Futures Study: End-Use Electric Technology Cost and Performance Projections through 2050." NREL/TP-6A20-70485. National Renewable Energy Laboratory.
2. ASHRAE Standard 90.1-2019, *Energy Standard for Buildings Except Low-Rise Residential Buildings*.
3. CEC. 2019. "2019 Building Energy Efficiency Standards for Residential and Nonresidential Buildings, Title 24, Part 6." California Energy Commission.
4. Raftery, P., A. Geronazzo, H. Cheng, G. Paliaga. 2018. "Quantifying energy losses in hot water reheat systems." *Energy and Buildings* 179:183–199. https://doi.org/10.1016/j.enbuild.2018.09.020
5. Mai, T., et al. 2018. "Electrification Futures Study: Scenarios of Electric Technology Adoption and Power Consumption for the United States." NREL/TP-6A20-71500. National Renewable Energy Laboratory.
6. MacCracken, M. 2020. "Electrification, heat pumps and thermal energy storage: using yesterday's waste energy for tomorrow's heating." *ASHRAE Journal* 62(7):32–39. ■