The Solution to Large Building Electrification: Heat Recovery Chillers and Condenser Water Storage

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Abstract

Transitioning space heating from fossil fuels to some form of electric heating is critical for decarbonization. Air-source heat pumps (ASHP) are a reasonable electrification option for small and medium sized buildings but not for large buildings given the high first cost and large outdoor footprint of ASHP. For most new construction large buildings the lowest cost and most efficient electrification option is Time-Independent Energy Recovery (TIER). TIER combines trim ASHPs with condenser water thermal energy storage (TES), and heat recovery chillers (HRC). Most of the heating loads are met by the HRC, which are about twice as efficient as ASHP. The TES allows the HRC to recover heat even when heating and cooling loads are not simultaneous. It can also reduce the peak load on of the ASHPs by about 80%, which makes TIER less expensive and smaller footprint than a conventional heat pump system. This paper compares TIER with other all-electric options and with TES systems that employ other storage options, including chilled water, hot water, and ice storage.

Heating Options

Historically large buildings have typically used natural gas for heating. Recently electric heating has become more common to meet decarbonization goals and because fossil fuel heating is now prohibited for new construction in many jurisdictions. There are currently four primary options for generating heat using electricity for large buildings such as offices, laboratories and universities:

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

Each of the above options has significant challenges related to equipment and installation costs, spatial requirements, energy efficiency, and carbon emissions.

Air-Source Heat Pumps (ASHPs)

Air-source heat pumps (also called air-to-water heat pumps) are probably more carbon-friendly than electric boilers or electric resistance coils[1]. Based on several equipment selections recently provided by vendors to the authors for projects on the boards, a typical heating coefficient of heating performance (COPh) is 2.1 when generating 120°F (49°C) water at 29°F 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, efficiency and capacity both drop rapidly as ambient temperature falls.

ASHPs are also very expensive per unit capacity (roughly $150 to $200/MBH vs $15 to $30/MBH for high quality condensing gas boilers) and, because they use ambient air to extract heat, 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 need a primary pump dedicated to each unit or bank of modules to meet their very high minimum flow rates, further adding to first costs. ASHP plants are also likely to experience higher ongoing maintenance costs than heat recovery chillers or electric resistance because of the quantity of devices involved and the complexity of the equipment itself; large ASHPs typically have 4 to 6 scroll compressors, at least 2 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, thereby 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 to 10 EER (1.2 to 1.3 kW/ton, 2.8 to 2.9 COP) at AHRI 550/590 conditions. Contrast this with a well-designed water-cooled chiller plant that operates at about 0.60 to 0.65 kW/ton (5.4 to 5.9 COP) at design conditions, including condenser water pumps and cooling towers. This reality makes it almost impossible to comply with either ASHRAE 90.1 or California Title 24 using the performance approach when replacing water-cooled plant cooling capacity with ASHP capacity since the baseline cooling system for large buildings with both standards is a chiller plant with water-cooled variable speed centrifugal chillers.

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 hot water supply temperatures of around 140°F (60°C), electric boilers can generate 180°F (82°C) like conventional natural gas boiler plants,

[1] Note that ASHPs described herein are very different from the small single zone air-to-air heat pumps used in residential (no hot water piping). These are large air-to-water heat pumps for generating hot water which is then piped to many zones.

and thus can benefit from the higher hot water ΔTs (e.g., 40°F, 22°C) and smaller pipe and pumps sizes that result from supplying hotter water.

A major benefit of zone level electric resistance heating coils is that they eliminate parasitic pipe heat losses inherent to all water-based designs. Recent research by Raftery et.al. (2018) found that piping losses can be as high as the amount of heat needed to condition the space.

Both electric resistance design strategies are, however, limited by thermodynamics to a peak COPh of 1. Even in 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 upsized at considerable expense.

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. This only works when there are simultaneous heating and cooling loads, but that typically 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 warmup) 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.

The Solution

The key to solving these issues 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. There are multiple versions of thermal energy storage systems, including:

- Condenser water storage (stratified and unstratified)
- Hot water storage
- Chilled water storage
- Ice storage
- 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 common components: 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 very cold climates or where roof space is limited). When 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 unintuitive, 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. Hot water storage can be used to illustrate this concept. In a design with a HW 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 *1* illustrates the energy flow paths for the HW storage system design.

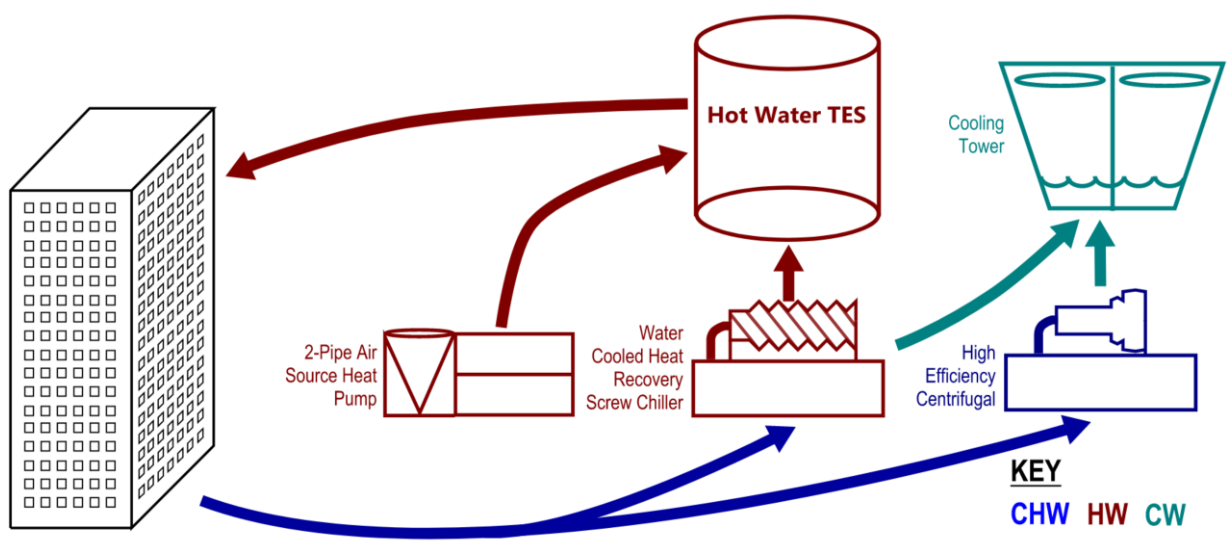

Figure 1. Direction of heat transfer for a hot water tier system

In a condenser water storage design, trim air-source heat pumps, 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 months, the heat recovery chillers, can be indexed to the chilled water loop to provide cooling. Figure *2* illustrates the energy flow paths for the condenser water storage system design.

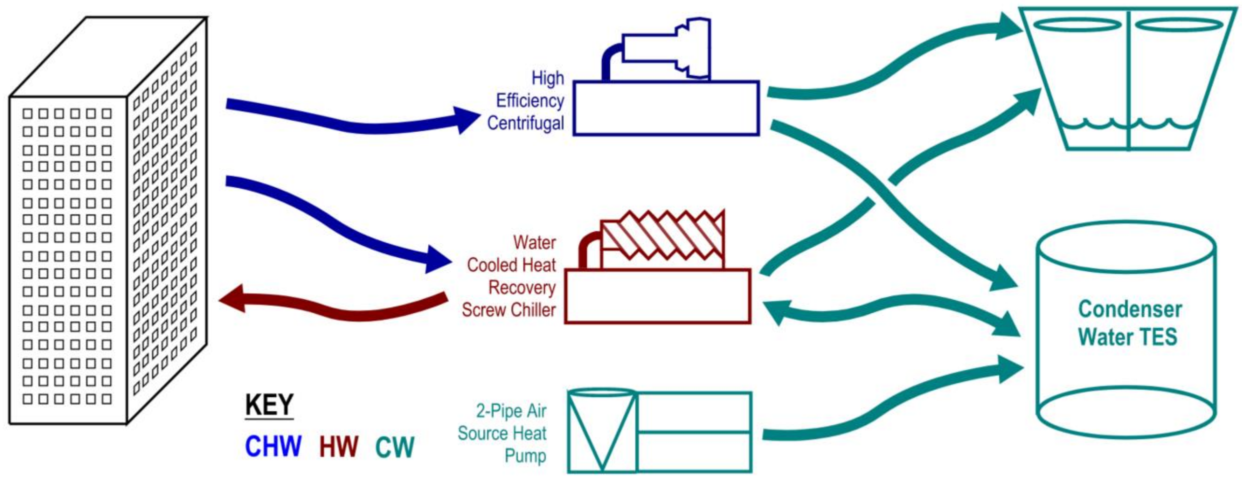

Figure 2. Direction of heat transfer for a condenser water TIER System

Understanding Condenser Water TIER

The remainder of this paper 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 paper.

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 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 mild coastal climate zones like California the energy recovered from cooling loads alone can satisfy heating loads. During the small fraction of the year when heat recovery alone cannot satisfy heating demand, trim ASHPs (or electric boilers) are used to charge the storage tank.

The schematics below, which are simplified and adapted from a project for which we employed 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, condenser water, and hot water are traced in each.

Figure 3 illustrates a typical cold morning operation condition during which the TES tank discharges. All the 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 air source heat pumps, 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).

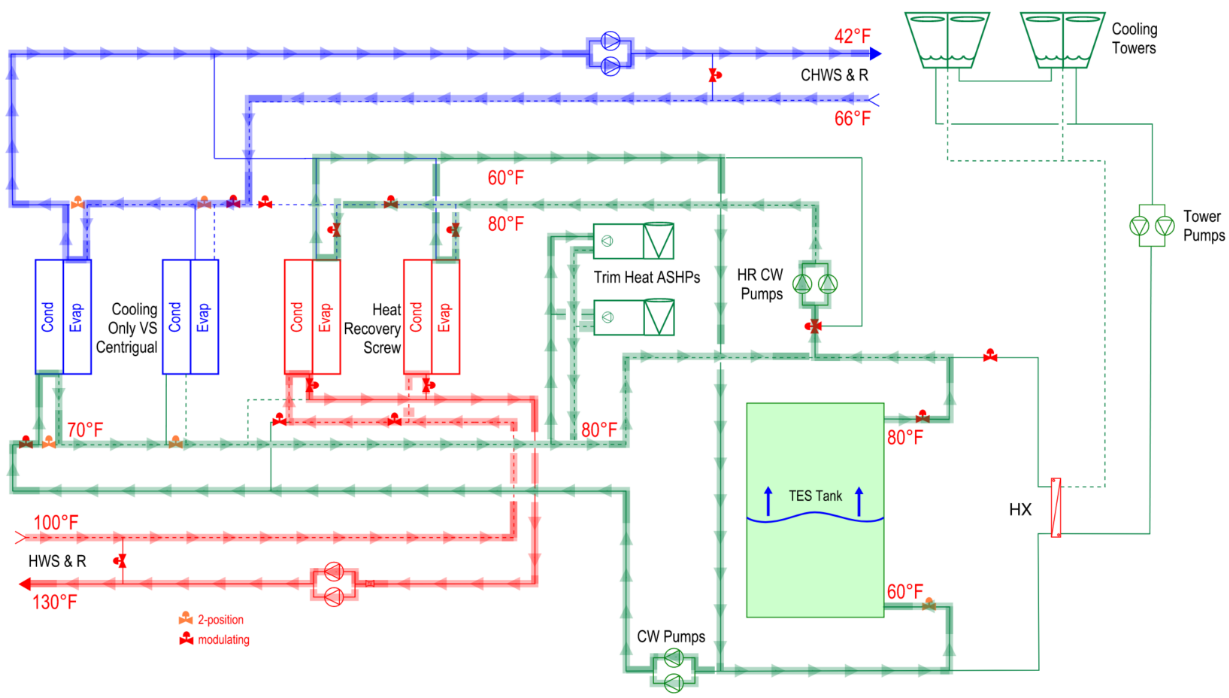

Figure 3. Cool day morning operation of a condenser water TIER System

Later during the same day, when heating loads decrease and cooling loads increase, the net result is that the tank charges (increasing in temperature). During the example condition in Figure 4, 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

80F.

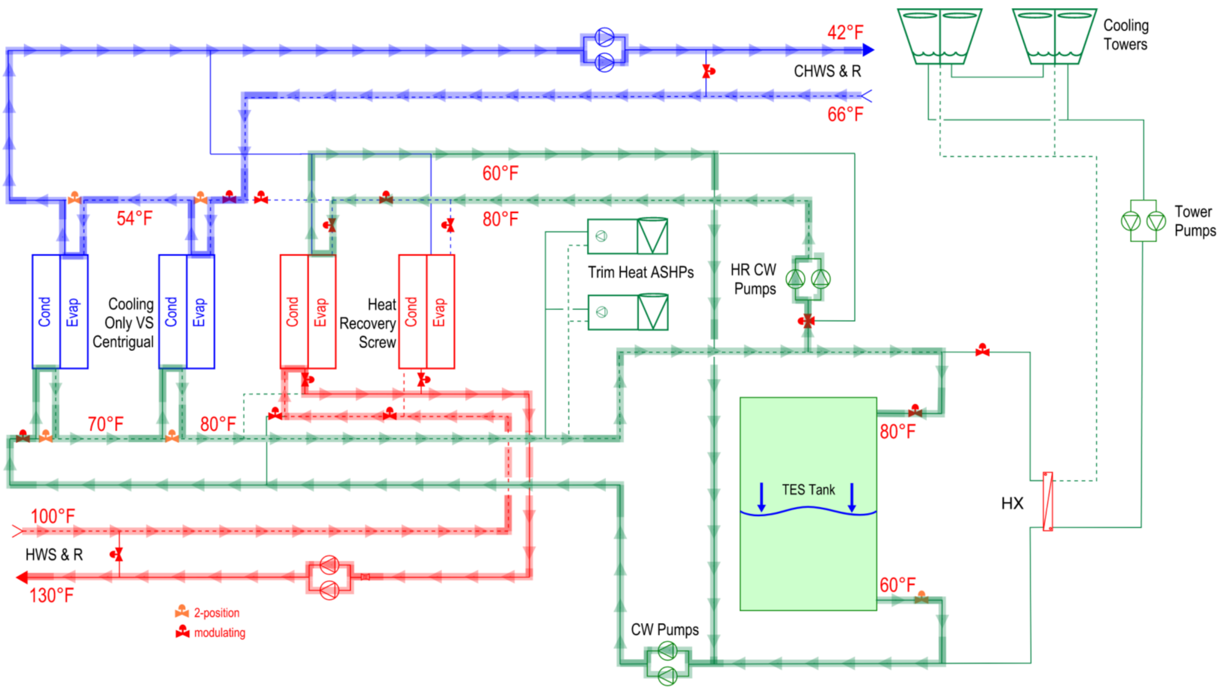

Figure 4. Cool day afternoon operation of a condenser water TIER System

Figure 5 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 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.

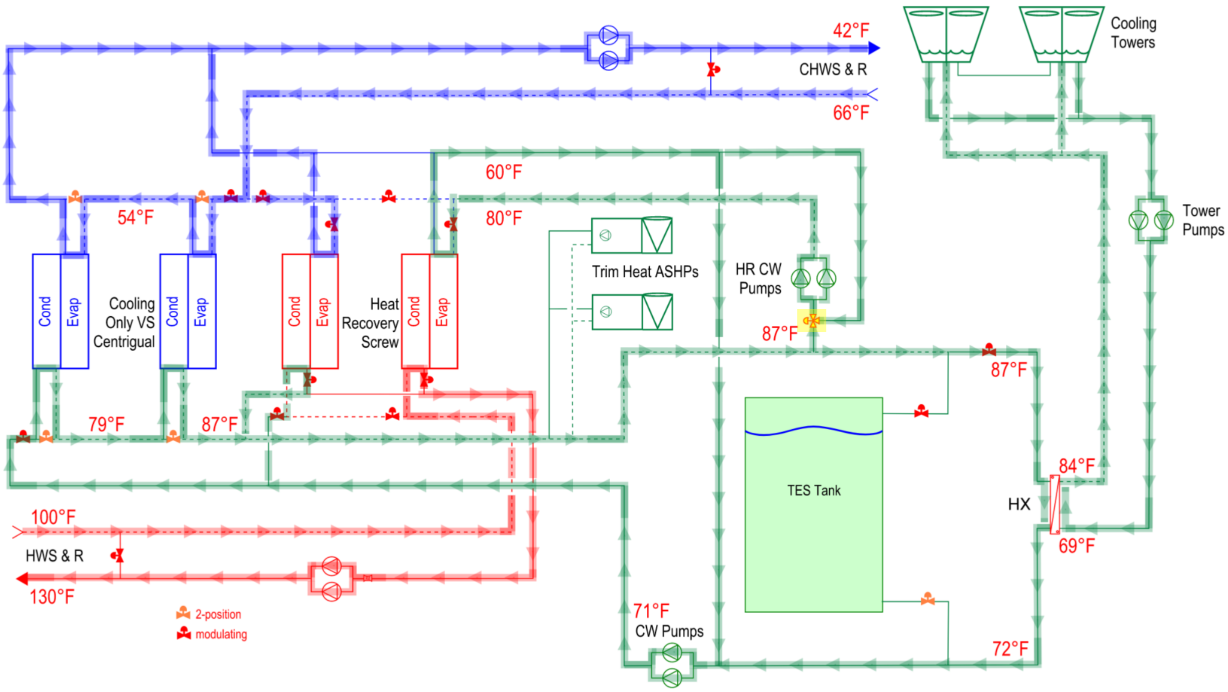

Figure 5. Warm day afternoon operation of a condenser water TIER System

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 change tank size by adjusting heat source capacity. More heat source capacity allows for smaller tanks 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 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 provide that capacity over a 24-hour period.

In the real example below, two ASHPs totaling 3,530 kBtu/h of capacity were proposed along with a condenser water tank providing 31,200 kBtu of storage for a 1.1M ft^2 office/dry computer lab complex with a design heating load of approximately 16,000 kBtu/h. A 110,000 gallon, 50' tall, 20' diameter tank was selected for the project. The TIER design allowed us to provide (2) ASHPs totaling 3,530 kBtu/h where (10) ASHPs totaling over 16,000 kBtu/h would have otherwise been required. The relative footprints of these two designs are shown below in Figure *6* and Figure *7*.

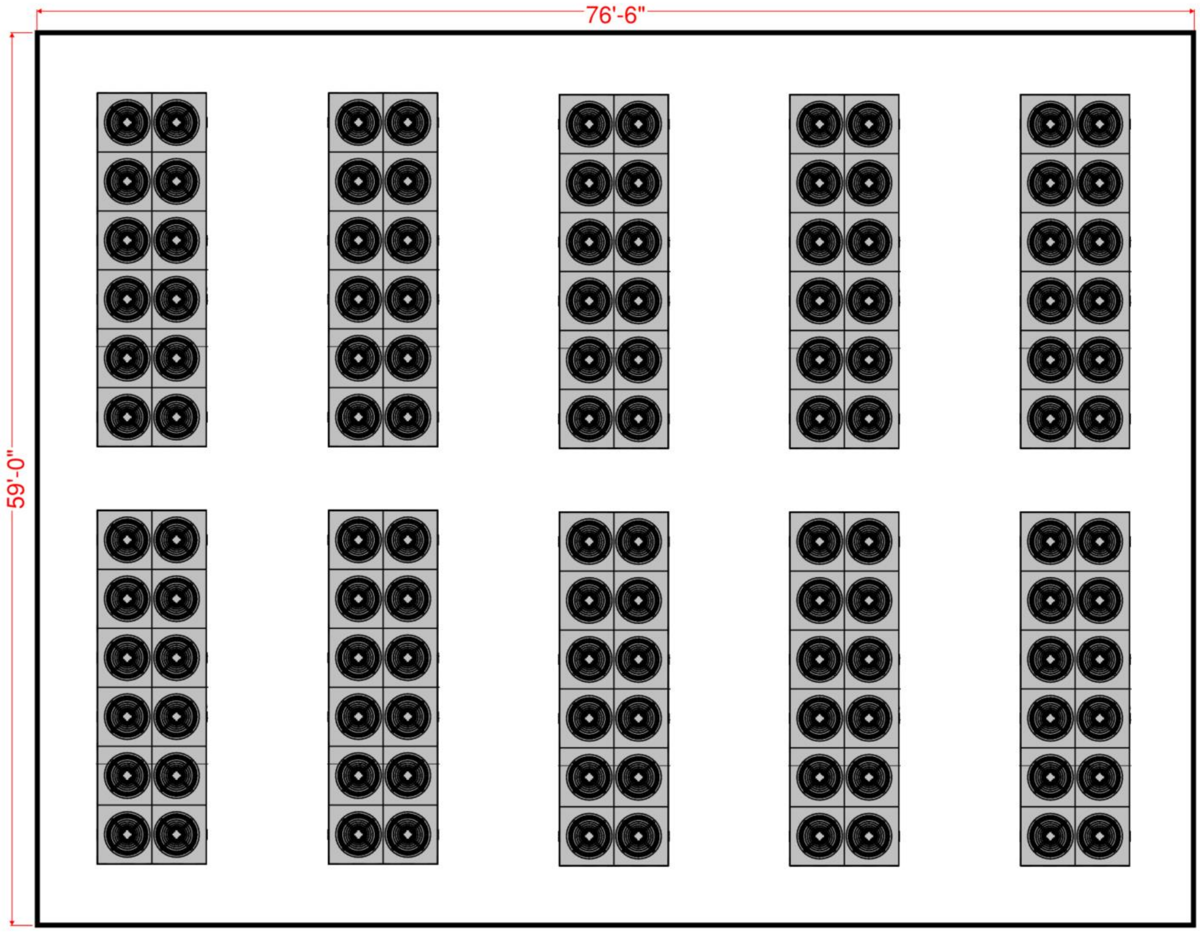

Figure 6. Footprint of conventional ASHP farm

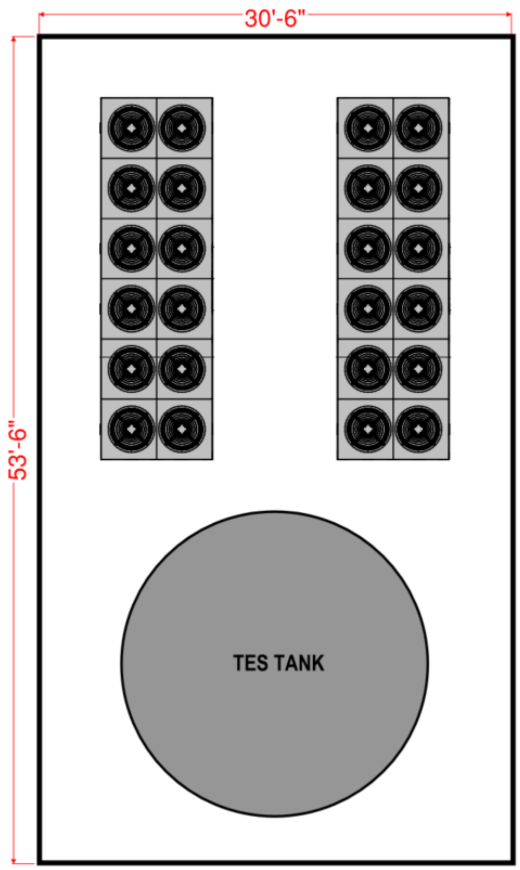

Figure 7. Footprint of TIER TES tank and ASHP alternative

TIER TES tanks are 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) so finding an optimal location for the tank can be a challenge. A great option is locating tanks in parking garages. Typically, the TES tank is smaller than the fire water storage tank needed for a high rise. We have been approved on a few projects 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, include HW and CHW: while a HW or CHW TES tank's capacity is limited by the ΔT of the loads it serves, a condenser water tank serves as a source for heat recovery chillers, so it can have a much higher ΔT. For a CHW TES system, ΔT typically ranges from18°F to 25°F (11°C to 14°C). A HW TES tank storing 130°F (54°C) hot water might similarly be limited to a 30°F (17°C) ΔT. Furthermore, both CHW storage and HW storage are vulnerable to "Degrading Delta-T Syndrome" (See Taylor, 2002). Degrading dT means the actual dT in normal operation is less than the design dT (due to 3-way valves and other flaws) and thus the storage capacity of the tank is reduced. A condenser water TES tank by contrast can readily be sized for 40°F (22°C) ΔT or more and is immune from Degrading Delta-T. The tank is intended to operate with a 20°F (11°C) ΔT 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. However, on design heating days, the tank can cycle through one more time down from 60°F (16°C) to 40°F (4.4°C). In other words, on cold days we can pull more heat out of the tank by running the heat recovery chiller until the tank is 40F. So we start with an 80F tank and end with a 40F tank. The overall ΔT 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; 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 net COPh is therefore roughly 3.7. 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 importantly, 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 byproduct of the heating process. The typical paradigm is to view the recovered heat from heat recovery chillers as a "free" byproduct 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".

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 there may be no cooling energy to recover and 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. Condenser water TIER was bid as an alternate system design option versus Conventional ASHPs on four recent Bay Area new construction projects. Table 1 shows the actual first cost savings of the TIER design based on GC bids. 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.

Table 1. Cost savings of TIER vs conventional ASHPs

Location	Santa Clara	Sunnyvale	San Jose	Oakland
Stories	3	3	6	27

Building area (ft2)	314,000	1,100,000	1,022,981	718,000
CHWcap (tons)	780	2,660	1,800	1,200
Tank capacity (gallons)	35,000	141,000	TBD	53,000
Tank doubles as fire water storage?	No	No	Yes	Yes
First Cost Savings ($)	*	1,500,000	6,725,003	2,200,000
First cost savings ($/ft2)	*	$1.36	$6.57	$3.06
*For the Santa Clara site, TIER was the base bid. The GC indicated that ASHPs was a net cost add but did not provide a hard bid, i.e., TIER was lower cost. The owner opted for TIER since it was lower cost, lower energy use, and lower maintenance.				

Alternative Thermal Energy Storage Approaches

Hot Water Storage

The energy flows for HW storage were conceptually shown in Figure 1 above. A supposed advantage of HW storage relative to CW storage is that it eliminates the cascade chiller configuration. Eliminating the cascade configuration would seem to yield a significant energy benefit, but in practice the difference is relatively small as Table 2 illustrates.

Table 2: Condenser water and hot water TIER design lift heating efficiency comparison

	Condenser Water Heat Recovery			Hot Water Heat Recovery		
Device	Source (°F/°C)	Sink (°F/°C)	COPh	Source (°F/°C)	Sink (°F/°C)	COPh
Cooling Only Chiller	40 (4.4)	80 (27)	12.72	-	-	-
Heat Recovery Chiller	60 (16)	140 (60)	4.2	40 (4.4)	140 (60)	3.5
Net Heating Efficiency			3.36			3.5

HW storage also 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 the same hot water temperature as the heat recovery chillers since they feed the same TES tank. This can be problematic since many ASHPs are limited to a maximum hot water supply temperature (HWST) of approximately 120°F (49°C). Lower hot water ΔTs 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.

HW 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.

HW storage tanks cannot be used as fire water storage. Lastly, HW 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. Thus HW storage is considerably more expensive and less efficient than CW storage.

Chilled Water Storage

Another interesting TIER alternative is chilled water storage, for which the energy flows are illustrated in Figure *8* below.

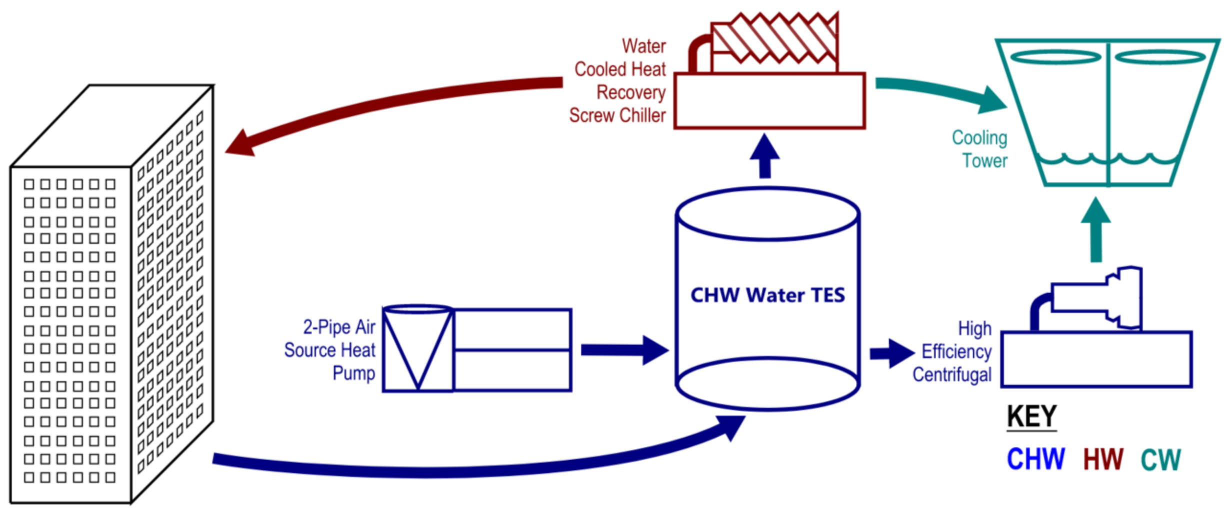

Figure 8. Direction of heat transfer for a chilled water TIER System

Many large campuses have existing chilled water storage tanks because utility rates encouraged generating chilled water at night when rates are low and not running chillers in during hot days when rates are high. Chilled water TIER integrates well with this conventional peak shifting but as the generation mix includes more solar power, conventional peak shifting is less desirable for utilities. In the future CW or HW storage will be better suited for peak shifting rather than CHW storage.

CHW TES 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 delta-T 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. In contrast, condenser storage allows for demand based chilled water supply setpoint reset.

Lastly, 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 creating 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 (31°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 overlaps 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 *8* 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, ice only requires about one fourth the volume of CW storage and only about one eighth the volume of CHW storage.. 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 introduction of glycol to the cooling loop to prevent freezing, which reduces heat exchange efficiency and adds a maintenance complication; 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, CHW storage in any of the schematics in Figure 1, Figure 2, or Figure 8 above. 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 significantly more expensive than any of the other TES options and therefore may not be life cycle cost effective.

Heat Recovery Chiller and CW TIER Retrofits

Heat recovery chillers and/or CW TIER can dramatically reduce or eliminate natural gas use in thousands of existing buildings and can do so more cost-effectively than other electrification options, such as ASHPs or electric boilers, in most cases. Figure 9 shows the heat flows for a proposed CW TIER retrofit at a manufacturing facility in San Diego. There was a perfect spot near the existing cooling towers to add a 50,000 gallon CW tank ($200k installed price), a heat exchanger, and a heat recovery screw chiller, capable of supplying 170ºF hot water. For simplicity and cost the new HRC is not connected to the chilled water system. This design is predicted to reduce space heating gas use by over 90%.

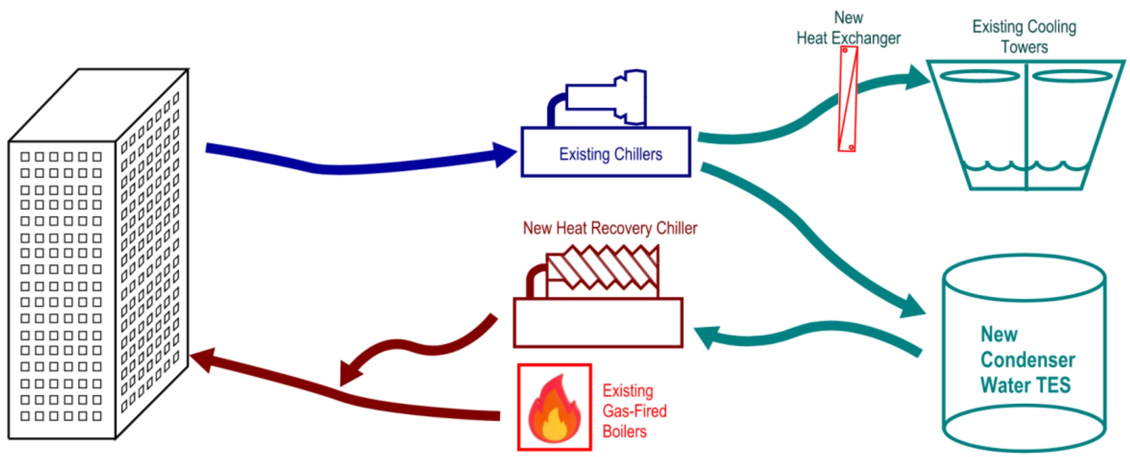

Figure 9. CW TIER retrofit

Figure 10 shows the heat flows for a proposed heat recovery chiller retrofit at a hospital in San Francisco. A new HR chiller and a new exhaust air heat recovery coil are proposed. In cold weather, when the air handlers do not use chilled water, the HR chiller will cool the exhaust air and thus recover energy to heat the building. There was no available space to locate a thermal energy storage tank but there was also no need for one: the HRC will reduce annual space heating gas use by over 90%. Without the heat recovery coil annual space heating gas can still be reduced by 60%.

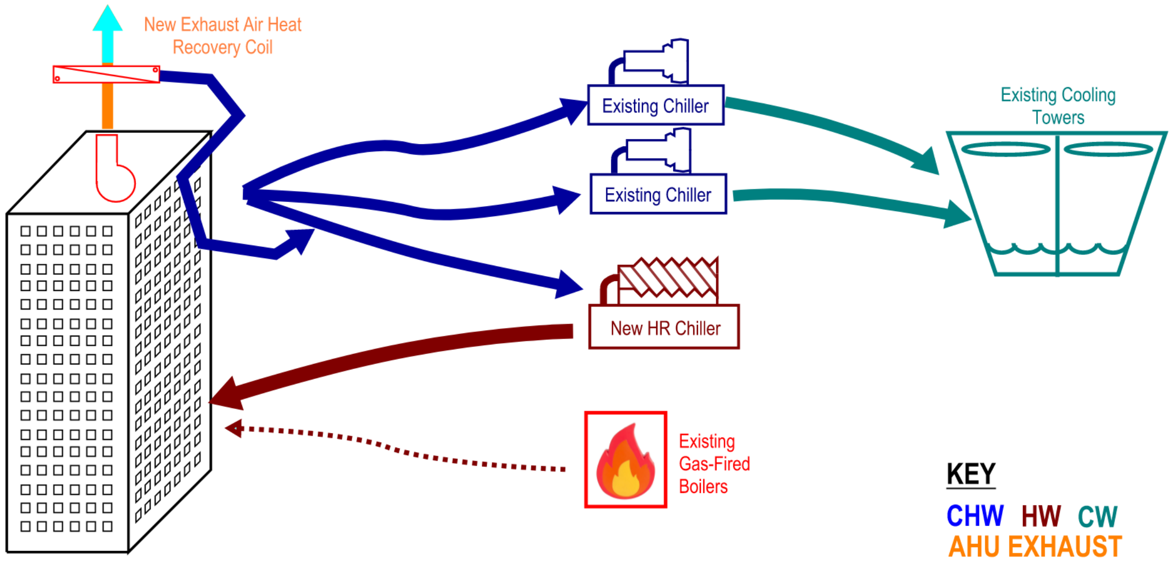

Figure 10. Heat s

Conclusion

Time-Independent Energy Recovery (TIER) using heat recovery chillers and condenser water storage is the lowest cost and most efficient option for electrification of most new construction, large commercial buildings (e.g. above 200,000 ft^2). TIER is not the best electrification option for most small commercial buildings (single zone heat pumps are often the best option). Heat recovery chillers and/or CW TIER can also reduce or eliminate natural gas use in large existing buildings and can do so more cost-effectively than other electrification option, but full electrification of existing buildings is generally not cost-effective. A far more cost-effective option for reducing gas use in existing buildings is retro-commissioning with ASHRAE Guideline 36 sequences and replacing oversized non-condensing boilers with smaller condensing boilers (Raftery 2024).

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