Large District Plant

Commissioning an Existing Heat Recovery Chiller System

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A heat recovery chiller (HRCHLR) is a special type of chiller. A chiller typically operates with a condenser water supply temperature of 55°F to 85°F (13°C to 29°C) and a condenser leaving water temperature of 60°F to 95°F (16°C to 35°C) (Heemer et al. 2011). This leaving water temperature is too low to be used effectively as a heating source for heating, ventilation, and air conditioning (HVAC) loads. However, a heat recovery chiller can be used to generate chilled water and work at condenser water temperatures of 155°F (68.3°C) and higher, which makes it possible for condenser water to be used as a heating hot-water source in HVAC systems.

The use of recovered heat from chillers can be an efficient and cost-effective means of providing heating hot water (HHW) or domestic hot water to buildings (Johnson 2007; Temos 2006). A typical utility plant includes both chillers and boilers running year round to meet the building's heating and cooling demands. A heat recovery chiller may be installed to allow for the recovery of waste heat from the chiller condenser water to in order to generate heating hot water for plant heating in addition to generating useful chilled water. This allows for a reduced load on hot-water boilers, cooling tower fans, and cooling tower water makeup.

However, not every central plant is a candidate for installing heat recovery chillers. Coincident heating hot-water and chilled- water loads are required unless thermal storage tanks are used or alternative heat sources are available (e.g., sea/lake/river water and geothermal). Esen et al. (2017), Yang et al. (2015), and Dai et al. (2015) studied the performance of solar assisted ground source heat pump systems. Liu et al. (2017) evaluated the feasibility of a hybrid ground-source heat pump system for an office building in a heating dominated climate zone in China. Luo et al. (2015) analyzed the performance of a ground-source heat pump system in southern Germany. Sebarchievici and Sarbu (2015) conducted a study of an experimental ground-coupled heat pump system for heating, cooling, and domestic hot-water application. Wu et al. (2014) simulated the performance of a ground-source absorption heat pump to produce heating, cooling, and domestic hot water. Buker and Riffat (2016) conducted a systematic review of solar assisted heat pump systems for low temperature water heating applications.

Both cooling and heating demands are required simultaneously to make the installation of a heat recovery chiller applicable. ASHRAE's Chiller Heat Recovery

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Application Guide provides a comprehensive reference manual for the design of heat recovery chiller systems. The information in the guide assists engineers, owners, and system operators in evaluating the potential of integrating chiller heat recovery into their cooling and heating systems. The primary focus of the guide is on new construction and applications where a chiller is being replaced due to inefficiency and high operating and maintenance costs.

Campbell et al. (2012) provide guidelines for building owners and designers who are interested in applying large- capacity water-to-water heat pumps in their facilities. Hubbard (2009) discusses several system design aspects of heat pump application, including energy consumption calculations, green technology benefits, and capital constraint issues. Although the electrical demand charge is a significant portion of most commercial electric utility bills, a blended electricity rate was used for cost savings calculations to simplify the calculation. Heemer et al. (2011) discuss various design issues associated with the heat pump application that need to be given serious consideration because they are not typical of plant design without a heat pump. They state that one critical design issue is to size the heat pump properly by looking at coincident chilled-water and heating hot-water loads on an annual basis. For simplicity, a blended annual electrical rate was used in the paper to show potential savings of a heat pump installation. Using a blended rate is acceptable for the design analysis, but variation in electric utility rates should be reviewed carefully for optimizing the heat recovery chiller performance.

Liu (2013) presented an approach for controlling outdoor air economizer and energy recovery systems based on minimizing the total energy input to the HVAC system, including chiller, pumps, and fans. Gong et al. (2012) developed a thermodynamic simulation model by using a popular toolbox to simulate the performance of a single-stage centrifugal heat recovery chiller. The theoretical analysis showed the waste heat recovery technique to be a type of sustainable energy technique. Edwards and Finn (2015) developed a control strategy to predict optimal ground-source heat pump water flow rates under part-load operation. Montagud et al. (2014) conducted an experimental study of the influence of the water circulation pump's frequency on the indoor and outdoor loops in the overall system performance of a ground-source heat pump plant.

Most existing studies of control strategies concentrate on other types of heat recovery or heat pump applications. Only a few researchers have focused on how to optimally design a utility plant with heat recovery chillers. They give virtually no guidance to operators on running a heat recovery chiller system efficiently at different operating conditions. Factors that need to be considered include summer demand charges and utility rate structure, heating hot-water supply temperature (HHWST) reset schedule, heating hot-water loop ΔT degeneration, chilled-water (CHW) loop ΔT degeneration, and the performance of other variable-speed drive (VSD) chillers in the plant. Tremblay and Zmeureanu (2014) developed benchmarking models using measurements from the building automation system (BAS) for the ongoing commissioning of the heat recovery process in a cooling and heating plant. The performance indices of the heat recovery process were compared with benchmarks to evaluate system performance.

This article presents several energy conservation measures (ECMs) for a heat recovery chiller application at a large district plant in central Texas. The measured data from the BAS are used to develop a regression model to evaluate the savings potential for each ECM.

TABLE 1 Design specifications of heat recovery chiller.

NO.	ITEM	VALUE/UNIT
1	Heating Capacity	20 MMBtu/h
2	Cooling Capacity	1,178 tons
3	Input Power	1,812 kW
4	COP (Heating Only)	3.235
5	COP (Heating and Cooling)	5.521
6	CHW Flow	2,348 gpm
7	HHW Flow	2,039 gpm
8	HHW Entering Temperature (ET)	135°F
9	HHW Leaving Temperature (LT)	155°F
10	CHW LT	42°F
11	CHW ET	54°F

Design Information

Heat Recovery Chiller

A centrifugal heat recovery chiller is used in a large district plant to produce heating hot water and chilled

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FIGURE 1 COP of case study HRCHLR at different heating partial-load ratios (PLRs).

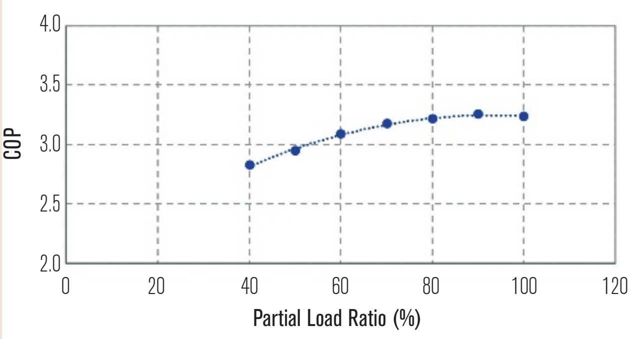
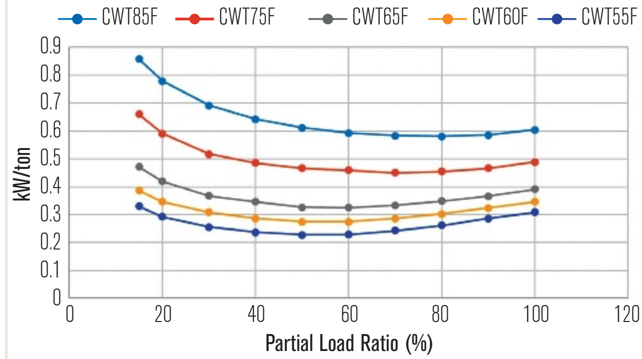


FIGURE 2 Conventional chiller performance curve.



water simultaneously for campus buildings. The design specifications of the centrifugal heat recovery chiller are shown in *Table 1,* Page 45.

According to the heat recovery chiller submittal document, the manufacturer suggests the minimum heating load ratio be 40%; otherwise the HRCHLR may be suffer surge and low efficiency. *Figure 1* presents the design coefficient of performance (COP) curve when the heating load ratio is between 40% and 100%.

Conventional Chiller Performance Curve

A conventional chiller model is used to calculate the electricity consumption for producing the same amount of CHW as the HRCHLR produces. The design performance curve of the conventional chiller is shown in *Figure 2*.

Piping Arrangements

A simultaneous heating and cooling application can have many options for piping arrangements, such as heat pump side-stream to boiler, heat pump parallel to boiler, hybrid side-stream/parallel system, etc. The case-study project uses a hybrid side-stream/parallel system as shown in *Figure 3*.

This system can be used in either heating priority or cooling priority mode, depending on the specific requirements, and make very effective use of the HRCHLR for heating and cooling.

HRCHLR Sizing Strategy

A HRCHLR condenser must always be able to fully reject the heat absorbed in the evaporator plus the compressor work. For this reason, a heat pump should never be oversized for the load. If the facility's heating requirement is insufficient to accept the minimum heat output of a given heat pump, the temperature in the hot-water loop will rise uncontrollably until the heat pump shuts down. In simultaneous heating and cooling applications, the sizing of the heat pump requires careful analysis. Consider a facility with year-round heating and cooling requirements. *Figure 4* shows the heating and cooling load profiles, plus the curve of heat rejection from the condenser, which is the sum of the cooling load and the compressor work. Campbell et al. (2012) suggested the heat pump design capacity be selected as Point A in *Figure 4*.

FIGURE 3 HRCHLR system diagram.

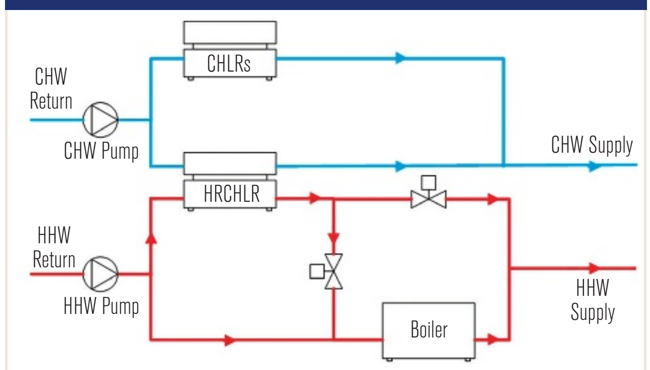


FIGURE 4 Selection point for the simultaneous heating and cooling heat pump (Campbell et al. 2012).

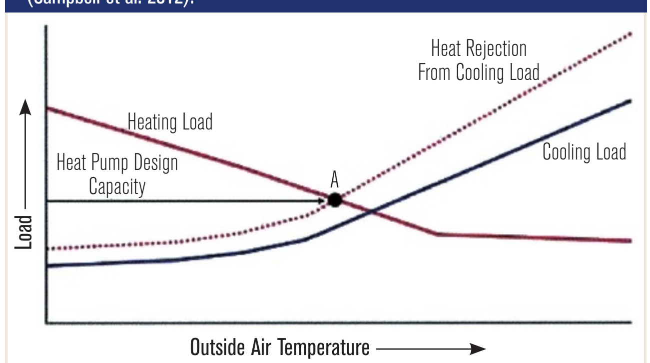


FIGURE 5 Campus heating and cooling load regression models.

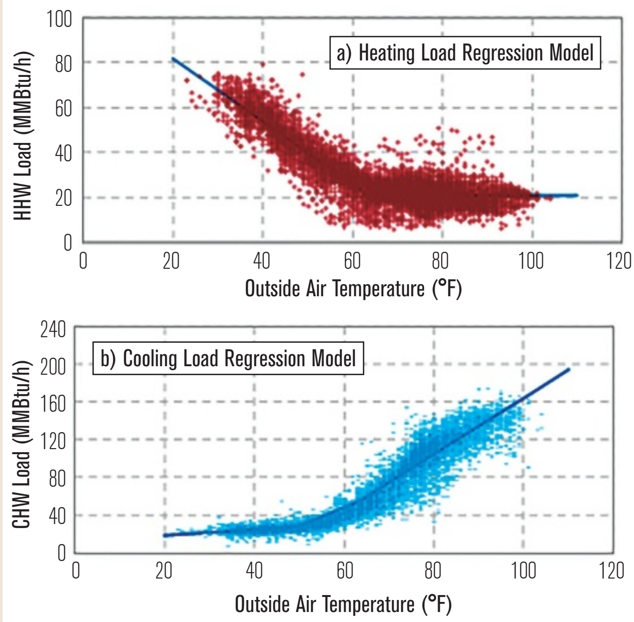
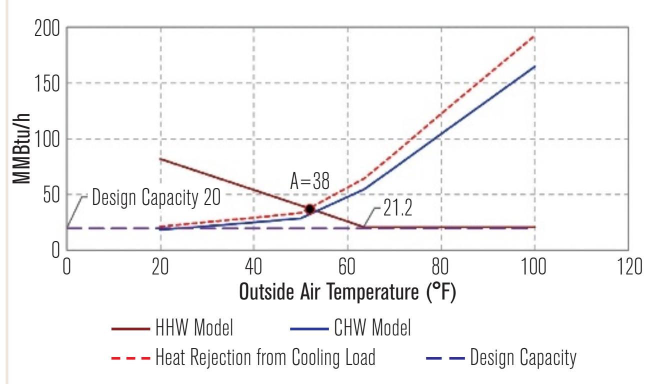


Figure 5 presents the measured data from Sept. 1, 2014 through Aug. 31, 2015 and models for HHW load and CHW load, respectively. The measured parameters and the range of their values are as follows:

- Outdoor air temperature: 23°F and 104°F (–5.0°C and 40°C);
- CHW load: 8.05 and 175.8 MMBtu/h (2360 and 51,524 kW); and
- HHW load: 6.1 and 79.2 MMBtu/h (1794 and 23,220 kW).

FIGURE 6 Design point of HRCHLR.



Because the plant services different types of buildings, such as offices, research laboratories, data server rooms, etc., the campus has cooling load in winter and heating load in summer. The design capacity of the HRCHLR is 20 MMBtu/h (21.1 GJ/h), which is less than the Point A value of 38 MMBtu/h (40.1 GJ/h) suggested by Campbell, et al. (2012) (*Figure 6*).

Campbell, et al. (2012) suggested that a HRCHLR should never be oversized for the load. In order to run the HRCHLR at full load as much as possible, the designers sized the case-study HRCHLR capacity equal to the base heating load in summer. The expected annual energy cost savings using baseline operating practices of maintaining the HHW supply temperature at 155°F (68.3°C) and condenser water flow at design of 2039 gpm (128.7 L/s) is $342,060, as shown in *Table 2*.

TABLE 2 Predicted HRCHLR energy cost savings using baseline operating practices.

MONTH	NATURAL GAS RATE, $/MMBTU	ELECTRICITY RATE $/KWH	HHW MMBTU	CHW TON/H	ELECTRICITY INPUT KWH	HRCHLR COST $	CHLR + COOLING TOWER KWH	BOILER NATURAL GAS MMBTU	CHLR + NATURAL GAS BOILER $	ENERGY SAVINGS $
9	4.257	0.0503	14,400	848,160	1,304,640	$65,623	504,415	17,143	$98,349	$ 32,726
10	4.257	0.0441	14,880	876,432	1,348,128	$59,452	435,671	17,714	$94,623	$35,170
11	4.257	0.0516	14,400	848,160	1,304,640	$67,319	314,117	17,143	$89,186	$21,866
12	4.257	0.045	14,880	876,432	1,348,128	$60,666	324,921	17,714	$90,031	$29,365
1	4.257	0.0466	14,880	876,432	1,348,128	$62,823	289,786	17,714	$88,914	$26,091
2	4.257	0.0491	13,440	791,616	1,217,664	$59,787	280,910	16,000	$81,905	$22,117
3	4.257	0.0538	14,880	876,432	1,348,128	$72,529	351,216	17,714	$94,305	$21,776
4	4.257	0.0375	14,400	848,160	1,304,640	$48,924	435,160	17,143	$89,296	$40,372
5	4.257	0.0404	14,880	876,432	1,348,128	$54,464	495,059	17,714	$95,410	$40,946
6	4.257	0.0494	14,400	848,160	1,304,640	$64,449	528,521	17,143	$99,086	$34,637
7	4.257	0.0734	14,880	876,432	1,348,128	$98,953	571,275	17,714	$117,341	$18,389
8	4.257	0.0716	14,880	876,432	1,348,128	$96,526	554,767	17,714	$115,131	$18,605
Annual	4.257	0.0511	175,200	10,319,280	15,873,120	$811,517	5,085,819	208,571	$1,153,576	$342,060

However, the performance of the HRCHLR is also impacted by other factors in addition to the HRCHLR capacity, such as:

- Heating hot-water supply temperature;
- Turndown (available load reduction);
- Heating water loop ΔT;
- Maximum condenser and evaporator water flows; and
- Electrical rates and demand charges.

In the following sections, the details for each factor are discussed, and ECMs to optimize HRCHLR operation are presented.

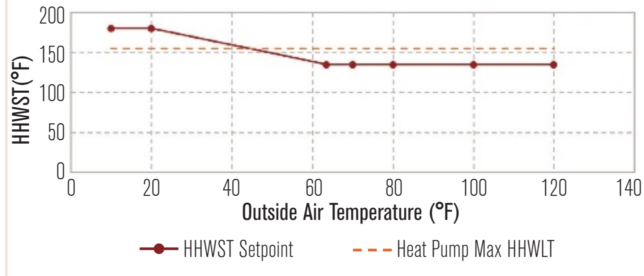
Heating Hot-Water Supply Temperature Reset

Typical HVAC heating hot-water temperature for conventional plants is 180°F (82°C) supply and 160°F (71°C) return. At these temperatures, it is difficult to find a heat pump that will be effective from a combined capital cost and operating cost standpoint. The lower the HHW temperature, the more applicable a heat pump is, because efficiency increases as HHW temperature drops. However, as temperature of the hot water drops, additional hot-water coil surface is required to heat the supply air to the temperature required for heating the space. For an existing heating system, it is impossible to increase coil surface, so an alternative solution to increase efficiency is to reset HHWST during partial load as shown in *Figure 7*.

When the outdoor air temperature (OAT) is higher than 45°F (7.2°C), the heat pump can operate parallel to boilers. When the HHWST setpoint is higher than 155°F (68.3°C), the boilers need to be used to boost the HHW temperature. If the HHWST setpoint is reset to a lower temperature, the heating hot-water return temperature (HHWRT) may be below the condensing temperature. For a noncondensing boiler, a means of protecting the boiler must be provided. For the case-study plant, a mixing valve is used to prevent the boiler entering water temperature from dropping below 135°F (57.2°C), and a loop bypass valve is used to maintain the loop HHWST at a lower setpoint.

The heat pump performance can be impacted by the PLR and the lift temperature. Although reducing the HHWST can reduce lift temperature, it also reduces the HHW loop ΔT and heat pump load ratio, which reduces heat pump COP and increases distribution pump energy consumption. Therefore, a trade-off needs to be conducted to optimize the HHWST reset schedule.

FIGURE 7 HHWST setpoint reset schedule.



Turndown and Heating Water Loop Heating Hot-Water Loop ΔT

Because a heat pump operates at essentially constant lift (meaning the entering CHW temperature and the leaving heating hot-water temperature [LHWT] remain relatively constant), the turndown (available load reduction) can be significantly less than the turndown for water chillers. For centrifugal compressors, the minimum capacity may be only 40% to 50% of the design heating load before the refrigerant flow may become unstable, otherwise known as "surge." The minimum capacity requires the return temperature of the HHW loop to be lower than 147°F (63.9°C) at the design flow of 2039 gpm (128.7 L/s) for the case-study chiller, with a leaving temperature of 155°F (68.3°C).

The HHW loop return temperature is impacted by many factors, such as the HHWST, heating load, heating coil design ΔT, etc. For a central HHW loop, a good solution is to create a regression model based on the historical loop data. The measured parameters and the range of their values of HHW loop data from 9/1/2014 through 8/31/2015 are as follows:

- HHWST: 123.5°F and 171.9°F (50.8°C and 77.7°C); and
- HHWRT: 114°F and 146.5°F (45.5°C and 63.6°C).

Figure 8 shows a linear relationship between HHWST and HHWRT.

Figure 9 presents key temperatures of the HHW loop as a function of outdoor air temperature. These include the HHW loop supply temperature, HHW loop return temperature, heat pump maximum leaving temperature, and heat pump turn-down entering temperature.

When the outside air temperature (OAT) is less than 45°F (7.2°C), the required HHW loop supply setpoint is

FIGURE 8 HHWRT regression model.

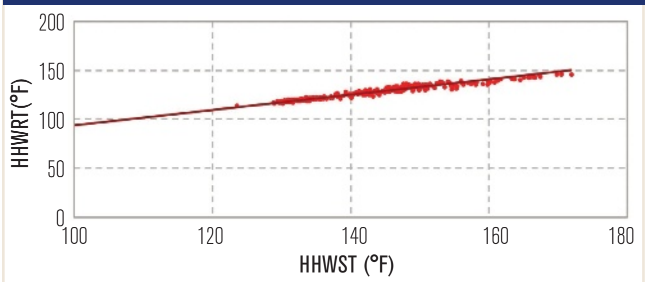
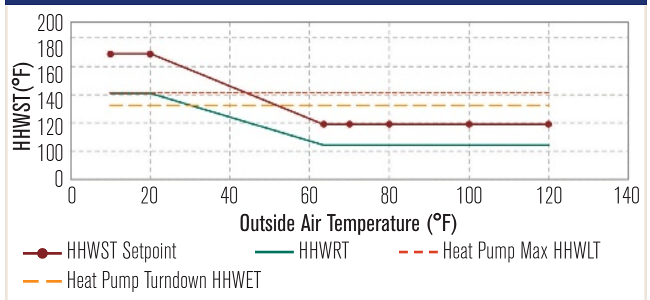


FIGURE 9 Key HHW temperatures versus OAT.



higher than the maximum leaving temperature of the heat pump, 155°F (68.3°C). The heat pump can still run at side-stream to the boilers until the OAT is less than 30°F (–1.1°C) or the loop return temperature is higher than 147°F (63.9°C). When the HHW loop return temperature is higher than 147°F (63.9°C), the ΔT across the condenser is only 8°F (4.44°C), because the maximum HHW leaving temperature of HRCHLR is 155°F (68.3°C). An 8°F (4.44°C) ΔT is 40% of design ΔT 20°F (11.11°C). Therefore, when the condenser flow is at the design flow of 2039 gpm (128.7 L/s), the HRCHLR will turn down when the HHW loop return temperature is higher than 147°F (63.9°C) or the OAT is lower than 30°F (–1.1°C), although the HHW load on campus is as high as 68 MMBtu/h (71.75 GJ/h). In order to take advantage of the heat pump at low OATs, it is recommended to increase HHW flow as high as possible to increase heat pump load ratio with a higher pumping energy consumption penalty. This recommendation is evaluated in Scenario 3 in the Operation Cost Saving Analysis section.

Optimize Heating Hot-Water Flow

If the condenser water flow could be modulated without any limit, the HHW loop ΔT would not impact heat pump performance. However, due to the hydronic pump head and condenser design parameter limits, the water flow range normally is between 60% and 130% of the design flow. With 130% of design HHW flow, the turndown entering HHW temperature can increase from 147°F to 148.8°F (63.9°C to 64.9°C) without exceeding the maximum heating hot-water leaving temperature (HHWLT).

Electrical Rates and Demand Charges

Electrical rates change dramatically from month to month. For example, the minimum blended rate during a fiscal year was \$0.0375/kWh in April, while the maximum blended rate was \$0.0716/kWh in August. Another important factor is the peak demand during summer months (June through September). The annual demand charge was \$46,000/MW for the fiscal year. The Four Coincidental Peaks (4CP) is a value measured by all the regulated utilities and coops in Texas that is used to directly or indirectly capture regulated transmission and transmission cost recovery factor rates from end-use customers. It is calculated based on Electric Reliability Council of Texas's (ERCOT) system peak demand during the months of June, July, August, and September in coincidence with clients loads, as measured over a 15-minute interval. Historically, ERCOT's measured weekday peak demand has occurred between 3:45 p.m. and 5 p.m. Therefore, heat pump operation should be avoided during these hours for 4CP days.

Operation Cost Saving Analysis

As discussed previously, the operation strategy impacts the operating cost significantly. In this section, the energy cost savings are calculated for the following four scenarios with same expected thermal comfort level for buildings:

- Scenario 1: Maintain HHWST at 155°F (68.3°C) and HHW flow at design flow, 2,039 gpm (128.7 L/s);
- Scenario 2: Implement HHW supply reset schedule and maintain HHW flow at design flow, 2,039 gpm (128.7 L/s);
- Scenario 3: Implement HHW supply reset schedule and increase the maximum HHW flow to 130% of design flow, 2650 gpm(167.2 L/s); and
- Scenario 4: Scenario 3 + turn off HRCHLR during 4CP hours.

Model and Assumptions

The following HRCHLR model is based on the manufacturer's performance data:

$$COP_{normal} = 1.5321PLR^2 + 2.8546PLR + 1.9182 \quad (1)$$

$$Correct_T = 0.00010998T^2 + -0.04186T + 4.8422 \quad (2)$$

$$COP = COP_{normal} \times Correct_T \quad (3)$$

where PLR is the part-load ratio $\frac{HHW\ Load}{Capacity_{design}}$

and T is heating hot water supply temperature.

A conventional chiller model below is based on manufacturer's performance data:

$$ELE_{ratio} = (0.319589x^2 - 0.07587x + 0.1759y^2 - 0.101093y) + 0.602307xy + 0.075846 \quad (4)$$

$$ELE_{input} = ELE_{design} \times ELE_{ratio} \quad (5)$$

where x is part-load ratio: $\frac{CHW\ Load}{Tonnage_{design}}$

and y is lift temperature ratio: $\frac{CWET - CHWLT}{(CWT - CHWLT)_{design}}$

where

$CWET$ = condensing water entering temperature

$CHWLT$ = chilled-water leaving temperature

ELE_{design} = design electricity input.

The following describes the assumptions for the model:

- Natural gas (NG) boiler efficiency: 82%
- Cooling tower: kW/ton = 0.08
- HHW pump efficiency: 75%
- Motor efficiency: 95%
- 4CP hours: 3:00 PM to 5:00 PM for weekdays from June through September
- Rates: $Rate_{NG}$ = \$4.257/MMBtu; $Rate_{ELE} \cong$ \$0.0375 to \$0.0734/kWh (as seen in *Table 2*)

For each scenario, two operating costs are calculated. One is calculated using the models of a HRCHLR and HHW pump. The other is calculated by using the models of conventional chiller, HHW pump, natural gas boiler, and cooling tower to produce the same amount of CHW and HHW energy. The difference between these two costs is the savings of each scenario.

The calculated annual energy cost savings and electricity demand cost increase for each scenario are shown in *Figure 10*.

Although reducing the HHWST can improve heat pump efficiency, the HHW loop ΔT decreases at the same time. If the HHW flow is maintained at the design flow, the load ratio of the HRCHLR will decrease, which reduces the HRCHLR savings potential. The savings results show that if HHWST reset is implemented without increasing condenser water flow, the cost savings of Scenario 2 is less than that of Scenario 1 if demand cost is not considered. If condenser water flow is increased to 130% of design flow, the savings (Scenario 3) significantly increase to \$371,428.

The electricity demand charge also impacts cost savings significantly. If the HRCHLR operates at design conditions during 4CP hours, the demand cost will be \$43,673. If the HRCHLR is turned off during 4CP hours, the energy savings will be reduced from \$397,701 (Scenario 3) to \$389,182 (Scenario 4), but Scenario 4 can avoid \$26,273 in demand charges. The overall cost savings of Scenario 4 is the highest. Hence, the optimal operation strategy is to combine the following three ECMs:

- Implement the HHWST reset schedule;
- Increase condenser water flow to 130% of design flow; and
- Turn off the HRCHLR during 4CP hours.

The optimal operation of Scenario 4 can achieve \$389,182/year in savings at current natural gas and electricity rates, which is 43.9% more than the savings currently being realized with operations using Scenario 1.

Conclusion

A heat recovery chiller can be a very effective means of lowering natural gas consumption. Although coincident heating hot-water and chilled water loads are essential to justify the economics of a heat recovery chiller application, designers and operators also need to consider the following factors to operate the HRCHLR system more efficiently:

- Heating hot-water supply temperature reset;
- Turndown (available load reduction);
- Heating hot-water and chilled-water loop ΔT;
- Maximum condenser and evaporator water flow; and
- Electrical rates and demand charges.

Four scenarios are studied in this paper. The results show that Scenario 4 can achieve the most savings. The proposed optimal operation strategy for the case-study HRCHLR is to combine the following three measures:

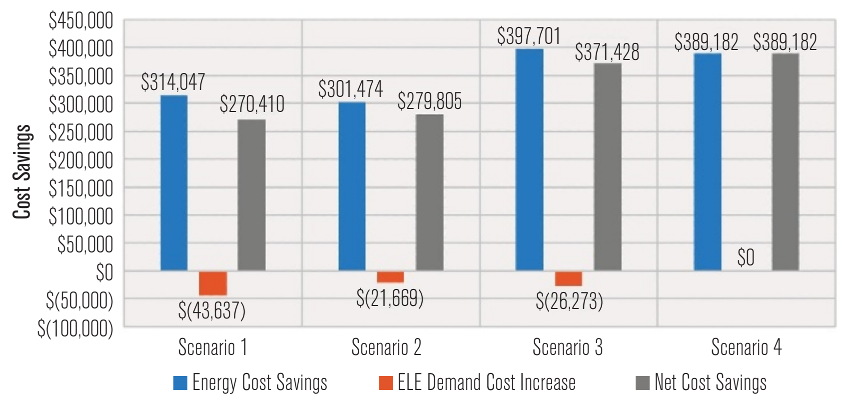
- Implement the HHWST reset schedule;
- Increase condenser water flow to 130% of design

flow; and

- Turn off the HRCHLR during 4CP hours

An overall savings of $389,182 can be achieved at the given natural gas and electricity rates, which is 43.9% more savings than the current strategy.

FIGURE 10 Energy cost savings for different scenarios.



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