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Case Study of the ARRA-Funded GSHP Demonstration at Lancaster County Adult Detention Facility Thermal Plant

Mini Malhotra, PhD
Xiaobing Liu, PhD

October 2016

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Energy and Transportation Science Division

CASE STUDY OF THE ARRA-FUNDED GSHP DEMONSTRATION AT DISTRICT ENERGY CORPORATION THERMAL ENERGY PLANT

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ACRONYMS

ADF	Adult Detention Facility
ANSI	American National Standards Institute
ARRA	American Recovery and Reinvestment Act of 2009
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CO_2e	equivalent CO_2 emissions
COP	coefficient of performance
$COPc_{eqp}$	coefficient of performance of heat pump in cooling mode
$COPh_{eqp}$	coefficient of performance of heat pump in heating mode
COP_{sys}	coefficient of performance of GSHP system
DEC	District Energy Center
DHW	domestic hot water
DOE	Department of Energy
EER	energy efficiency ratio
FCW	flow rate in the chilled water loop
FGL	flow rate in the ground loop
FHW	flow rate in the hot water loop
GHX	ground heat exchanger
gpm	gallon per minute
GSHP	ground source heat pump
HVAC	heating, ventilation, and air conditioning
OA	outdoor air
ORNL	Oak Ridge National Laboratory
PLR	part-load ratio
QC	heat pump cooling delivered
QGL	heat transfer with the ground
QGL_c	heat rejected to ground loop
QGL_h	heat extracted from ground loop
QH	heat pump heating delivered
SCADA	supervisory control and data acquisition
TD	temperature differential
TCWR	chilled water loop return temperature
TCWS	chilled water loop return temperature
TGLS	ground-loop supply water temperature
TGLR	ground-loop return water temperature
THWR	hot water loop return temperature
THWS	hot water loop supply temperature
TMY3	typical meteorological year
$TSSL_n$	source-side leaving water temperature

VAV	variable air volume
VFD	variable frequency drive
WAHP	water-to-air heat pump
WHPc	heat pump power consumption in cooling mode
WHPh	heat pump power consumption in heating mode
WP	(total) pump power consumption
WPCW	chilled water loop pump power consumption
WPGL	ground loop pump power consumption
WPHW	hot water loop pump power consumption
WWHP	water-to-water heat pump

EXECUTIVE SUMMARY

With funding provided by the American Recovery and Reinvestment Act, 26 ground-source heat pump (GSHP) projects were competitively selected in 2009 to demonstrate the benefits of GSHP systems and innovative technologies for cost reduction and/or performance improvement. One of the selected demonstration projects is a central GSHP system installed in the Lancaster County Adult Detention Facility thermal plant in Lincoln, Nebraska.

The building is a newly constructed 270,000 square foot, 786-bed, secure detention facility served by a variable-air-volume (VAV) system for space heating and space cooling. Hot and chilled water to the building's VAV system is supplied by a GSHP system located in a separate thermal energy plant at the site. The GSHP system consists of a closed-loop vertical bore ground heat exchanger, six banks of water-to-water heat pump (WWHP) units, and three central variable speed pumping stations to circulate water through the ground loop and primary hot water and chilled water loops. The primary hot water and chilled water loops are isolated from the detention facility's secondary loops by heat exchangers.

The actual performance of the GSHP system was analyzed based on available measured data from October 14, 2013, through July 21, 2014. Using a curve-fit of available measured data, the annual energy consumption of the GSHP system was predicted for the year 2014 and compared with that of a baseline system—a central plant with hot water boilers and water-cooled chillers with the minimum allowed efficiencies specified in ASHRAE Standard 90.1-2013 (i.e., 79% efficiency for steam boilers and 0.585 kW/ton efficiency for water-cooled chillers) (ASHRAE 2013a). The comparison was also made for source energy savings, operating cost savings, cost-effectiveness, and environmental benefits. Finally, recommendations presented for further improving the operational efficiency of the GSHP systems.

Energy Performance and Cost-Effectiveness

The annual operational efficiencies, evaluated by the coefficient of performance (COP) of the GSHP systems, ranged between 3.6 and 5.7 for heating and 5.1 and 7.9 for cooling. If heating and cooling operation are combined, the effective efficiency of the GSHP system averaged 3.5 in the winter and 5.3 in the summer.

Accounting for the pumps in the primary loops, the pumping power fraction averaged as low as 8.3% in the winter, 11.7% in the summer and as high as 13% in the shoulder seasons.

Compared with the baseline system, the GSHP system achieved 27.3% source energy savings and 25.5% CO_2 emission reductions, annually. The reduced energy consumption from the GSHP system would result in a $67,927 operating cost savings per year.

In addition, the GSHP system would save 3.106 million gallons of water per year by eliminating the cooling tower. Using nonresidential water rates for the city of Lincoln, Nebraska, (i.e., $1.344 per unit for up to 80 units and $1.911 per unit for additional units of water usage; 1 unit = 748 gallons), the GSHP system would save $7,590 per year in avoided costs for water.

Table ES.1 provides a performance comparison between the two systems. Based on the cost of the baseline system estimated in the original feasibility study for this GSHP demonstration project, and the actual installed cost of the GSHP system, the cost of installing the GSHP system would be $379,651 less than the cost of installing the baseline system, which would result in an instantaneous payback.

Table ES.1. Performance comparison between the GSHP system and the baseline system

	Baseline system	GSHP system	Savings
Source energy use (MMBtu)	41,019	29,804	27.3%
CO_2 emissions (1,000 lb)	5,920	4,413	25.5%
Total annual energy cost ($)	289,343	221,416	23.5%
Cooling tower water use (million gallon)	3.106	–	100%
Cost of water ($)	7,590	–	100%

Lessons Learned and Issues

- The site provided data from April 2013 through September 2015. However, pump data after July 21, 2014, were invalid because data from the pump's variable frequency drive and the kilowatt-hour accumulator in the drives had maxed out and needed to be reset. Such issues should be avoided by regularly checking monitored data.

- The pattern in the relationship between the flow rate and the temperature differential (TD) of the ground loop is a useful indicator for evaluating the variable-speed pumping performance. Ideally, the pattern should be a "U" shape, which indicates that the TD is kept at the design value while the flow rate varies within a large range. At this site, the measured data show a "V" shape pattern with a decidedly high minimum flow rate, which indicates excessive pumping at part-load conditions.

Recommendations for Further Improvements

- The measured hot water supply temperature ranged between 115° and 125.7°F even in the summer. To further improve the operational efficiency of the GSHP system, the hot water supply temperature in summer could be reduced. According to the specification of the WWHP, the normal range of the hot water supply temperature is 110–135°F. Lowering the hot water supply temperature from 120 to 110°F would result in a ~0.5 COP improvement (a 9–14% increase).

- The performance of ground loop pumps can be improved by varying the flow rate to keep the TD in the ground loop at the design value. The ground loop measured data show a small TD at part-load conditions and a high minimum flow rate, which indicates excessive pumping at part-load conditions.

1. PROJECT OVERVIEW

The Lancaster County Adult Detention Facility (ADF) (Figure 1) is a newly constructed 270,000 square foot, 786-bed, secure detention facility in Lincoln, Nebraska. Under the American Recovery and Reinvestment Act (ARRA) of 2009, District Energy Corporation (DEC)—a nonprofit entity that provides innovative, efficient, low-cost utility services to facilities owned by the city, the county, and the state of Nebraska (DEC Energy 2016a)—was awarded $5 million in stimulus funds for a demonstration project to install a ground-source heat pump (GSHP) system to provide space heating and cooling and preheating of domestic hot water (DHW) for the detention facility. A central thermal energy plant constructed as a separate building on the site (marked on Figure 2) houses the central plant of the GSHP system, as well as emergency power generation equipment to provide backup electricity to the detention facility. The central plant was commissioned in May 2012 (DEC Energy 2016b). The detention facility was opened in July 2013 (Lancaster County 2016).

Figure 1. Lancaster County Adult Detention Facility.

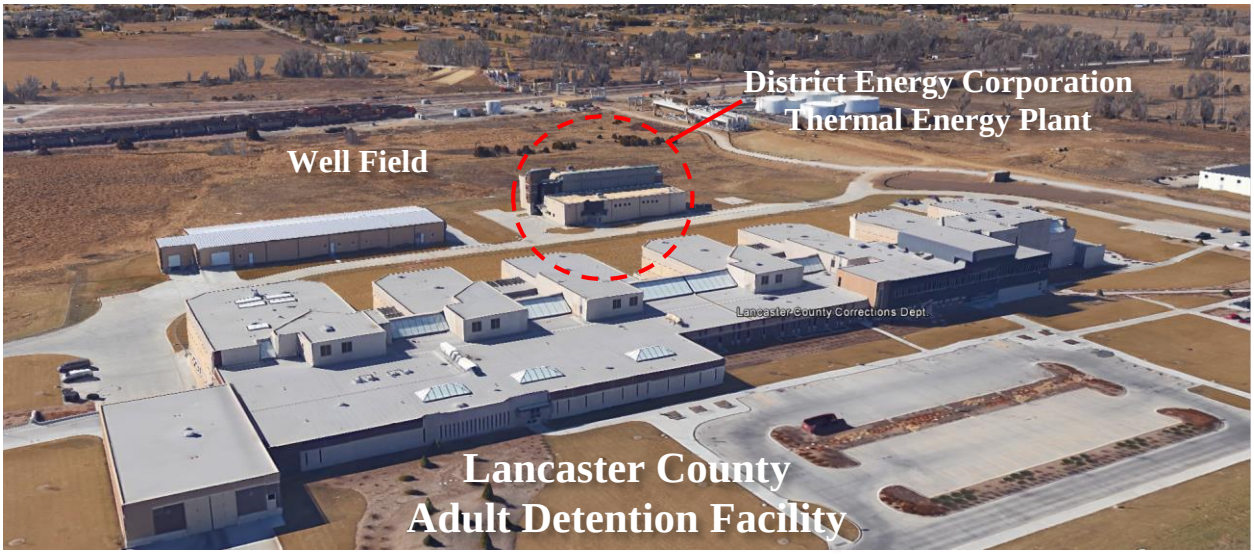

Figure 2. Aerial view of the GSHP demonstration site showing the detention facility, thermal energy plant and well field.

Figure 3 shows the location of the demonstration site within the United States. The site is located in climate zone 5A, which is defined as cool-humid (ASHRAE 2013b). The climate is characterized as a highly variable four-season humid continental climate with cold and relatively dry winters and hot and occasionally humid summers. The TMY3 (Typical Meteorological Year) monthly statistics show a −9.9°F (−23.3°C) minimum and 24.6°F (−4.1°C) daily average dry-bulb temperature in January and a 102°F (38.9°C) maximum and 77.7°F (25.4°C) daily average dry-bulb temperature in July.

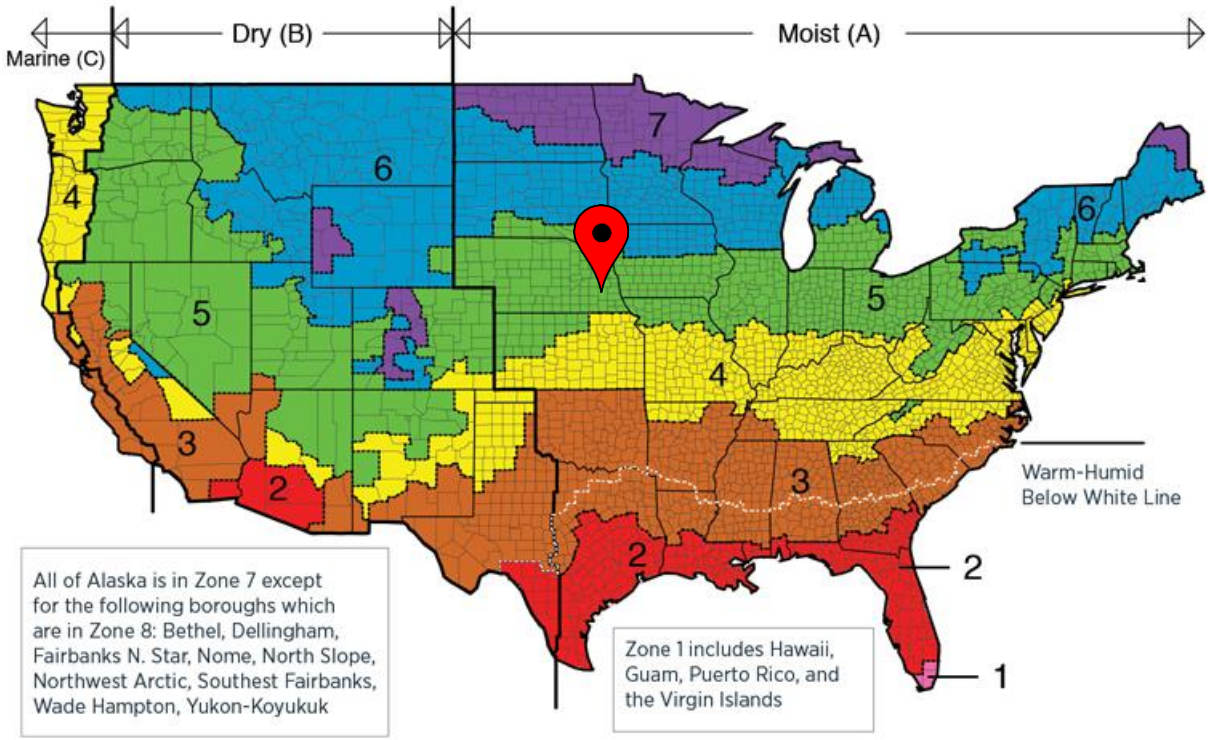

Figure 3. Location of the GSHP demonstration site within the United States.

Measured data for the GSHP systems became available starting in April 2013 as the detention facility became operational. Based on measured data and other relevant information, this case study evaluated a few key performance metrics, including the electricity end use levels of all major equipment of the GSHP system (i.e., heat pumps and pumps), energy efficiency of the overall GSHP system, benefits achieved by the GSHP system (i.e., energy and operating cost savings, water savings, and carbon emission reductions) compared with a baseline system, and cost-effectiveness of the GSHP installation. This case study also identified areas for further improving the operational energy efficiency of the GSHP system. The following sections describe the building and GSHP system.

1.1 BUILDING INFORMATION

The Lancaster County ADF is a newly constructed 270,000 square foot, 786-bed secure detention facility in Lincoln, Nebraska. The facility was built to address the issues of safety, security, and inmate overcrowding in previous facilities, as well as accommodate facility expansion, if needed. It contains 6 general population pods, 12 general population dormitories, a segregation pod, and a pod for inmates with special needs due to mental/ behavioral health concerns, cognitive deficits, or developmental disabilities (Lancaster County 2016).

1.2 GSHP SYSTEM

The GSHP system is located in a separate thermal energy plant. It is designed to provide (1) hot and chilled water to the detention facility's variable-air-volume (VAV) system for space heating and cooling and DHW preheating, (2) hot water to the radiant floor system and 12 unit heaters in the thermal energy plant, and (3) groundwater to 14 water-to-air heat pump (WAHP) units in the thermal energy plant. Figure 4 shows a schematic of the GSHP system installed in the DEC thermal energy plant and the location of data collection points for ongoing performance monitoring. Table 1 explains the data collection points.

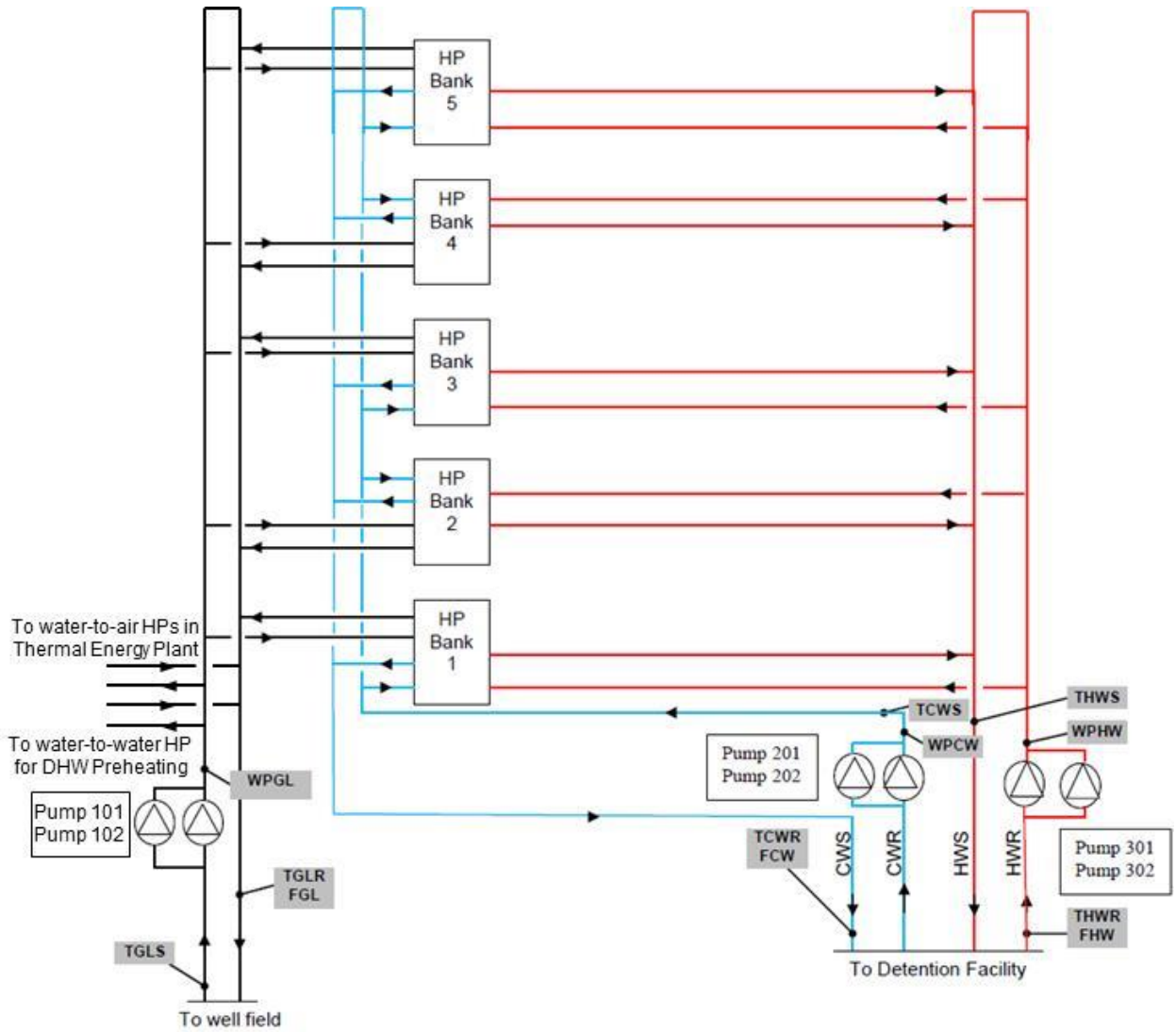

Figure 4. GSHP system schematic and data collection points.

Table 1. GSHP system data collection points

No.	Data point	Description	Units
1	TGLS	Ground loop supply temperature	°F
2	TGLR	Ground loop return temperature	°F
3	FGL	Ground loop flow	gpm
4	THWS	Supply temperature hot water	°F
5	THWR	Return temperature hot water	°F
6	FHW	Flow hot water	gpm
7	TCWS	Supply temperature chilled water	°F
8	TCWR	Return temperature chilled water	°F
9	FCW	Flow chilled water	gpm
10	WPGL	Ground loop pump 101 power	kWh
11	WPGL	Ground loop pump 102 power	kWh
12	WPHW	Hot water loop pump 301 power	kWh
13	WPHW	Hot water loop pump 302 power	kWh
14	WPCW	Chilled water loop pump 201 power	kWh
15	WPCW	Chilled water loop pump 202 power	kWh
16	TOA	Ambient temperature	°F
17	WT	Total power District Energy thermal energy facility	kWh

The GSHP system consists of a closed-loop vertical bore ground heat exchanger, six banks of WWHP units, and three central variable speed pumping stations to circulate water through the ground loop and primary hot water and chilled water loops. The primary hot water and chilled water loops are isolated from the detention facility's secondary loops by heat exchangers. The components of the GSHP system are described in the following sections.

1.2.1 Ground Heat Exchanger

Figure 5 shows the location of the current and future bore fields for the ground heat exchanger (indicated as "Geothermal Wellfield" in the figure). The current bore field consists of 667 bores, each approximately 300 ft deep, resulting in a total of 200,100 bore-feet and a normalized bore depth of 160 bore-feet/ton (of cooling).

Based on the published annual earth temperatures near the site, the undisturbed soil temperature is 53–57°F. The thermal conductivity of the ground (in kilogram) tested in December 2008 on a 305 ft deep test loop at this location was calculated at 1.56 Btu/h-ft·°F. Based on published data and actual test well logs, the thermal diffusivity was estimated to be 1.34 ft^2/day. Heat extraction from the ground for preheating DHW and space heating is expected to moderate the ground loop temperatures during heat rejection (cooling) periods.

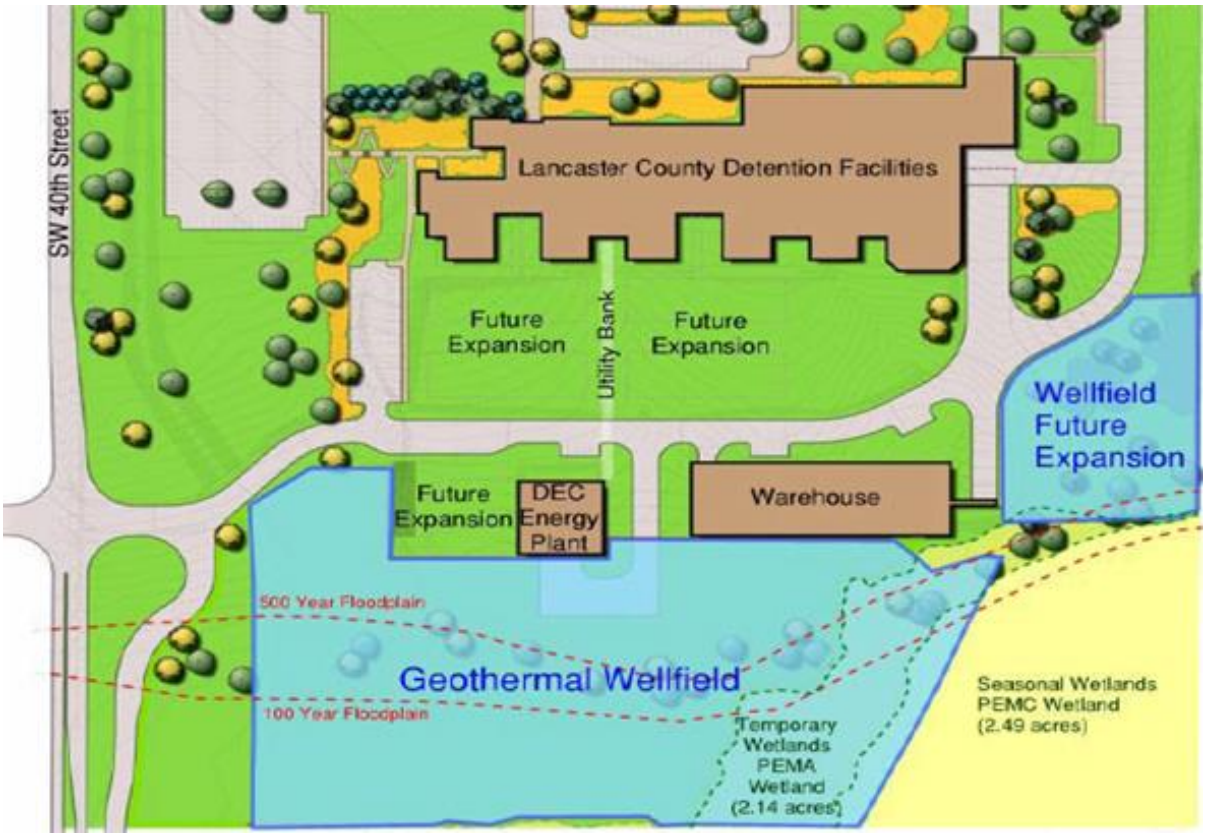

Figure 5. Layout of the bore fields.

1.2.2 Water-to-Water Heat Pump

Figure 4 shows the GSHP system schematic. The thermal energy plant has five banks of WWHPs for space heating and cooling. Each of these banks has five 50-ton Multistack WWHPs. In addition, another bank of three 50-ton Multistack WWHPs located in the basement of the detention facility is used for preheating DHW. The combined nominal capacity of the entire system is 1,400 tons.

The five banks of WWHPs for space conditioning provide both chilled and hot water to a four-pipe hydronic piping system to satisfy the mixed heating and cooling demands in different parts of the facility. Each bank of WWHPs can produce only chilled water or hot water at a given time. All five banks of WWHPs use the ground loop as a heat sink or heat source. When a bank of WWHP produces chilled water, it is blocked from the hot water loop, and vice versa. When a bank of WWHPs is not called on, it is blocked from the ground loop, as well as the chilled and hot water loops.

Typical peak summer operation is anticipated to utilize three banks of WWHPs (up to 750 tons) for cooling, one bank (250 tons) for space heating or reheating, and one bank as a spare. As the heating load increases, more banks of WWHPs switch from cooling operation to heating operation to satisfy the increased heating load.

The sixth bank of WWHPs in the basement (150 tons) use heat extracted either from the ground loop or from the chilled water return loop to preheat the cold municipal water entering the preheat tank. The operation of these WWHPs was not fully functional at the time when this study was conducted, so their performance is not evaluated in this report.

1.2.3 Pumping Stations

Water in the ground loop is circulated by two redundant 75 hp pumps (one primary and one backup), each providing 3,350 gpm of water flow. The chilled water loop can circulate 2,550 gpm of water flow using two 50 hp pumps (one primary and one backup) and the hot water loop is circulated using two 25 hp pumps that provide 1,350 gpm of water flow. All circulating pumps have 100% redundancy and are controlled by variable-frequency drives (VFDs) to maintain a constant pressure across the supply-return mains of each individual water loop.

The control system operates the pumps based on the required differential pressure to maintain adequate flow through the WWHP units. When a unit is not in operation, it is valved off and less flow is required to match the load. This arrangement is intended to minimize pump energy consumption.

1.3 DATA COLLECTION

Performance monitoring and data collection for the WWHP system were provided by DEC using the on-site Wonderware SCADA (supervisory control and data acquisition) system. The SCADA system polls the sensors once per second and provides 15 minute totals or averages of each sensor, depending on the sensor type. The data are recorded in a column-oriented, comma-delimited file (CSV format) along with a corresponding date/time stamp for the data record. Table 1 lists the data collection points. The locations of these data collection points are shown in Figure 4. Note that the power consumption of the WWHPs was not measured. A method was developed to estimate the power consumption of the WWHPs based on the available data; it is discussed in Section 2.3.1.

1.3.1 Data Quality

Measured data for the GSHP systems are available from April 2013, through December 2015. However, there were a few system operation and data quality issues.

- The ground loop supply temperature sensor and chilled water flow rate sensor were reporting null data until June 2013.

- The measured ground loop temperature and flow rate data show that the GSHP system was not fully operational until October 14, 2013. However, the chilled water loop temperature and flow rate data show that cooling was delivered to the facility during this period. Also, there were a few periods of missing data spanning from a few hours in a day to as long as a week.

- The ground loop and chilled water loop pump power data after July 21, 2014, and hot water loop pump power data after January 2015 are missing. For pump power, the data acquisition system uses data from the pump's VFD; however, the kilowatt-hour accumulator in the drives had maxed out and was not reset, so the system stopped collecting data.

Figure 6 and Figure 7 show the hours per day for which measured data from different sensors are available and valid. In Figure 6, loop temperature and flow rate data are marked valid if they have a reasonable temperature differential (i.e., for the hot water loop, a supply temperature higher than the return temperature; for the chilled water loop, a supply temperature lower than the return temperature; and for the ground loop, at least a 1°F temperature differential). In Figure 7, pump power data above the threshold power consumption (0.1 kWh per 15 min) are marked valid.

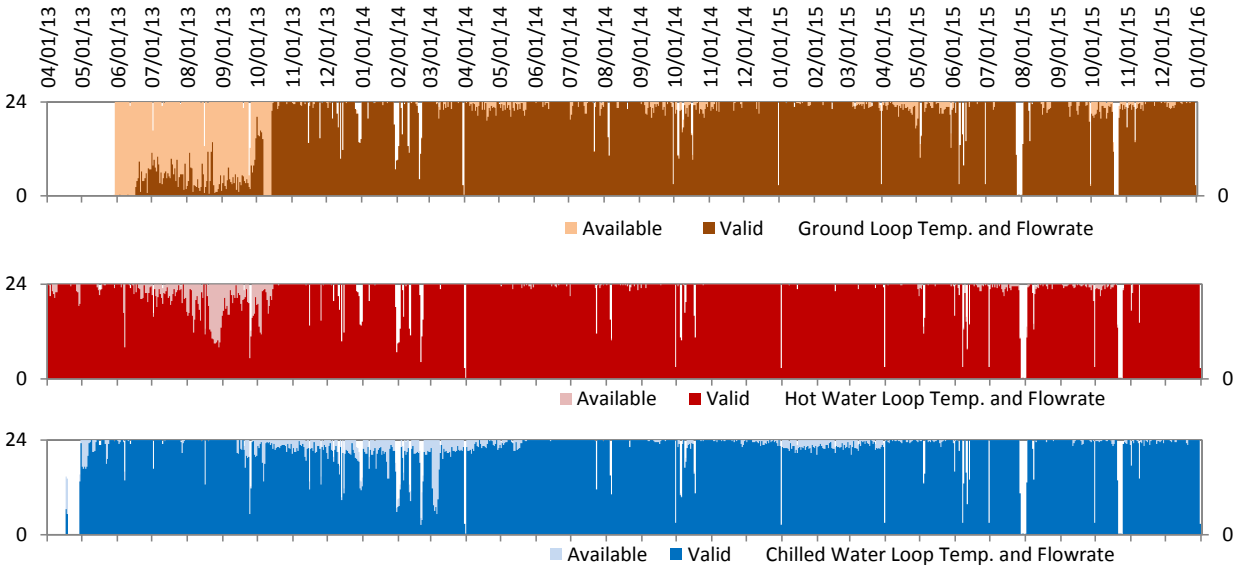

Figure 6. Measured loop temperature and flow rate data availability.

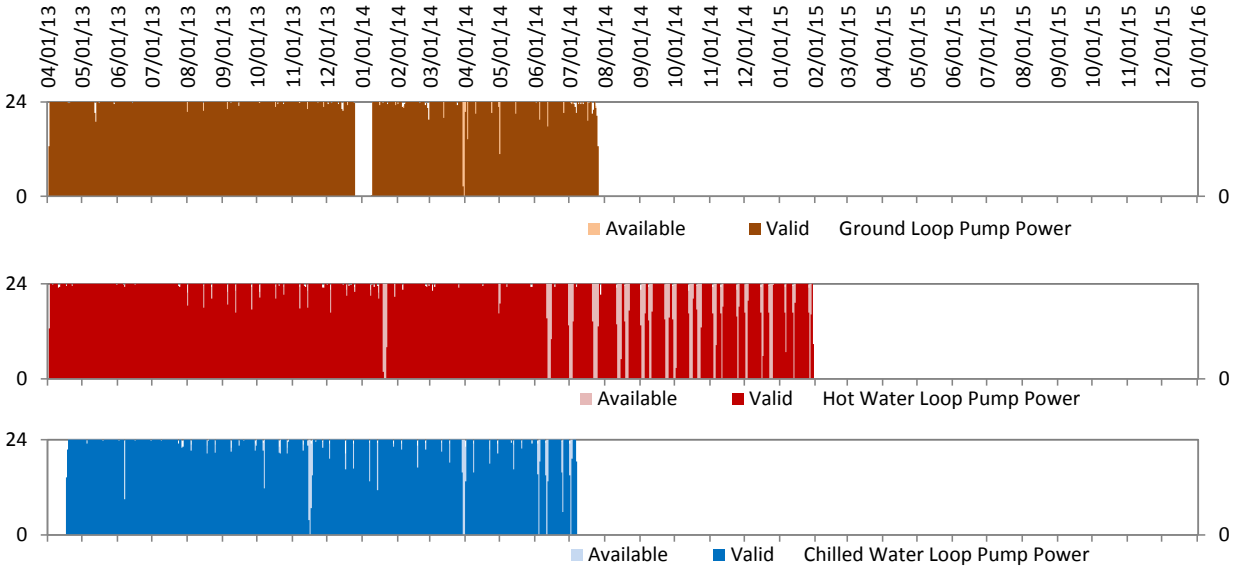

Figure 7. Measured pump power data availability.

2. ANALYSIS OF MEASURED PERFORMANCE OF GSHP SYSTEM

Based on the completeness and quality of the available data, analysis of the measured data was conducted for a partial year starting October 14, 2013, and ending July 21, 2014. The analysis is primarily based on the hourly aggregation of the 15 minute measured data provided by the on-site monitoring system. However, for analyzing the pump power data, daily aggregation was used.

For the analysis period, the performance of the GSHP system was analyzed for (1) source-side operation, including the trend of the measured water temperature and flow rate in the ground loop, the pumping energy use, and the heat transfer in the ground loop; (2) load-side operation, including the heating and cooling output and the operational efficiency of the WWHPs; and (3) the overall GSHP system, including the operational efficiency of the pumping system and the entire GSHP system.

In analyzing the measured data, faults and abnormalities in operation and potential improvements in GSHP system performance were identified.

2.1 SOURCE-SIDE OPERATION

2.1.1 Ground Loop Water Temperature Trend and Flow Rate

The hourly average ground loop supply and return water temperature and flow rate are shown in Figure 8. The overlying supply and return water temperature data collected until October 14, 2013, indicate that the WWHPs were not operating properly during this period, although the ground loop pump was running. Excluding this period, a scatter plot of loop temperature and flow rate versus outdoor air (OA) temperature is shown in Figure 9. From these figures, the following observations can be made:

- The ground-loop supply water temperature, which is the entering water temperature at the source side of the WWHPs, ranged between 43 and 75°F.

- The ground-loop return water temperature, which is the mixing temperature of all the water leaving from the source side of each bank of the WWHPs, ranged between 39.6 and 87°F.

- The ground loop flow rate ranged between 260 and 2,900 gpm, with a maximum of 2,300 gpm in the winter, 2,900 gpm in July, and lower flow rates of up to 1,000 gpm during the shoulder months and most of the summer. The summer cooling loads were higher than the winter heating loads; and so more WWHPs ran in cooling mode, thereby requiring higher flow rates. The seasonal variation in the flow rate was due to variation in the system operating conditions, as well as the differential pressure set point selected by the operator.

- Compared with the OA temperature range (i.e., −9.7 to 100.7°F), the ground loop supply water temperature was more stable. It was up to 53.7°F higher than the OA temperature on the coldest day and up to 31.6°F lower than the OA temperature during the analysis period.

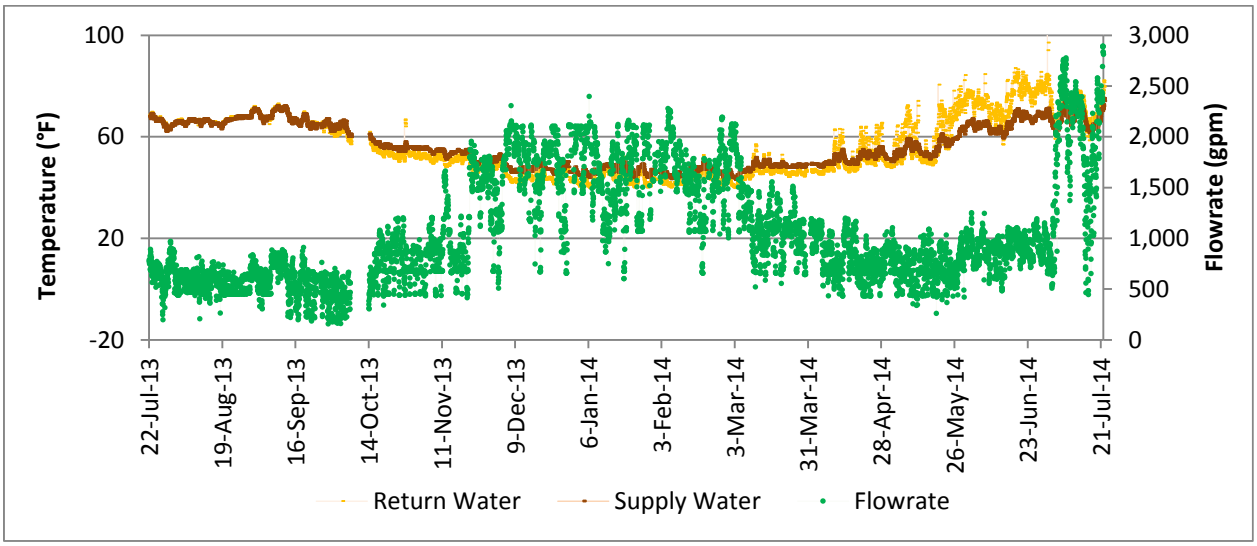

Figure 8. Ground loop water temperature and flow rate.

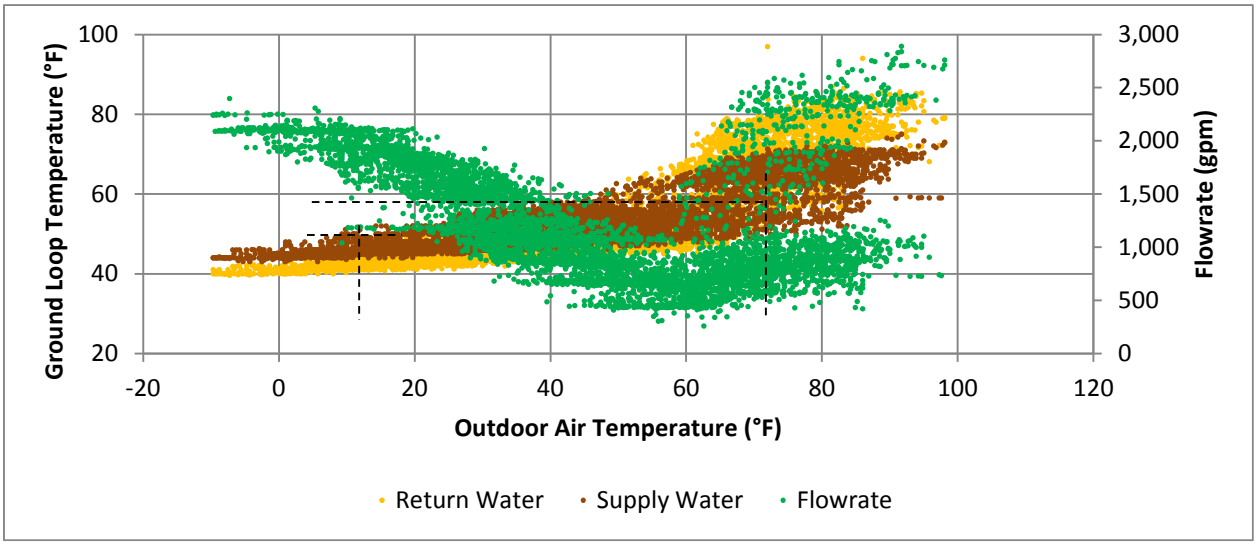

Figure 9. Ground loop water temperature and flow rate versus outdoor air temperature.

Figure 10 shows the ground loop TD (in red) and flow rate (in green) as the OA temperature varied during the analysis period. The TD of the ground loop ranged between 20°F in the summer and 5°F in the winter. In the summer, two trends were observed for a similar OA temperature range—a higher TD exceeding 15°F and a flow rate of up to 1,000 gpm during May–June and a lower TD of up to 7°F and flow rate of up to 2,900 gpm during July.

Since, at the design flow rate, the WWHPs usually operated with a 10°F TD in cooling mode and a 6°F TD in heating mode, the measured TD indicates expected operation in the winter and July but underflow in the ground loop during May–June.

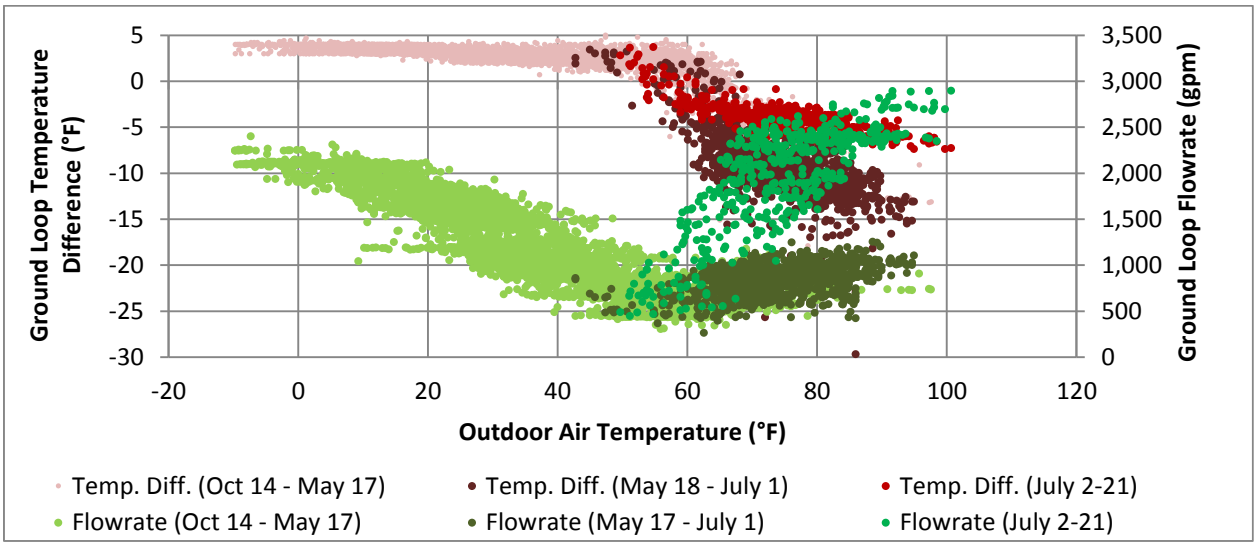

Figure 10. Ground loop temperature differential and flow rate versus outdoor air temperature.

Figure 11 shows the histogram of the ground loop TD in 2°F bins and the average flow rate for each bin. The ground loop TD in the winter was within 2–4°F most of the time, with a bin average flow rate of 1,272 gpm. During the summer, the TD was equally distributed in a wider range between −2 and −12°F (lower TD in July with a higher average loop flow rate exceeding 1,500 gpm and higher TD in May–June with a lower average flow rate within 800–1,100 gpm).

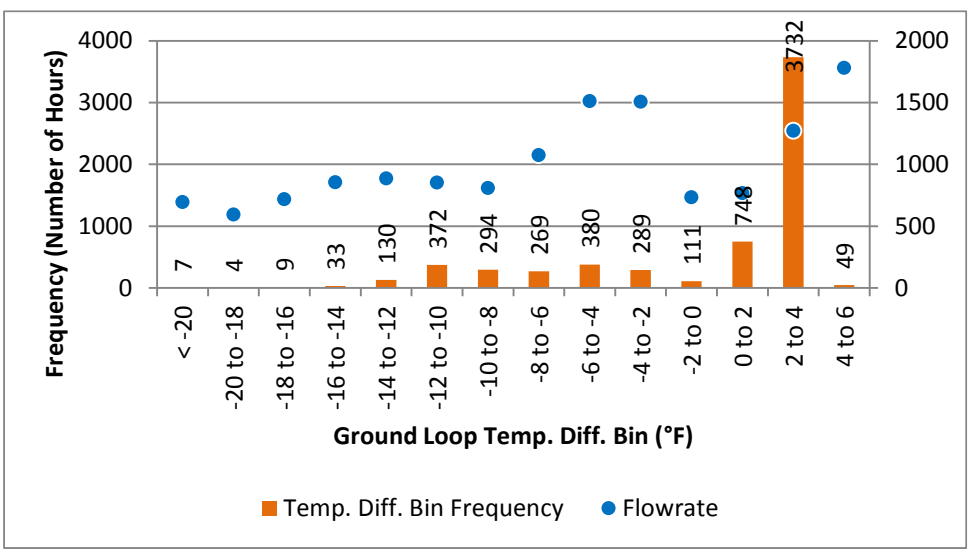

Figure 11. Histogram of ground loop temperature differential and bin average flow rate.

The relationship between the measured flow rate and ground loop TD for the analysis period is explored in Figure 12 through a scatter plot. During the winter, the pump operation was close to what is expected (i.e., following a U-shape curve, meaning a constant TD was maintained when the flow rate varied in a large range). In July, the scatter plot shows a somewhat linear relationship between the flow rate and the TD, which indicates excessive pumping at part-load conditions. The unexpected operation pattern of the

pump during May–June with a very low flow rate is evident by the large change in the TD. This might be due to the control scheme for the pump (i.e. changes in the differential pressure set point).

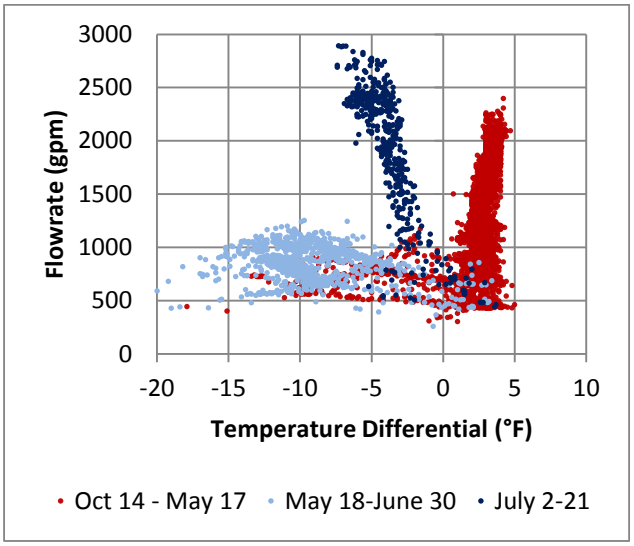

Figure 12. Ground loop flow rate versus temperature differential.

2.1.2 Pumping Electricity Consumption

Figure 13 shows a scatter plot of daily average ground loop pumping power versus average flow rate. The trend line of pumping power as a function of flow rate shows the best fit for the daily average data to represent the ground loop pump performance. The observed trend derived from the aggregated data deviates from the characteristic pump curve and does not indicate how well the pumping control worked.

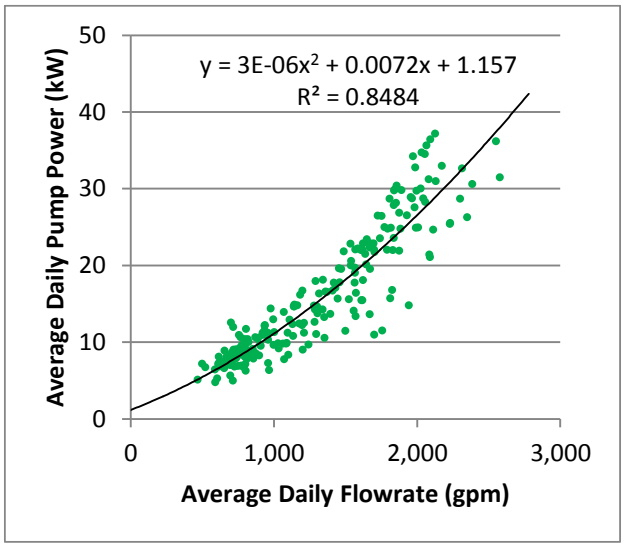

Figure 13. Daily ground loop pumping electricity consumption versus flow rate.

2.1.3 Heat Transfer in Ground Loop

Heat transfer through the ground loop was calculated using the ground loop flow rate and the ground loop supply and return water temperatures, as expressed in Eq. (1):

$$QGL = k * FGL * \frac{(TGLS - TGLR)}{1000} \quad , \tag{1}$$

where

QGL = heat transfer to the ground (kBtu/h) (extraction >0, rejection <0),
FGL = ground loop flow rate (gpm),
TGLS = ground loop supply temperature to heat pump (°F),
TGLR = ground loop return temperature from heat pump (°F),
k = a factor that incorporates conversion factors and the specific gravity of the fluid, which is 500 Btu/h-gpm·°F for pure water.

The calculated hourly heat transfer rate of the ground loop is plotted in Figure 14 as a time series plot. The heat extraction rate is shown with positive values (red) and the heat rejection rate with negative values (blue). During the analysis period, the maximum ground loop heat extraction rate was 5.0 MMBtu/h (98.5 MMBtu/day), and the maximum ground loop heat rejection rate was 10.6 MMBtu/h (184.4 MMBtu/day). Aggregating for the analysis period (which was less than a year by 84 days of the summer and fall seasons), the cumulative heat extraction amounted to 8,449 MMBtu, and the cumulative heat rejection amounted to 7,027 MMBtu.

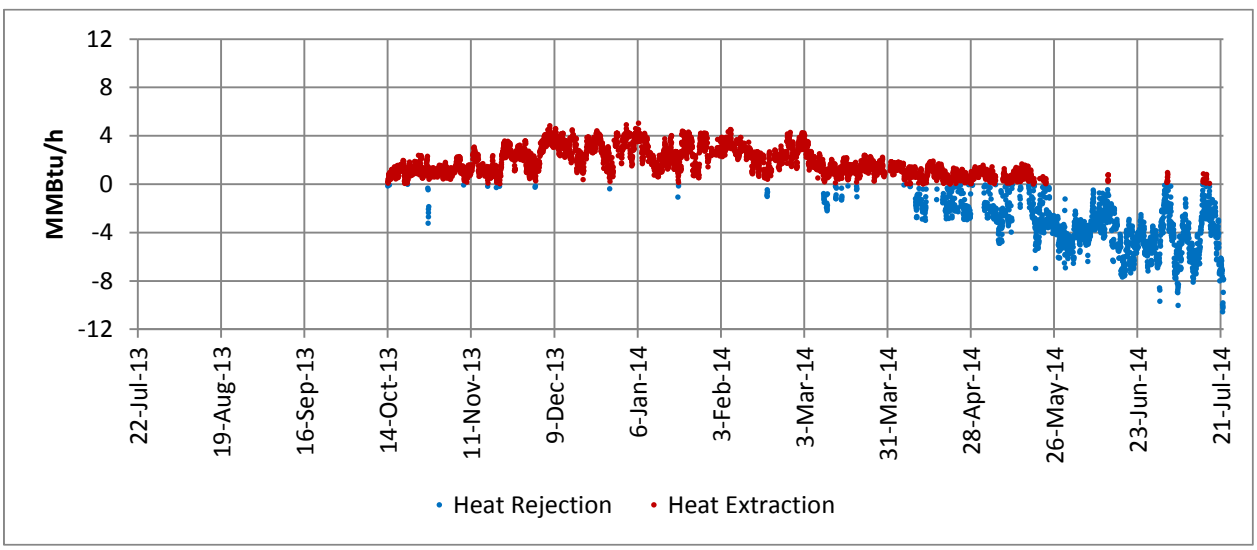

Figure 14. Ground-loop heat transfer rate.

Figure 15 shows a scatter plot of the ground loop heat transfer rate versus OA temperature. Clearly, the ground loop heat extraction and heat rejection rates are linearly related to OA temperature. The correlation between ground loop heat transfer rate and OA temperature is more scattered in the summer than in the winter. This is the case because additional factors in the summer, such as solar radiation and the simultaneous heating and cooling loads of the various banks of WWHPs, affect the building loads and heat transfer rate to the ground loop.

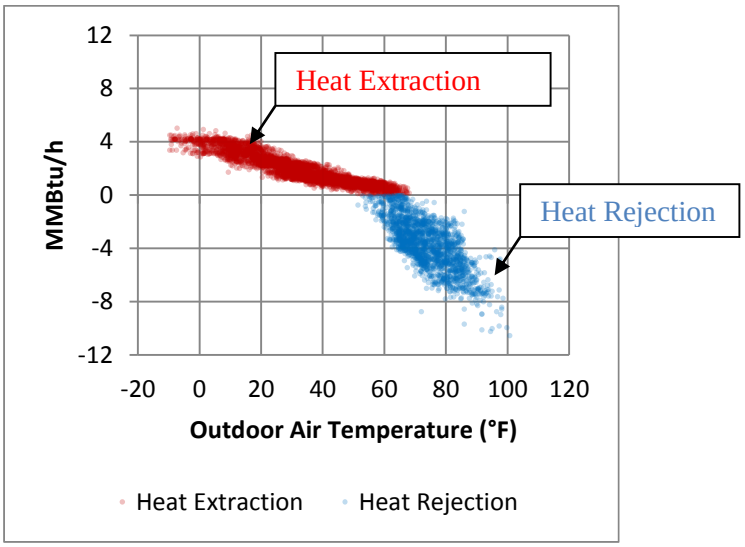

Figure 15. Ground-loop heat transfer rate versus OA temperature.

2.2 LOAD-SIDE OPERATION

2.2.1 Hot Water Loop Temperature Trend and Flow Rate

The hot water loop supply and return temperatures and flow rate are shown in Figure 16. The minimum flow rate, as well as the overlying supply and return temperatures, until October 14, 2013, show that the WWHPs were not in operation during this period. Excluding this period, a scatter plot of loop temperature and flow rate versus OA temperature is shown in Figure 17. Figure 18 shows the histogram of the hot water loop TD in 2°F bins and the average flow rate for each bin. From these figures, the following observations can be made:

- The GSHP system provided hot water during all seasons, including the summer.
- The hot water loop supply temperature—which is the mixing temperature of the hot water leaving the load-side of each bank of WWHPs that operates in heating mode—ranged between 115 and 125.7°F. During the winter, the hot water supply temperature varied within a narrow range of ~2°F around 120°F. With higher OA temperatures, the hot water supply temperature varied in a wider range of ~10°F (i.e., 115–125°F), possibly as a result of more intermittent operation of the WWHPs. The higher hot water supply temperature in summer was due to the higher return temperature resulting from the smaller cooling loads.
- The hot water return temperature ranged between 95 and 121.6°F, with a TD of up to 15°F in the winter and up to 9°F in the summer.
- The hot water loop flow rate ranged between 190 and 1,400 gpm, with the maximum approaching 1,400 gpm in the winter, 750 gpm during the shoulder season, and 600 gpm in the summer.
- The hot water loop TD was within 4–12°F most of the time. The minimum TD of 4°F at a minimum flow rate is significant. As confirmed by the site, typically at least one WWHP operated in heating mode to provide heat for reheating. At times, the same heating source was used for DHW heating. For

the higher TD bin, the bin average flow rate was 400 gpm (which occurred in the summer and shoulder months). For lower TD bins, the bin average flow rate was 660–950 gpm.

- The hot water supply temperature in summer could be reduced with an OA temperature reset schedule to improve the WWHP operating COP in heating mode. According to the specification of the WWHP, the normal range of the hot water supply temperature is 110–135°F. Lowering the hot water supply temperature from 120 to 110°F would improve the COP by ~0.5.

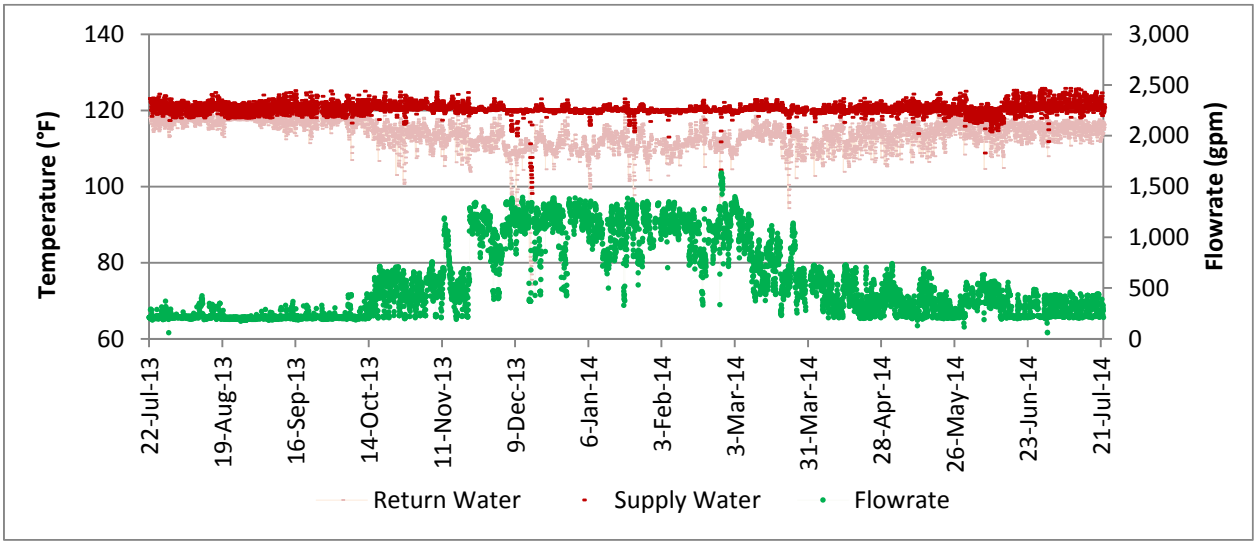

Figure 16. Hot water loop temperature (red) and flow rate (green).

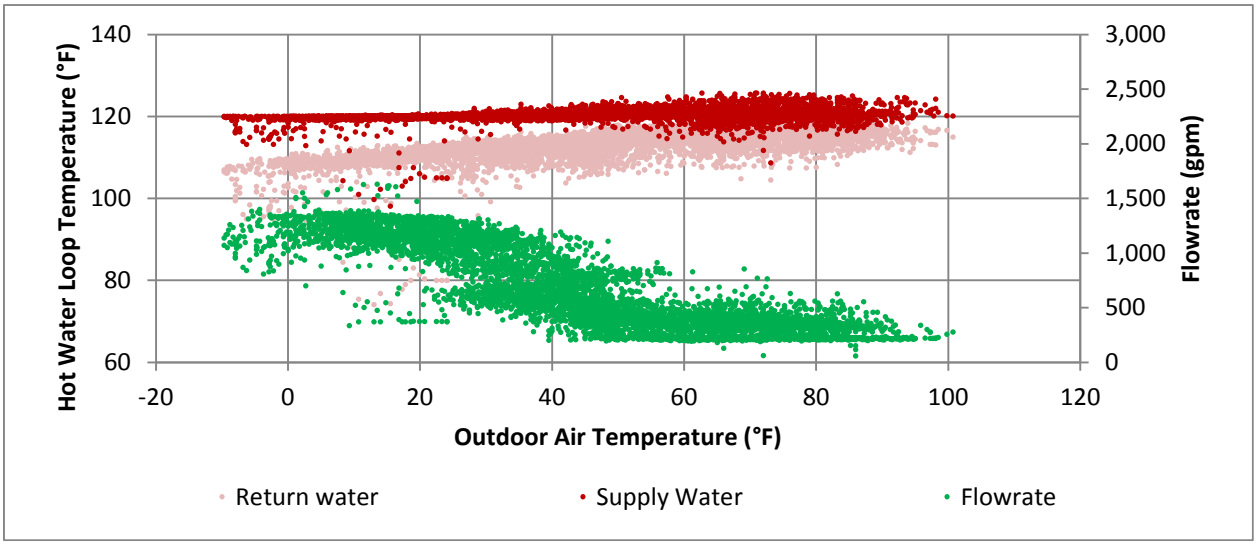

Figure 17. Hot water loop temperature (red) and flow rate (green) versus OA temperature.

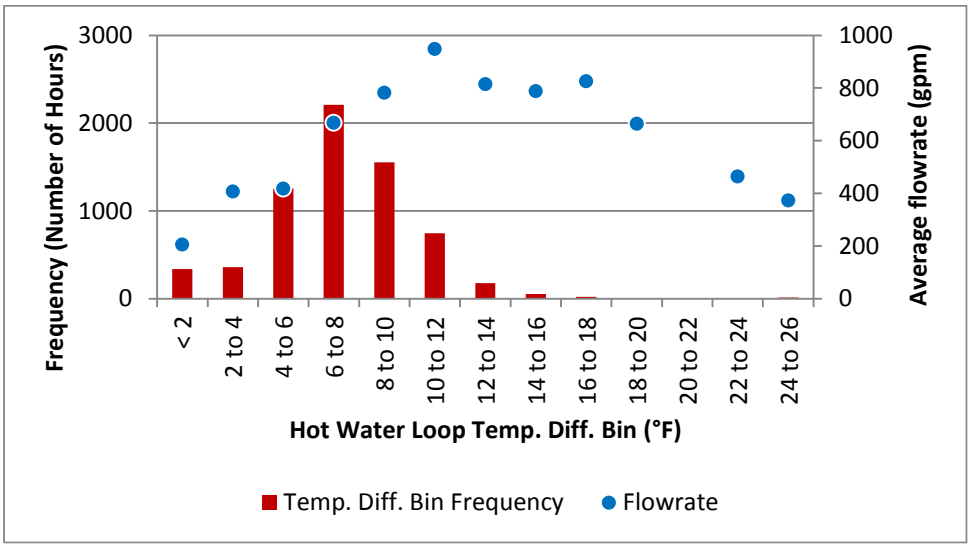

Figure 18. Histogram of hot water loop temperature differential and bin average flow rate.

2.2.2 Chilled Water Loop Temperature Trend and Flow Rate

The chilled water supply and return temperature and flow rate are shown in Figure 19. The flow rate data show that the WWHPs were intermittently operational before October 14, 2013. Excluding this period, a scatter plot of loop temperature and flow rate versus OA temperature is shown in Figure 20. Figure 21 shows the histogram of the chilled water loop TD in 2°F bins and the average flow rate for each bin. From the figures, the following observations can be made:

- No cooling was provided during the winter.
- The chilled water loop supply temperature, which is the leaving temperature of the chilled water from the load-side of each bank of the WWHPs running in cooling mode, ranged between 39.8 and 47°F. At OA temperatures above 85°F, the WWHPs supplied chilled water at temperatures within a close range of 42°F. At lower OA temperatures (i.e., between 55 and 85°F), the chilled water supply temperature had a wider range owing to the intermittent operation of the WWHPs. The chilled water return temperature ranged between 43.8 and 67.1°F.
- The chilled water loop flow rate ranged between 155 and 1,600 gpm, with the maximum approaching 1,600 gpm in the summer and 630 gpm during the shoulder season.
- Most often in the summer, the chilled water loop TD was within −12 and −8°F with an average flow rate of 960–980 gpm. However, on a few days in the summer, the TD was within −4 to −2°F with a very low average flow rate of 200 gpm.

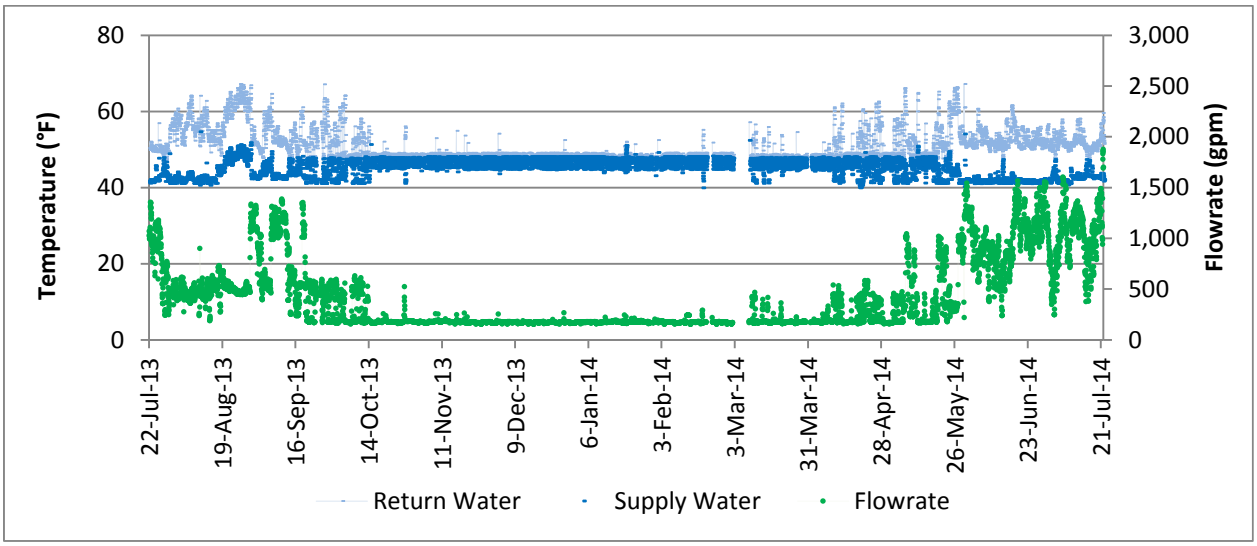

Figure 19. Chilled water loop temperature (blue) and flow rate (green).

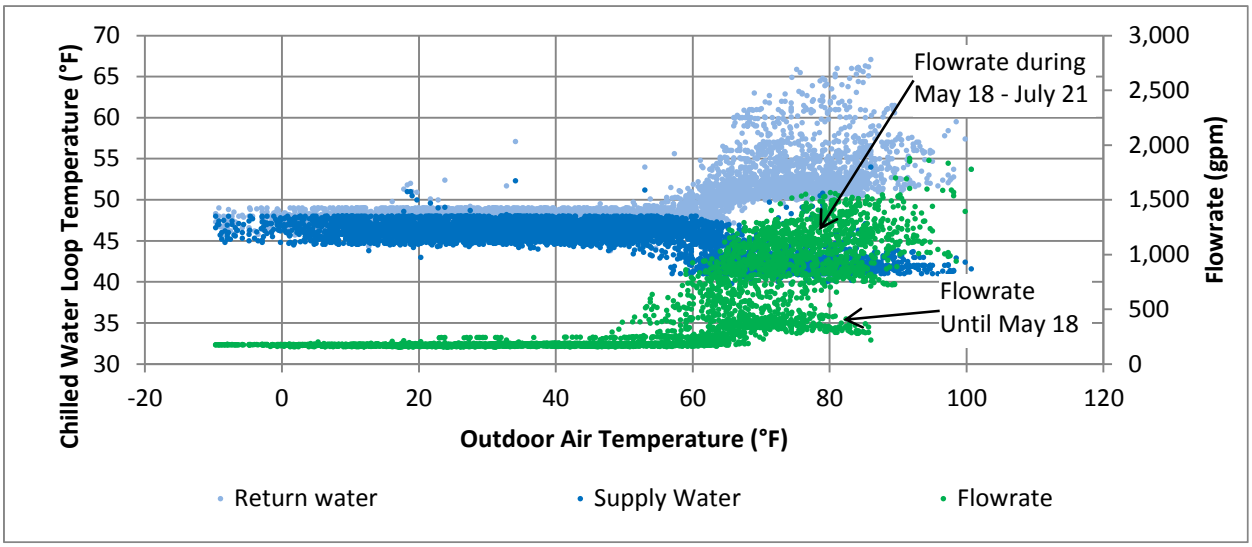

Figure 20. Chilled water loop temperature (blue) and flow rate (green) versus OA temperature.

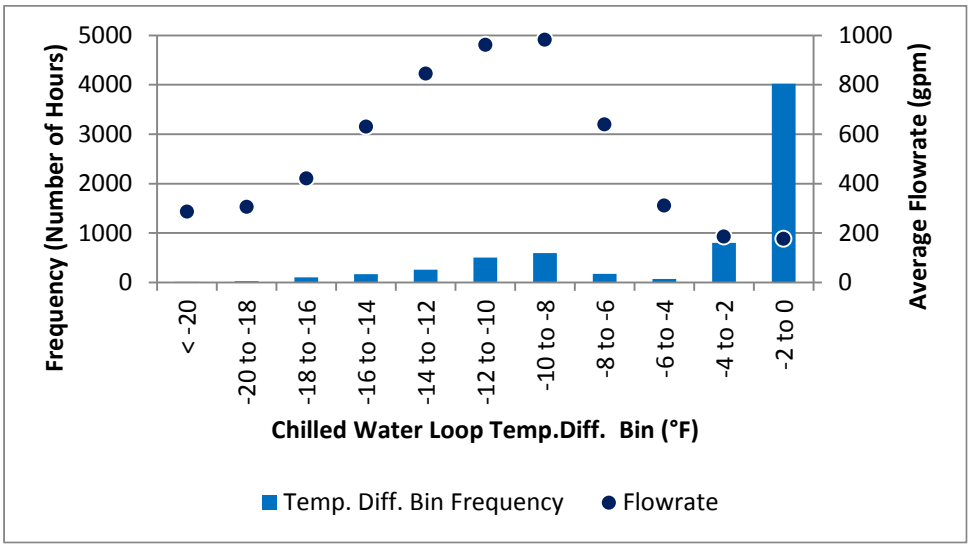

Figure 21. Histogram of chilled water loop temperature differential and bin average flow rate.

2.2.3 Hot Water and Chilled Water Loop Pump Power

Figure 22 shows scatter plots of the daily average pumping power of the circulation pumps in the hot water and chilled water loops versus the average flow rate. The observed trend derived from the aggregated data also shown in this figure.

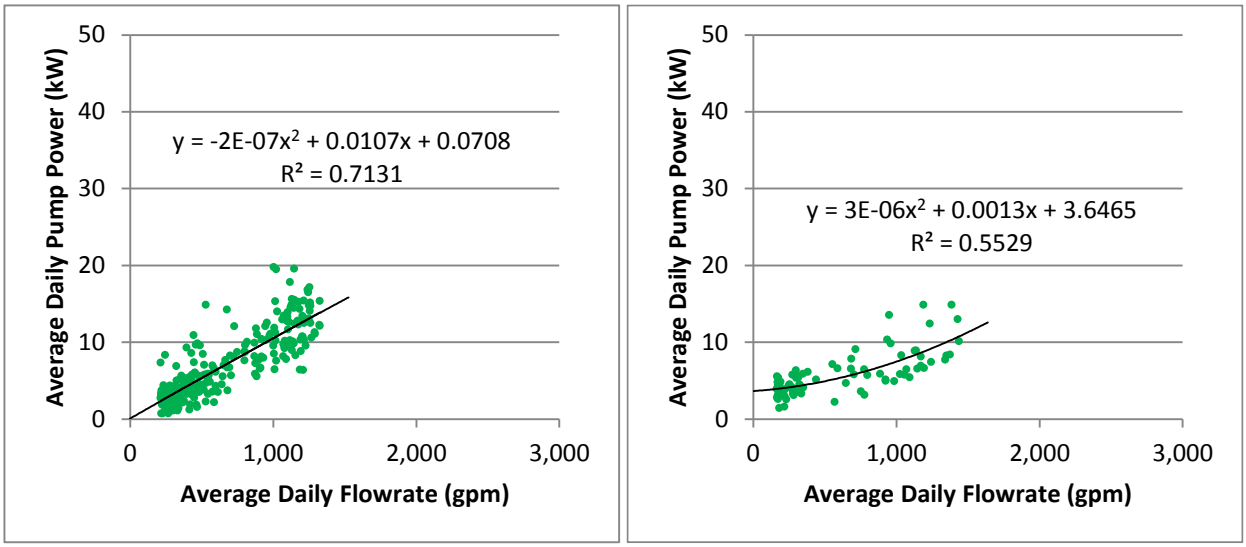

Figure 22. Daily average pump power versus flow rate for hot water loop (left) and chilled water loop (right).

For further investigation of pumping control, the flow rate and TD in the distribution loops were analyzed. Figure 23 shows the histogram of the flow rate in the hot water loop (red) and chilled water loop (blue) and the average TD in each 100 gpm bin of the flow rate in the two loops. It demonstrates a U shape, which indicates good variable-speed pumping control.

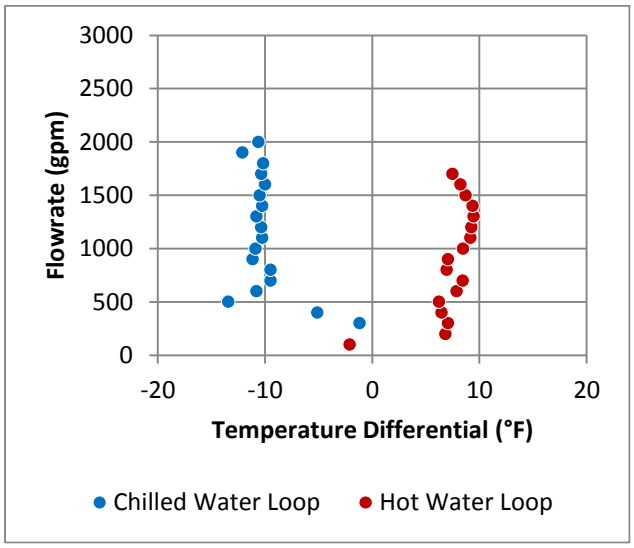

Figure 23. Histogram of hot water (red) and chilled water (blue) loop flow rates and bin average temperature differential.

2.2.4 Heat Transfer in the Load-side Loops

The heat delivered by the WWHPs through the hot water loop was calculated using the hot water loop flow rate and supply and return water temperatures, as expressed in Eq. (2):

$$QH = k * FHW * \frac{(THWS - THWR)}{1000} \quad . \tag{2}$$

Similarly, cooling delivered by the WWHPs through the chilled water loop was calculated using the chilled water loop flow rate and supply and return water temperatures, as expressed in Eq. (3):

$$QC = k * FCW * \frac{(TCWS - TCWR)}{1000} \quad , \tag{3}$$

where

QH = heating delivered by WWHPs (kBtu/h),
FHW = hot water loop flow rate (gpm),
THWS = hot water loop supply temperature (°F),
THWR = hot water loop return temperature (°F),
QC = cooling delivered by WWHPs (kBtu/h),
FCW = chilled water loop flow rate (gpm),
TCWS = chilled water loop supply temperature (°F),
TCWR = chilled water loop return temperature (°F),
k = a factor that incorporates conversion factors and the specific gravity of the fluid, which is 500 Btu/h-gpm·°F for pure water.

The resulting heating and cooling delivered by the WWHPs are plotted in Figure 24 as time series plots. The hourly heating delivered is shown in red and the cooling delivered is shown in blue. The heat pump provided heating during all seasons. During the analysis period, the maximum heating delivered was 8.6 MMBtu/h (178.7 MMBtu/day) in the winter and 2 MMBtu/h in the summer. The maximum cooling delivered was 10.8 MMBtu/h (196.6 MMBtu/day). Aggregating for the analysis period (which is less than

a year by 84 days of the summer and fall seasons), the heating delivered amounted to 17,469 MMBtu and the cooling delivered amounted to 8,472 MMBtu.

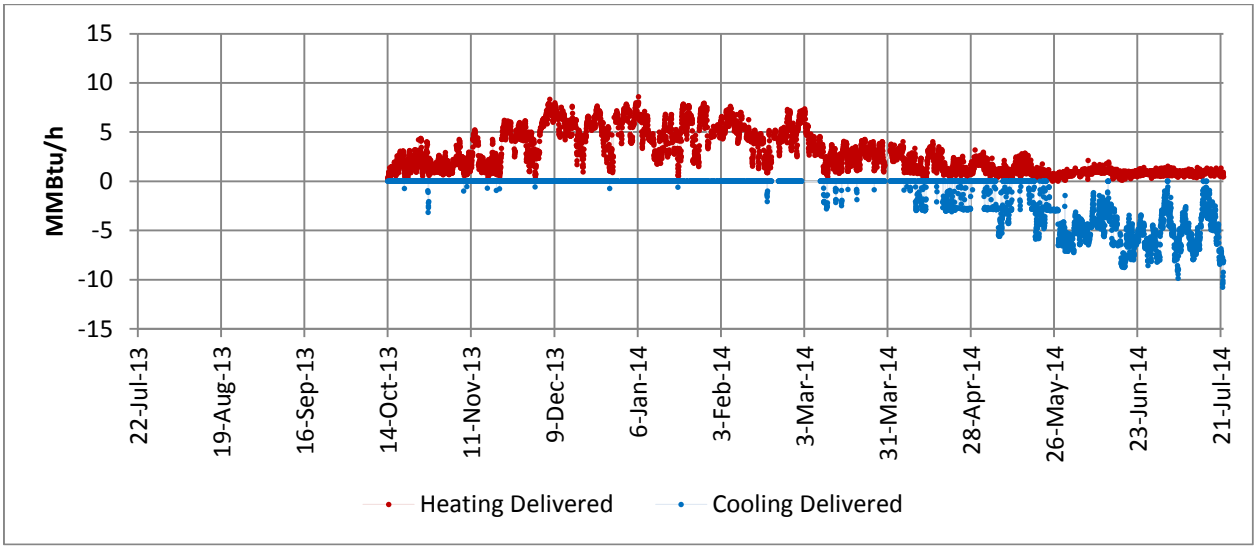

Figure 24. Heating (red) and cooling (blue) delivered by the water-to-water heat pumps.

Figure 25 shows a scatter plot of heating and cooling delivered versus OA temperature. It shows that the WWHPs provided both chilled water and hot water simultaneously when the OA temperature was above 52°F. The scatter plots demonstrate an approximately linear relationship between the delivered heating or cooling energy and OA temperature, with the slope of the heating output changing at about 52°F OA temperature. This trend is useful in predicting values during the periods for which the GSHP system was not in operation or the performance data were missing.

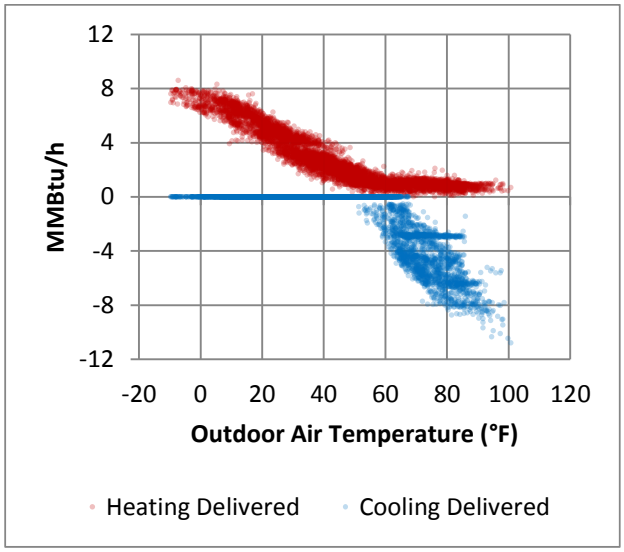

Figure 25. Heating (red) and cooling (blue) delivered by WWHPs versus OA temperature.

By comparing Figure 15 and Figure 25, it is noted that the heat extracted from the ground is smaller than the heating delivered by the WWHPs (because the power consumed by the WWHPs is delivered as heat to the facility), but there is little difference between cooling output and the ground loop heat rejection owing to the simultaneous heating and cooling operation. This helps reduce the ground loop temperature and thus improve the cooling efficiency of the GSHP system.

2.3 GSHP SYSTEM EFFICIENCY

2.3.1 Analysis Methodology

A challenge for evaluating the operational efficiency of this GSHP system is that the power consumption of the WWHPs was not measured. Therefore, it has to be calculated with available measured data and other related information. The approach for calculating the heat pump power consumption, and thereby evaluating the operational efficiency of the GSHP system, is shown in Figure 26.

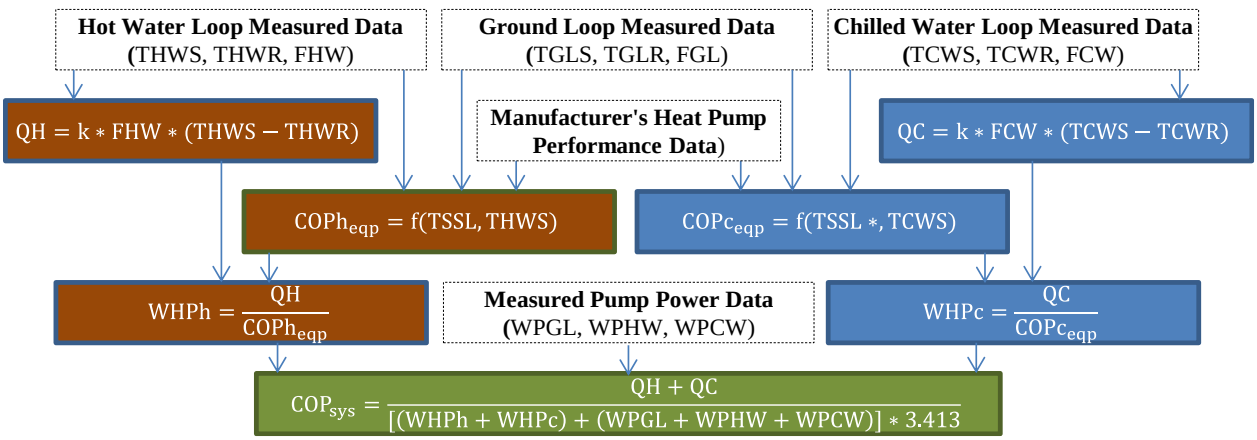

Figure 26. Procedure for determining GSHP system efficiency.

The process can be broken down into four steps:

1. Calculate the heating and cooling delivered to the building (QH and QC, respectively) from the measured supply and return water temperature and flow rate of the hot water loop (THWS THWR, FHW) and the chilled water loop (TCWS TCWR, FCW) as described in Section 2.2.4.

2. Determine the operational efficiency of the WWHPs in heating and cooling modes ($COPh_{eqp}$ and $COPc_{eqp}$), based on the manufacturer's catalog data, with the measured heat pump leaving water temperature on the load-side (THWS or TCWS) and the source-side. Since the source-side leaving water temperature of each individual WWHP ($TSSL_n$) was not measured, it was approximated with the measured ground loop supply water temperature (TGLS) and a fixed TD. Typically, for WWHPs running in the cooling mode, TSSL is 10°F higher than TGLS; and for WWHPs running in heating mode, TSSL is 6°F lower than TGLS.

3. Compute the power consumption of the WWHPs running in heating mode (WHPh) using QH and $COPh_{eqp}$, and power consumption in cooling mode (WHPc) using QC and $COPc_{eqp}$.

4. Finally, calculate the annual operational efficiency of the GSHP system (COPsys) as the ratio of cumulative heating and cooling delivered and the sum of the power consumption of the WWHPs and

circulation pumps (i.e., WPGL, WPHW and WPCW—corresponding to the ground loop pump, hot water loop pump, and chilled water loop pump, respectively), as shown in Eq. (4).

$$\mathrm{COP_{sys}} = \frac{\sum(\mathrm{QH+QC})}{3.413*\sum(\mathrm{WHPh+WHPc+WPGL+WPHW+WPCW})} \ . \tag{4}$$

2.3.2 Operational Efficiency of Heat Pump Equipment

The operational efficiency of the WWHPs (i.e., the COP for heating and COP or EER (energy efficiency ratio) for cooling) was determined based on the heat pump manufacturer's performance data (plotted in Figure 27) and the measured or approximated leaving water temperature at both the source- and load-side of each WWHP (assuming the design water flow rate was delivered to each WWHP).

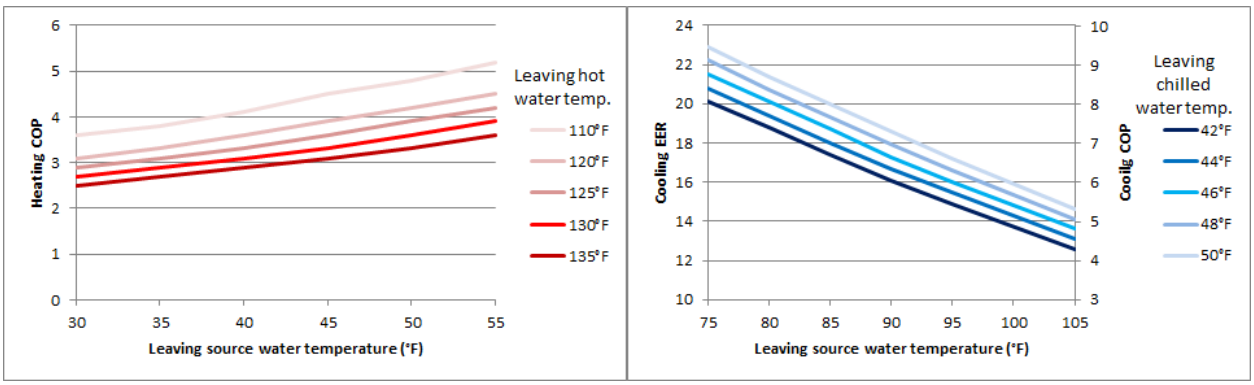

Figure 27. Heat pump efficiency curves in heating (left) and cooling (right) modes.

COP and EER can be determined as a function of the leaving source-side water temperature (T_1) and the leaving load-side water temperature (T_2), as expressed in Eq. (5). The coefficients (C_0 through C_5) in Eq. (5) for the heating and cooling modes are derived from the data shown in Figure 27 and listed in Table 2.

$$\mathrm{COP} = f(T_1, T_2) = C_0 + C_1T_1 + C_2{T_1}^2 + C_3T_2 + C_4{T_2}^2 + C_5T_1T_2 \ , \tag{5}$$

where T_1 is the measured ground-loop supply temperature minus 6°F (TGLS − 6°F) in the heating mode and measured ground-loop supply temperature plus 10°F (TGLS + 10°F) in the cooling mode; and T_2 is the measured hot water supply temperature for the heating mode and measured chilled water supply temperature for the cooling mode.

Table 2. Coefficients in the curve-fit equation for heat pump efficiency at non-rating conditions

Coefficient	Heating COP	Cooling EER
C0	8.240096459	24.93852381
C1	0.118150579	−0.292371429
C2	0.000542857	0.000980952
C3	−0.083025886	0.45452381
C4	0.000279332	0.001488095
C5	−0.000897297	−0.003214286

The heating COP and cooling COP (i.e., cooling EER/3.413) are shown in Figure 28 as a time series plot and in Figure 29 as a scatter plot versus OA temperature. For the analysis period, the heating COP ranged between 3.6 in the winter and 5.7 in the summer. The cooling COP ranged between 5.1 in the summer and 7.9 in the shoulder season.

The heating COP was relatively constant when the OA temperature was lower than 50°F and increased with OA temperature. The COP increase with OA temperature resulted from the heat recovery in the ground loop due to the simultaneous heating and cooling operation of different WWHPs (i.e., warm water rejected from WWHPs running in cooling mode provides heat to the WWHPs running in heating mode).

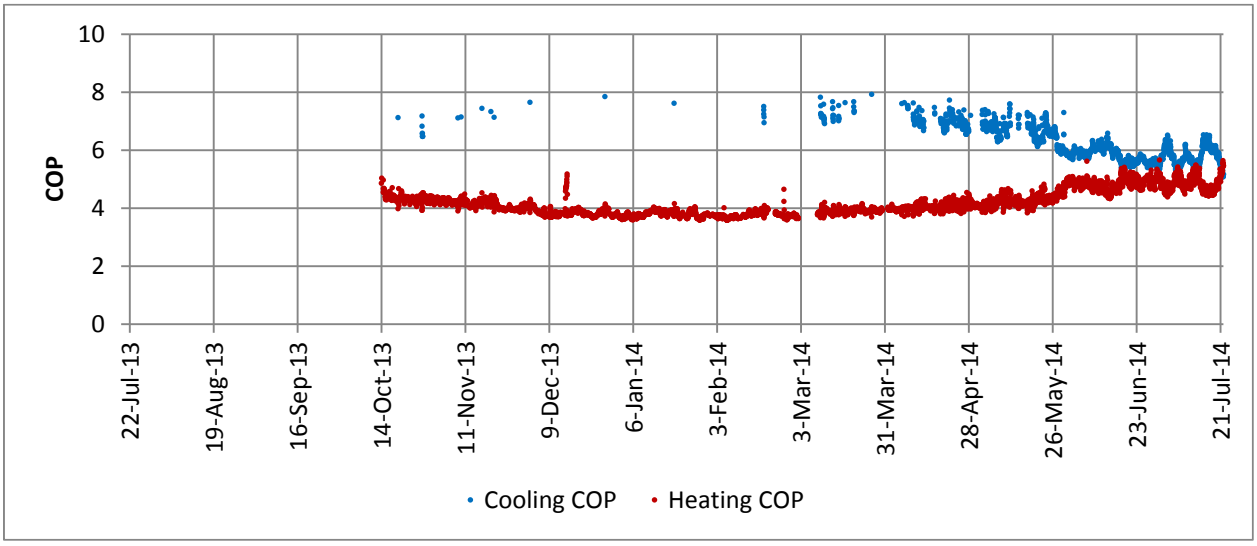

Figure 28. Heat pump COP in heating and cooling modes.

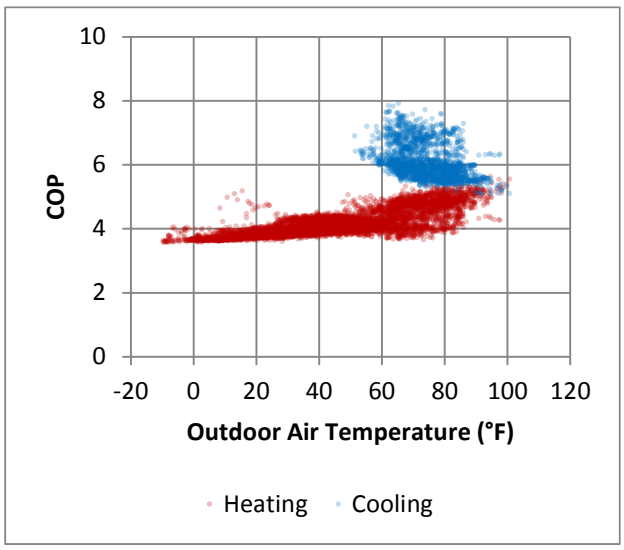

Figure 29. Heat pump COP in heating and cooling modes versus OA temperature.

2.3.3 Heat Pump Power Consumption

The calculated hourly power consumption of the WWHPs in heating and cooling modes is shown in Figure 30 as a time series plot and in Figure 31 as a scatter plot versus OA temperature. The red markers indicate heating mode and blue markers indicate cooling mode. In heating mode, the maximum power consumption was 650 kWh in the winter and 100 kWh in the summer. In cooling mode, the maximum power consumption was 500 kWh in the summer. Aggregating for the analysis period (which is less than a year by 84 days of the summer and fall seasons), the total heat pump power consumption was 1,283,291 kWh in heating mode and 421,424 kWh in cooling mode. The scatter plot in Figure 32 shows that the heat pump power consumption in cooling mode increased almost linearly with the increase in OA temperature. The pattern was reversed for heating operation when OA temperature was below 60°F.

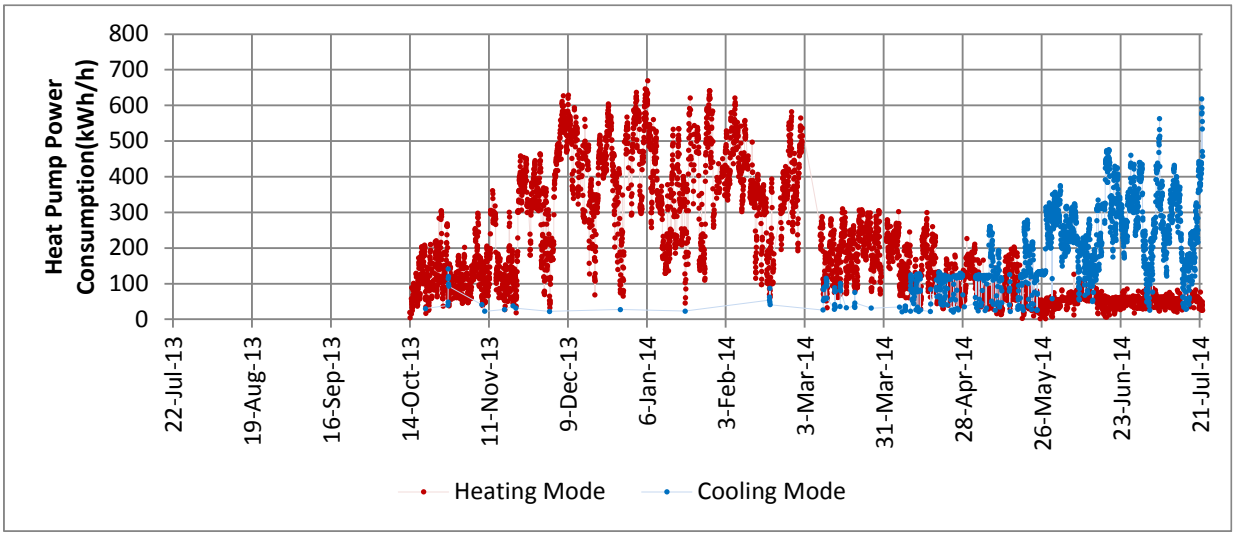

Figure 30. Heat pump power consumption in heating and cooling modes.

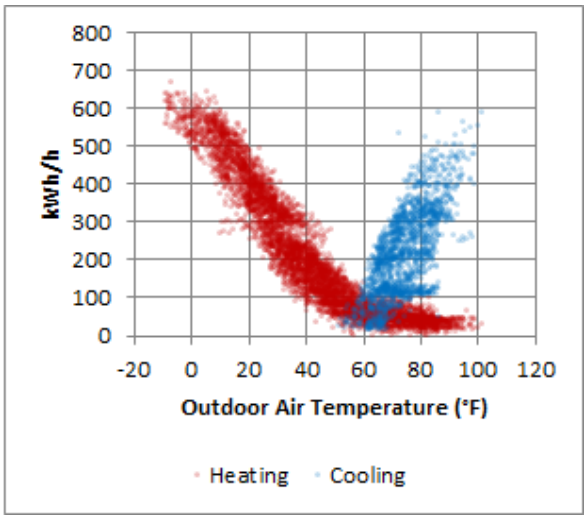

Figure 31. Heat pump power consumption in heating and cooling modes versus OA temperature.

2.3.3.1 Verification of Calculated Heat Pump Power Consumption

To verify that the calculated heat pump power consumption was reasonable, it was compared with the measured total power consumption in the DEC thermal plant facility (as listed in Table 1). In addition to the electricity use by the WWHPs (i.e., the variable to be compared), the DEC total power consumption included GSHP system pump electricity use, electricity use for the 14 WAHPs serving the thermal energy plant, auxiliary electricity use for the radiant floor heating system, and lighting and equipment electricity use in the facility. Figure 32 shows the daily aggregation of measured total power consumption in the DEC thermal energy plant and calculated heat pump power consumption. Figure 33 shows the calculated and measured values versus the OA temperature. As expected, an offset for fixed electricity use (such as lighting) and a small difference in slope for seasonal heating, ventilation, and air-conditioning (HVAC) -related loads in the facility can be observed. Figure 32 shows that the differences are higher during May 18 through June 30; the increase is possibly due to anomalous operation of the GSHP system.

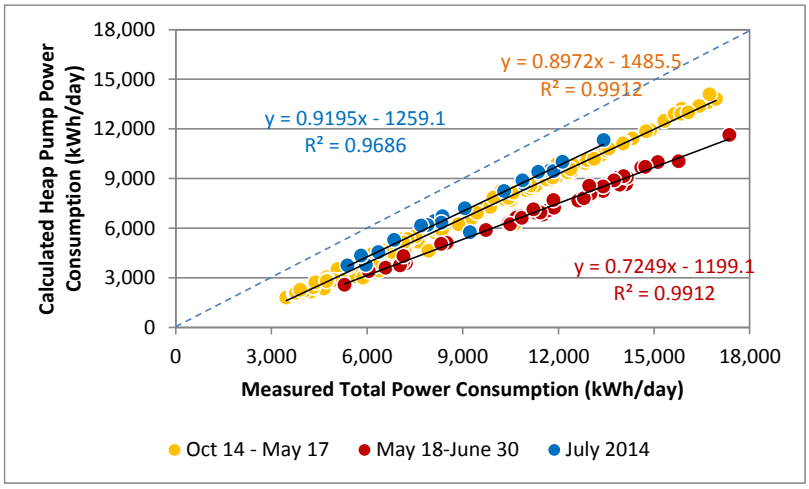

Figure 32. Calculated heat pump power consumption versus measured total DEC power consumption.

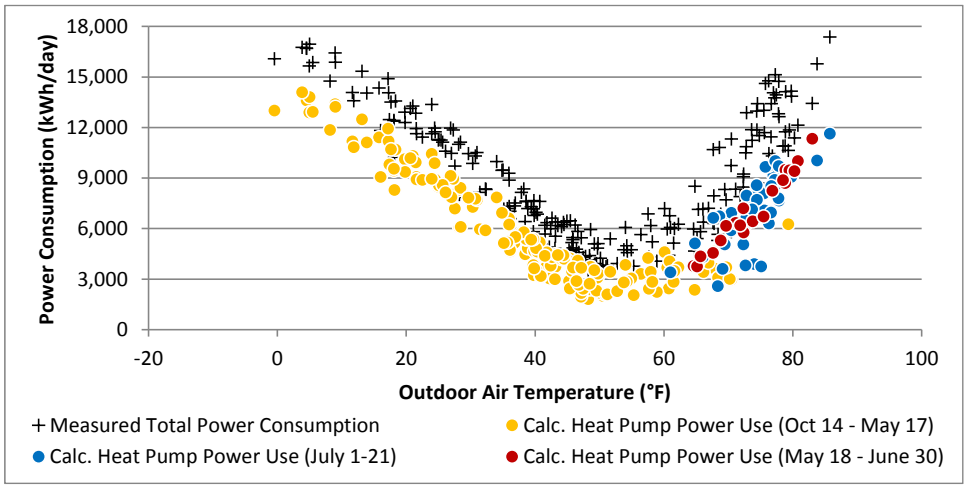

Figure 33. Calculated heat pump power consumption and measured total DEC power consumption versus OA temperature.

2.3.4 Pump Power Consumption

Figure 34 shows the stacked area plot for hourly pumping power consumption in the primary hot water loop, chilled water loop, and ground loop. The maximum hourly total pumping power was over 100 kWh in January and over 70 kWh in July. (It was up to 35 kWh in May and June, which was considered anomalous operation.)

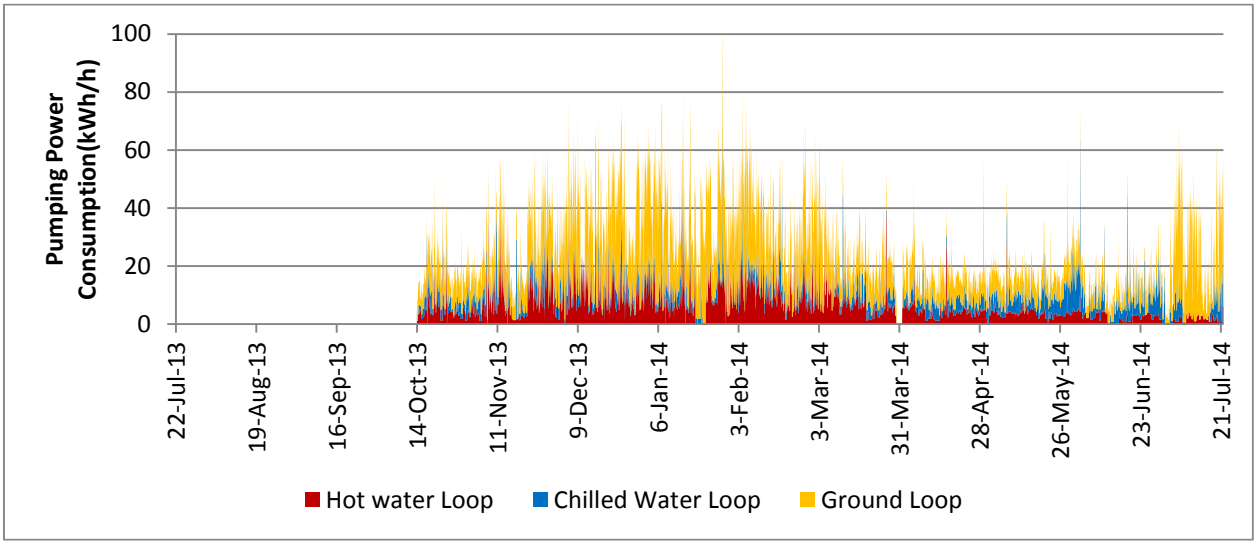

Figure 34. Pumping power consumption in the ground loop, hot water loop and chilled water loop.

Pumping performance is evaluated as the ratio of pumping power consumption to total GSHP system power consumption (referred to as "pumping power fraction"). Figure 35 shows the scatter plot of hourly pumping power fraction versus OA temperature. Excluding May and June (which represent anomalous operation), the pumping power fraction ranged between 3 and 40% and was higher at moderate temperatures.

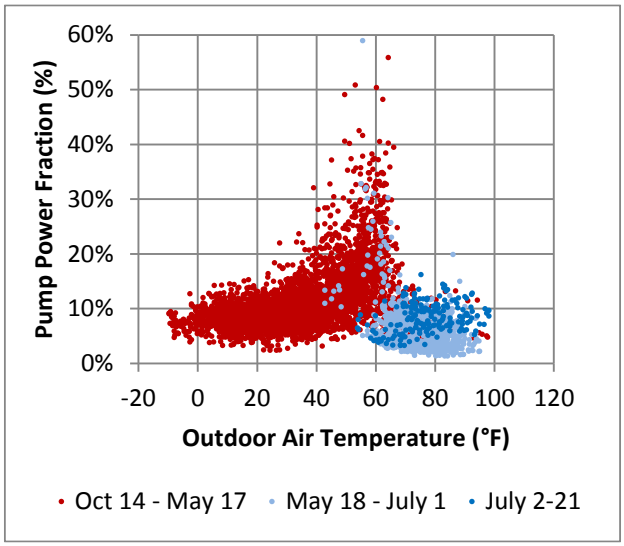

Figure 35. Pumping power fraction versus outdoor air temperature.

Figure 37 shows the histogram of the pumping power fraction for 10°F OA temperature bins. Most of the time, the bin average pumping power fraction was between 8.3 and 13%, with higher bin averages occurring in the 60–70°F bin. The latter finding indicates that the GSHP system may be during the shoulder months.

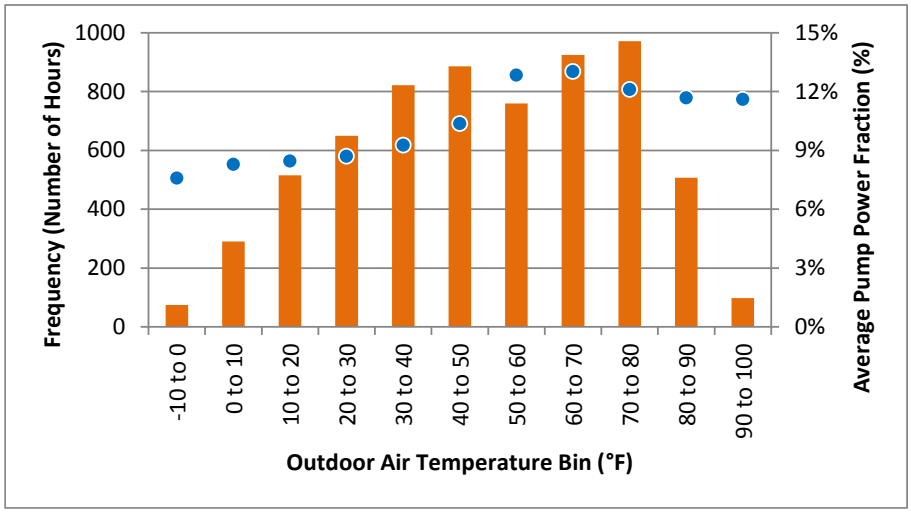

Figure 36. Histogram of bin average pumping power fraction.

2.3.5 Overall Efficiency of GSHP System

Figure 37 shows the overall COP of the GSHP system versus OA temperature. During periods when cooling was not provided, the system COP ranged between 2.8 and 4.7, averaging 3.52. During the shoulder season and summer, when heating and cooling were provided simultaneously by different WWHPs, the system COP ranged between 3.4 and 6.4, averaging 5.3. Note the step change in system effective COP, which accounts for both heating and cooling outputs, at the moderate OA temperature range, when the system starts providing simultaneous heating and cooling.

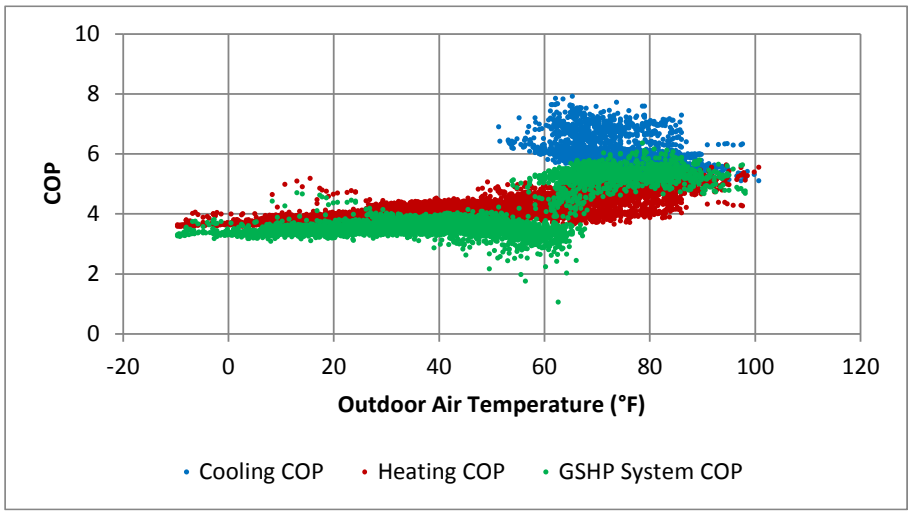

Figure 37. GSHP system COP versus OA temperature.

3. COMPARISON BETWEEN GSHP SYSTEM AND BASELINE SYSTEM

To determine energy savings and other benefits of the GSHP system at this site, the performance of the GSHP system was compared with that of a baseline system in terms of annual energy savings, operating cost savings, and emission reductions. To accomplish this, first the baseline system characteristics were determined. A complete set of measured data for the GSHP system is available for October 14, 2013, through July 21, 2014, which is less than a year, and there exist several periods of missing data. Therefore, it was necessary to predict relevant loads and energy use using known variables such as OA temperature. To do so, curve-fits of daily aggregations of relevant measured and calculated variables were derived based on the data for October 14, 2013, through July 21, 2014. These curve-fits were first validated by comparing the sum of the measured values with the sum of the predicted values; they were then used to determine the energy use for the baseline and GSHP systems for the year 2014.

3.1 BASELINE SYSTEM CHARACTERISTICS

The characteristics of the baseline HVAC system were adopted from the feasibility report prepared by the GSHP system designers for this site (DEC 2009). The baseline HVAC system uses boilers and a water-cooled chiller to provide hot water and chilled water to the detention facility. The pumps in the chilled water and hot water loops and the HVAC system inside the detention facility were assumed to be the same as those that exist for the GSHP system.

As assumed in the feasibility report (DEC 2009), the baseline HVAC system has (1) three 3,450 kBtu/h gas-fired boilers; (2) three 500 ton electric centrifugal chillers (including a redundant chiller). The condenser of the chillers receives cooling water from an axial fan, open-circuit cooling tower through a cooling water loop, with the pumps controlled by a VFD.

The efficiencies of the baseline equipment were determined in accordance with ASHRAE Standard 90.1-2013 minimum performance requirements, which include 79% boiler efficiency, 0.585 kW/ton chiller efficiency, and 40.2 gpm/hp cooling tower efficiency.

3.2 METHODOLOGY FOR DETERMINING ANNUAL ENERGY USE

3.2.1 Procedure for Determining GSHP System Electricity Use

The GSHP system uses electricity to operate the heat pumps and pumps in the ground loop, hot water loop, and chilled water loop to deliver chilled water and hot water to the detention facility. Figure 38 shows the procedure for determining the annual energy use for the GSHP system. From the daily aggregation of measured and calculated data, curve-fits were derived for (1) daily heat pump power consumption as a function of daily average OA temperature and (2) daily total pumping power consumption as a function of the daily heat pump power consumption. Using the OA temperature for 2014, the heat pump and total pumping power consumption were estimated and summed to obtain the annual electricity use for the GSHP system.

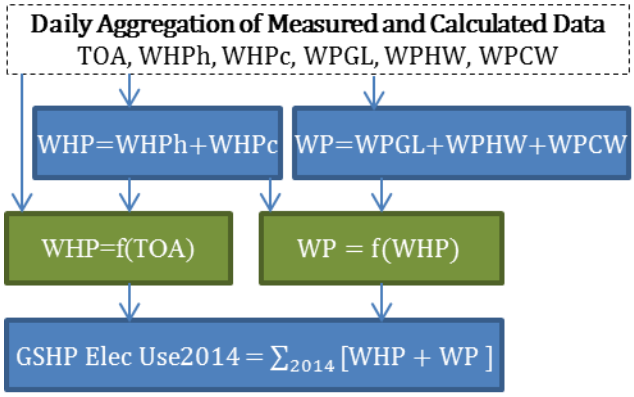

Figure 38. Procedure for determining GSHP system annual electricity use.

3.2.2 Procedure for Determining Baseline Natural Gas Use

The baseline system would use natural gas for the boiler to deliver hot water to the detention facility. Figure 39 shows the procedure for determining the annual natural gas use for the baseline system. From daily aggregation of measured and calculated data, curve-fits were derived for the daily heating load as a function of daily average OA temperature. Using the average OA temperatures for 2014, heating loads were estimated and part-load ratios (PLR) were determined. Using the boiler steady-state efficiency and part-load performance curve (Appendix A; obtained from the DOE-2 simulation program), the boiler operating efficiency was determined. From heating loads and boiler operating efficiency, natural gas use was calculated and summed for the entire year to obtain the annual natural gas use for the baseline system.

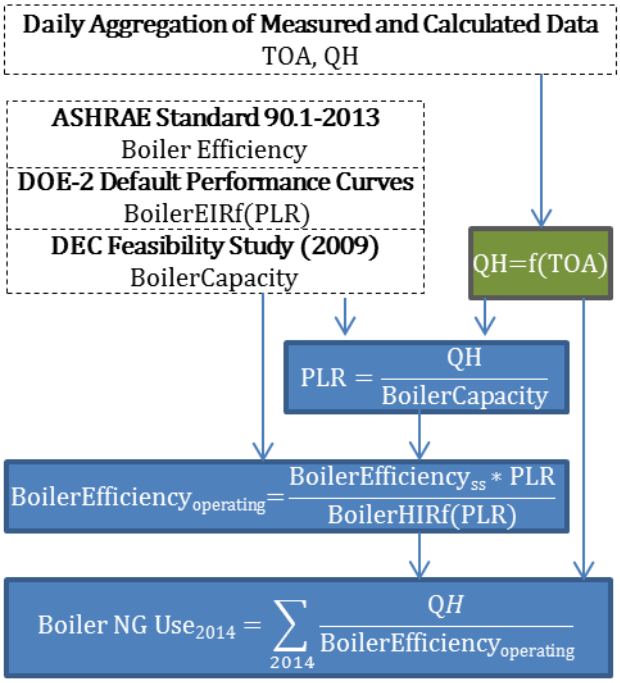

Figure 39. Procedure for determining baseline system annual natural gas use.

3.2.3 Procedure for Determining Baseline Electricity Use

The baseline system would use electricity to operate the chiller, cooling tower, cooling water pump, and hot and chilled water distribution pumps to deliver chilled water and hot water to the detention facility. Figure 40 shows the procedure for determining the annual electricity use of the baseline system. The electricity use of the chiller, cooling tower, cooling water pump, and hot and chilled water distribution pumps were calculated separately and added to determine the total electricity use.

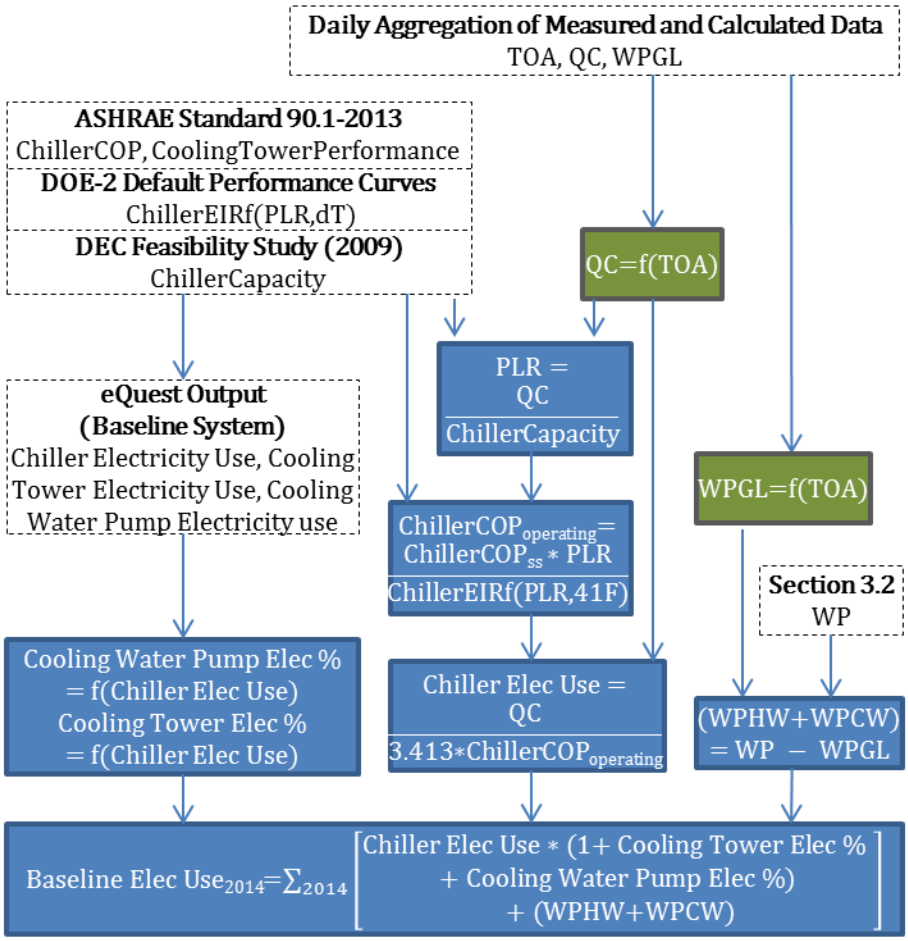

Figure 40. Procedure for determining baseline system annual electricity use.

- From the daily aggregation of measured and calculated data, curve-fits were derived for the daily cooling load as a function of the daily average OA temperature. Using OA temperatures for 2014, cooling loads were estimated and part-load ratios were determined. Using the chiller efficiency at nominal operating conditions (i.e., entering cooling water at 85°F and leaving chilled water at 44°F) and the part-load performance curve for the centrifugal chiller (Appendix A, obtained from the DOE-2 simulation program), the chiller operating efficiency was determined. From the cooling loads and chiller operating efficiency, the electricity use for the chiller was calculated.

- To determine the electricity use for the cooling tower and cooling water pump, an eQUEST simulation of the baseline system was performed to derive the ratios of the cooling tower electricity use and cooling water pump electricity use as a function of the chiller electricity use.

- Since the hot water and chilled water distribution pumps for the baseline system were assumed to be the same as those of the GSHP system, the electricity use by these pumps can be estimated using relevant curve-fits of measured pump power consumption. The measured data show weak correlations of hot and chilled water loop pump power consumption with the possible known variables, and a better correlation between the ground loop pump power consumption and OA temperature. Therefore, the electricity use for the hot and chilled water distribution pumps was calculated by subtracting the predicted ground loop pump electricity use from the total pumping energy use (as determined above for the GSHP system).

The chiller electricity use, cooling tower and cooling water pump electricity use, and hot and chilled water distribution pump electricity use were added and summed for the entire year to obtain the annual electricity use for the baseline system.

3.3 CURVE-FITS OF MEASURED DATA

Curve-fits of daily aggregations of certain measured and calculated variables were needed to determine the annual energy use for the GSHP and baseline systems. To derive these curve-fits, only those days were used for which hourly data were available for 24 hours of the day. The relevant variables for which curve-fits were derived include the following:

- Daily heating and cooling delivered as a function of OA temperature (Figure 41), which are required to predict the baseline energy use. Daily cooling delivered, accounting for only 57 days in the summer, showed a reasonably good linear correlation with OA temperature. For daily heating delivered, because of the small reheating needs during the summer, a second-order polynomial curve fit was found to be a good fit for the 249 data points acquired during the analysis period.

- Daily heat pump power consumption in heating and cooling modes as a function of OA temperature (Figure 42), which is needed to predict the GSHP system energy use. Daily heat pump power consumption in cooling mode, accounting for only 57 days in the summer, showed a reasonably good linear correlation with OA temperature. For daily heat pump power consumption in heating mode, a second-order polynomial curve fit was found to be a good fit for the 230 data points acquired during the analysis period.

- Daily total pumping power consumption as a function of heat pump power consumption (as determined above) (Figure 43), which is needed for predicting the GSHP system energy use. Daily total pumping power consumption, accounting for 172 days, showed a good linear correlation with heat pump power consumption (excluding May 17–June 30 because of anomalous ground loop data and days when heating and/or cooling was provided but the hot or chilled water distribution pump power data were missing).

- Daily ground loop power consumption as a function of OA temperature (Figure 44), which, when subtracted from the total pumping power (as determined above), would provide the hot water and chilled water distribution pump energy use needed to predict the baseline energy use. For daily total pumping power consumption, accounting for 262 days (excluding May 17–June 30 because of anomalous ground loop data), a five-parameter model showed a good correlation with OA temperature.

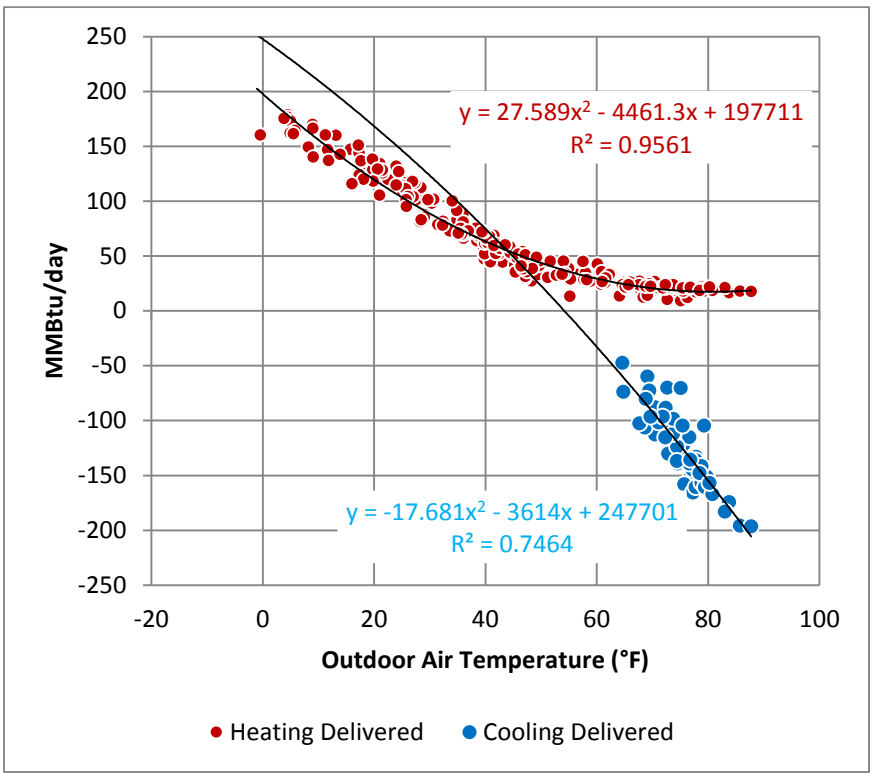

Figure 41. Daily heating and cooling delivered versus OA temperature.

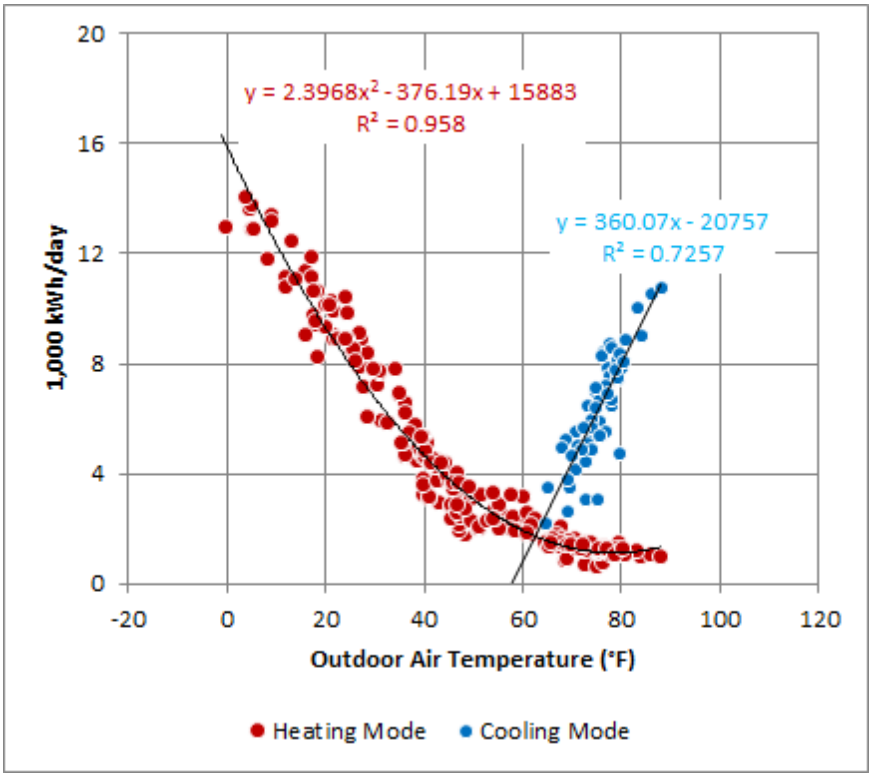

Figure 42. Daily heat pump power consumption in heating and cooling modes versus OA temperature.

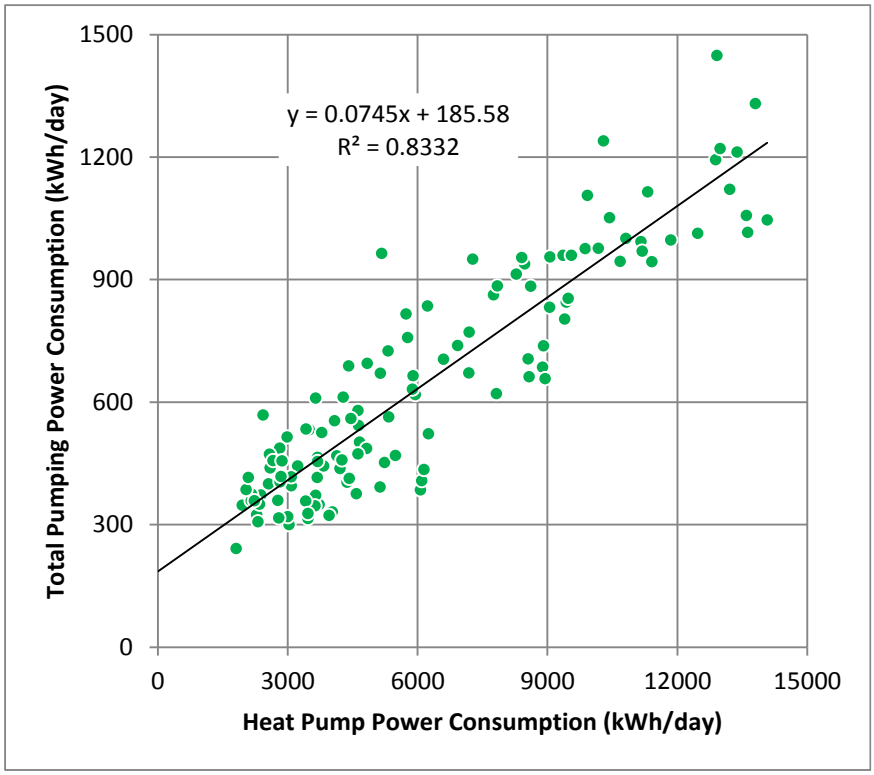

Figure 43. Daily total pumping power consumption versus heat pump power consumption.

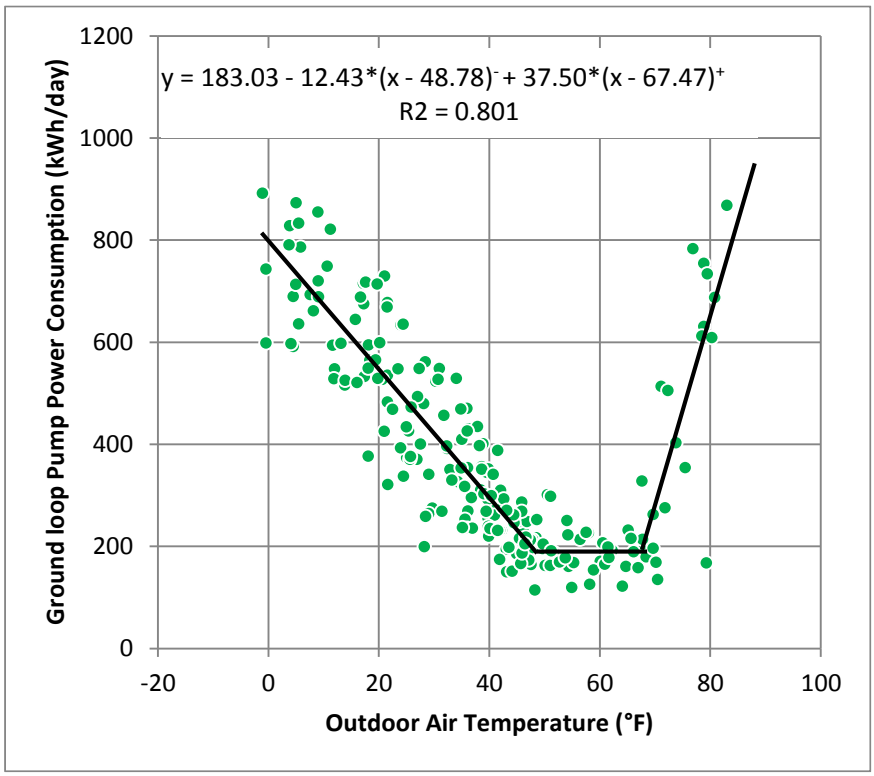

Figure 44. Daily ground loop pumping power consumption versus OA temperature.

Table 3 compares the measured sums and predicted sums for the variables. The sums include days for which measured data are available for 24 hours of the day. Days with partially missing hourly data were excluded. It shows that the percentage difference between the measured and predicted sums is negligible for all variables, except for the 2.4% difference in the total pumping power consumption. This is because the predicted daily total pumping power consumption is based on daily heat pump power consumption, which itself is predicted based on another curve-fit. Although the sum of the predicted heat pump power consumption matches with the sum of the measured values, the daily predicted values differ from the measured values. Overall, the small difference, if any, between the sums of the predicted and measured values validates that these curve-fits are a good tool for predicting the annual energy consumption of the GSHP and baseline systems.

Table 3. Difference between measured and predicted variables

Predicted variables	Curve-fit	No. of days accounted for	Sum of measured values	Sum of predicted values	Percentage difference
Heating delivered, kBtu	QH = f(TOA)	249	15,400,707	15,400,206	0.0%
Cooling delivered, kBtu	QC = f(TOA)	57	7,134,532	7,134,716	0.0%
Heat pump power consumption in heating mode, kWh	WHPh = f(TOA)	230	1,003,686	1,003,599	0.0%
Heat pump power consumption in cooling mode, kWh	WHPc = f(TOA)	57	362,683	362,651	0.0%
Total pumping power consumption, kWh	WP = f(WHP)	172	81,405	77,883	2.4%
Ground loop pump power consumption, kWh	WPGL = f(TOA)	262	82,435	82,435	0.0%

3.4 ANNUAL ENERGY USE FOR GSHP SYSTEM

Using the curve-fits for heat pump energy use (Figure 41) and total pumping energy use (Figure 42) as a function of daily average OA temperature, the annual energy use for the GSHP system was predicted for 2014. Figure 45 and Figure 46 show the predicted monthly heat pump electricity use and total pumping electricity use for 2014. The predicted heat pump electricity use was as high as 272,764 kWh/month in January and 238,009 kWh/month in August, and as low as 95,303 kWh/month in October. For linearly related total pumping electricity use, the pumping power fraction was as low as 8.7% in January and 9% in July and August, and as high as 11.9% in October.

Annually, the electricity use for the GSHP system was 2,536,264 kWh, which included 2,297,373 kWh consumed by the WWHPs and 238,891 kWh used by the pumps. The pumps accounted for 9.42% of the total electricity use of the GSHP system.

As shown in Figure 47, the average COP of the GSHP system ranged between 3.4 and 5.2, with higher COPs in the shoulder months because of the simultaneous heating and cooling. The annual average COP of the GSHP system was 4.3. These estimates are close to the COPs calculated using the measured data, confirming the validity of the approach used to predict the annual energy use.

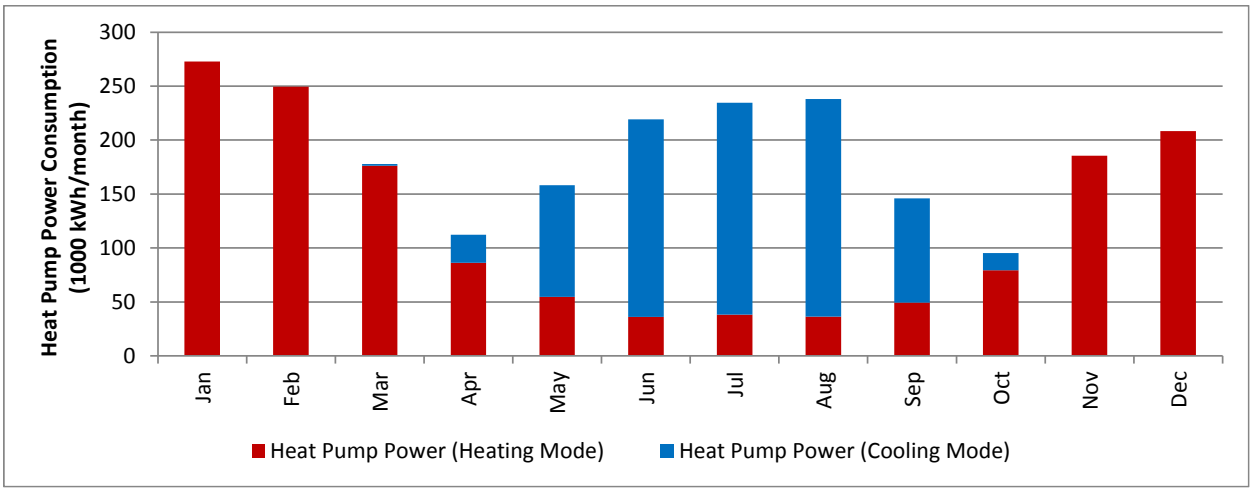

Figure 45. Predicted monthly heat pump power consumption (year 2014).

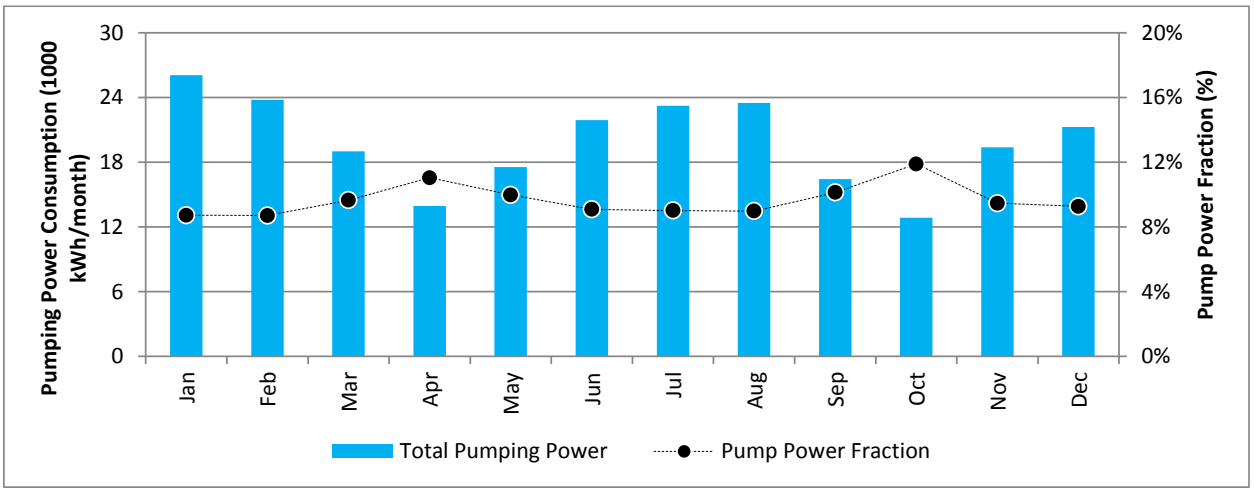

Figure 46. Predicted monthly power consumption of the circulation pumps (year 2014).

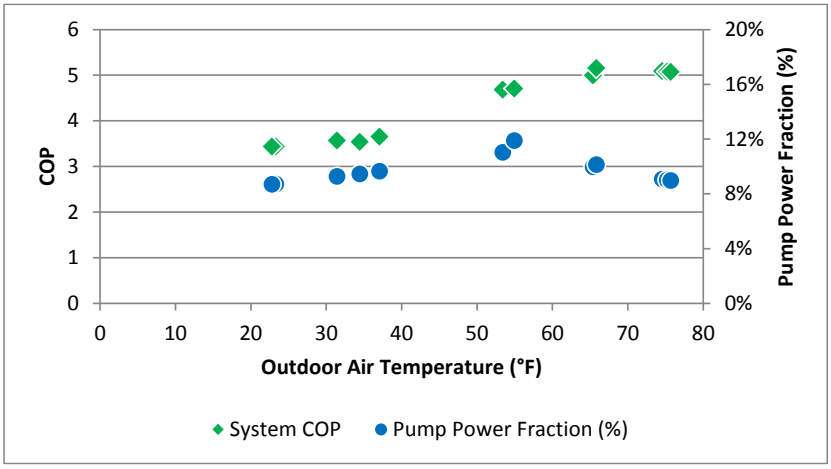

Figure 47. Predicted COP and pump power fraction (year 2014).

3.5 ANNUAL ENERGY USE FOR BASELINE SYSTEM

Using the procedure described in Sections 3.2.2 and 3.2.3 and using curve-fits, the annual natural gas and electricity use for the baseline system was predicted for 2014. Figure 48 and Figure 49 show the predicted monthly natural gas use and electricity use by its components for 2014. The natural gas use was as high as 4,439 MMBtu/month in January, was as low as 723 MMBtu/month in June, and added up to 25,234 MMBtu/year.

The predicted electricity use was as high as 250,537 kWh/month in August, was as low as 9,423 kWh/month in December, and added up to 1,145,753 kWh/year. That amount included 803,986 kWh/year for the chillers, 12,256 kWh/year for the cooling tower, 228,419 kWh/year for the cooling water pump, and 101,091 kWh/year for the hot and chilled water distribution pumps.

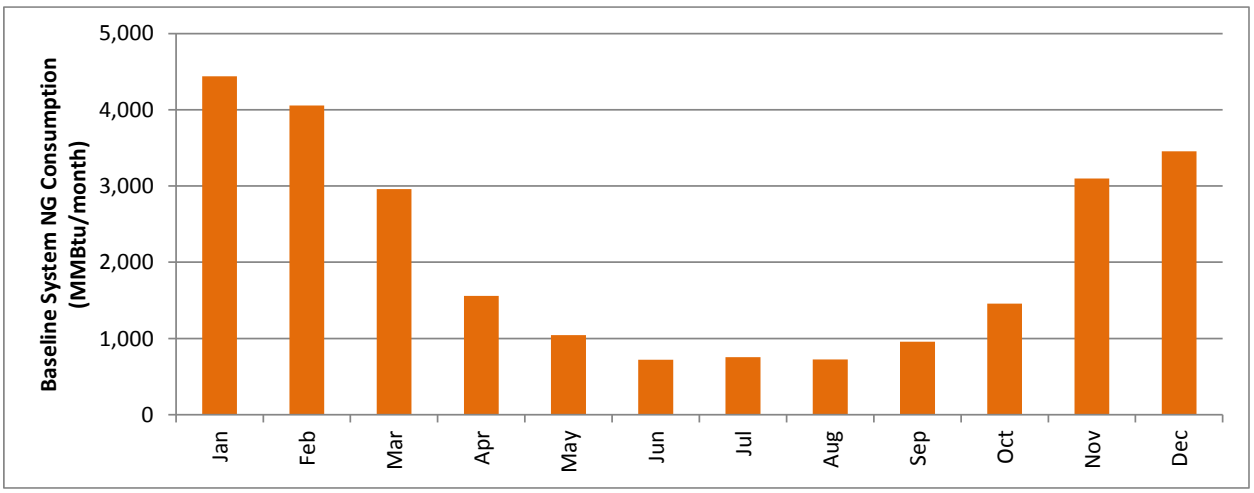

Figure 48. Predicted monthly natural gas consumption by baseline system (year 2014).

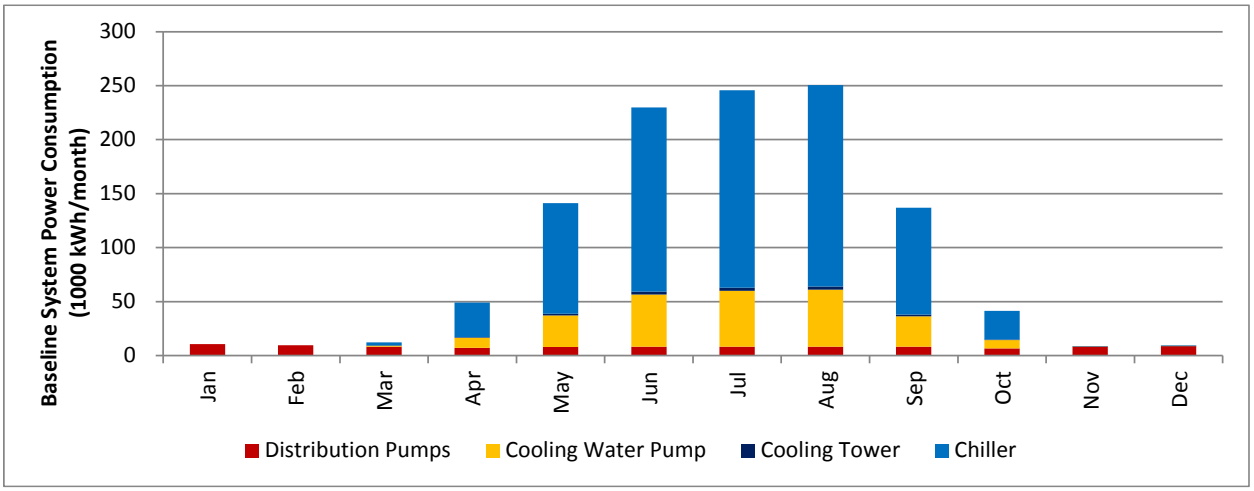

Figure 49. Predicted monthly power consumption by baseline system (year 2014).

3.6 COMPARISON BETWEEN THE GSHP SYSTEM AND BASELINE SYSTEM

3.6.1 Savings Associated with Energy Use

Figure 50 compares the annual source energy use, energy costs, and CO_2 emissions resulting from the two systems while they satisfy the same demands for space heating and space cooling. The source energy consumption and the equivalent CO_2 emissions (CO_2e) for the two systems were calculated using the source energy conversion factors and CO_2e emission factor for electricity and natural gas in the region where the demonstration project is located. According to a report from the National Renewable Energy Laboratory (Deru and Torcellini 2007), the site-to-source energy conversion factor for electricity is 3.443 per unit of delivered electricity and 1.092 for natural gas. The CO_2e emission factors are 1.74 lb/kWh of delivered electricity for the eastern region, and 27.8 lb for pre-combustion plus 123 lb for on-site combustion per 1,000 ft^3 of natural gas. The energy costs for the two systems were calculated on the basis of a $0.0873/kWh price for electricity and a $7.27/MCF price for natural gas for commercial customers in Nebraska in 2014, as reported by the Energy Information Administration (EIA 2016a, b).

Annually, the GSHP system saved 11,215 MMBtu of source energy and reduced CO_2 emissions by 1,507,514 lb compared with the baseline system. Percentage-wise, the **GSHP system reduced source energy consumption by 27.3% and CO_2 emissions by 25.5% compared with the baseline system**. The resulting energy cost savings were $67,927 per year (23.5% savings).

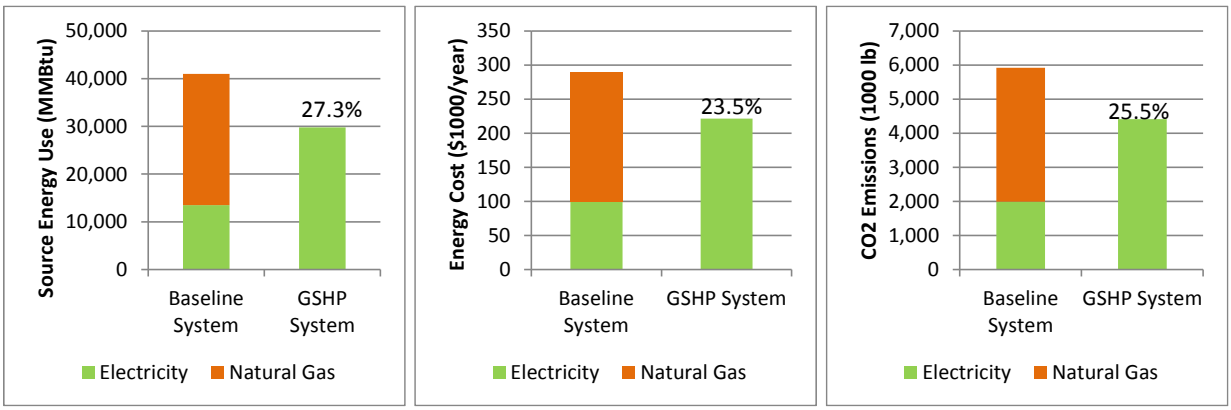

Figure 50. Comparison of source energy use (left), energy cost (center), and CO_2 emissions (right) for the GSHP and baseline systems.

3.6.2 Savings Associated with Water Use

The GSHP system would also result in water savings because it does not use a cooling tower and thus does not consume any water. To determine the avoided cost of water for the GSHP system, water usage by the baseline cooling tower was estimated based on the cooling loads, and the cost of water was calculated using water rates in the city of Lincoln, Nebraska.

Figure 51 shows the histogram of the cooling loads. In 2014, the chillers of the baseline system would operate for a total of 4,335 hours and provide up to 973 tons of cooling. Assuming a 10°F condenser approach temperature (i.e., the difference between the condenser refrigerant temperature and the exiting condenser water temperature) and five cycles of concentration for the cooling tower (Boyd 2011), the makeup water needed by the baseline cooling tower was calculated for each bin of the cooling load (also shown in Figure 51), which adds up to 3.106 million gallons/year.

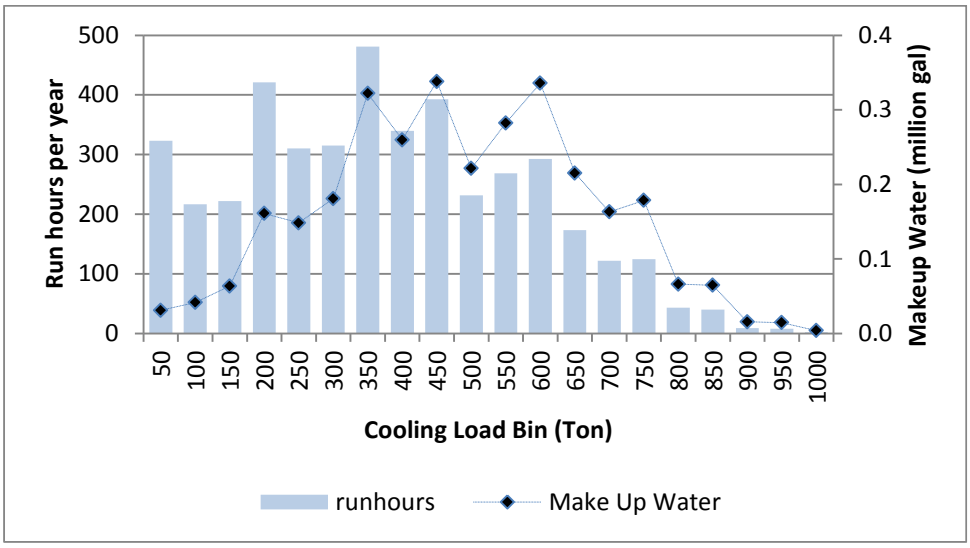

Figure 51. Histogram of cooling loads and bin total water use in the cooling tower of the baseline system.

Figure 52 shows the estimated water use of 3.106 million gallons/year (i.e., 4,153 units/year; 1 unit = 748 gallons) distributed over 12 months in proportion to the monthly cooling loads. Using nonresidential water rates for the city of Lincoln, Nebraska, (i.e., $1.344 per unit for up to 80 units and $1.911 per unit for additional units of water usage) (figures from city of Lincoln, 2016), the monthly cost for water was calculated (also shown in Figure 52). The monthly water costs added up to $7,590 per year. This was the estimated water cost savings resulting from the installation of the GSHP system.

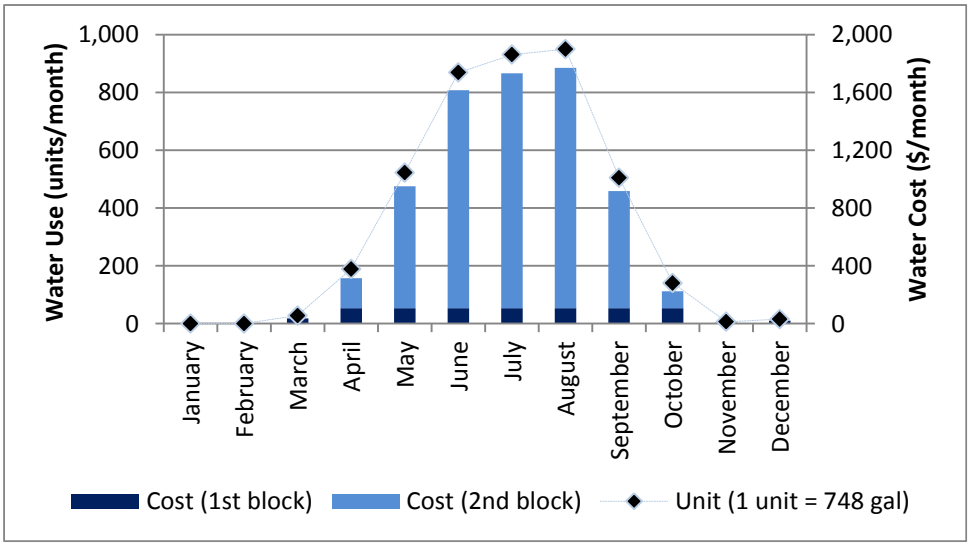

Figure 52. Monthly water use and cost of water for the baseline system.

3.6.3 Installation Cost and Cost Effectiveness

Based on the cost data provided by DEC, the installation cost of the GSHP system was $2,981,188, i.e., $2,130/ton or $11/ft^2. That cost was lower than the typical GSHP system installation cost, since it

accounted only for the equipment in the energy plant and excluded the HVAC system in the detention facility. The cost breakdown is shown in Figure 53. The cost of the geothermal loop field (i.e., drilling vertical bores and installing the ground heat exchangers) was $1,393,481 (i.e., $6.96 per foot of bore depth), which contributed a significant portion (46.7%) of the total cost. The cost of the heat pumps accounted for 32.6% of the total cost.

The baseline plant cost was adopted from the feasibility study report provided by DEC (DEC 2009). It included $1,241,190 (36.9% of the total) for the cooling equipment, $560,100 (16.7%) for the heating equipment, $850,200 (25.3%) for piping, and $709,340 (21.1%) for pumps and VFDs. Based on these figures, the cost of installing the GSHP system would be $379,651 less than the cost of installing the baseline system, which would result in an instantaneous payback. [1]

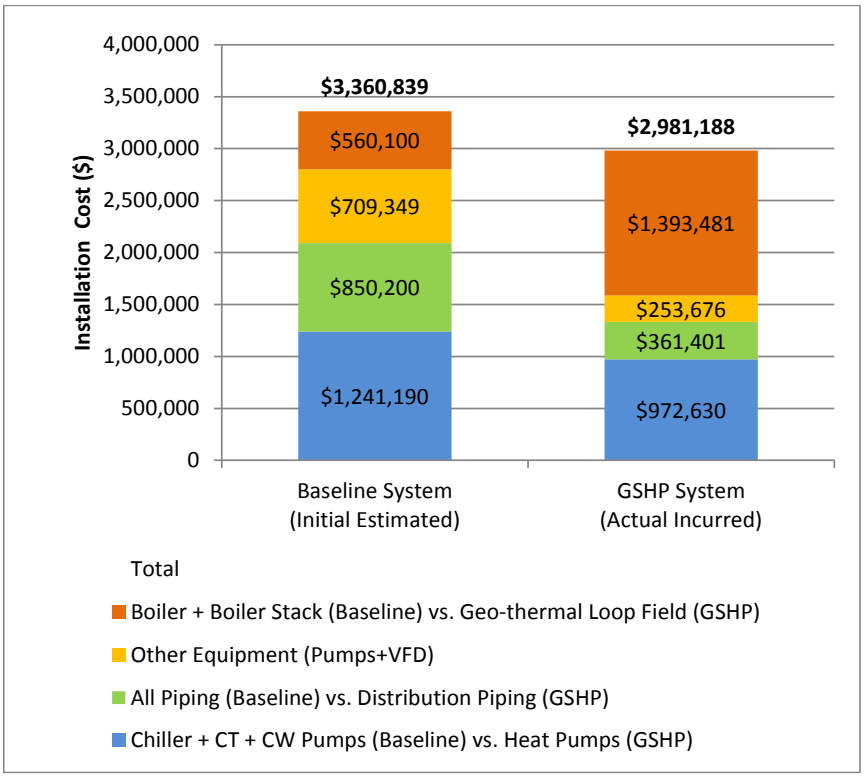

Figure 53. GSHP system installation cost breakdown.

[1] The actual costs of the baseline equipment were also obtained from 2016 RSMeans data to verify the estimated costs in the feasibility study compared with average costs in North America. Based on the RSMeans cost data for the chiller, cooling tower, and boiler, the total baseline system cost would be 18.5% less than that estimated in the feasibility report. The result of that comparison was a cost premium of $242,641 for the GSHP system. Based on this cost premium and the annual energy and water cost savings of $75,517 calculated in Sections 3.6.1 and 3.6.2, the simple payback period for the demonstrated GSHP system would be 3.2 years.

4. CONCLUSION

This case study analyzes the energy performance of a central GSHP system installed in the Lancaster County ADF thermal plant in Lincoln, Nebraska. The actual performance of the GSHP system was analyzed based on available measured data from October 14, 2013, through July 21, 2014. Using curve-fits of available measured data, the annual energy consumption of the GSHP system was predicted for the year 2014 and compared with that of a baseline system—a central plant with hot water boilers and water-cooled chillers with the minimum allowed efficiencies specified in ASHRAE Standard 90.1-2013. The comparison was also made in terms of source energy savings, operating cost savings, cost-effectiveness, and environmental benefits. The following sections summarize the results of the analysis, the lessons learned, and recommendations for improving the system operation and performance data collection.

4.1 ENERGY PERFORMANCE AND COST-EFFECTIVENESS

The ground loop supply water temperatures ranged between 43 and 75°F and were relatively stable compared with the OA temperature (up to 53.7°F higher than the OA temperature on the coldest day and up to 31.6°F lower than the highest OA temperature during the analysis period). Predicting over a one-year period (2014), the GSHP system rejected more heat to the ground than it extracted from the ground. The estimated ratio between heat rejection and heat extraction was 1.6.

The maximum TD of the ground loop was up to 5°F in the heating season, up to 7°F in July, and over 15°F in May–June. Since each individual heat pump usually operated with a 10°F TD in cooling mode and a 6°F TD in heating mode, the measured TD indicates the expected operation in the winter and July but underflow in the ground loop during May–June. The flow rates in the ground loop were 1,000 gpm in May–June and the shoulder season, 2,900 gpm in July, and up to 2,300 gpm in the winter.

The pumping power fraction based on measured data averaged as low as 8.3% in the winter, 11.7% in the summer, and as high as 13% in the shoulder season. Note that this pumping power accounts only for the primary pumps in the energy plant; the distribution pump in the detention facility is not included owing to a lack of data.

Aggregated by month, the operational heating and cooling efficiency of the heat pump was 3.6–5.7 and 5.1–7.9, respectively. If heating and cooling operation are combined, the effective efficiency of the GSHP system, which accounts for the power consumption of the primary pumps, averaged 3.5 in the winter and 5.3 in the summer.

Compared with the baseline system (i.e., steam boilers with 79% thermal efficiency and water-cooled chillers with 0.585 kW/ton efficiency), the GSHP system achieved 27.3% source energy savings and 25.5% CO_2 emission reductions, annually. Based on a commercial electricity rate of $0.0873/kWh and a natural gas rate of $7.27/MCF in Nebraska, the reduced energy consumption resulting from GSHP system would result in a $67,927 operating cost savings per year.

In addition, the GSHP system would save 3.106 million gallons of water per year as a result of eliminating the need for a cooling tower. Using nonresidential water rates for the city of Lincoln, Nebraska, (i.e., $1.344 per unit for up to 80 units and $1.911 per unit for additional units of water usage; 1 unit = 748 gallons), the GSHP system would result in $7,590 per year of avoided costs for water.

Table 4 provides a comparison between the two systems. Based on the cost of the baseline system estimated in the original feasibility study for this GSHP demonstration project, and the actual installed

cost of the GSHP system, the cost of installing the GSHP system would be $379,651 less than the cost of installing the baseline system, which would result in an instantaneous payback.

Table 4. Summary comparison between the GSHP system and the baseline system

	Baseline system	GSHP system	Savings
Source energy use (MMBtu)	41,019	29,804	27.3%
CO_2 emissions (1,000 lb)	5,920	4,413	25.5%
Total annual energy cost ($)	289,343	221,416	23.5%
Cooling tower water use (million gallons)	3.106	–	100%
Cost of water ($)	7,590	–	100%

4.2 LESSONS LEARNED

- The site provided data from April 2013 through September 2015. However, pump data after July 21, 2014, were invalid because data from the pump's VFD and the kWh accumulator in the drives had maxed out, and they needed to be reset. Such issues should be avoided by regularly checking monitored data.

- The pattern in the relationship between the flow rate and the TD of the ground loop is a useful indicator for evaluating the variable-speed pumping performance. Ideally, the pattern should be a U shape, which indicates that the TD is kept at the design value while the flow rate varies within a large range. At this site, measured data showed a V-shape pattern with a substantially high minimum flow rate, which indicated excessive pumping in the ground loop at part-load conditions.

4.3 RECOMMENDATIONS FOR FURTHER IMPROVEMENTS

- The measured hot water supply temperature ranged between 115 and 125.7°F, even in the summer. To further improve the operational efficiency of the GSHP system, the hot water supply temperature could be reduced in the summer. According to the specification of the WWHP, the normal range of the hot water supply temperature is 110–135°F. Lowering the hot water supply temperature from 120 to 110°F would result in a ~0.5 COP improvement (a 9–14% increase).

- The performance of the ground loop pumps could be improved by varying the flow rate to keep the TD in the ground loop at the design value. The ground loop measured data show a small TD at part-load conditions and a substantially high minimum flow rate, which indicate excessive pumping at part-load conditions.

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APPENDIX A. DOE-2 EQUIPMENT PERFORMANCE CURVES

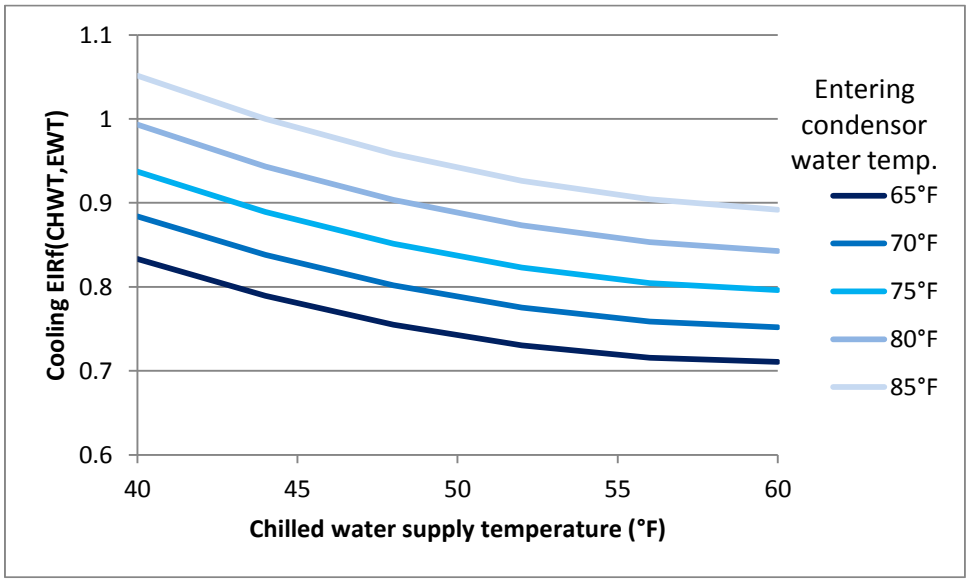

Figure A-1. Chiller performance curve: ChillerEIRf(CHWT, EWT).

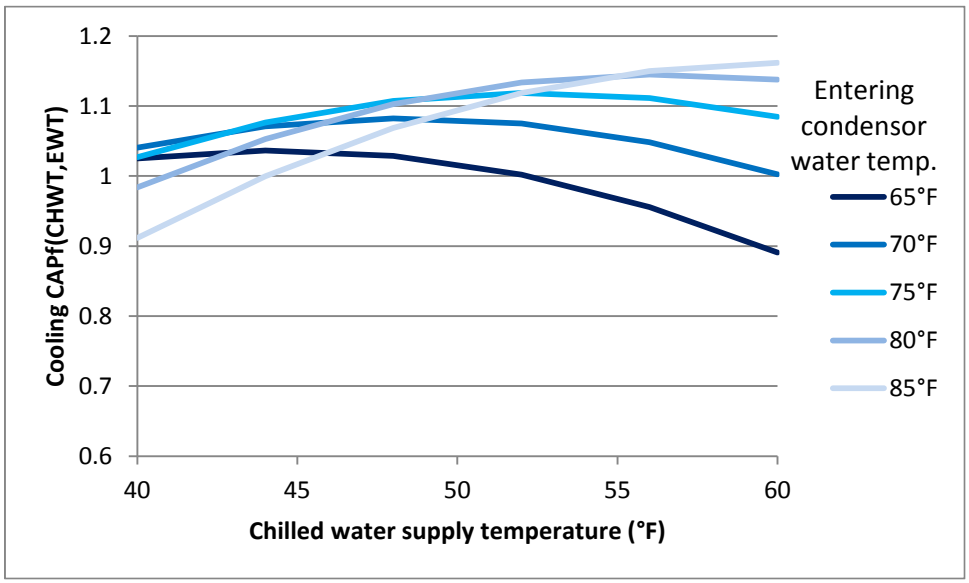

Figure A-2. Chiller performance curve: ChillerCapacityf(CHWT, EWT).

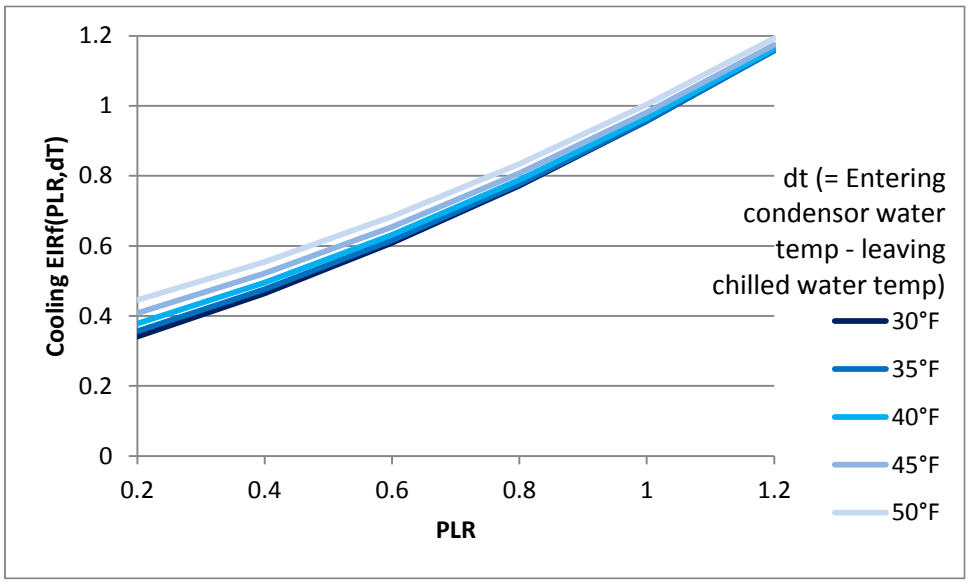

Figure A-3. Chiller performance curve: ChillerEIRf(PLR, dT).

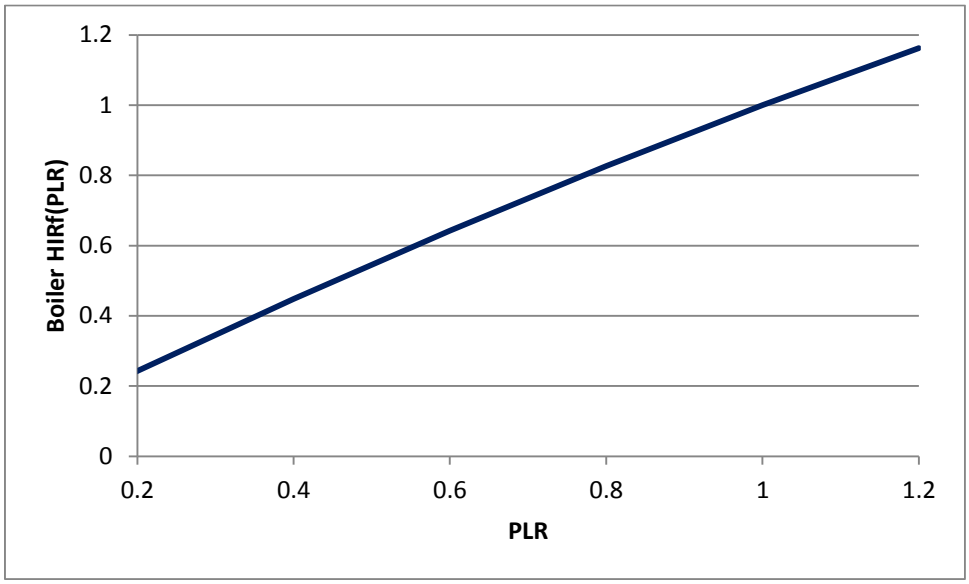

Figure A-4. Boiler performance curve: BoilerEIRf(PLR).