ORNL/TM-2016/644

Final Report:

Case Study for the ARRA-Funded Ground-Source Heat Pump Demonstration at Ball State University

Piljae Im
Xiaobing Liu
Hugh Henderson

October 2016

Approved for public release. Distribution is unlimited.

OAK RIDGE NATIONAL LABORATORY
MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

DOCUMENT AVAILABILITY

Reports produced after January 1, 1996, are generally available free via US Department of Energy (DOE) SciTech Connect.

Website http://www.osti.gov/scitech/

Reports produced before January 1, 1996, may be purchased by members of the public from the following source:

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
Telephone 703-605-6000 (1-800-553-6847)
TDD 703-487-4639
Fax 703-605-6900
E-mail info@ntis.gov
Website http://www.ntis.gov/help/ordermethods.aspx

Reports are available to DOE employees, DOE contractors, Energy Technology Data Exchange representatives, and International Nuclear Information System representatives from the following source:

Office of Scientific and Technical Information
PO Box 62
Oak Ridge, TN 37831
Telephone 865-576-8401
Fax 865-576-5728
E-mail reports@osti.gov
Website http://www.osti.gov/contact.html

ORNL/TM-2016/644

Energy and Transportation Science Division

FINAL REPORT: CASE STUDY FOR ARRA-FUNDED GROUND-SOURCE HEAT PUMP DEMONSTRATION AT BALL STATE UNIVERSITY

Piljae Im
Xiaobing Liu
Hugh Henderson

Date Published: October 2016

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37831-6283
managed by
UT-BATTELLE, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

ACKNOWLEDGEMENT

This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE). The specific organization overseeing this report is the Building Technologies Office under contract number DE-AC05-00OR22725.

Special thanks to Dr. James Lowe and his colleagues at the Ball State University for providing data and inputs needed for this study.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

ACRONYMS

ARRA	American Recovery and Reinvestment Act
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BSU	Ball State University
CFB	circulating fluidized bed
CHW	chilled water
COP	Coefficient of Performance
CW	cooling water
DESN	District Energy Station North
DESS	District Energy Station South
DHW	domestic hot water
ECOP	Effective Coefficient of Performance
EMS	energy management system
GSHP	ground-source heat pump
HDD	heating degree days
HDPE	high-density polyethylene
HR	heat recovery
HW	hot water
TD	temperature difference

EXECUTIVE SUMMARY

With funding provided by the American Recovery and Reinvestment Act (ARRA), 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 district central GSHP system installed at Ball State University (BSU) in Muncie, IN.

Prior to implementing the district GSHP system, 47 major buildings in BSU were served by a central steam plant with four coal-fired and three natural-gas-fired steam boilers. Cooling was provided by five water-cooled centrifugal chillers at the District Energy Station South (DESS). The new district GSHP system replaced the existing coal-fired steam boilers and conventional water-cooled chillers. It uses ground-coupled heat recovery (HR) chillers to meet the simultaneous heating and cooling demands of the campus.

The actual performance of the GSHP system was analyzed based on available measured data from August 2015 through July 2016, construction drawings, maintenance records, personal communications, and construction costs. Since Phase 1 was funded in part by the ARRA grant, it is the focus of this case study. The annual energy consumption of the GSHP system was calculated based on the available measured data and other related information. It was compared with the performance of a baseline scenario— a conventional water-cooled chiller and natural-gas-fired boiler system, both of which meet the minimum energy efficiencies allowed by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE 90.1-2013). The comparison was made to determine source energy savings, energy cost savings, and CO_2 emission reductions achieved by the GSHP system. A cost analysis was performed to evaluate the simple payback of the GSHP system. The following sections summarize the results of the analysis, the lessons learned, and recommendations for improvement in the operation of this district GSHP system.

ENERGY PERFORMANCE AND COST-EFFECTIVENESS

- The annual heating and cooling outputs of the GSHP system (from August 2015 through July 2016) were 149,738 MMBtu and 140,032 MMBtu, respectively. An analysis of the heating and cooling outputs reveals that 78% of the heating outputs were recovered during the simultaneous heating and cooling operation of the HR chillers. As a result, the heat rejection and heat extraction at the ground loop were imbalance—the heat rejection (71,684 MMBtu) is 4.5 times larger than the heat extraction (16,120 MMBtu) on an annual basis.

- The analysis shows that the average system Effective Coefficient of Performance (ECOP) during the monitoring period was about 3.74±0.2, while the chiller ECOP is about 4.28±0.2. The fraction of the central system plant monthly pumping power (excluding the local pumps in each building to distribute the hot water [HW] and chilled water [CHW] provided by the district GSHP system) ranges from 8% to 18% of the total system power consumption. Lower pumping power percentages during the heating season are due to the low flow rate in the ground loop resulting from the low heat extraction load.

- An analysis of the historical steam productions at BSU indicates a 40% reduction after the GSHP system was implemented in the campus. By converting from the original steam heating system to HW heating loops in the buildings, heat loss from the distribution pipelines was significantly reduced.

- It is calculated that the baseline system would have consumed 10,970 MWh of electricity and 212,806 MMBtu of natural gas to produce the same heating and cooling outputs as the GSHP system. The total electricity use for the GSHP system during the same period was 22,698 MWh. The source energy savings achieved by the GSHP system is about 96,281 MMBtu, or 27% of the baseline source energy consumption. Carbon emission reduction achieved by the GSHP system is about 8,494,540 lb, or 19% of the baseline carbon emission. The net cost savings (assuming $ 0.08/kWh for electricity and $8/MMBtu for natural gas) would be about $764,200, or 30% of the baseline energy cost.

- Based on the actual installation cost of the GSHP system and the estimated baseline system installation cost, a cost-effectiveness of the GSHP system was evaluated. The total installed costs of the GSHP system and the baseline system are $17,261,241 and $5,234,750, respectively. The simple payback of this GSHP system is about 16 years. However, if the ground heat exchangers were installed at the regional average cost, the payback would have been 9 years.

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

Annual Energy	
GSHP-Source Energy (MMBtu)	266,720
Baseline Source Energy (MMBtu)	363,001
Source Energy Savings (MMBtu)	96,281 (27% savings)
Annual CO_2 Emissions	
GSHP-CO_2 Emissions (lb)	37,224,227
Baseline-CO_2 Emissions (lb)	45,718,767
CO_2 Emission Reductions (lb)	8,494,540 (19% reduction)
Annual Cost	
Cost-GSHP ($)	1,815,816
Cost-Baseline ($)	2,580,016
Cost Savings ($)	764,200 (30% savings)

LESSONS LEARNED AND RECOMMENDATIONS

- The HR chillers could operate very efficiently when both the produced CHW and HW are fully utilized to satisfy the cooling and heating demands of the campus. However, the current configuration and control of the chillers resulted in over-production of hot or chilled water. This is extremely wasteful and has resulted in not only excessive chiller energy use but also unexpectedly high ground loop temperatures. As a result, the operational efficiency of the GSHP system is lower than expected.

- The operational efficiency could have been higher if the two chillers were operated separately—one dedicated to producing CHW and the other operating as a heat pump to produce only HW. In this case, there would be no need to dump excess CHW or HW to the ground. In addition, the ground loop loads would be more balanced since more heat would be extracted from the ground to satisfy the heating demand.

- If the installed chillers cannot be operated separately as suggested above, adding storage tanks in both the CHW and HW loops would help reduce or eliminate the need to dump excess CHW or HW to the ground and thus improve the operational efficiency of the GSHP system.

1. INTRODUCTION

In 2009, 26 projects were competitively selected and funded with American Recovery and Reinvestment Act (ARRA) grants to demonstrate innovative ground-source heat pump (GSHP) technologies. Ball State University (BSU) in Muncie, IN, was one of the 26 demonstration project sites. The demonstrated system is a district GHP system that ultimately replaced the existing coal-fired steam boilers and conventional water-cooled chillers on the campus. It was expected that the district GSHP system would reduce carbon emission by 85,000 tons once it w fully implemented.

This case study evaluates the performance of the demonstrated district GSHP system based on measured performance data, utility bills, and other relevant information. The evaluated performance metrics include the energy efficiency of the overall GSHP system, electricity usage of all major equipment of the GSHP system, the achieved benefits (e.g., energy and cost savings) compared with a new conventional HVAC system, and the cost-effectiveness of the GSHP system. This case study also identifies opportunities for improving the operational efficiency of the system.

1.1 BACKGROUND

Prior to implementing the district GSHP system, 47 major buildings in BSU were served by a central steam plant with four coal-fired and three natural-gas-fired steam boilers. Cooling was provided by five water-cooled centrifugal chillers at the District Energy Station South (DESS). Faced with the need to eliminate the coal-fired boilers and aging chillers, BSU considered several options, including circulating fluidized bed (CFB) coal boilers and improved stack controls. Finally, a geothermal system with HR chillers was selected to meet the simultaneous heating and cooling needs in the campus. The district GSHP system has a cost premium compared to other options, but it has higher energy savings potential, which was one of major reasons for the final selection.

BSU planned to replace the existing coal-fired boilers and aging chillers in the campus in two phases. Phase 1 included building a geothermal field with 1803 vertical boreholes as well as a new District Energy Station on the north side of the campus (referred as DESN). This energy station includes two new 2,500 ton two-stage HR chillers connected to the Phase 1 geothermal fields (**Error! Reference source not found.**). These chillers can simultaneously produce 42°F chilled water (CHW), and hot water (HW) up to 150°F.[1] In Phase 2, the existing conventional chillers in DESS were replaced with two additional 2,500 ton two-stage HR chillers tied with the Phase 2 geothermal fields on the south end of the campus. Two of the original water-cooled chillers with their cooling towers remained in place and are used in conjunction with the new HR chillers to provide CHW to the campus.

Since Phase 1 was funded in part by an American Recovery and Reinvestment Act (ARRA) grant, it is the focus of this case study.

The installation and implementation of GSHP systems on the BSU campus was completed in two phases, and buildings were incrementally added to the new system over several years. Therefore, the control and operation of the GSHP systems has been modified and improved by the facility operators as the number of buildings connected to the system has incrementally increased. As a result, the heating and cooling loads imposed on DESN did not always show a consistent seasonal trend from year to year. This will also be discussed in a later section of this case study.

[1] https://steadystatecollege.files.wordpress.com/2013/03/heat-pump-chiller-brochure.pdf

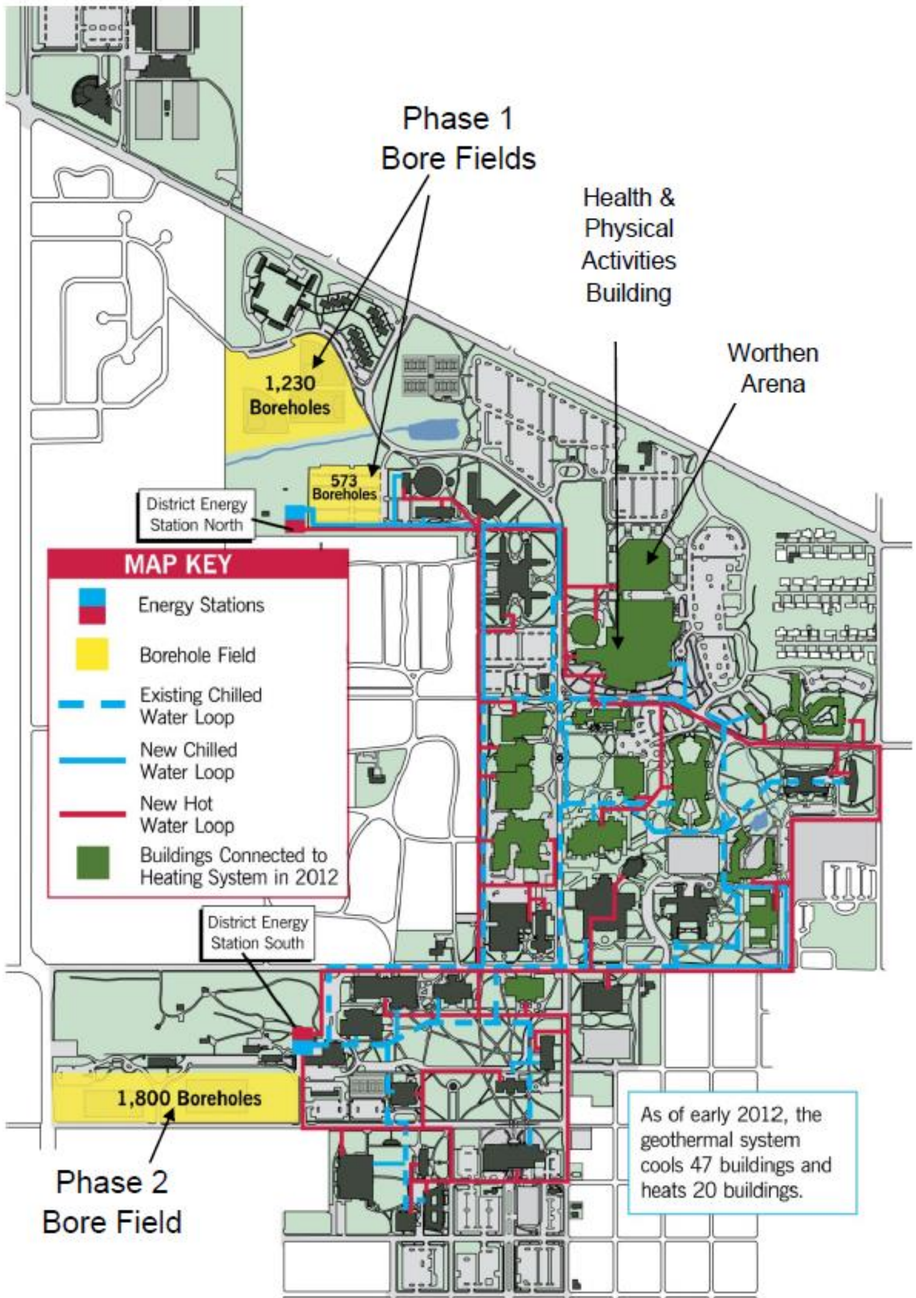


Fig. 1. Ball State University campus map with borehole fields.

1.2 DISTRICT GEOTHERMAL HEAT PUMP SYSTEM (PHASE 1, DESN)

The district GSHP system in Phase 1 includes two geothermal borehole fields, two HR chillers, three pumping stations, and a new campus-wide control system with monitoring capabilities. All the chillers and pumps are installed in a new DESN building. CHW and HW are produced at both the DESN and DESS plants and distributed to campus. As of May 2106, there were 47 buildings on the CHW loop and 30 buildings on the new campus HW loop. Note that only 20 buildings were connected to the new HW loop in 2012. Expansion of the HW piping to other buildings is still ongoing. Some buildings still use steam from the central plant and will not be converted to the HW heating system in near future according to BSU. The steam supplied to these buildings is now produced by the natural gas boilers.

1.2.1 Geothermal Fields

Two geothermal fields were constructed during Phase 1 (DESN), as shown in Fig. 1. One field includes 1,230 bores, and the other has 573 bores. Each bore is 400 ft deep, and the bores are spaced on 15 ft centers. Two 1-in.-diameter high-density polyethylene (HDPE) U-shape loops filled with water are inserted into each bore, which is then grouted with a mixture of sand, bentonite, and water. All the HDPE pipes are routed into DESN through five major circuits that are buried 5 ft under the ground.[2]

A ground formation thermal conductivity test was performed at the site, and the in situ borehole test report shows that the thermal conductivity is about 1.68 Btu/hr-ft-°F, and the undisturbed underground temperature is about 55 to 56°F.

1.2.2 Heat Recovery Chiller

There are two 2,500 ton JCI/York HR chillers (CH #1 and CH #2), which can simultaneously provide 42°F CHW and 110–155°F HW. **Error! Reference source not found.** is a screenshot from the energy management system (EMS) of the North Plant (DESN), which shows the schematic of the HR chillers with the associated control valves and pumps. In the South Plant (DESS), there are also two HR chillers (CH #3, and CH #4) connected to Phase 2 geothermal fields and two conventional centrifugal chillers (CH #5, and CH #6) from the original plant connected to cooling towers. Currently the two plants (DESN and DESS) are operated together to meet the CHW and HW loads. The chiller staging has been to use CH #1 (or CH #2) first and then CH #3, CH #5, and CH #6 in a sequence. Monitored data shows that on no occasion were CH #1 and CH #2 operated together. It appears the CH #1 was usually running as lead chiller, and CH #2 was only sporadically used as the lead chiller.

During typical operation, CH #1 is run to produce both HW and CHW. If the HW load is smaller than the available heating capacity, then the HW return temperature starts to rise above the desired set point. The HW return is then directed to the geothermal field (referred as ground loop hereinafter), and water from ground loop is directed back to the chiller. This is achieved by opening valves CV-LFW-01, CV-LFW-03, and CV-CW-04, which connect the ground loop to the condenser side of the chiller. Then control valves CV-CW-02 and CV-CW-03 modulate in opposite directions to control the HW return temperature back to the (condenser side of the) chiller at the desired temperature (the set point was 106°F on May 18, 2016). As the second stage of control, the circulation pumps for the ground loop modulate their speed to maintain the desired return HW set point.

If the CHW load is too small, then water from the ground loop is added into CHW loop. When the CHW temperature drops, valves CV-LFW-02 and CV-LFW-04 are opened to let ground loop water into CHW loop. These valves are not modulated but instead are cycled ON and OFF to increase the CHW return temperature into the CH #1 evaporator.

When in dual-stage operation (both compressors on), the chiller is controlled to maintain a HW supply temperature. In single-stage mode (i.e., conventional chiller mode), the chiller is controlled to maintain the desired CHW supply set point and condenser heat is rejected to the ground loop.

[2] Another 1,800 boreholes were drilled during Phase 2 for the District Energy Station South.

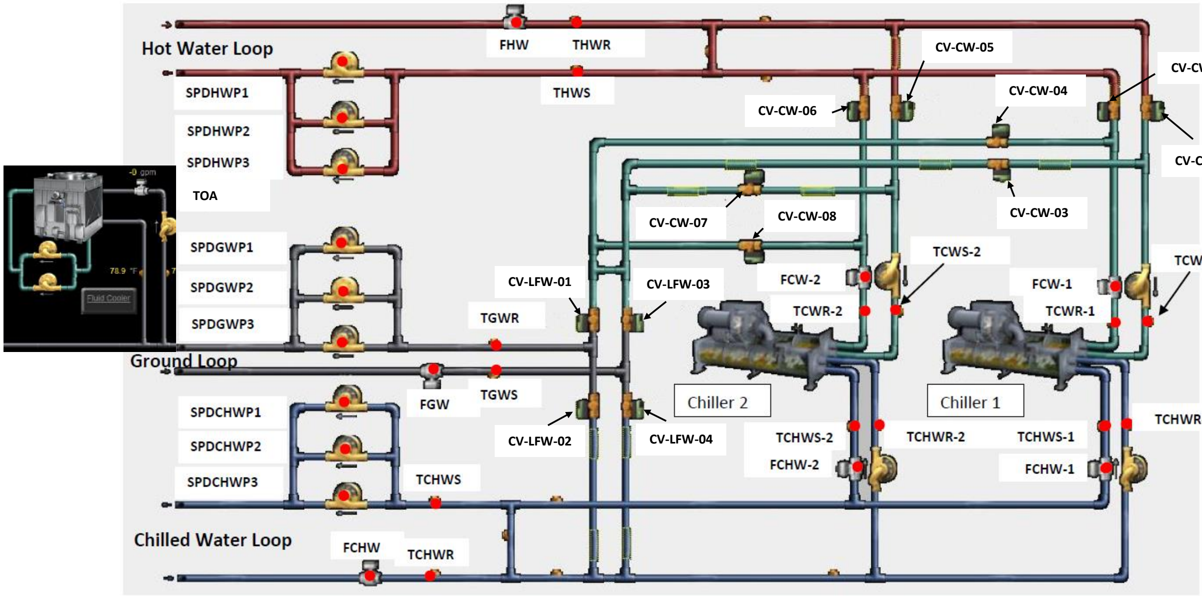


Fig. 2. DESN system schematic shown in EMS system.

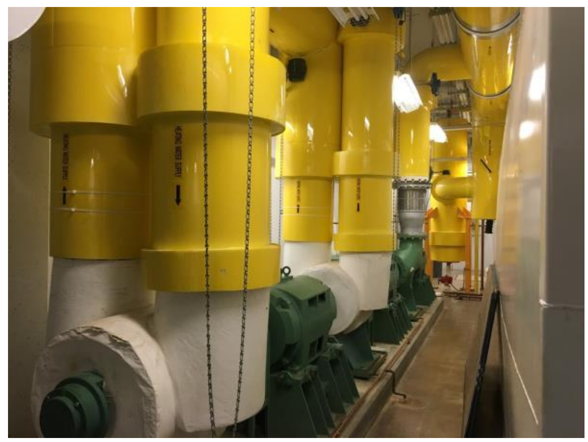
1.2.3 Pumps in DESN

The system in DESN has three sets of pumps for the ground loop, CHW loop, and HW loop, respectively. Each hydronic system has its own set of two or three pumps that operate as required. In addition, the condenser and evaporator sides of each chiller have their own dedicated (primary) pumps, so there are three ground loop pumps (250 HP each), two primary CHW pumps (125 HP each), two cooling water (CW) pumps (125 HP each), three secondary CHW pumps (350 HP each), and three secondary HW pumps (300 HP each) (**Error! Reference source not found.**). **Error! Reference source not found.** lists more information on each pump. The CHW secondary pumps that send water throughout the campus are coordinated between the North and South chiller plants. Pumps in DESN are controlled based on the distribution system pressure difference. Each building has its own internal loop circulation (tertiary) pumps with a hold back valve (commonly referred to as a "bridge") to maintain a significant temperature difference across the building and the loop. Therefore, the loop pressure does not change significantly with load on either the campus HW or CHW distribution loops. As a result, the speed of the secondary CHW and HW pumps in each plant does not change significantly.

2500 ton HR chiller

Three ground loop circulation pumps

Three district chilled water pumps



Three district hot water pumps

Fig. 3. Chiller and pumps in DESN.

Table 1. Pumps in DESN

Dwg. Labels	Description	Design Flow (GPM)	Head (FT)	Motor (HP)
LFWP-1	Geothermal Loop Field Pump -1	6,000	100	250
LFWP-2	Geothermal Loop Field Pump -2	6,000	100	250
LFWP-3	Geothermal Loop Field Pump -3	6,000	100	250
PCHWP-1	Primary Chilled Water Pump - 1 (CH#1)	6,000	60	125
PCHWP-2	Primary Chilled Water Pump - 2 (CH#2)	6,000	60	125
SCHWP-1	Secondary Chilled Water Pump -1	5,800	150	350
SCHWP-2	Secondary Chilled Water Pump -2	5,800	150	350
SCHWP-3	Secondary Chilled Water Pump -3	5,800	150	350
CWP – 1	Condenser Water Pump - 1 (CH#1)	6,400	55	125
CWP – 2	Condenser Water Pump - 2 (CH#2)	6,400	55	125
HWP-1	Hot water Pump - 1	3,600	205	300
HWP-2	Hot water Pump - 2	3,600	205	300
HWP-3	Hot water Pump - 3	3,600	205	300

1.2.4 Supplemental Heat Rejection

A fluid cooler (a wet cooling tower) was added to the ground loop in spring 2015 in an effort to reduce the excessively high ground loop temperatures. It has a dedicated pump that pulls a side stream of water from the supply water going into the loop field. The fluid cooler was added because of the excessively high ground loop temperatures that were experienced in the first few years of operation. The reasons for the high ground loop temperatures are discussed in detail in the following section.

Because the data from the fluid cooler was not available throughout the monitoring period, the energy impact of the fluid cooler was not included in the data analysis.

2. MONITORING AND DATA ANALYSIS

2.1 DATA COLLECTION

The BSU provides performance monitoring and data collection for the GSHP systems using the on-site building automation system. Fig. 2 shows the measured data points on the DESN's system schematic. The system polls the sensors every few seconds and provides 30 minute averages of each sensor. High-quality temperature and flow rate measurements are taken using a BTU meter (Onicon System 10 with F-series turbine meters). The data are recorded in a column-oriented, comma-delimited file (CSV format) along with a corresponding date/time stamp and are sent to a server on a daily basis. Table 2 gives a brief description of each data point. Note that the power consumption of the pumps was not measured. Only the speed (percentage of the full speed) of each pump was measured. The power consumption of each pump is estimated using correlations between the speed and the actual power consumption measured with a handheld true power meter during an on-site visit. The details of this estimation will be explained later.

Table 2. GSHP system monitoring points

No.	Data Point	Description	Units	Accuracy
1	TGWS	Ground Loop Supply Temperature	°F	~ ±0.15°F
2	TGWR	Ground Loop Return Temperature	°F	~ ±0.15°F
3	FGW	Ground Loop Flow Rate	gpm	~ ±1%
4	SPDGWP1	LFWP-1 VFD % Speed to its full speed	%	NA
5	SPDGWP2	LFWP-2 VFD % Speed to its full speed	%	NA
6	SPDGWP3	LFWP-3 VFD % Speed to its full speed	%	NA
7	TCWS-2	Condenser Water Supply to Chiller 2	°F	~ ±0.15°F
8	TCWS-1	Condenser Water Supply to Chiller 1	°F	~ ±0.15°F
9	TCWR-2	Condenser Water Return from Chiller 2	°F	~ ±0.15°F
10	TCWR-1	Condenser Water Return from Chiller 1	°F	~ ±0.15°F
11	FCW-2	Condenser Water Flow to Chiller 2	gpm	~ ±1%
12	FCW-1	Condenser Water Flow to Chiller 1	gpm	~ ±1%
13	TCHWS-2	Chiller 2 Chilled Water Supply Temperature	°F	~ ±0.15°F
14	TCHWS-1	Chiller 1 Chilled Water Supply Temperature	°F	~ ±0.15°F
15	TCHWR-2	Chiller 2 Chilled Water Return Temperature	°F	~ ±0.15°F
16	TCHWR-1	Chiller 1 Chilled Water Return Temperature	°F	~ ±0.15°F
17	FCHW-2	Chiller 2 Chilled Water Flow	gpm	~ ±1%
18	FCHW-1	Chiller 1 Chilled Water Flow	gpm	~ ±1%
19	TCHWR	District Chilled Water Return Temperature	°F	~ ±0.15°F
20	TCHWS	District Chilled Water Supply Temperature	°F	~ ±0.15°F
21	FCHW	District Chilled Water Flow Rate	gpm	~ ±1%
22	SPDCHWP1	CHWP-1 VFD % Speed to its full speed	%	NA

23	SPDCHWP2	CHWP-1 VFD % Speed to its full speed	%	NA
24	SPDCHWP3	CHWP-1 VFD % Speed to its full speed	%	NA
25	THWR	District Hot Water Return Temperature	°F	~ ±0.15°F
26	THWS	District Hot Water Supply Temperature	°F	~ ±0.15°F
27	FHW	District Hot Water Flow Rate	gpm	~ ±1%
28	SPDHWP1	HWP-1 VFD % Speed to its full speed	%	NA
29	SPDHWP2	HWP-2 VFD % Speed to its full speed	%	NA
30	SPDHWP3	HWP-3 VFD % Speed to its full speed	%	NA
31	Ambient Temperature		°F	

In some cases, data were missing. In total, 17 days of data were missing during the data collection period, and all these missing data were filled with the average daily value of the particular measurement during the month, in which the data were missing.

In addition, it appears the entire system in DESN was not running from October 19, 2015, through November 2, 2015, since the system DESS provided all district HW and CHW to the campus. No attempt was made to fill or replace the data during this period. The baseline system and GSHP system had the same scenario (no system running during this time period), so that the comparison analysis is valid.

2.2 DATA ANALYSIS PLAN

For the data analysis, the following quantities were evaluated with the measured data as explained below.

- **Ground Loop Heat Exchanger**

$$Q_{GL} = K \times FGW \times (TGWR\text{-}TGWS)/1{,}000, \quad (1)$$

where

Q_{GL} = Ground loop heat transfer rate (kBtu/h)
(negative values for heat rejection, positive values for heat extraction),
K = 495 for water at 110°F (Btu/h-GPM-°F),
FGW = Ground loop flow rate (GPM),
TRWS = Ground loop supply temperature (°F), and
TRWR = Ground loop return temperature (°F).

- **District Chilled Water Loop (to Campus)**

$$Q_{CHW} = K \times FCHW \times (TCHWR\text{-}TCHWS)/1{,}000, \quad (2)$$

where

Q_{CHW} = Chilled water load (kBtu/h) (cooling >0),
K = 500 for chilled water (Btu/h-GPM-°F),
FCHW = Chilled water flow rate (GPM),
TCHWS = Chilled water loop supply temperature (°F), and
TCHWR = Chilled water loop return temperature (°F).

- **District Hot Water Loop (to Campus)**

$$Q_{HW} = K \times FHW \times (THWS\text{-}THWR)/1000, \quad (3)$$

where

Q_{HW}	=	Hot water load (kBtu/h) (heating >0),
FHW	=	Hot water loop flow rate (GPM),
THWS	=	Hot water loop supply temperature (°F),
THWR	=	Hot water loop return temperature (°F), and
K	=	494 for water at 120°F (Btu/h-GPM-°F).

- **Chiller 1 Evaporator**

$$Q_{CHW1} = K \times FCHW1 \times (TCHWR1\text{-}TCHWS1)/1{,}000, \quad (4)$$

where

Q_{CHW1}	=	Chiller 1 evaporator load (kBtu/h),
FCHW1	=	Chilled water flow from Chiller 1 (gpm),
TCHWR1	=	Chilled water supply temperature from Chiller 1 (°F),
TCHWS1	=	Chilled water return temperature to Chiller 1 (°F), and
K	=	500 for water (Btu/h-GPM-°F).

- **Chiller 1 Condenser**

$$Q_{CW1} = K \times FCW1 \times (TCWR1\text{-}TCWS1)/1{,}000, \quad (5)$$

where

Q_{CW1}	=	Chiller 1 condenser load (kBtu/h),
FCW1	=	Hot water flow from Chiller 1 (gpm),
TCWR1	=	Hot water supply temperature from Chiller 1 (°F),
TCWS1	=	Hot water return temperature to Chiller 1 (°F), and
K	=	500 for water (Btu/h-GPM-°F).

- **Chiller 2 Evaporator**

$$Q_{CHW2} = K \times FCHW2 \times (TCHWR2\text{-}TCHWS2)/1{,}000, \quad (6)$$

where

Q_{CHW2}	=	Chiller 2 evaporator load (kBtu/h),
FCHW2	=	Chilled water flow from Chiller 2 (gpm),
TCHWR2	=	Chilled water supply temperature from Chiller 2 (°F),
TCHWS2	=	Chilled water return temperature to Chiller 2 (°F), and
K	=	500 for water (Btu/h-GPM-°F).

- **Chiller 2 Condenser**

$$Q_{CW2} = K \times FCW2 \times (TCWR2\text{-}TCWS2)/1{,}000, \quad (7)$$

where

Q_{CW2}	=	Chiller 2 condenser load (kBtu/h),
FCW2	=	Hot water flow from Chiller 2 (gpm),
TCWR2	=	Hot water supply temperature from Chiller 2 (°F),
TCWS2	=	Hot water return temperature to Chiller 2 (°F), and
K	=	500 for water (Btu/h-GPM-°F).

Because the HR chillers provide simultaneous heating and cooling, the Effective Coefficient of Performance (ECOP) was calculated for the chillers and the entire GSHP system, respectively, based on the cumulative heating and cooling outputs and the associated power consumptions.

- **ECOP of Chiller and District GSHP System**

$$\text{Chiller ECOP} = (Q_{CHW} + Q_{HW})/((WCH1+WCH2) * 3.413)), \quad (8)$$

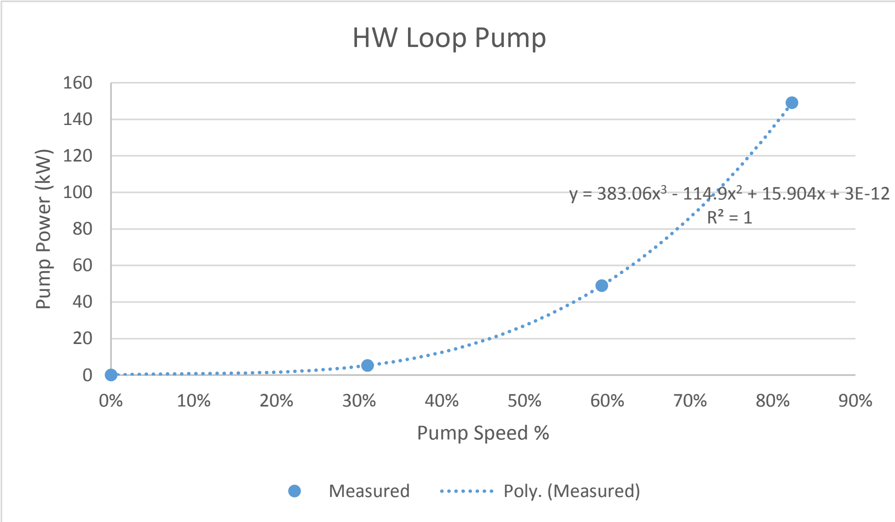
$$\text{System ECOP} = (Q_{CHW} + Q_{HW})/((WCH1+WCH2+GLP+ DHWP + DCHWP + CCP + CEP) \times 3.413)), \quad (9)$$

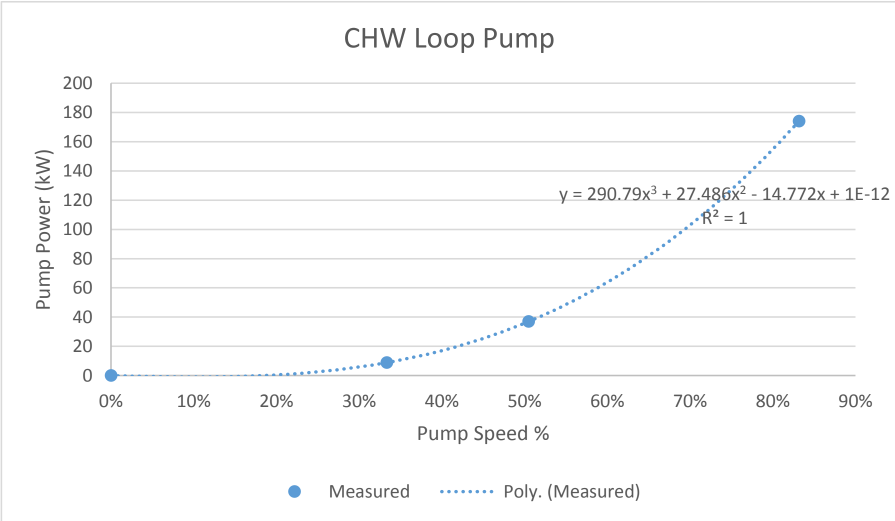
where

GLP	=	Ground loop pump power (kW),
DHWP	=	District hot water pump power (kW),
DCHWP	=	District chilled water pump power (kW),
CCP	=	Chiller condenser pump power (kW), and
CEP	=	Chiller evaporator pump power (kW).

- **Pumping Power**

Pump power consumptions were estimated based on the measured pump speed and a correlation derived to relate pump power to speed for each pump. The correlation was developed by taking two or three measurements of actual pump power consumption (with ±1% accuracy) with handheld true-power meter at different speeds during a site visit. One pump was chosen from each of the CHW, HW, and ground loops, respectively. The symbols on Fig. 4 shows the measured pump power and speed for each of the three pumps. The line on each plot represents the curve-fit to the data. The resulting regression models were used to estimate the pump power based on the measured pump speed at each time interval.





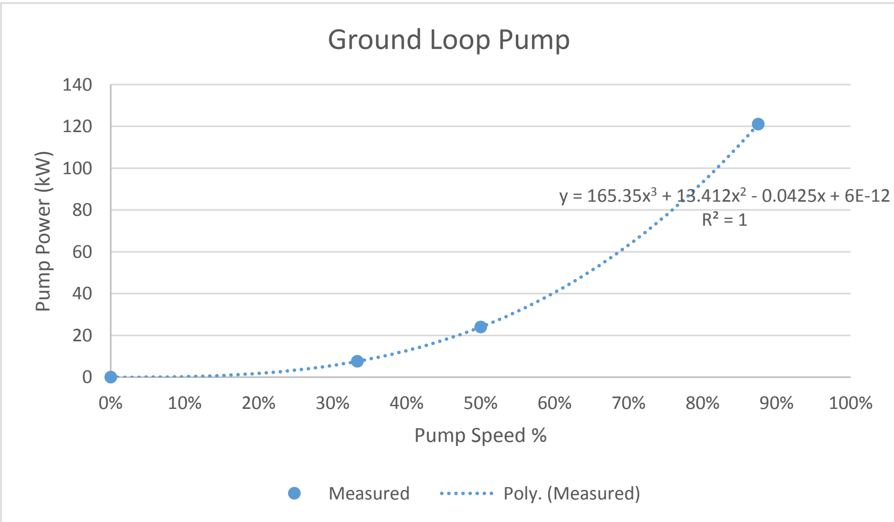


Fig. 4. Measured pump power draws at various pump speeds.

3. DATA ANALYSIS RESULTS

3.1 GROUND LOOP PERFORMANCE ANALYSIS

Monthly average ground loop temperature at DESN and outdoor air temperature from August 2015 through July 2016 are plotted in Fig. 5. As shown in this figure, the absolute values of the difference between the average monthly supply temperature and the return temperature during summer and winter ranged within 5 to 7ºF and 2 to 4ºF, respectively. It is observed that the overall ground temperature during this period is lower than those from August 2014 through July 2015 (Fig. 6). For example, the monthly average ground loop supply temperature for 2014-2015 was 89 to 101°F, while the range for 2015-2016 was 82 to 95°F. The lower ground loop supply temperature could be due to the installation of the additional fluid cooler to the ground loop in April 2015[3], or more campus heating in 2016 as the HW distribution piping was expanded to serve more buildings, or just a general shift of loads from DESN to DESS.

In both periods shown in the figures, the monthly average ground loop supply temperature is significantly higher than the outdoor air temperature all year-round, which is good for extracting heat in winter but results in low cooling efficiency due to the elevated condensing temperature of the chiller in summer. As discussed below, the high ground loop temperature is due to several reasons, including the operation of the chiller in the HR mode with both compressors on and the lower-than-expected heating loads.

The hourly ground loop supply and return temperatures for the same period are plotted in Fig. 7. The hourly data shows that the ground loop supply temperatures were close to the outdoor air temperature in summer and that they were often lower than the outdoor air temperature when it peaked during a day. Throughout the year, the average hourly temperature differences (TDs) between the supply and return water of the ground loop for heat rejection and heat extraction were 6.0°F and 7.4°F, respectively. For further investigation, a histogram chart of TD distribution was plotted in Fig. 8. In this plot, the x axis represents the TD (i.e., ground loop supply temperature – ground loop return temperature) and the y axis represents the frequency (hours). In case of heat rejection (i.e., negative TD), TD is between 0 to -10°F most of the time, while for heat extraction, the TD range is wider. The data also show that the frequency of a TD between -5°F to 5°F is about 72% of the total hours. Although the average TD for heat extraction was bit higher than that for heat rejection, the loop flow rate during heat extraction in winter months was much lower than that during heat rejection in summer months (Fig. 9). The flow rate difference results in a lower heat extraction rate from the ground loop compared to the heat rejection rate to the loop.

As introduced in Section 1.2.2., the ground loop flow rate is controlled in different ways depending on whether the cooling or heating output of the chiller exceeds the demand. When heating demand is less than the heating output of the chiller, some of the return HW is directed to the ground loop to dump the excess heat. The ground loop flow is modulated to keep the HW return temperature (going to the condenser side of the chiller) at the desired set point. On the other hand, if the cooling demand is less than the cooling output of the chiller, the valves in the ground loop are cycled on and off to direct (some of) the CHW to go through the ground loop to increase the CHW return temperature to the set point. The operators report that cycling these ON/OFF valves causes fluctuations of 5 to10°F in CHW loop. The site is considering changing these valves to modulating valves for better control.

[3] Performance data of the fluid cooler were not measured, and the impact of the fluid cooer cannot be quantified.

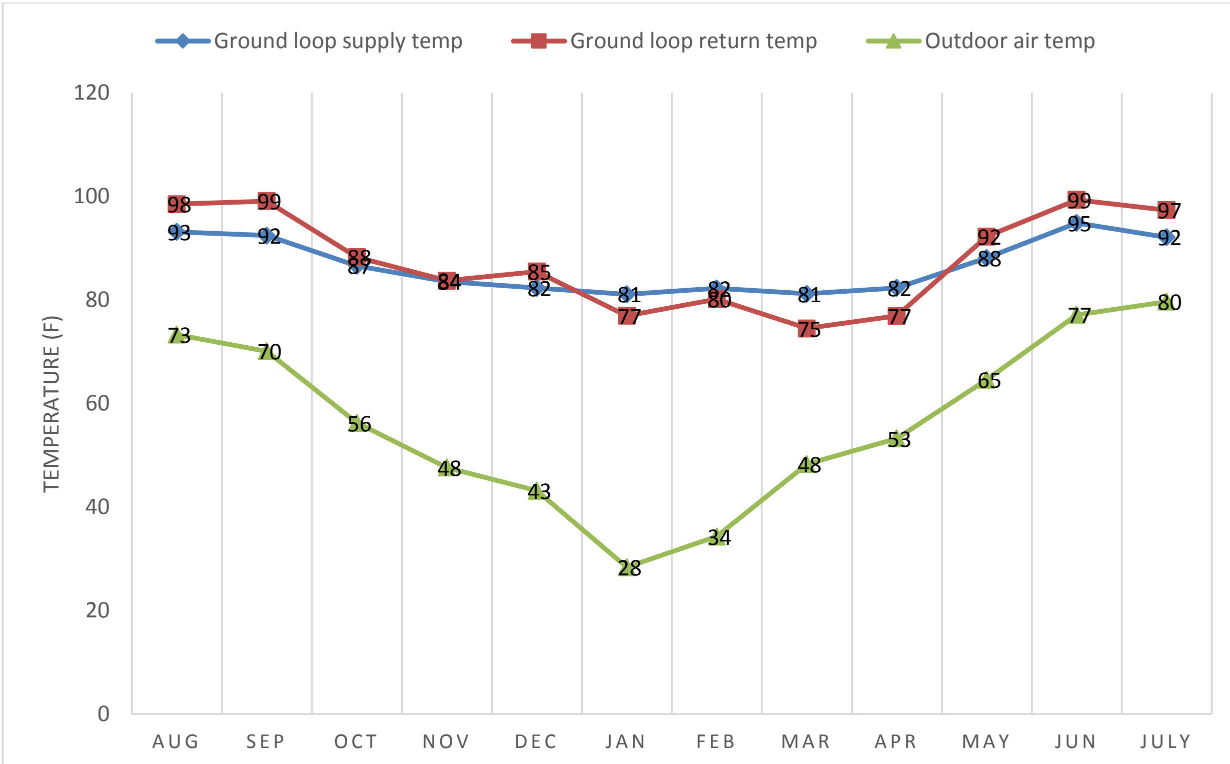


Fig. 5. Monthly ground loop supply and return water temperature at DESN and the outdoor air temperature (August 2015 through July 2016).

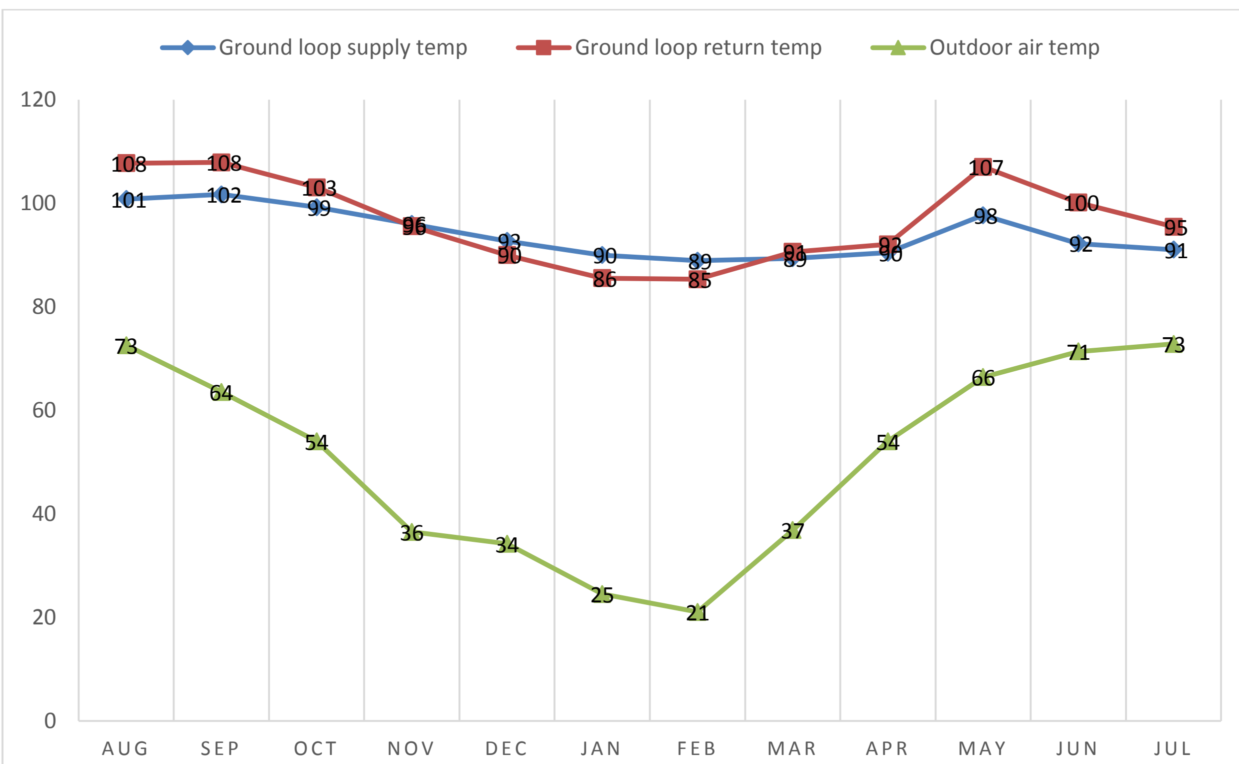


Fig. 6. Monthly ground loop supply and return water temperature at DESN and the outdoor air temperature (previous year: August 2014 through July 2015).

Both of these control practices are extremely wasteful, resulting in increased chiller energy use and unexpectedly high ground loop temperatures. When HW production exceeds the heating demand, the unneeded HW (higher than 100°F) is rejected to the ground loop. Similarly, over production of CHW results in cooling being transferred to the ground loop. Since the chiller compressors use electricity to produce the unneeded HW and CHW, the chiller energy use is much higher than what is necessary.

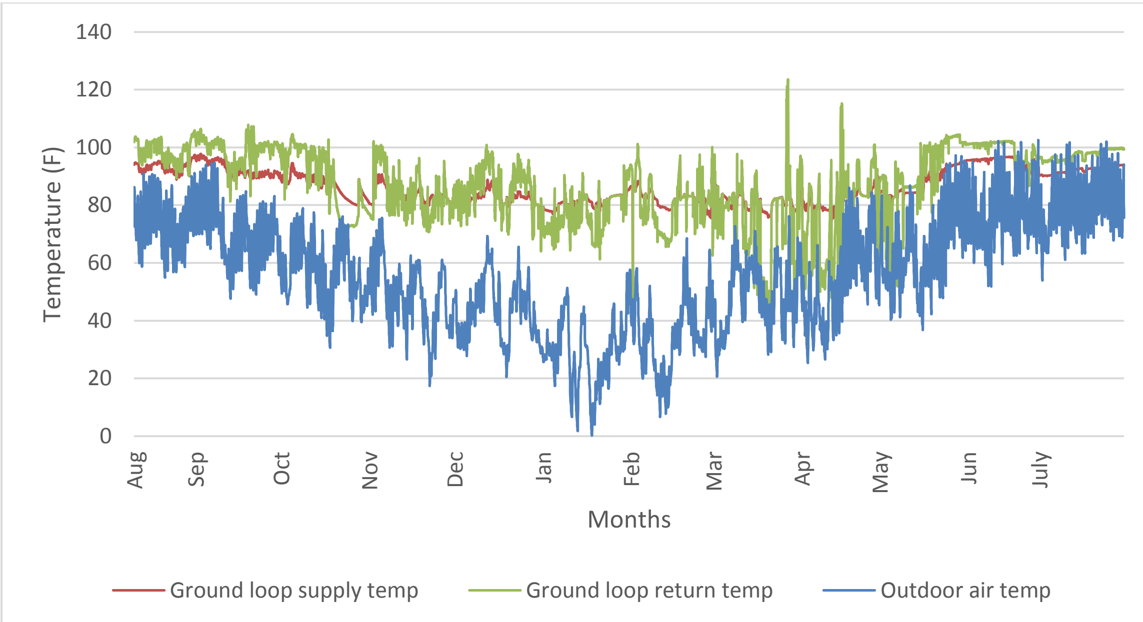


Fig. 7. Hourly ground loop supply and return water temperature and the outdoor air temperature.

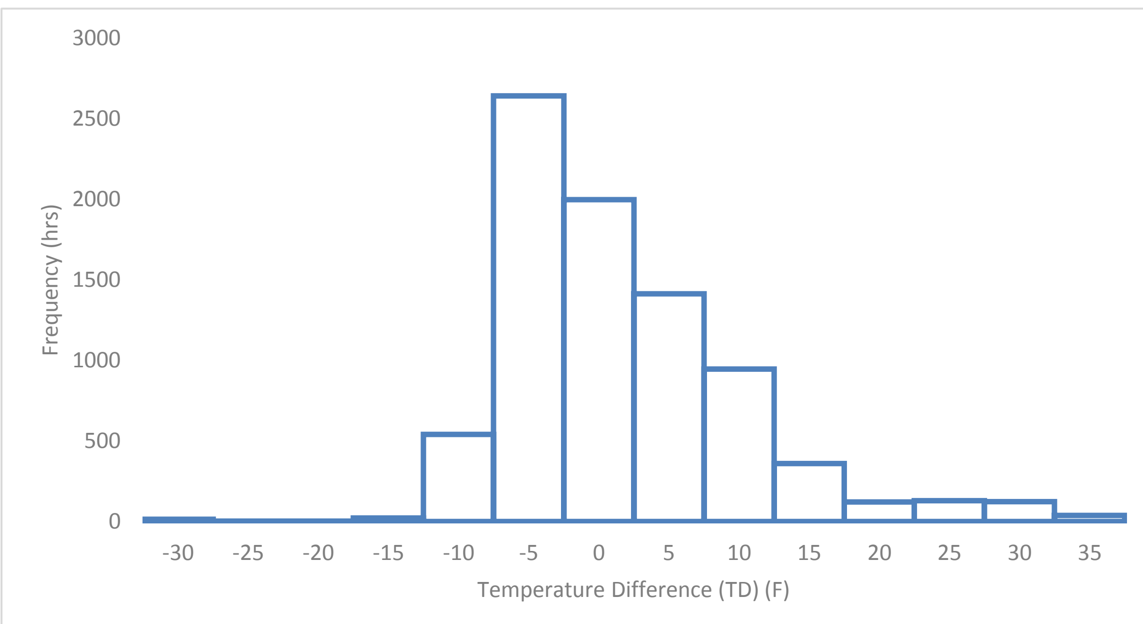


Fig. 8. Histogram of temperature difference in the ground loop.

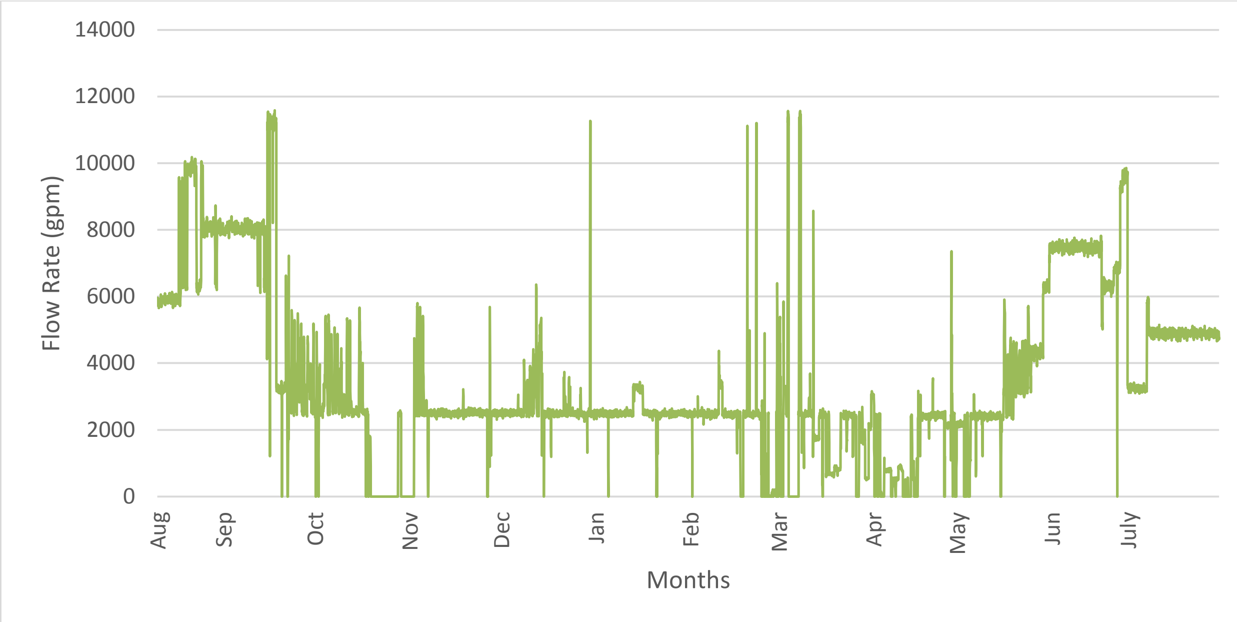


Fig. 9. Hourly ground loop flow rate.

Fig. 10 presents the hourly heat rejection/extraction to/from the ground loop as outdoor air temperature changes. It appears mixed heat rejection/extraction occurred when the outdoor air temperature was between 30 and 60°F. As shown in Fig. 10, heat rejection to the ground loop is much greater than heat extraction. During the monitoring period, total heat rejection and extraction were 71,684 MMBtu and 16,120 MMBtu, respectively, or the heat rejection was 4.5 times greater than the heat extraction. During the same time period of the previous year, heat rejection to the ground loop was 5.7 times greater than heat extraction from the ground.

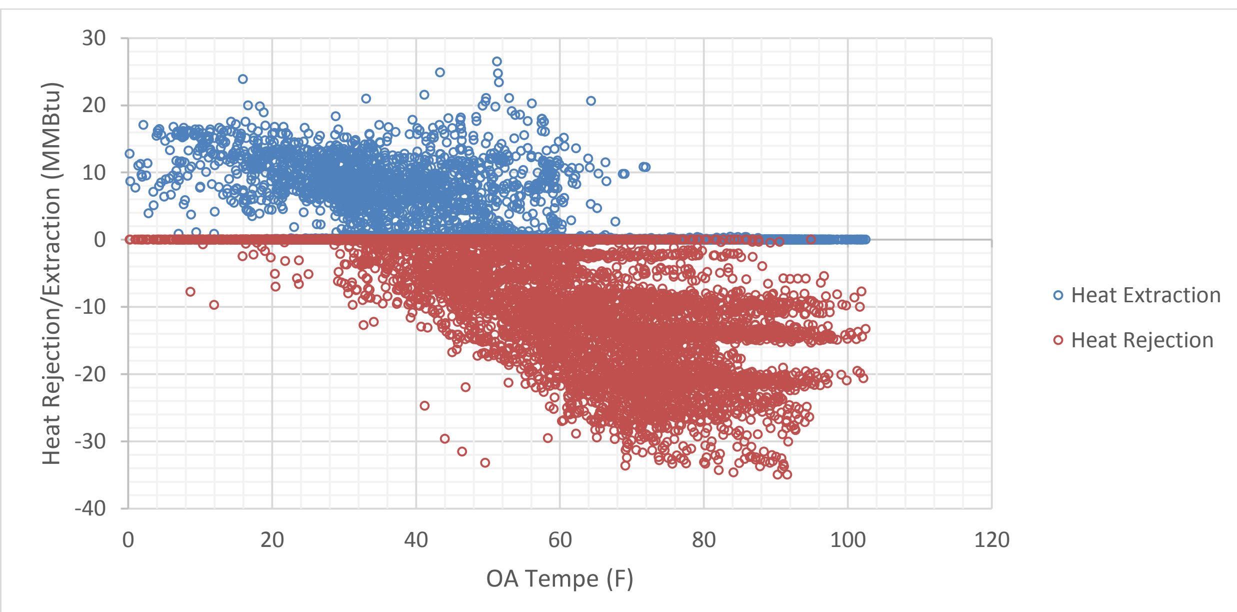


Fig. 10. Hourly heat rejection/extraction to/from ground loop for DESN.

3.2 HEATING AND COOLING OUTPUTS ANALYSIS

Fig. 11 shows the average daily output from DESN to the district CHW loop and HW loop in each 5°F outdoor air temperature bin. There were cooling and heating demands on the campus throughout the year. The year-round cooling demands were due to high internal heat gains, which can be typically seen in the variety of buildings on a university campus. The year-round heating demands are due to domestic hot water (DHW) preheating and reheating needs in the variable air volume (VAV) space conditioning systems used in the campus buildings. The total heating and cooling provided by the HR chillers in DESN to the district HW and CHW loops were 149,738 MMBtu and 140,032 MMBtu, respectively, which were fairly balanced. During the same period in previous years (August 2014 through June 2015), the total heating and cooling outputs were 79,124 MMBtu and 108,065 MMBtu, respectively. Hence, there has been a significant increase (i.e., about 89%) in heating demand.

3.2.1 Heat Recovery

While the heating and cooling demands were balanced in 2015–2016, this was not the case for heat extracted from and rejected to the ground loop. The annual heat rejection to and extraction from the ground loop were 71,684 MMBtu and 16,120 MMBtu, respectively. The heat rejection to the ground was only about 52% of the total cooling outputs, and the heat extracted from the ground was only about 11% of the total heating output. Based on the heat balance of the chiller, the heat recovered from the condenser of the chiller can be calculated by following equation.

$$Q\_HR = Q\_clg + Q\_clg / COP\_clg - Q\_rej\_GL, \tag{10}$$

where

Q_HR (kBtu) = Heat recovered from the condenser of the chiller when it produces CHW,
Q_clg (kBtu) = cooling outputs of the chiller,
COP_clg = Average cooling COP of the HR chiller (2.92)[4], and
Q_rej_GL = Heat rejected to the ground loop.

The annual Q_HR was calculated as 116,304 MMBtu, which means about 78% of the heating output was recovered from the simultaneous heating and cooling operation of the HR chillers instead of being extracted from the ground loop. Although this is an expected benefit of the HR chiller, it may not be the optimal for the GSHP system. Because a large portion of the heating output is from the recovered heat at the condenser, the GSHP system does not extract much heat from the ground. As a result, the heat rejected to and extracted from the ground is significantly unbalanced even with relatively balanced heating and cooling demands from the buildings. A significantly unbalanced load from the ground is one of the reasons for the elevated ground loop temperature, which resulted in a lower-than-expected cooling efficiency.

[4] According to chiller manufacturer’s catalog data, when the HR chiller produces both 40°F CHW (from evaporator) and 125°F HW (from condenser), they use 1.2 kW of electricity per each ton of cooling supplied.

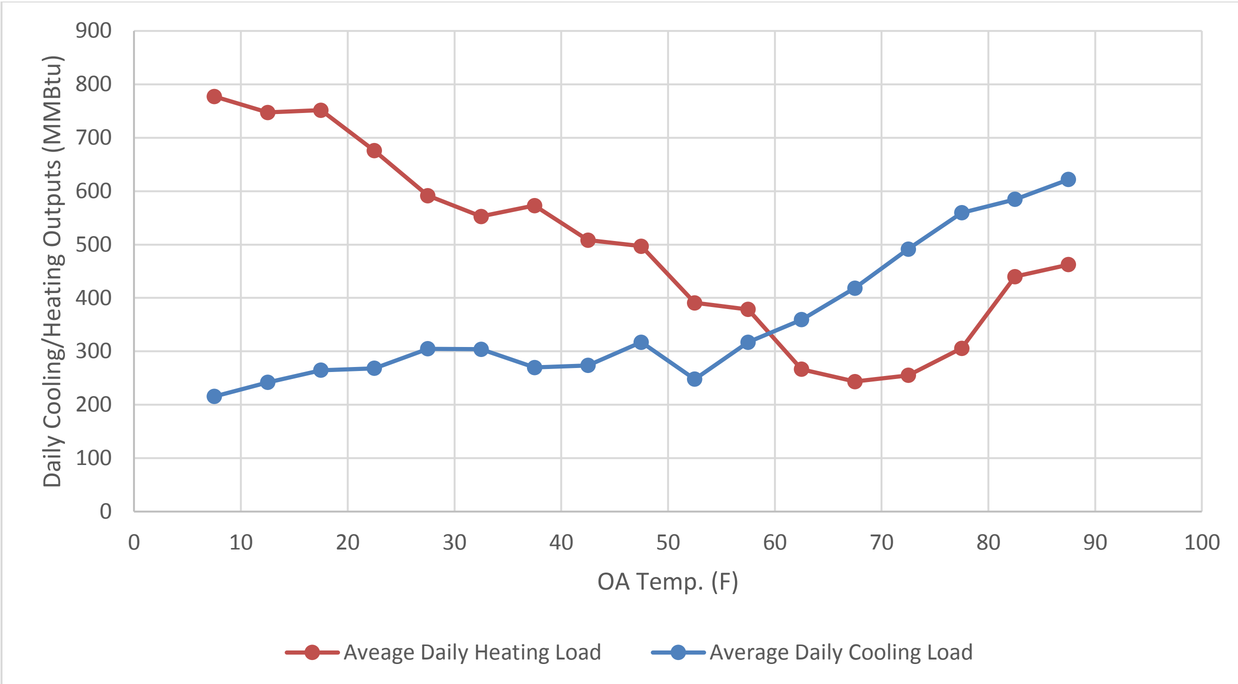


Fig. 11. Daily heating and cooling outputs.

Fig. 12 shows the monthly heating and cooling outputs of the GSHP system. As shown in this figure, the heating outputs in spring and summer of 2016 increased by a large margin compared to last year (i.e., note the differences in heating output between August 2015 and July 2016). Further investigation reveals that DESS provided most of district HW in the summer of 2015, while DESN provided most of district HW in the summer of 2016. The relatively low heating and cooling outputs in October were due to a shutoff of the GSHP system in DESN from October 19, 2015, through the rest of the month, when the district HW and CHW was provided by DESS.

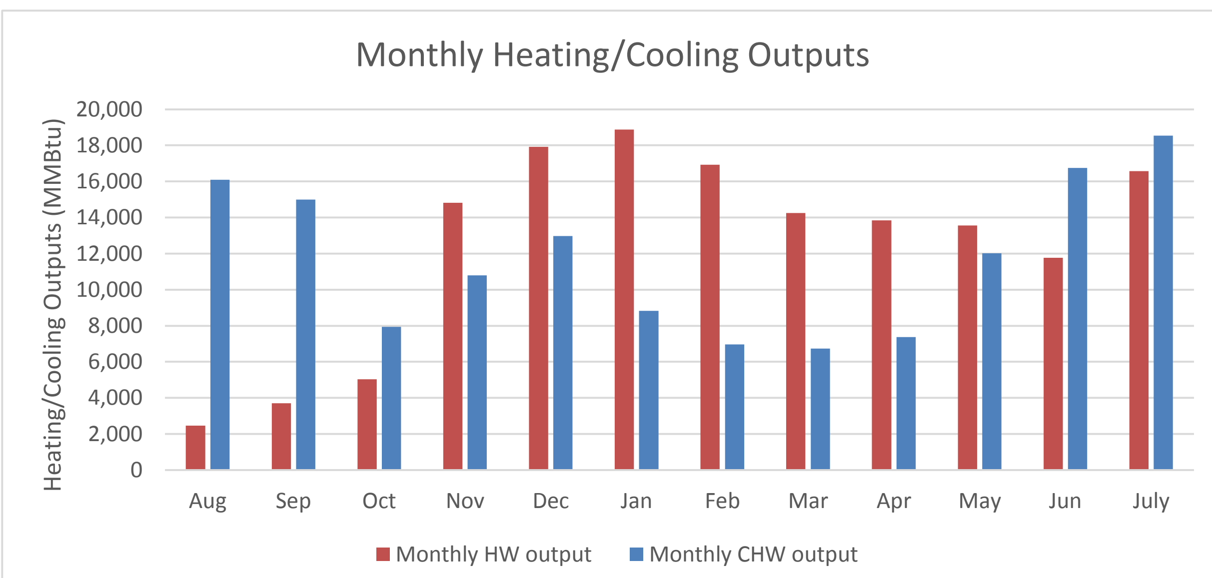


Fig. 12. Monthly heating and cooling outputs (August 2015 through July 2016).

3.2.2 Campus-Wide Steam Consumptions Before and After the GSHP Retrofit

The annual demand for steam before the GSHP retrofit was about 700 million lb/yr (with an average hourly demand of 80,000 lb/h). The geothermal HR chillers have replaced the coal-fired steam boilers at BSU. However, the existing HVAC systems of many buildings on the campus have not been retrofitted to use the HW produced by the district GSHP system. Currently, a few natural gas-fired boilers are still running to meet the remaining steam demand on the campus and a nearby hospital.

Fig. 13 shows monthly steam production data for the campus from January 2009 through July 2016. This figure shows a reduction in steam production in January 2012 when the Phase 1 GSHP system was implemented. It appears further reduction occurred when the new energy station in the south part of the campus (Phase 2) was added in January 2015, but that reduction is less obvious.

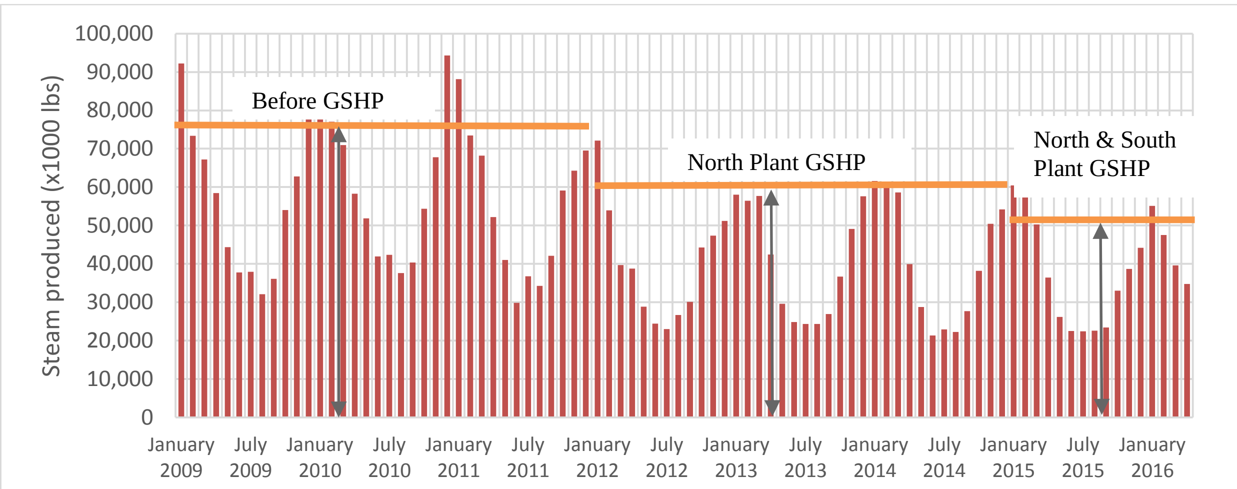


Fig. 13. Historical monthly steam production on the campus.

Fig. 14 shows the trends of monthly steam production before and after the GSHP retrofit versus the heating degree days (HDD) in the month. As can be seen in Fig. 14, the monthly steam production demonstrates a strong linear trend with HDD. The trend indicates a 34% reduction in annual steam production since the GSHP system in DESN started operating and an additional 6% reduction when the GSHP system in DESS came online. Assuming that steam energy content is (at 150 psig) is about 1,194 Btu/lb, the 34% reduction of 700 million lb/year of steam is equivalent to about 284,172 MMBtu/year. However, the measured district HW heating output for August 2015 through July 2016 is only about 198,737 MMBtu (total district HW from DESN and DESS), which is only about 70% of the reduction in steam production. The difference is thought to be due to the smaller heat loss associated with HW distribution. Typical steam distribution loss in a district steam heating system is 5 to 30% (DOE 2015).

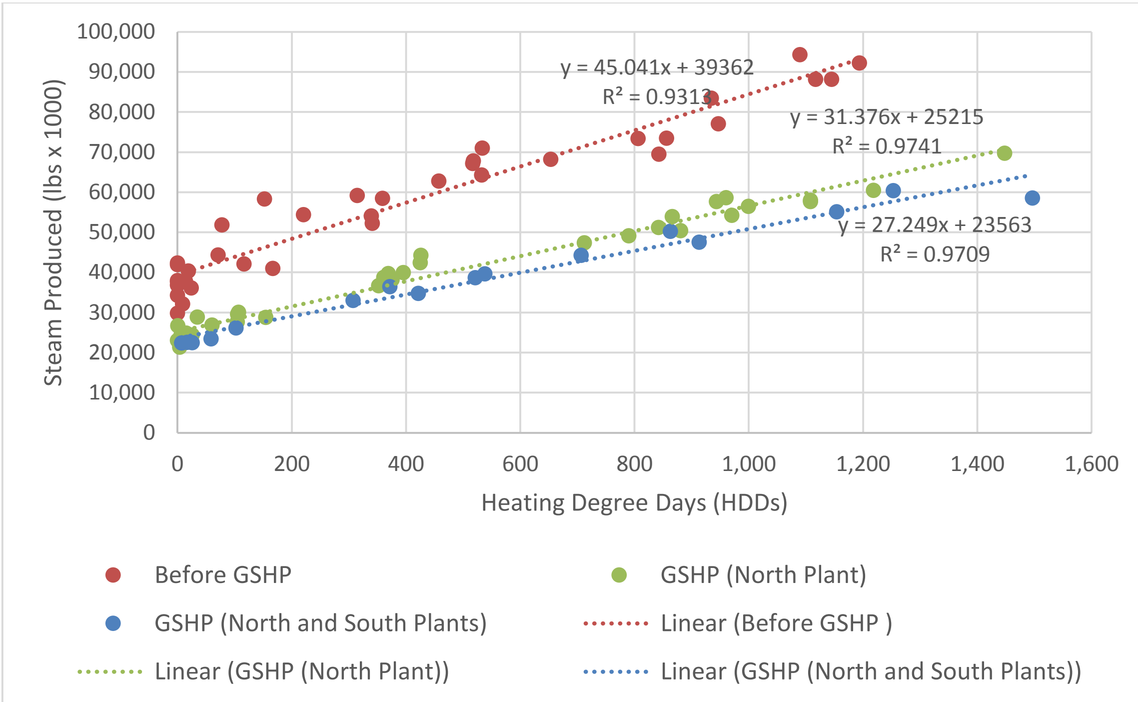


Fig. 14. Monthly steam production versus the associated heating degree days.

About 30% of the current steam production is for the hospital. BSU estimated that 60% of the heating demand in the hospital could be converted to HW, which could be helpful in balancing the currently unbalanced heat rejection/extraction load of the ground loop. The remaining 40% heating demand will probably always have to be served by 150 psig steam. Therefore, BSU has no plan to discontinue operation of its steam system.

3.3 GSHP SYSTEM POWER CONSUMPTION ANALYSIS

In order to evaluate the GSHP system performance, the power consumptions of the GSHP system were analyzed and discussed as follows.

3.3.1 Chiller Power Consumption

The chiller power consumption has been measured continuously since August 2015. To verify the quality of the calculated chiller power before that point in time, the measured monthly chiller power consumption was compared with that calculated using a heat balance equation (Eqs. 11 and 12). Fig. 15 shows that the calculated and the measured monthly chiller power consumptions match closely. It indicates that the calculated values are consistent with the measured values and are therefore reliable.

$$WCH1 = (Q_{CW1} - Q_{CHW1})/3.413, \tag{11}$$

$$WCH2 = (Q_{CW2} - Q_{CHW2})/3.413, \tag{12}$$

where WCH1 and WCH2 are the power draws (in kW) of Chillers #1 and #2, respectively.

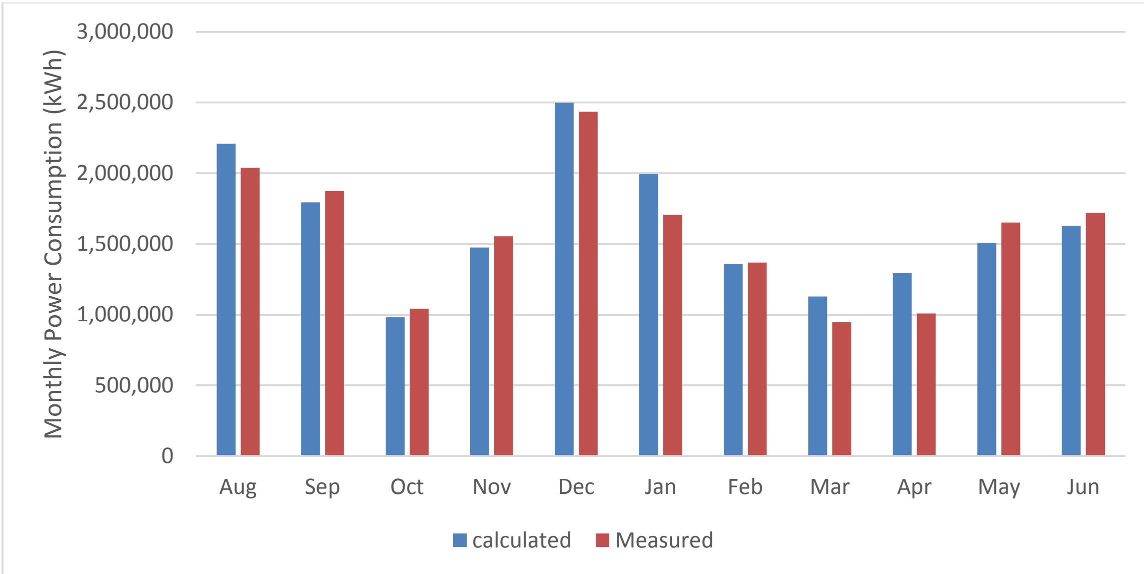


Fig. 15. Calculated and measured monthly chiller power consumption.

3.3.2 Pump Power Consumptions

Fig. 16 shows the monthly pump power consumption in each water loop. Analysis shows that the pumping power of the ground loop is responsible for only 1.6% of the total power consumption of the entire district GSHP system. Accounting for all the pumps, the pumping energy was 12.6% to the total power consumption of the entire district GSHP system at DESN. The local pumping energy used in each building served by the district GSHP system is not accounted for in this analysis.

It shows that the group loop pumping power from October through May was lower than that in other months, which is due to the heat recovery from the simultaneous heating and cooling of the chiller, as discussed before.

Note that the monthly district HW pumping power is nearly constant year-round, although the monthly HW output varied. Because the TD of the district HW varied by season (i.e., much higher TD for heating season than cooling season), the flow rate of the district HW appeared to be constant. The HW pumping power may be reduced if the HW pump is controlled to maintain a constant TD instead by varying the HW flow rate in response to the varying HW demand.

Although the TD for district CHW also varied by season, it was not as large as the TD for district HW and thus resulted in varying pumping energy use of the district CHW pump over the year.

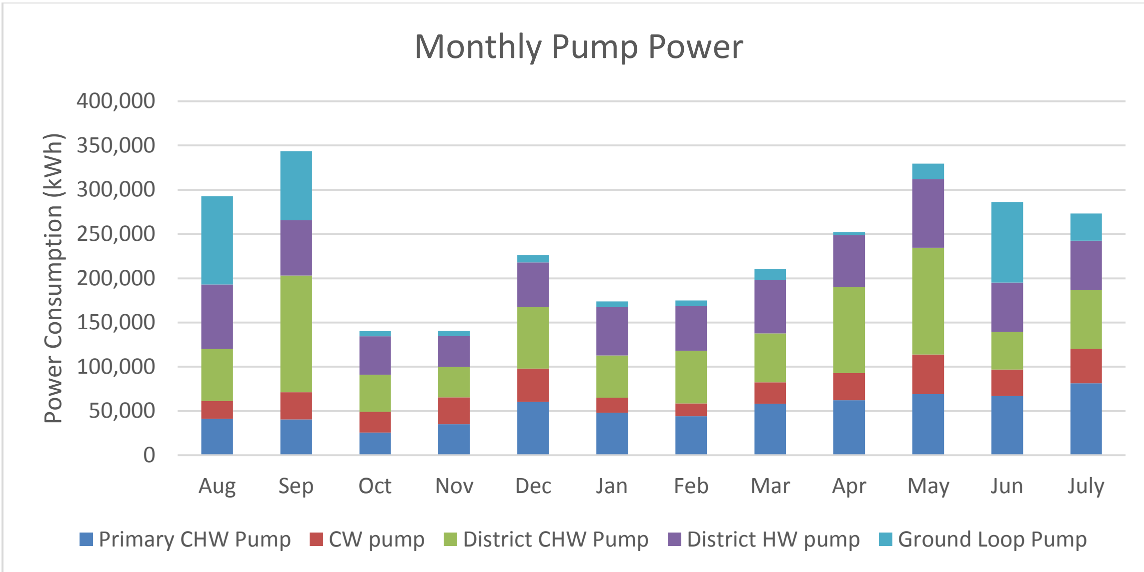


Fig. 16. Monthly pumping power for each pumping station.

Error! Reference source not found. summarizes the monthly heating/cooling outputs, the power consumptions of the chillers and all the circulation pumps (for primary CHW, CW, district CHW, district HW, and ground loop), as well as the ECOPs of the chiller and the GSHP system. The analysis shows that the average ECOP of the district GSHP system at DESN during the investigated period was about 3.74±0.2, while the ECOP of the chiller was about 4.28±0.2 (excluding any pumping power). The ±0.2 uncertainty of ECOPs is calculated based on the accuracies of the sensors (Table 2) following the procedure described in ASHRAE Guideline 14-2014 – Measurement of Energy, Demand, and Water Savings (ASHRAE 2014). The contribution of the pumping power to the total GSHP system power consumption ranges from 8% to 18% across the different months. The lower contribution during heating season is due to the lower pumping energy use in the ground loop and the district CHW loop.

Table 3. Monthly energy consumption and ECOPs for the chiller and the GSHP system

Year	***Month***	***QCHW (MMBtu)***	***QHW (MMBtu)***	***GSHP Chiller (kWh)***	***Total Pump (kWh)***	***Total System GSHP (kWh)***	***Chiller Effective COP***	***System Effective COP***	***Pump (%)***
2015	Aug	16,091	2,469	2,209,493	292,637	2,502,130	2.46	2.17	11.7%
2015	Sep	14,998	3,698	1,793,464	343,700	2,137,164	3.06	2.56	16.1%
2015	Oct	7,945	5,040	983,526	140,324	1,123,850	3.87	3.39	12.5%
2015	Nov	10,797	14,811	1,474,227	140,450	1,614,677	5.09	4.65	8.7%
2015	Dec	12,977	17,912	2,499,243	226,218	2,725,461	3.62	3.32	8.3%
2016	Jan	8,828	18,872	1,993,210	173,819	2,167,029	4.07	3.75	8.0%
2016	Feb	6,968	16,935	1,359,116	174,844	1,533,960	5.15	4.57	11.4%
2016	Mar	6,738	14,255	1,127,858	210,555	1,338,413	5.46	4.60	15.7%
2016	Apr	7,376	13,835	1,292,560	252,180	1,544,740	4.81	4.02	16.3%
2016	May	12,021	13,556	1,507,827	329,548	1,837,375	4.97	4.08	17.9%
2016	Jun	16,754	11,777	1,629,536	286,166	1,915,702	5.13	4.36	14.9%
2016	Jul	18,540	16,579	1,983,840	273,358	2,257,199	5.19	4.56	12.1%
Total		**140,032**	**149,738**	**19,853,900**	**2,843,799**	**22,697,699**	**4.28**	**3.74**	**12.5%**

3.4 BASELINE PERFORMANCE

The energy savings achieved by the district GSHP system compared with a baseline system was evaluated. The baseline system includes a conventional water-cooled chiller and a natural gas boiler, which have the minimum energy efficiency as specified in ASHRAE Standard 90.1-2013 (ASHRAE 2013). The baseline chiller efficiency was 0.57 kW/ton, and the boiler efficiency was 80%. It was assumed that there were two 2,500-ton water-cooled centrifugal chillers, four 1,000-ton cooling towers, and three 20,000 MBH natural gas boilers.

The baseline energy use was estimated using the measured cooling and heating outputs of the GSHP system, and the generic performance curves of centrifugal chiller and gas-fired boiler provided in DOE-2/eQUEST (Hirsch 2016). The pumping energy use of the district CHW and HW loop (i.e., the secondary loop pumps) was assumed to be the same between the baseline and the GSHP systems. A generic school building with a similar baseline system and at the same location was modeled using eQUEST, an hourly building energy simulation program (Hirsch 2016), and the annual percentages of the energy use of the cooling tower and the primary loop pumps (i.e., the cooling water pump) to the energy use of the chillers were determined based on the simulation results. These percentages were used to calculate the annual energy use of the cooling tower and the primary loop pumps for the baseline system. The simulation results indicate that the annual cooling tower and pump energy use is 5.62% and 27.84% of the annual chiller energy use, respectively. As shown in Table 4, the baseline system would consume 10,970 MWh electricity and 212,806 MMBtu natural gas, respectively, in a year to provide the same heating and cooling outputs as the GSHP system.

Table 4. Baseline system energy consumption

Year	*Month*	*Baseline Chiller (kWh) (From Chiller 5)*	*Primary Pumps (kWh)*	*Secondary Pumps (kWh)*	*Cooling Tower (kWh)*	*Total Chiller System (kWh)*	*Total Boiler (MMBtu)*
2015	Aug	738,976		131,622			4,600
2015	Sep	679,219		194,169			6,217
2015	Oct	491,183		85,191			7,946
2015	Nov	545,428		69,038			20,508
2015	Dec	617,338		120,043			24,426
2016	Jan	492,364		102,611			25,570
2016	Feb	432,271		109,856			22,723
2016	Mar	441,150		115,598			19,774
2016	Apr	456,569		155,912			19,241
2016	May	591,777		198,330			18,936
2016	Jun	670,925		98,386			16,524
2016	July	933,958		122,188			26,343
Total		**7,091,157**	**1,974,262**	**1,502,945**	**401,236**	**10,969,600**	**212,806**

3.5 ENERGY SAVINGS POTENTIAL

Table 5 shows the annual savings in source energy, energy cost, and CO_2 emissions. The source energy factor for delivered electricity is 3.443, which is an average value for U.S. Eastern Interconnection according to Deru and Tocellini (2007). The same literature also provides the CO_2 emission factors for delivered electricity (i.e., 1.64 lb of pollutant per kWh of electricity) and natural gas (including 119.0 lb per 1 MMBtu from on-site combustion and 11.3 lb per 1 MMBtu from pre-combustion). These emission factors are used to calculate the CO_2 emissions. The annual source energy savings achieved by the GSHP system is about 96,281 MMBtu, or 27% of the baseline source energy consumption. The annual energy cost savings (assuming $ 0.08/kWh for electricity and $8/MMBtu for natural gas) would be about $764,200, or 30% of the baseline energy cost. In addition, the GSHP system has reduced CO_2 emission by about 8,494,540 lb each year, which is a 19% reduction compared with the baseline system.

Table 5. Source energy and cost savings from the district GSHP system

Annual Energy	
GSHP-Source Energy (MMBtu)	266,720
Baseline Source Energy (MMBtu)	363,001
Source Energy Savings (MMBtu)	96,281 (27% savings)
Annual CO_2 Emissions	
GSHP-CO_2 Emissions (lb)	37,224,227
Baseline-CO_2 Emissions (lb)	45,718,767
CO_2 Emission Reductions (lb)	8,494,540 (19% reduction)
Annual Cost	
Cost-GSHP ($)	1,815,816
Cost-Baseline ($)	2,580,016
Cost Savings ($)	764,200 (30% savings)

3.6 COST-EFFECTIVENESS

The district GSHP system's estimated cost premium is the difference between the installed cost of the GSHP system and the estimated cost of the baseline system described in Sect. 3.4. Cost information for the GSHP system was provided by BSU, and the baseline system cost was estimated based on RSMeans Construction Cost Data (RSMeans 2016) and other sources.

Table 6 summarizes the itemized costs of the installed district GSHP system and the baseline system. The costs for the district pipelines for distributing the CHW and HW are not included since they are the same for both the baseline and the GSHP systems. The total installed costs of the district GSHP system and the baseline system are $17,261,241 and $5,234,750, respectively. The cost premium is $12,026,491. Given the tonnage of the chiller and the total linear length of boreholes, the normalized cost of the GSHP system is about $3,452/ton and the normalized cost of the ground heat exchanger is $18.7/ft, which is higher than the $12/ft average cost in the Midwest region (Battocletti and Glassley 2013).

Based on the annual energy cost savings discussed in previous section, the simple payback of this system is about 16 years. However, if the ground heat exchangers were installed at the regional average cost, the payback would have been 9 years.

Table 6. Cost-effectiveness of the GSHP system compared with the baseline system

Cost item	Baseline System	GSHP System (as built)	GSHP System (regional average)
Baseline chillers	$ 2,117,750		
Baseline setting, rigging, and installation	$ 300,000		
Baseline pumps	$ 852,600		
Baseline cooling towers	$ 450,000		
Baseline boilers	$ 1,514,400		
HR chillers (including pumps and installations)		$ 3,750,000	$ 3,750,000
Boreholes		$ 13,511,249	$ 8,670,320
Total	$ 5,234,750	$ 17,261,241	$ 12,420,320
Cost premium		$ 12,026,491	$ 7,185,570
Simple payback (years)		16	9

3.7 SYSTEM EFFICIENCY

According to the catalog data, when HR chillers produce both 40°F CHW (from evaporator) and 125°F HW (from condenser), they use 1.2 kW of power per each ton of cooling supplied. So, for each ton of cooling supplied, there are 1.34 tons of heating available. The HR chiller could operate very efficiently (i.e., with an ECOP as high as 6.83) if both the produced CHW and HW were fully used to satisfy the cooling and heating demands of the campus. However, if the produced CHW and HW are not fully used, and the surplus heat energy is thus dumped to the ground (as discussed before), the ECOP of the HR chiller would be lower than the Coefficient of Performance (COP) of conventional chiller or heat pump.

Error! Reference source not found. shows ECOP of the HR chiller vs. the ratio of the daily HW demand (the actual output to the district HW loop) to the daily CHW demand (the actual output to the district CHW loop). Each data point in this figure is the average of ECOPs within a 0.2 bin of the HW/CHW ratio. As shown in **Error! Reference source not found.**, the minimum average daily ECOP of the chiller is about 2.5 when the HW/CHW ratio is less than 0.2 (i.e., most of the HW is dumped into the ground loop), and the ECOP is around 5 when the HW/CHW ratio is 1.2. The ECOP varies between 4 and 5 when the HW/CHW ratio is higher than 1.2.

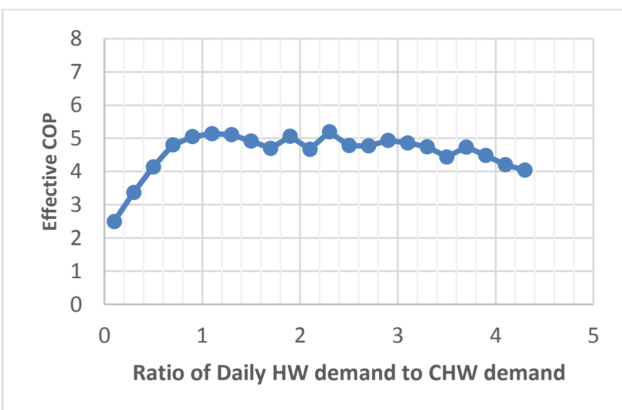


Fig. 17. Average ECOP of the heat recovery chiller.

The operational efficiency of the GSHP system could have been higher if the two chillers were operated separately—with one chiller always producing CHW, and the other operating as a heat pump to produce HW. Given the relatively balanced HW and CHW loads, these separate operations would reduce the ground loop temperature because (1) the (heat pump) chiller would produce only the needed HW necessary to satisfy the heating demand, eliminating dumping of 125°F HW into the ground loop; and (2) heat extraction from and heat rejection to the ground would be more balanced since the (heat pump) chiller would extract more heat from the ground.

The power consumptions of the two separately operated chillers for satisfying the same heating and cooling demands are calculated. The average cooling and heating COPs of the chiller are obtained from the catalog data of a modular heat pump chiller, as shown in Fig. 18. The average heating COP is 4 assuming a 50°F leaving source water temperature and a 125°F leaving HW temperature; and the average cooling COP is 6 assuming a 90°F leaving source water temperature and a 42°F leaving CHW temperature. To satisfy the same heating and cooling demands (140,032 MMBtu cooling and 149,738 MMBtu heating), the two separately operated chillers would consume 17,806 MWh electricity. Compared with the electricity consumption of the as-built/operated GSHP system (19,853 MWh), the separate operations will save 2,047 MWh electricity. However, the separate operations will result in an increase in ground loop pumping energy use since more heat is extracted from and rejected to the ground loop. The current annual ground loop pumping energy use is 334,430 kWh for the 87,804 MMBtu ground loop loads. Assuming that the pumping energy use is proportional to the ground loop loads, the new ground loop pumping energy use is calculated with the anticipated ground loop loads (i.e., heat rejected plus heat extracted) resulting from the separated operation. The calculation result indicates that the ground loop pumping energy use will increase by 715 MWh, so the net savings in electricity is 1,331 MWh, which is of $106,557 value given the $0.08 per kWh electricity rate. This additional electricity savings will shorten the payback of the GSHP system to 14 years with the as-built ground heat exchanger cost, and in the case of regional average ground heat exchanger cost, the payback is shortened to 8 years.

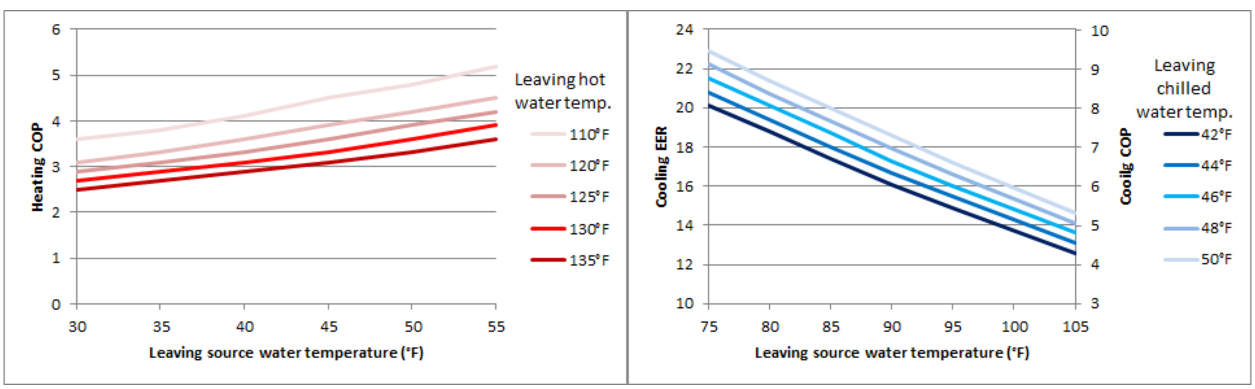


Fig. 18. Efficiency curves of a heat pump chiller in heating and cooling modes.

4. CONCLUSIONS AND LESSONS LEARNED

This case study analyzes the energy performance of a district central GSHP system installed at BSU in Muncie, IN. The GSHP system replaced existing coal fired steam boilers and conventional water-cooled chillers. Upon completion of the Phase 2 GSHP retrofit, the entire central heating and cooling system of

BSU will be transitioned to a central GHP system using HR chillers. This case study is conducted based on the available measured performance data from August 2015 through July 2016, construction drawings, maintenance records, personal communications, and construction costs. The annual energy consumption of the GSHP system was calculated based on the available measured data and other related information. It was compared with the performance of a baseline scenario— a conventional water-cooled chiller and natural-gas-fired boiler system, each with the minimum energy efficiencies allowed by ASHRAE 90.1-2013. The comparison was made to determine source energy savings, energy cost savings, and CO_2 emission reductions achieved by the GSHP system. A cost analysis was performed to evaluate the simple payback of the GSHP system. The following sections summarize the results of the analysis, the lessons learned, and recommendations for improvement in the operation of this district GSHP system.

4.1 ENERGY PERFORMANCE AND COST-EFFECTIVENESS

- The annual heating and cooling outputs of the GSHP system (from August 2015 through July 2016) were 149,738 MMBtu and 140,032 MMBtu, respectively. An analysis of the heating and cooling outputs reveals that 78% of the heating outputs were recovered during the simultaneous heating and cooling operation of the HR chillers. As a result, the heat rejection and heat extraction at the ground loop were imbalance—the heat rejection (71,684 MMBtu) is 4.5 times larger than the heat extraction (16,120 MMBtu) on an annual basis.

- The analysis shows that the average system Effective Coefficient of Performance (ECOP) during the monitoring period was about 3.74±0.2, while the chiller ECOP is about 4.28±0.2. The fraction of the central system plant monthly pumping power (excluding the local pumps in each building to distribute the hot water [HW] and chilled water [CHW] provided by the district GSHP system) ranges from 8% to 18% of the total system power consumption. Lower pumping power percentages during the heating season are due to the low flow rate in the ground loop resulting from the low heat extraction load.

- An analysis of the historical steam productions at BSU indicates a 40% reduction after the GSHP system was implemented in the campus. By converting from the original steam heating system to HW heating loops in the buildings, heat loss from the distribution pipelines was significantly reduced.

- It is calculated that the baseline system would have consumed 10,970 MWh of electricity and 212,806 MMBtu of natural gas to produce the same heating and cooling outputs as the GSHP system. The total electricity use for the GSHP system during the same period was 22,698 MWh. The source energy savings achieved by the GSHP system is about 96,281 MMBtu, or 27% of the baseline source energy consumption. Carbon emission reduction achieved by the GSHP system is about 8,494,540 lb, or 19% of the baseline carbon emission. The net cost savings (assuming $ 0.08/kWh for electricity and $8/MMBtu for natural gas) would be about $764,200, or 30% of the baseline energy cost.

- Based on the actual installation cost of the GSHP system and the estimated baseline system installation cost, a cost-effectiveness of the GSHP system was evaluated. The total installed costs of the GSHP system and the baseline system are $17,261,241 and $5,234,750, respectively. The simple payback of this GSHP system is about 16 years. However, if the ground heat exchangers were installed at the regional average cost, the payback would have been 9 years.

4.2 LESSONS LEARNED AND RECOMMENDATIONS

- The HR chillers could operate very efficiently when both the produced CHW and HW are fully utilized to satisfy the cooling and heating demands of the campus. However, the current configuration and control of the chillers resulted in over-production of hot or chilled water. This is extremely wasteful and has resulted in not only excessive chiller energy use but also unexpectedly high ground loop temperatures. As a result, the operational efficiency of the GSHP system is lower than expected.

- The operational efficiency could have been higher if the two chillers were operated separately—one dedicated to producing CHW and the other operating as a heat pump to produce only HW. In this case, there would be no need to dump excess CHW or HW to the ground. In addition, the ground loop loads would be more balanced since more heat would be extracted from the ground to satisfy the heating demand.

- If the installed chillers cannot be operated separately as suggested above, adding storage tanks in both the CHW and HW loops would help reduce or eliminate the need to dump excess CHW or HW to the ground and thus improve the operational efficiency of the GSHP system.

5. REFERENCES

ASHRAE Standard 90.1-2013, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, ASHRAE standard.

ASHRAE Guideline 14-2014, *Measurement of Energy, Demand, and Water Savings*, ASHRAE standard.

Battocletti, E. C. and W. E. Glassley. 2013. Measuring the Costs and Benefits of Nationwide Geothermal Heat Pump Deployment. Bob Lawrence & Associates, Inc. DE-EE0002741. Available online at http://www.osti.gov/geothermal/servlets/purl/1186828. Database available online at https://gdr.openei.org/submissions/180

Deru, M., and P. Tocellini. 2007. *Source Energy and Emission Factors for Energy Use in Buildings*, Technical Report, NREL/TP-550-38617, National Renewable Energy Laboratory, Golden, Colorado.

Hirsch J. et al. (2016). eQUEST program. www.doe2.com

U.S. Department of Energy (DOE). 2015 Energy Efficient District Heating Systems, Advanced Manufacturing Office (AMO), U.S. DOE Energy Efficiency & Renewable Energy (EERE).

RSMeans. 2016. “Cost Data Online,” http://www.rsmeans.com/RSMeans_Online.asp