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Final Report:

Case Study for the ARRA-funded Ground Source Heat Pump (GSHP) Demonstration at Wilders Grove Solid Waste Service Center in Raleigh, NC

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CASE STUDY FOR THE ARRA-FUNDED GROUND SOURCE HEAT PUMP (GSHP) DEMONSTRATION AT WILDERS GROVE SOLID WASTE SERVICE CENTER IN RALEIGH, NC
(Final Report)

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ABSTRACT

High initial costs and lack of public awareness of ground-source heat pump (GSHP) technology are the two major barriers preventing rapid deployment of this energy-saving technology in the United States. Under the American Recovery and Reinvestment Act (ARRA), 26 GSHP projects have been competitively selected and carried out to demonstrate the benefits of GSHP systems and innovative technologies for cost reduction and/or performance improvement. This paper highlights the findings of a case study of one of the ARRA-funded GSHP demonstration projects, a distributed GSHP system for providing all the space conditioning, outdoor air ventilation, and 100% domestic hot water to the Wilders Grove Solid Waste Service Center of City of Raleigh, North Carolina. This case study is based on the analysis of measured performance data, construction costs, and simulations of the energy consumption of conventional central heating, ventilation, and air-conditioning (HVAC) systems providing the same level of space conditioning and outdoor air ventilation as the demonstrated GSHP system. The evaluated performance metrics include the energy efficiency of the heat pump equipment and the overall GSHP system, pumping performance, energy savings, carbon emission reductions, and cost-effectiveness of the GSHP system compared with conventional HVAC systems. This case study also identified opportunities for reducing uncertainties in the performance evaluation and improving the operational efficiency of the demonstrated GSHP system.

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ACRONYMS

APWA	American Public Works Association
ARRA	American Recovery and Reinvestment Act
CAV	constant air volume
COP	coefficient of performance
CPI	consumer price index
CVRMS	coefficient of variation of the root mean square error
DBT	dry bulb temperature
DHW	domestic hot water
DOAS	dedicated outdoor air system
DOD	US Department of Defense
DOE	US Department of Energy
DPT	dew point temperature
EA	exhaust air
EER	energy efficiency ratio
EFT	entering water/fluid temperature
EIA	Energy Information Administration
ERV	energy recovery ventilator
GHX	ground heat exchanger
GSHP	ground source heat pump
HVAC	heating, ventilation, and air conditioning
IMT	inverse modeling tool
LEED	Leadership in Energy & Environmental Design
NMBE	normalized mean bias error
OA	outdoor air
TSP	total static pressure
VAV	variable air volume
VFD	variable frequency drive
WAHP	water-to-air heat pump
WAHP	water-to-air heat pump
WSHP	water source heat pump
WWHP	water-to-water heat pump

1. INTRODUCTION

In 2009, the Wilders Grove Solid Waste Service Center of Raleigh, NC, was chosen by the US Department of Energy (DOE) to demonstrate a ground source heat pump (GSHP) system. The GSHP system provides space conditioning and water heating services to a new 24,000 ft^2 Solid Waste Services Administration Building, which is located on a 27 acre site adjacent to the closed Wilders Grove Landfill. The location of the demonstration site is shown in Fig. 1.

The demonstrated GSHP system is in a distributed configuration, with a vertical closed-loop ground heat exchanger as the heat sink and source. Figure 2 is an aerial photo of the building site and its borehole field. The objectives of this demonstration project include:

- demonstration of viability of the GSHP system for a cooling-dominated building
- contributing to the goal of 20% reduction of fossil fuel use in Raleigh
- helping meet minimum Leadership in Energy & Environmental Design (LEED) Silver certification standards in the design and commissioning of new facilities, with the goal of achieving LEED Platinum
- achieving energy savings of 30% compared to conventional systems
- providing a comfortable work environment for the staff

The design of the building was completed in 2010, and the construction of the building was completed in March 2012. A centralized building energy management and control system was implemented to provide a means for the facility to monitor and control its energy use. The GSHP system began to be monitored in July 2012; monitoring is expected to end in July 2016.

Fig. 1. Location of Wilders Grove Solid Waste Services Facility GSHP demonstration site.

Fig. 2. Aerial photo of the building (shaded in red) and its borehole field (the rectangular array of orange dots in the south side of the building).

This case study evaluates the performance of the demonstrated GSHP system using measured performance data and other relevant information. The performance metrics include the overall energy efficiency of the GSHP system in heating and cooling modes; electricity uses of all major equipment of the GSHP system; and benefits from using the GSHP system (e.g., energy and cost savings, carbon emission reductions, and peak electric demand reductions). The cost effectiveness of the GSHP system is also analyzed based on the achieved energy cost savings and the cost premium compared with a new conventional HVAC system. This study also identifies some areas that need further improvement so that the GSHP system can achieve higher operational efficiency, as well as issues in collecting data to monitor and evaluate the performance of this distributed GSHP system.

1.1 BUILDING INFORMATION

The Solid Waste Services Administration Building incorporates many renewable, sustainable, and energy efficient systems and measures. In addition to achieving LEED Silver status, which is required by the city council of Raleigh for all new projects, particular effort was made to meet the LEED Platinum level, including incorporating solar photovoltaic roof panels and measurement devices to track actual performance of the building and to enhance commissioning services (APWA 2013).

Figure 3 shows the external appearance of the Solid Waste Services Administration Building. Figure 4 shows the interior spaces of the building, which include offices, gathering/training rooms and locker rooms.

Fig. 3. The Solid Waste Services Administration Building.

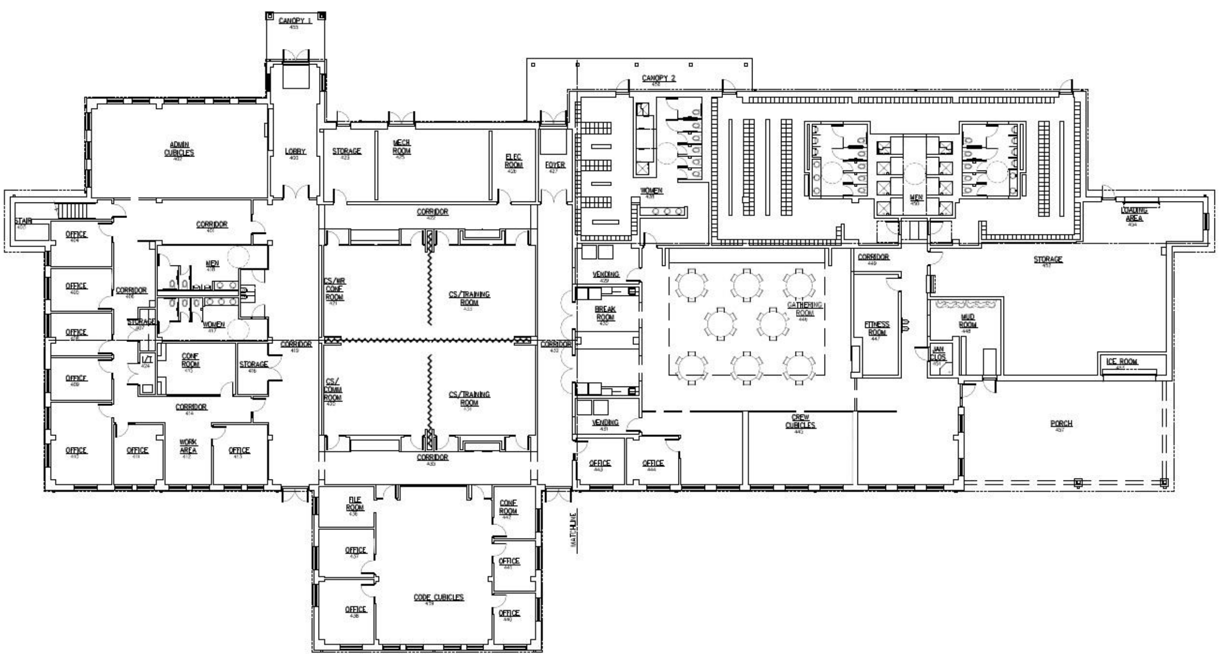

Fig. 4. Floor plan of the building.

1.2 GSHP SYSTEM

The GSHP system provides climate control to the building in a distributed configuration—packaged water source heat pump (WSHP*) units are used in each individual zone of the building. The WSHP units are attached to a common two-pipe water loop. The water loop is connected to a ground loop field, which was installed under a parking lot on the south side of the building. Figure 5 is schematic of the GSHP system and the location of data collection points (Walburger and Perriello 2013). Descriptions of each data collection point are given in Table 2.

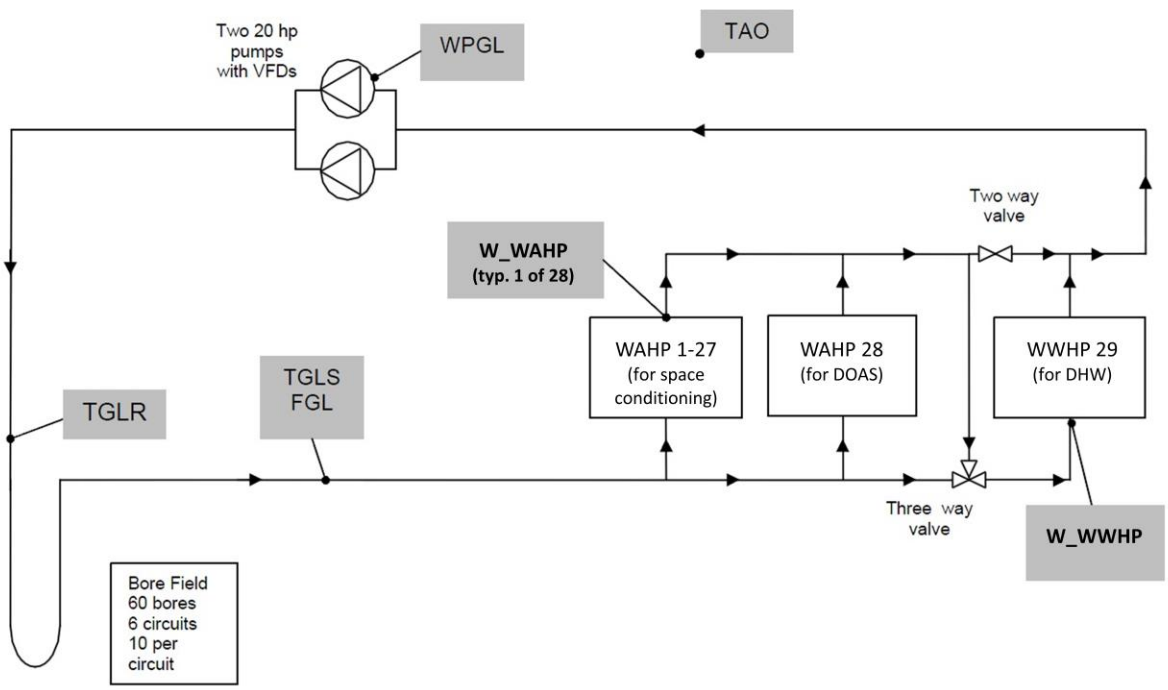

Fig. 5. GSHP system schematic with monitored data points.

The system contains 28 ClimateMaster water-to-air heat pump (WAHP) units, ranging in size from 0.75 to 50 tons, and a 25 ton Heat Harvester water-to-water heat pump (WWHP) for generating domestic hot water (DHW). Twenty-seven of the WAHP units are for space conditioning, and the other one is used to condition the outdoor air in a dedicated outdoor air system (DOAS). The combined cooling capacity of all the WAHP units is 134.5 tons. With the 25 ton WWHP, the total installed capacity of the entire GSHP system is 159.5 tons. The twenty-seven WAHP units operate independently of the DOAS to maintain room temperatures at desired levels (between 70 and 75°F in occupied hours and between 60 and 85°F in unoccupied hours). According to the “sequence of operation” provided by the building owner, the occupied hours are between 5:00 A.M. and 9:00 P.M., Monday through Thursday. The model numbers and cooling capacities of the 29 heat pump units and their service areas are given in Table 1.

As shown in Fig. 6, the ground loop field consists of 60 vertical bores in a 10 × 6 grid spaced on 25 ft centers.† Each bore is 335 ft deep and 6.3 in. in diameter. A 1.25 in. high-density polyethylene U-tube is inserted into each bore. The bores are grouped in six circuits, and each circuit has a shutoff valve that is accessible from a vault to isolate it from other circuits in case of water leakage. The ground loop heat

* This term includes both the water-to-air and water-to-water types of heat pump units.
† The planned boreholes shown in Fig. 6 had not been drilled at the time of this demonstration project.

exchanger uses water as a heat transfer carrier. An in-situ test was conducted in 2010, and the results indicated that the effective ground thermal conductivity along the depth of the vertical bore is 1.55 Btu/h·ft·°F; the estimated diffusivity for this formation is 1.07 ft^2/day; and the undisturbed ground temperature is estimated to be 62°F.

Table 1. Basic information of heat pump units

Heat Pump Number	Serving Room/Use	Manufacturer and Model	Size (tons)
WAHP-01	Lobby 400	Climatemaster Model TS24-ECM	2
WAHP-02	Admin Cubicles 402	Climatemaster Model TT064-Full Load	5
WAHP-03	Offices 404, 405, 408, 409	Climatemaster Model TT038-Full Load	3
WAHP-04	Offices 410	Climatemaster Model TT026-Full Load	2
WAHP-05	Offices 411, 412, 413	Climatemaster Model TT038-Full Load	3
WAHP-06	Conference Room 415	Climatemaster Model TS009-PSC	0.75
WAHP-07	Restrooms 417, 418	Climatemaster Model TT026-Full Load	2
WAHP-08	File Area 401, corridor	Climatemaster Model TS012-PSC	1
WAHP-09	File Room 436, Office 437	Climatemaster Model TS018-PSC	1.5
WAHP-10	Office 438	Climatemaster Model TS018-PSC	1.5
WAHP-11	Code Cubicles 439	Climatemaster Model TT064-Full Load	5
WAHP-12	Office 440	Climatemaster Model TS012-PSC	1
WAHP-13	Office 441	Climatemaster Model TS009-PSC	0.75
WAHP-14	Conference 442	Climatemaster Model TS012-PSC	1
WAHP-15	Comm. Room 420	Climatemaster Model TT038-Full Load	3
WAHP-16	WR. Conf. Room 421	Climatemaster Model TT038-Full Load	3
WAHP-17	Training Room 433	Climatemaster Model TT038-Full Load	3
WAHP-18	Training Room 434	Climatemaster Model TT038-Full Load	3
WAHP-19	Storage 416, Corridor	Climatemaster Model TT038-Full Load	3
WAHP-20	Break Rm. 430,429,431	Climatemaster Model TLV-192	15
WAHP-21	Office 443, 444	Climatemaster Model TS018-ECM	1.5
WAHP-22	Crew Cubicles 445	Climatemaster Model TS070-ECM	5
WAHP-23	Rooms 451, 446, 448, 449	Climatemaster Model TS070-ECM	5
WAHP-24	Fitness 447	Climatemaster Model TS018-ECM	1.5
WAHP-25	Loading Area 454	Climatemaster Model TS018-ECM	1.5
WAHP-26	Women’s Locker 428A, 428B	Climatemaster Model TT038-Full Load	3
WAHP-27	Men’s Locker 450A, 450B	Climatemaster Model TLV-120	7.5
WAHP-28	Condition outdoor fresh air	Climatemaster Model TOHW20	50
WWHP-29	Generate domestic hot water	Heat Harvester DS-40	25

A central variable-speed pump station is used to circulate water flow through the ground loop field and each individual WSHP unit via the common water loop. Two identical circulation pumps are installed and operate in a lead/lag configuration (alternating lead and lag every other week); under normal conditions, only one pump operates. According to the sequence of operation provided by the grantee, the variable-speed drives will modulate the pump speed to maintain the pressure differential across the supply and return lines of the ground loop at a user-specified (adjustable) set point. The set point specified in the original “sequence of operation” is 50 psi (see Appendix A), but it was lowered to 20 psi in 2013, according to the facility manager. Each of the two circulation pumps has a nominal power draw of 20 hp at the 407 gal/min maximum flow rate. To avoid overheating the pump, it should not run below 20% of its full speed. However, the as-built mechanical design indicates that the system circulates a minimum 357 gal/min water flow even when all the WSHP units are not in operation (see Appendix B). The minimum 357 gal/min water flow is a much higher flow rate than the pump can provide at 20% speed (81 gal/min).

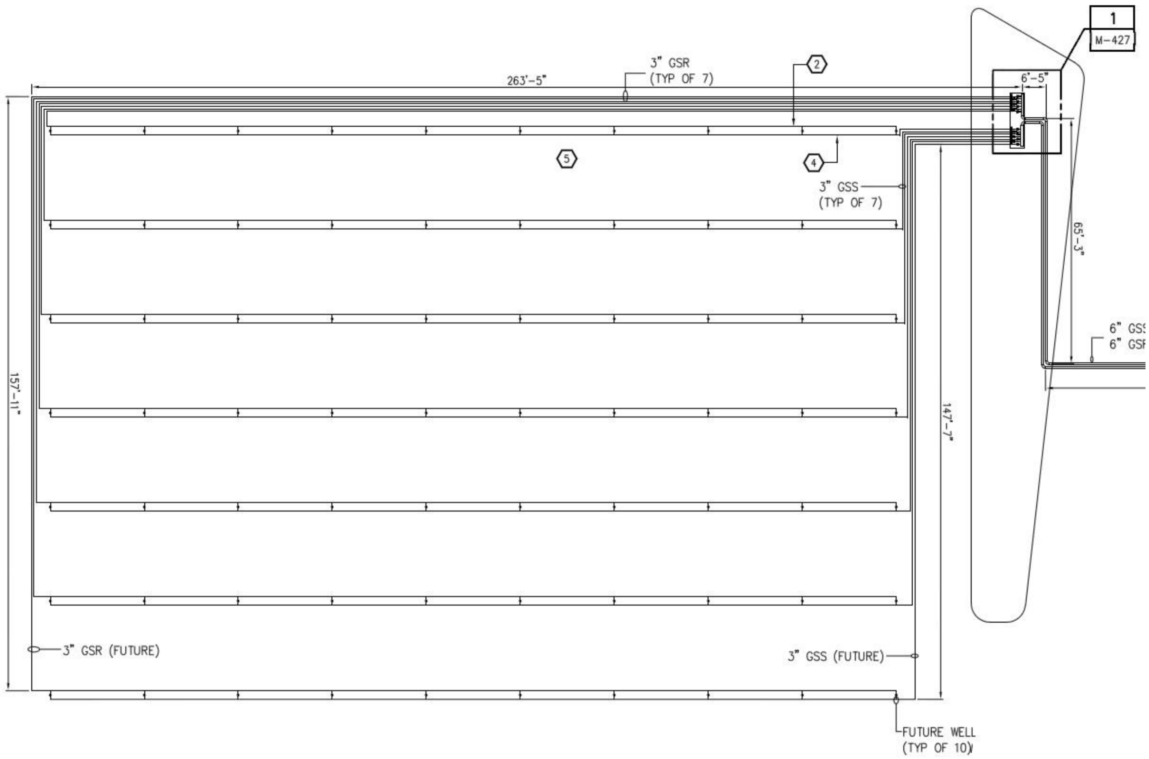

Fig. 6. Layout of the ground loop field.

The DOAS includes a ground-coupled refrigeration unit, a supply fan, an exhaust fan, an enthalpy recovery wheel, and an emergency electric heating coil. According to the sequence of operation provided by the building owner, the refrigeration unit is cycled on and off to maintain the supply air temperature at 55°F. Should the supply air temperature drop below 50°F, the emergency electric heating will be turned on to maintain the supply air temperature set point. The conditioned outdoor air (OA) is distributed to each zone through ductwork and a series of constant air volume (CAV) boxes and variable air volume (VAV) boxes. DOAS is designed to operate continuously to supply the 55°F OA to the building; and the air flow shall be reduced during unoccupied hours when the CAV boxes will be at their closed position but the VAV boxes continue to operate as in occupied mode. The VAV boxes are modulated to maintain the carbon dioxide levels in the conditioned space below 450 ppm. The building owner later indicated that "The DOAS always cools the air first based on a relative humidity algorithm to some temperature (not specified) and then reheats using hot gas to 70°F" (Black 2014). Based on measured data presented in Sect. 2.2.1, it is likely that the cooling operation of the DOAS is controlled by the dew point temperature (DPT) of the OA (leaving the DOAS). The DOAS cools the OA only when the DPT is higher than about 55°F.

According to the catalog data of the DOAS, two 7.5 hp variable speed fans are used for supplying 8,010 ft^3/min OA to the building and exhausting a slightly smaller amount of indoor air from the building. The supply fan is modulated to maintain a constant supply duct static pressure set-point (approximately

1.5 in. water). The exhaust fan is modulated to maintain the building at a slight positive pressure (not higher than 0.05 in. water column). A 0.25 hp motor is used to drive the enthalpy wheel.

DHW is generated by the WWHP unit. An emergency electric heating element is also provided in the hot water tank tied with the WWHP unit. The WWHP generates hot water between 9:00 P.M. and 5:00 A.M. (off-peak electrical hours). Two 1 hp pumps (which run alternate weeks) are used to circulate the flow in the DHW loop.

The WWHP will run during the operating hours until the water temperature at the lower portion of the tank reaches 140°F. Should the WWHP fail, the electric heating element will be activated to maintain the tank temperature at the set point. If the hot water temperature at the top of the tank falls below 120°F between 5:00 A.M. and 2:00 P.M., the electric heating element, WWHP, or both, will be started to heat up the tank. The source side supply water flow of the WWHP can be switched, depending on the operating mode of the HVAC system. When the HVAC system operates in heating mode, the leaving temperature from the ground loop will be routed to the source side inlet of the WWHP; when the HVAC system operates in cooling mode, the leaving fluid temperature of the WAHP units will be routed to the source side inlet of the WWHP. This switch is accomplished by switching a three-way return water valve and restricting the normal return water flow with a two-way valve, as shown in Fig. 5. The purpose of this switch is to use available warmer water as the heat source for the WWHP unit so that a higher water heating efficiency can be achieved.

1.3 DATA COLLECTION

A data collection plan was developed in accordance with the DOE protocol (ORNL/DOE 2010) for data collection from the American Recovery and Reinvestment Act (ARRA)–funded GSHP demonstration projects. Table 2 is a list of the measured data points and the sensors used to measure data from Wilders Grove Solid Waste Services Center. The measured data are collected by the grantee at 15 min intervals from the building energy management system at the facility and are automatically transferred to and stored on a server. Performance data collected from Sept. 1, 2012, to Aug. 31, 2013 (a whole year), are used in this case study. It was found that 4,067 out of 35,040 15 min interval data entries during the one year period were missing, which accounts for 12% of the total data entries.

Table 2. Wilders Grove Solid Waste Services Center GSHP system monitoring points

Data Point	Description	Units	Sensor
TGLS	Ground loop supply temperature	deg F	ACI (TS-2100-PH)
TGLR	Ground loop return temperature	deg F	ACI (TS-2100-PH)
TAO	Ambient temperature	deg F	ACI (TS-2100-PH)
FGL	Ground-loop flow	gal/min	Onicon (F-1111)
WPGL	Ground-loop pump energy	kWh	VSD register
W_WAHP_H (1-27)	Power consumption of each of the 27 WAHP units in heating mode	kWh	WattNode Pulse
W_WAHP_C (1-27)	Power consumption of each of the 27 WAHP units in cooling mode	kWh	WattNode Pulse
W_WAHP (28)	Power consumption of DOAS	kWh	WattNode Pulse
W_WWHP	Power consumption of WWHP	kWh	WattNode Pulse
WPDHW	Power consumption of DHW pump	kWh	WattNode Pulse

2. ANALYSIS OF MEASURED DATA

The objectives of this data analysis are to (1) evaluate the performance of the GSHP units, the entire GSHP system, the pumping system, and the DOAS; (2) assess the control and operation of these systems; (3) and identify faults in system operation/control, if there are any, and recommend solutions for performance improvement.

2.1 GROUND LOOP DATA ANALYSIS

The ground-loop supply and return temperatures, as well as the ambient temperature, are plotted against time in Fig. 7. As shown in Fig. 7, the ground loop supply fluid temperature varied in a very narrow range, from 61 to 77°F, compared to the ambient temperature fluctuating from 18 to 98°F. Given the moderate imbalance between the annual cumulative heat rejection and extraction loads imposed to the ground heat exchanger (GHX) as discussed later, the narrow range of the ground loop supply temperature indicates that the GHX is oversized. The grantee confirmed that the GHX was oversized to allow for future expansion of the facility.

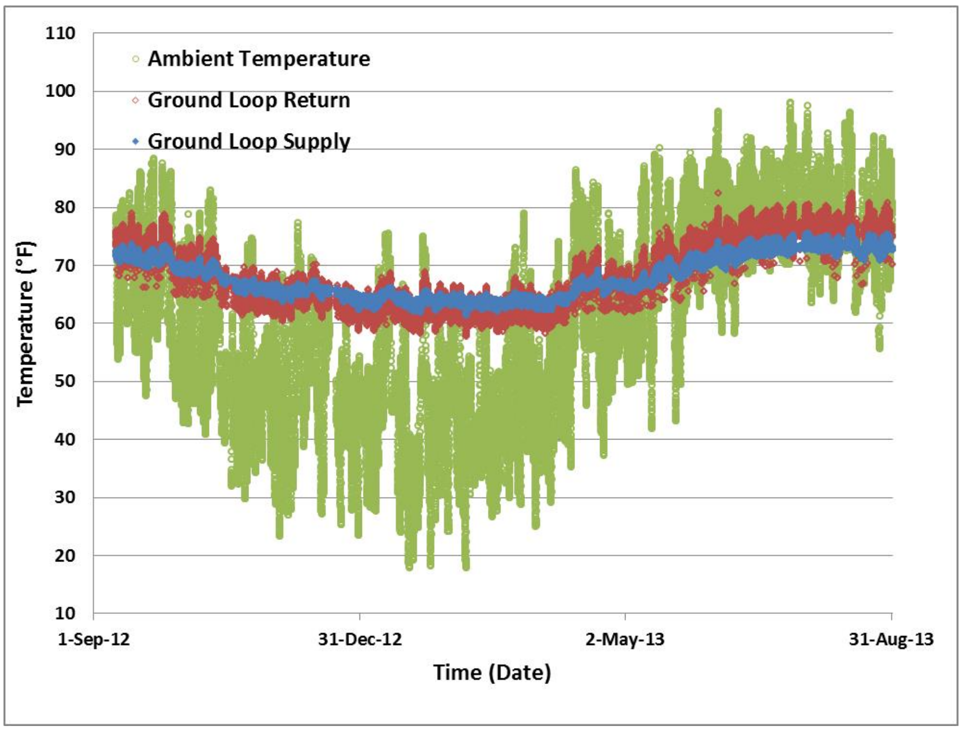

Fig. 7. Measured ground-loop supply, return fluid, and ambient temperatures from September 2012 to August 2013.

As shown in Fig. 8, the temperature differential between the supply and the return of the ground loop is less than 6°F for most of the time. The few outliers are the result of occasional abnormal operation of the data acquisition system. Among all the valid data points, the maximum temperature differential across the ground loop is 6.2°F, which occurred on Aug. 13, 2013 (Fig. 9). The highest ambient temperature was 92.3°F on that day, and about 90% of the 159.5 ton total installed cooling capacity of the entire GSHP system was used when the maximum temperature differential occurred. Since the typical design temperature differential between the supply and return of the ground loop is 10°F the measured low temperature differential indicates that the ground-loop pump circulated a more than adequate water flow (or in other words, the pump is oversized, or excessive pumping is provided).

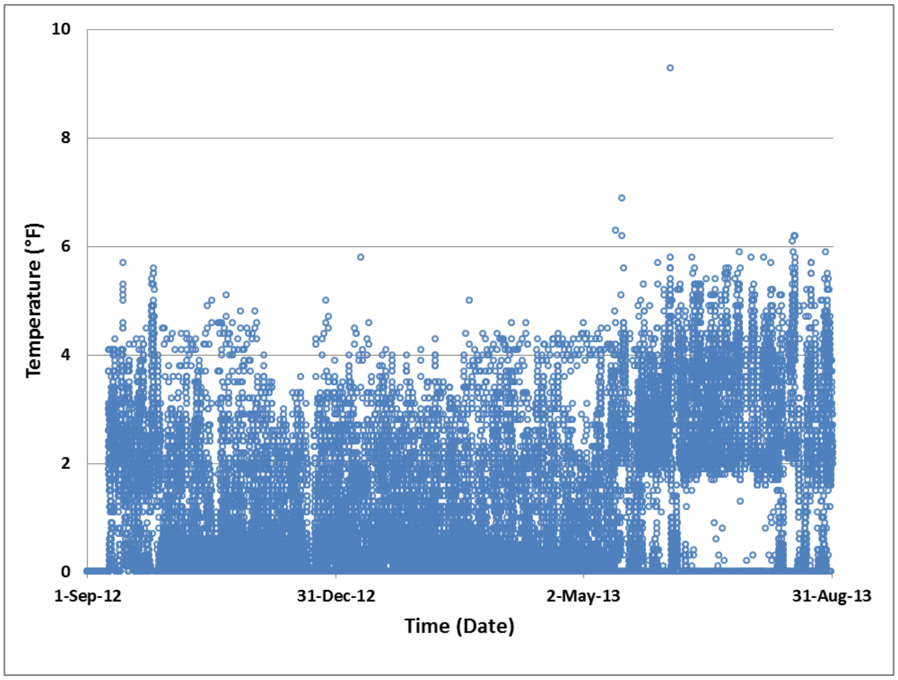

Fig. 8. Temperature differential between the supply and return of the ground loop over a 1-year period.

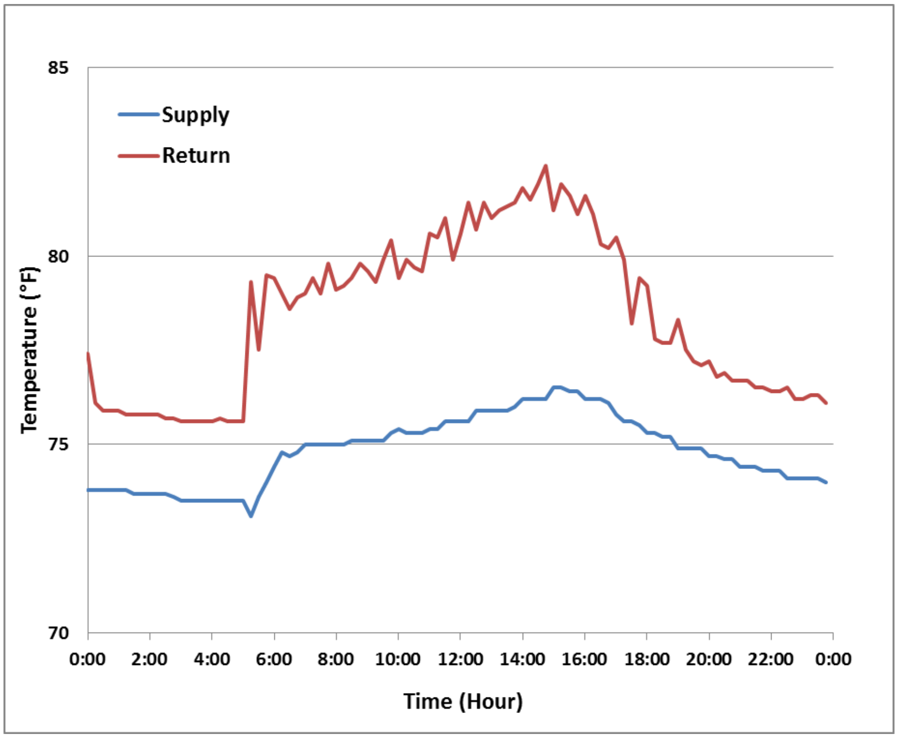

Fig. 9. Ground-loop supply and return temperatures at peak load day—Aug. 13, 2013.

Further analysis showed that the ground loop pump ran continuously all the time during the entire year and that the ground loop flow rate varied in a very narrow range. Figures 10 and 11 show the measured ground-loop flow rate against the coincidental combined power consumption of all heat pump units (including the 27 WAHP units, DOAS, and the WWHP unit) and the ground-loop temperature differential, respectively, in every 15 min interval during the year (when data are available). As shown in these figures, the ground-loop flow rate varied within a narrow range (300–400 gal/min), and it was more than 300 gal/min when the power consumption of the WSHP units or the loop temperature differential was zero. There is no clear relationship between the flow rate and the power consumption of the heat pumps or the ground loop temperature differential.

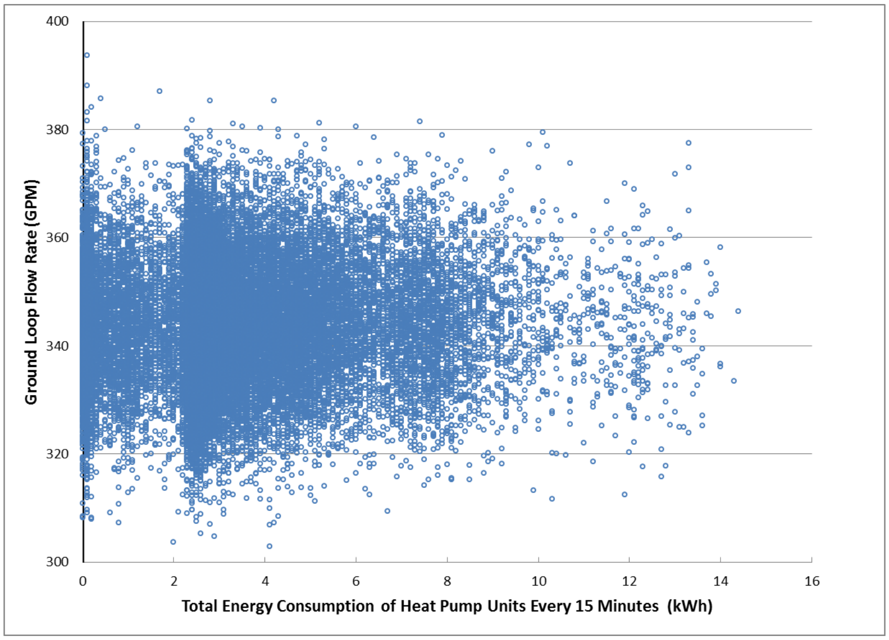

Fig. 10. Ground loop flow rate vs. total energy consumption of heat pump units.

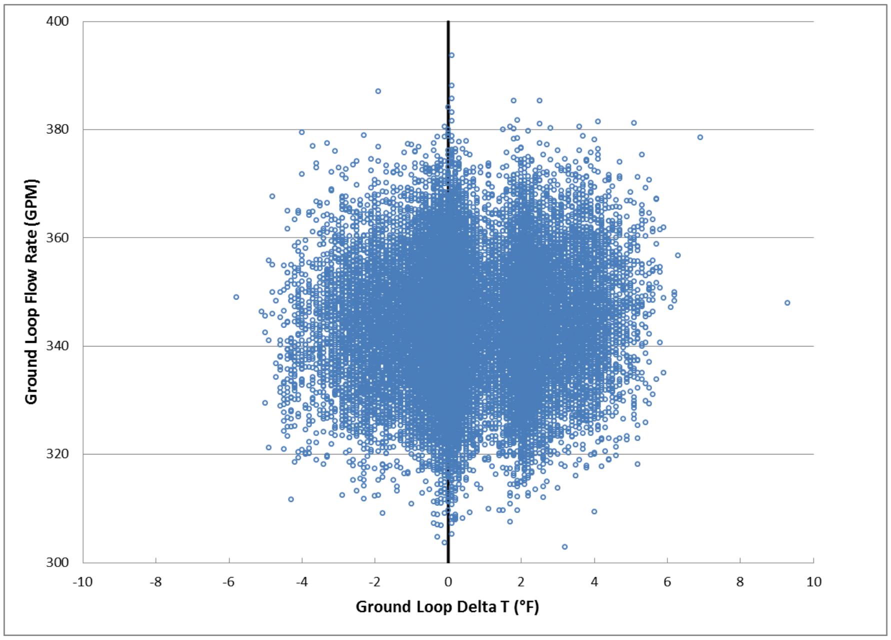

Fig. 11. Ground-loop flow rate *vs* ground-loop fluid temperature differential.

The reason for the small temperature differential and the narrow variation of the ground loop flow rate is thought to be the high bypass flow rate of the ground loop (a minimum 357 gal/min ground-loop flow is indicated in the design document). The authors have recommended the facility manager of the Wilders Grove Solid Waste Services to reduce the bypass flow in order to reduce the ground loop flow rate and the associated pumping power.

The ground loop loads are calculated with Eq. (1) and plotted in Fig. 12, with heat rejection loads (negative) in blue and heat extraction (positive) loads in red. During the 1 year period of data collection, the total amount of heat rejected to the ground was 2.6 times more than the amount of heat extracted. Significant heat rejection loads were observed in winter, which is common for an office building in Raleigh, NC, due to the mild weather in winter and substantial internal heat gains inside the building. On the other hand, heat extractions were observed during the summer, which is due to the operation of the WWHP for generating hot water.

$$Q_{GL} = k \cdot q \cdot \frac{(T_{out} - T_{in})}{1000} \tag{1}$$

where

Q_{GL} is ground loop load, in kBtu/h;

k is a factor that incorporates a conversion factor, specific heat and density of the fluid, which is estimated to be 480 Btu/(h-gal/min-°F) for 20% ethanol at 60°F in this case;

q is the ground loop fluid flow rate, in gal/min;

T_{out} is the outlet (supply) fluid temperature of the ground loop, in °F;

T_{in} is the inlet (return) fluid temperature of the ground loop, in °F.

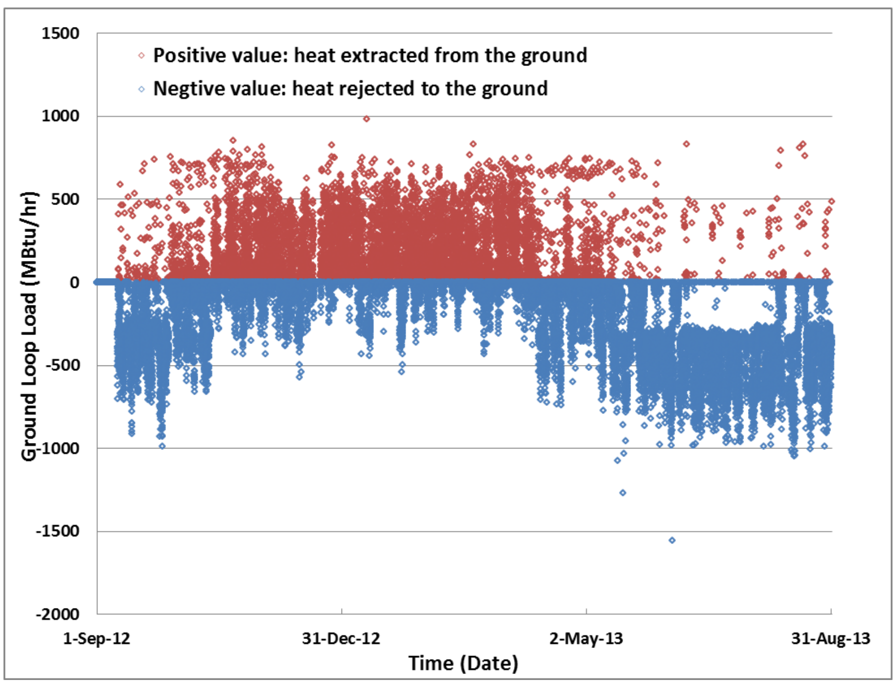

Fig. 12. Ground loop loads from September 2012 to August 2013.

To have a better understanding of the cause for the counter-season operations (i.e., extracting heat from the ground loop in summer, or rejecting heat to the ground in winter), the counter-season ground loop loads during a summer month (August) and a winter month (January) are plotted against the hours in a day in Figs. 13 and 14.

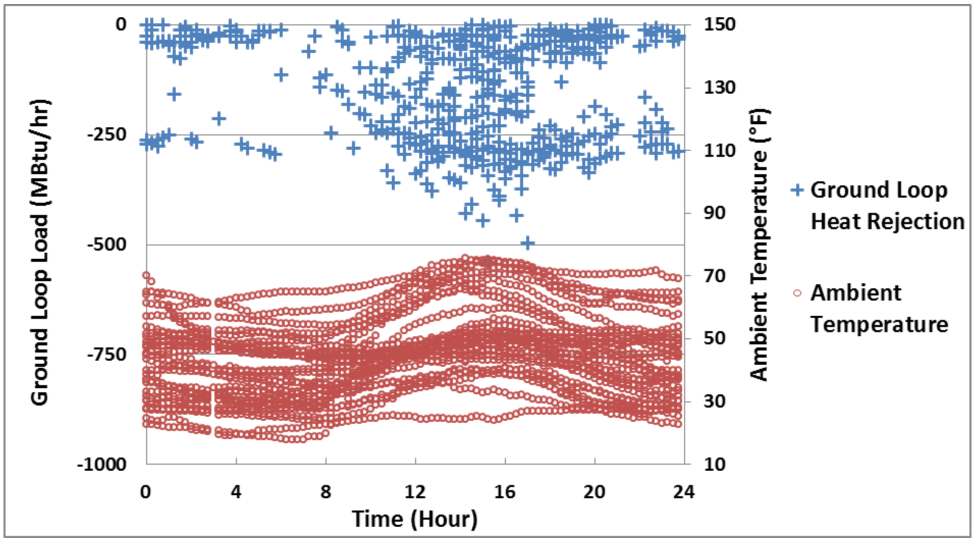

Fig. 13. Ground loop heat rejections in January 2013.

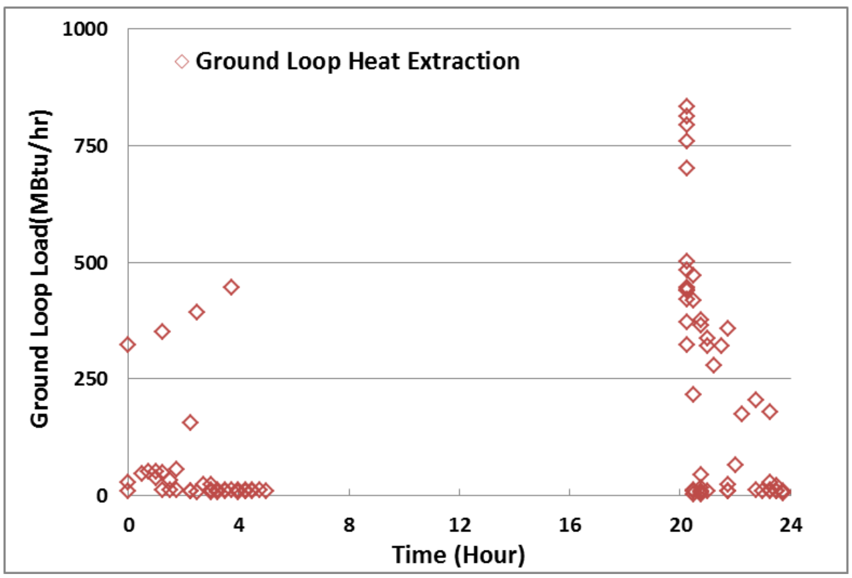

Fig. 14. Ground loop heat extractions in August 2013.

As shown in Fig. 13, most heat rejections in January happened from 12:00 P.M. to 8:00 P.M., when the weather was warmer during the day and the internal heat gains were high in the building (i.e., workers came back to building to take showers and change clothes). However, some heat rejections were also observed during the night time, which may be due to warm weather, or occasional high internal heat gains inside the building. Figure 14 shows that the heat extractions during the summer time happened from 9:00 P.M.to 5:00 A.M., which is consistent with the operating schedule for the WWHP as described previously.

2.2 POWER CONSUMPTION OF GSHP SYSTEM EQUIPMENT

Power consumption data for the DOAS, the twenty-seven WAHP units, the WWHP unit, and the circulation pump are analyzed separately in the following sections.

2.2.1 Power Consumption of DOAS

The hourly energy consumption of the DOAS and the ambient temperature in the monitored 1 year time period is plotted in Fig. 15. The annual total electricity use of the DOAS system was 64,929 kWh.

As shown in Fig. 15, the hourly power consumption of the DOAS was nearly constant at about 10 kWh from September 2012 to March 2013. It decreased to almost zero in April and May, then increased beginning in June. The hourly power consumption of the DOAS was higher than 10 kWh during most of the time from June to August. The most frequent hourly power consumption value (10 kWh) is consistent with the total nominal power draw of the two 7.5 hp fans and the 0.25 hp motor of the enthalpy wheel (11 kW in total). This rate of power consumption suggests that the supply and exhaust fans and the enthalpy wheel may run continuously at full speed except in April and May. In summer time, when outdoor air was warm, the refrigeration unit of the DOAS was turned on to cool it down to 55°F, and the hourly power consumption went above 10 kWh. It is not clear to the authors yet why the DOAS power consumption was nearly zero in April and May.

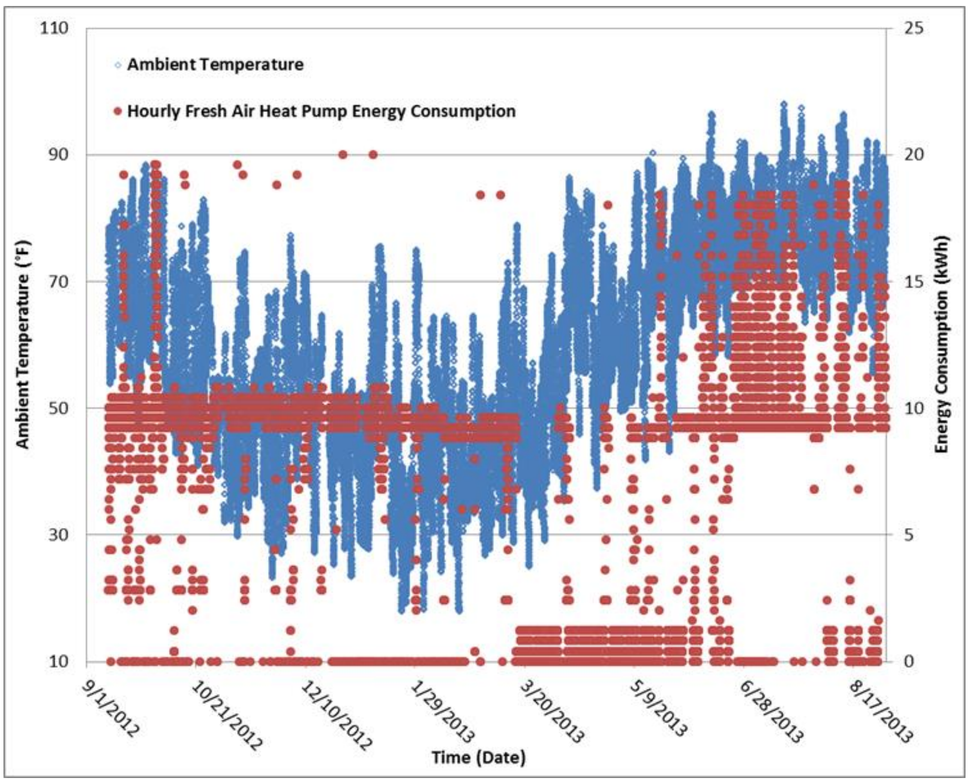

Fig. 15. Hourly energy consumption of the DOAS unit and the coincidental ambient temperatures.

We chose three typical weeks to demonstrate the detailed operation of the DOAS unit in different periods. The first week we picked is January 21–28. As Fig. 15 shows, from Sept. 1, 2013, to March 15, 2014, the hourly power consumption was nearly constant. The first week is a typical week in this period. The hourly energy consumption of the DOAS unit and the ambient temperatures during this week are shown in Fig. 16. As Fig. 16 shows, except for a short shut down of the unit at around 2:45 A.M. every day, the power consumption of the unit was maintained at an almost constant level. It is noticed that even when the ambient temperature dropped below 20°F, the hourly power consumption of the DOAS remained at about 10 kWh. Given the 70°F return air temperature and the typical heat transfer efficiency of the enthalpy wheel (76%), the OA temperature leaving the ERV at such ambient temperature would be below 40°F. It appears that the DOAS did not turn on the emergency electric heating to maintain the supply air temperature above 50°F as specified in the sequence of operation.

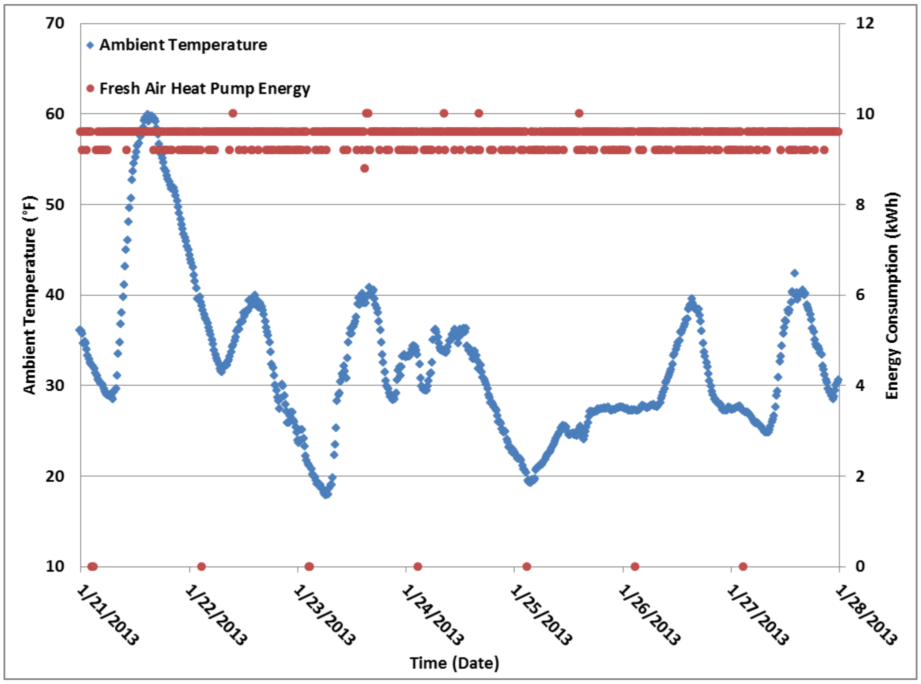

Fig. 16. Hourly DOAS energy consumption and ambient temperatures, Jan. 21–28, 2013.

The second week that we chose is April 15–22, 2013, which is a typical week in the "off" period from March through May. As Fig. 17 shows, the hourly energy consumption dropped to a low level, varying from 0 to 1.2 kWh. In addition, during this period, the operation of the DOAS unit appears to have been consistent with the occupied and unoccupied schedule of the building. Fig. 18 shows the hourly power consumption of the DOAS during a typical day (April 16, 2013). It demonstrates, in occupied condition (daytime), the power consumption was relatively high, from 0.8 to 1.2 kWh; but it decreased to 0.4 kWh in unoccupied condition (weekends, nighttime).

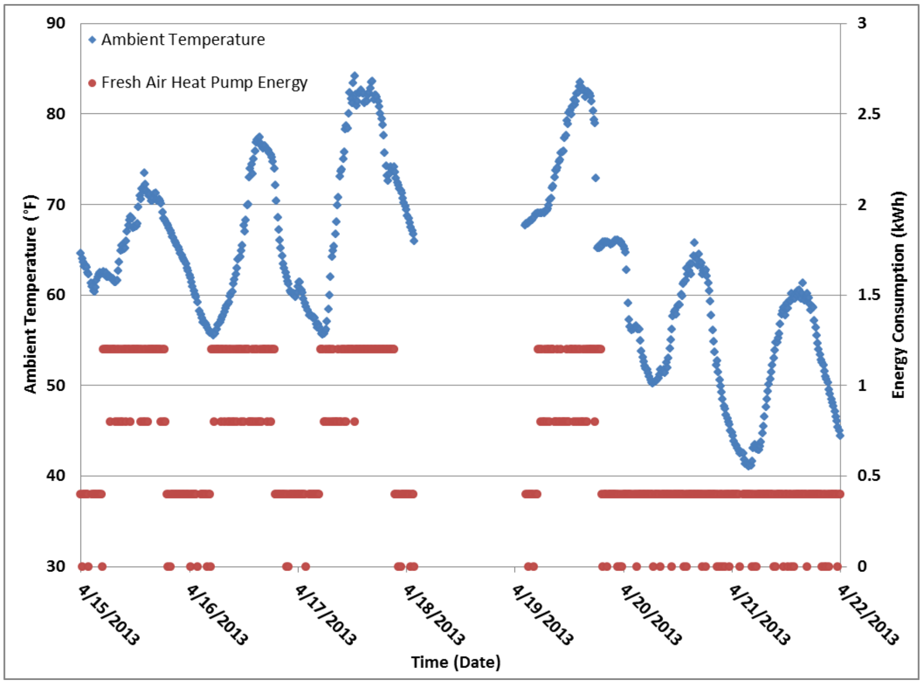

Fig. 17. Hourly DOAS energy consumption and ambient temperatures, Apr. 15–22, 2013.

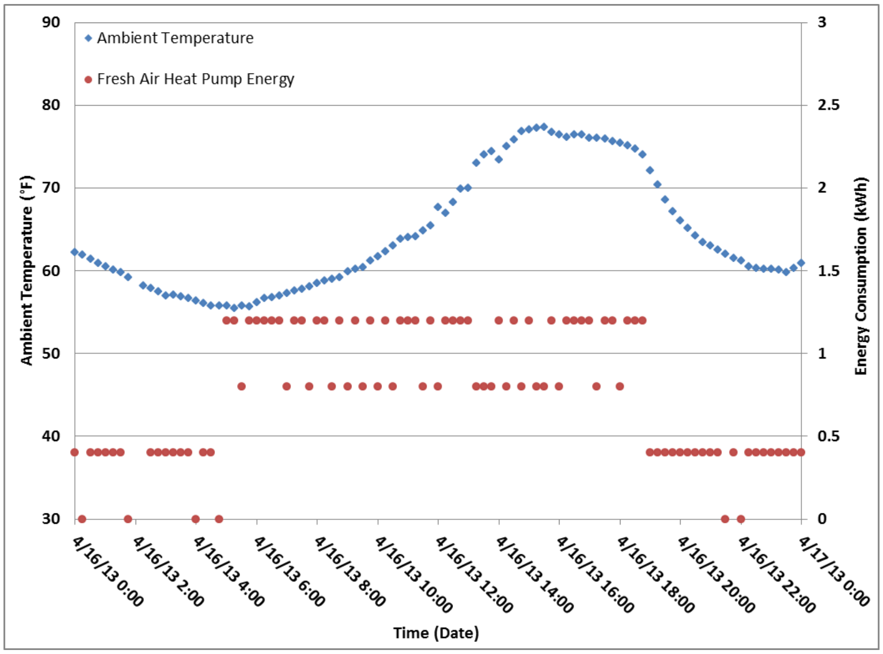

Fig. 18. Hourly DOAS energy consumption and ambient temperatures on Apr. 16. 2013.

The third typical week is July 1–7, 2013. Beginning in June, the power consumption of the DOAS unit returned to the high level and went even higher. As Fig. 19 shows, the hourly energy use of the DOAS was maintained at about 10 kWh level as shown in the first typical week. When the ambient temperature became high (usually higher than 75°F), the hourly power consumption of the DOAS unit increased to as high as 18 kWh due to the operation of the refrigeration unit in the DOAS. Just as shown in the first typical week, it is observed that the DOAS unit was only shut down for a very short time period (less than an hour) in every day.

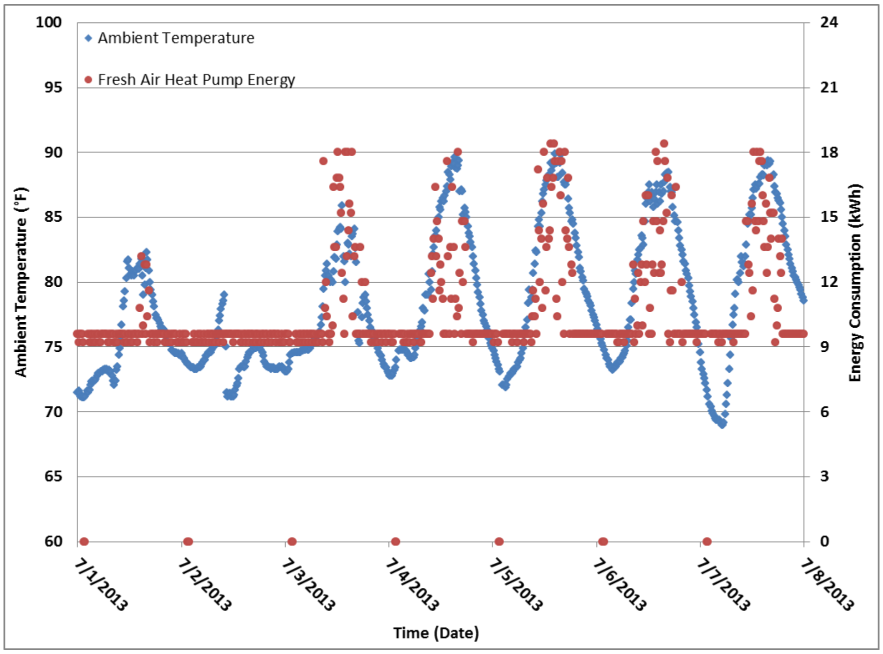

Fig. 19. Hourly DOAS energy consumption and ambient temperatures, July 1–7, 2013.

According to the sequence of operation, the OA supply to each individual zone should be adjusted based on the CO_2 level in that zone, and, in the meantime, to maintain the building slightly pressurized to prevent infiltration. However, it is uncertain whether the dampers of the VAV boxes of the DOAS operated properly since the hourly power consumption of the DOAS was very close to or higher than 10 kWh most of the time, which suggests the DOAS have been providing constant air flow to the building, even when the building was not occupied.

After being notified about issues with operation of the DOAS, the building owner changed a setting on its control in September 2013 so that the DOAS only operates when the building is occupied (from 5:00 A.M.to 6:00 P.M. during weekdays). We chose a typical week from July 14 to 21, 2014, to show the detailed operation of the DOAS unit operated with the new control strategy. As shown in Fig. 20, the DOAS was turned off during the unoccupied time period. When it was on during the occupied time, its fans ran at full speed (indicated by the nearly constant minimum power draws at about 10 kW). When the ambient temperature went up beyond about 75°F, the refrigeration unit of the DOAS was turned on, which led to higher power draws from the DOAS.

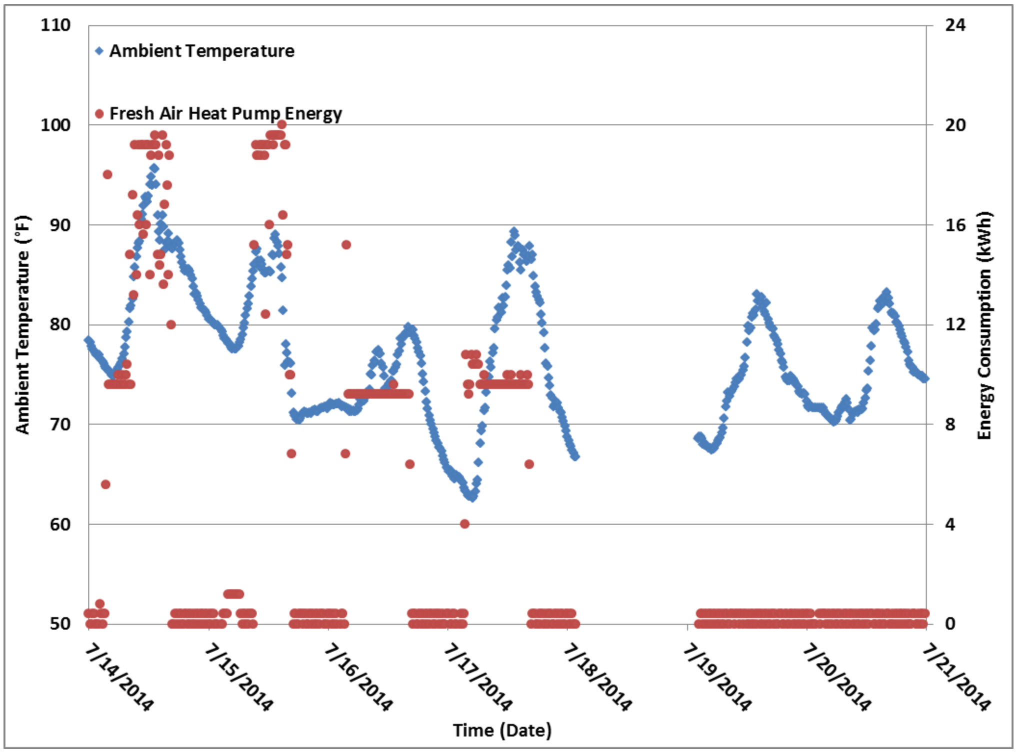

Fig. 20. Hourly DOAS power consumption and ambient temperatures, July 14–21, 2014.

Power consumption of the DOAS unit was reduced significantly after implementing the new control strategy. A comparison of the 1 year period from September 2013 through August 2014 with the year before shows that the annual power consumption of the DOAS was reduced by **37%,** from 64,929 to 40,648 kWh. The monthly DOAS power consumption in these two 1 year periods is compared in Fig. 21. After the new control strategy was implemented (the 2013–2014 data shown in the figure), the DOAS consumed much less power, except in April and May. As mentioned before, in April and May 2013, the DOAS unit also operated only during the occupied times, but with fans running at a much lower speed.

The annual power consumption of the entire GSHP system (including DOAS, DHW, and pumping) was also reduced after changing DOAS's control strategy. It was reduced by 20% from 140,078 kWh in year 2012–2013 to 112,212 kWh in year 2013–2014. The power use reduction of the DOAS accounts for 87% of the energy savings of the entire GSHP system; the rest is thought to be due to the relatively mild weather in the year 2013–2014. The average power consumption of the entire GSHP system within every 1°F bin of the ambient temperature during the 2 years is shown in Fig. 22. The GSHP system consumed less power in the year 2013–2014 than in the previous year at most ambient temperatures; the exception was when the ambient temperature was higher than 90°F. At these high temperatures, the control strategy of the DOAS does not make much difference. It is likely that when the ambient temperature is higher than 90°F, the energy saved by shutting down the DOAS during the unoccupied time (night and weekends) is offset by the increased power consumption of the heat pump units during the occupied time (to cool down the building at the beginning of weekdays).

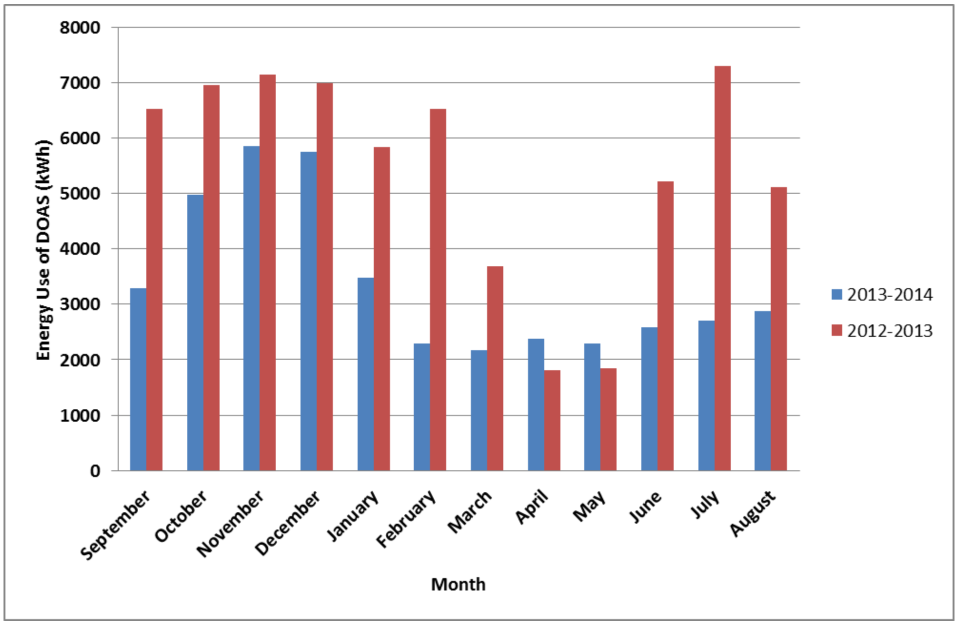

Fig. 21. Comparison of the monthly power consumption of the DOAS in 2013–2014 and 2012–2013.

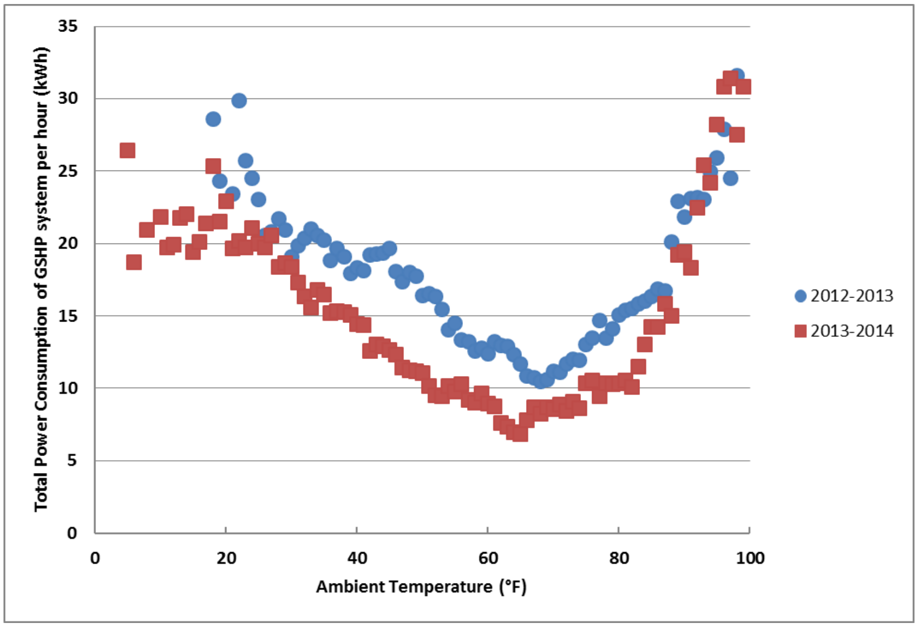

Fig. 22. Comparison of the average GSHP system energy use at various ambient temperatures.

2.2.2 Power Consumption of the 27 WAHP Units

Figure 23 shows the annual cumulative power consumption of each of the 27 WAHP units in heating and cooling modes, respectively, during the year, along with the cooling capacity of each WAHP unit. Although a few WAHP units operated mostly in cooling mode (e.g., unit 20), most of the WAHP units ran predominantly in heating mode. The combined annual power consumption of the 27 WAHP units is 15,171 kWh for cooling and 26,667 kWh for heating. This is not surprising since the nearly constant (8,010 ft^3/min) OA supplied by the DOAS provided a significant amount of cooling to the building. It thus reduced the cooling loads but increased the heating loads of the WAHP units.

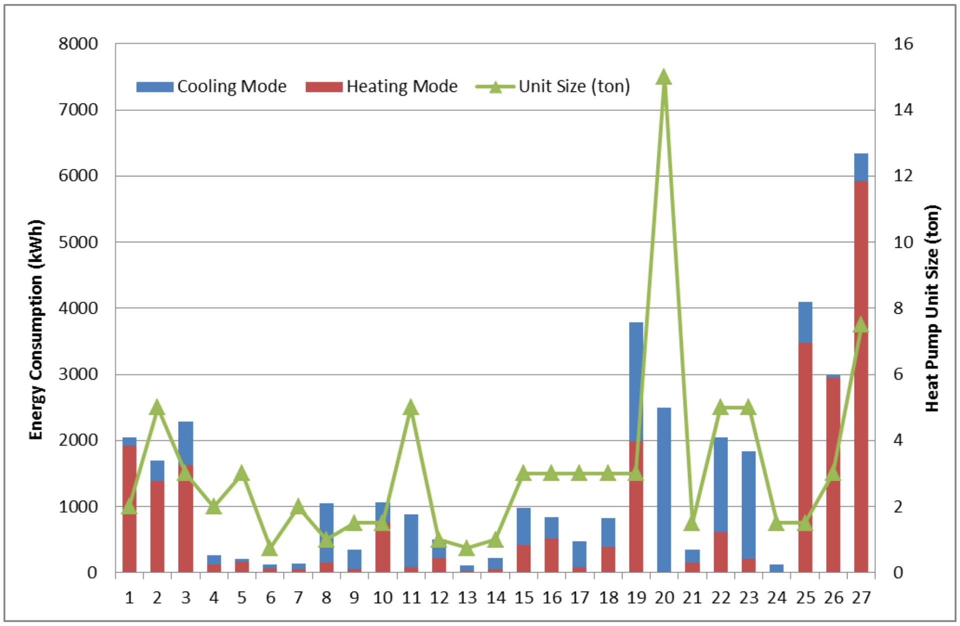

Fig. 23. Annual total electricity use of the 27 WAHP units for space conditioning.

Power consumption data indicate that a few WAHP units ran in heating mode during the cooling season, and some WAHP units ran in cooling mode in the heating season. As shown in Fig. 24, WAHP units 8, 15, and 20, which condition an inner zone (as shown in Fig. 26, with Zone 1 corresponding to WAHP unit 1; Zone 2 corresponding to WAHP unit 2; etc.), ran in cooling mode most of the time in the heating season, while WAHP units 25, 26, and 27, which serve peripheral zones (as shown in Fig. 26), ran only in heating mode during the same time period. This demonstrates a unique feature of the distributed GSHP system—it can satisfy varied heating and cooling demands in individual zones.

On the other hand, as shown in Fig. 25, WAHP units 15, 16, 17, and 18 used a significant amount of electricity for heating in the cooling season. The HVAC zoning plan (Fig. 26) shows that the zones conditioned by WAHP units 15, 16, 17, and 18 are surrounded by the zone conditioned by WAHP unit 19, which has the highest electricity use for cooling in the summer. It's very likely that WAHP unit 19 overcooled the inner zones served by WAHP units 15, 16, 17, and 18, and, as a result, triggered those units to run in heating mode to warm up the inner zones. It is possible that this conflicting simultaneous heating and cooling operation could be eliminated by adjusting the location and capacity of the WAHP units.

Although the distributed GSHP system can recover some energy through the common loop when various GSHP units run in heating and cooling mode simultaneously, it consumes power to run the compressors of different WAHP units. Proper zoning and partition among zones are desirable to avoid the conflicting simultaneous heating and cooling, and thus reduce energy consumption of the distributed GSHP system.

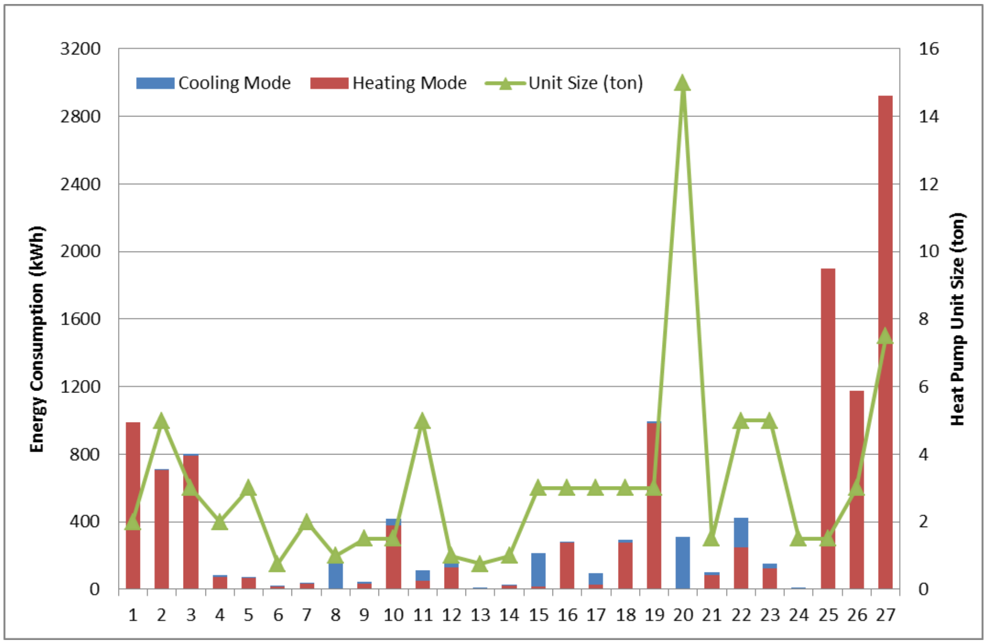

Fig. 24. Total electricity use of the 27 WSHP units in the heating season (December-February).

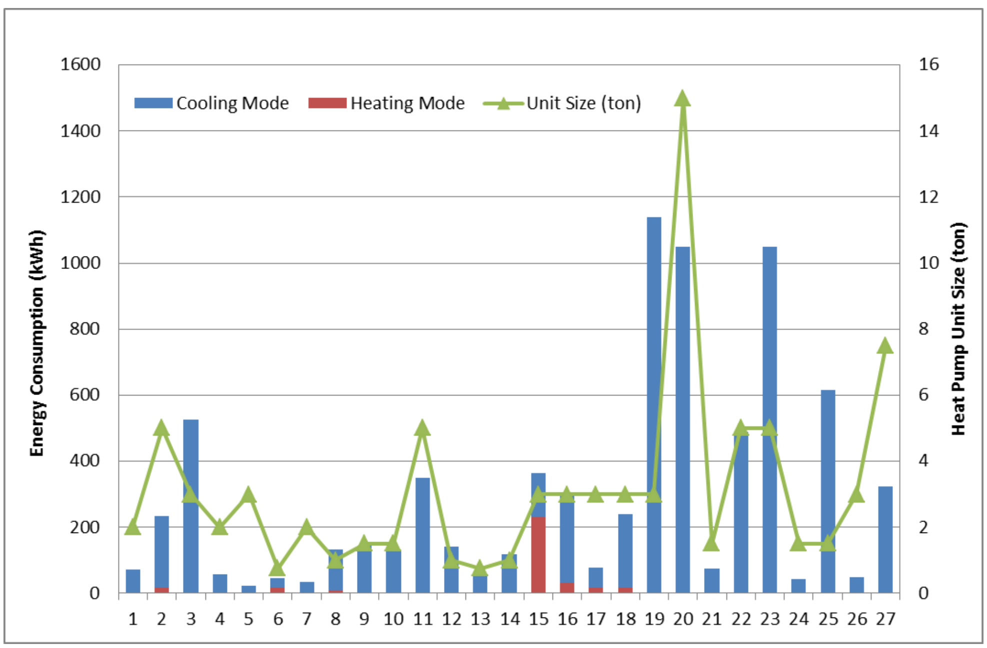

Fig. 25. Total electricity use of the 27 WSHP units in the cooling season (June-August).

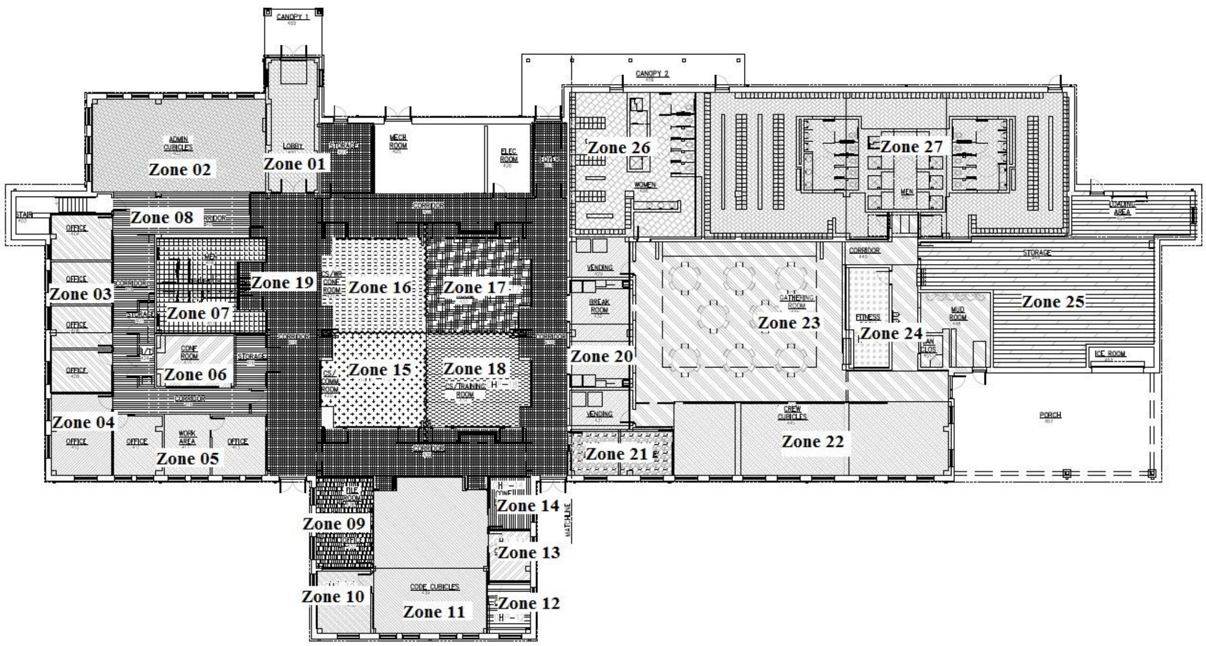

Fig. 26. The 27 zones associated with the 27 WAHP units.

2.2.3 Power Consumption of other Components or Subsystems

The power consumption of other components or subsystems of the GSHP system was measured and is available for this study. The energy consumed from September 2012 through August 2013 by ground loop pumping (20,456 kWh), DHW loop pumping (1,433 kWh), and the DHW heat pump unit (8,152 kWh) was calculated using the available data.

2.2.4 Percentage of Power Consumption by Each End Use

Using the power consumption data for each component or subsystem of the GSHP system described above, we calculated the percentage of the annual power consumption used by each major component or subsystem (Fig. 27). Power consumption by the 27 WAHP units in heating and cooling modes is accounted for separately in this figure. As shown in Fig. 27, the 27 WAHP units contribute 18.8% for space heating and 11.8% for space cooling to the total power consumption; the WWHP (for DHW) accounts for 6.5% of total power consumption; and the DOAS unit consumes 45.4% of the total power consumption. The two pumping systems—the ground loop pump and the DHW pump—consume 16.4% and 1.1% of the total power consumption, respectively. As Fig. 27 shows, the DOAS unit accounts for the largest portion of the total power consumption in the entire GSHP system from September 2012 through August 2013, which is due to its continuous operation as discussed before.

A new control strategy was implemented in September 2013, and the DOAS unit operates only when the building is occupied. The percentage of the entire GSHP system power consumption from each major component or subsystem from September 2013 to August 2014 is shown in Fig. 28. The percentage of DOAS power consumption decreased significantly after the new control strategy was implemented, from 45.4% to 37%. All the percentages of the other components and subsystems increased as expected, except for the ground loop pumping. The total power consumption from ground loop pumping decreased by 30% during the period from September 2013 to August 2014, compared with that in previous year (from September 2012 to August 2013). However, as shown in Fig. 29, there was no significant change in the ground loop flow rate during the last two years. The decrease in power consumption was likely due to lowering of the pressure differential set point after August 2013.

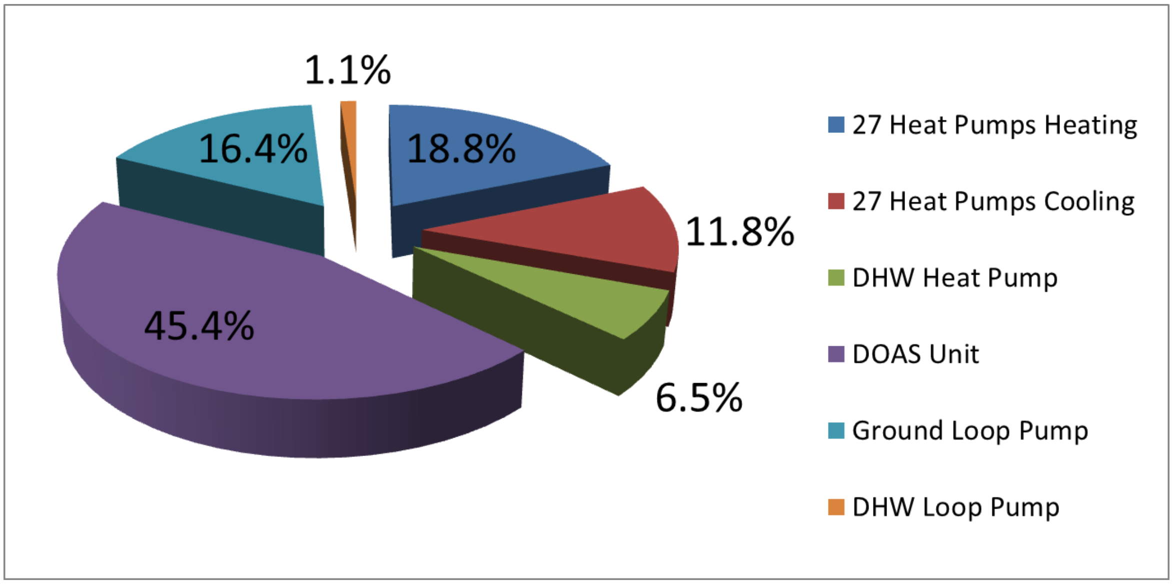

Fig. 27. Percentage of power used by major components and subsystems in the GSHP system from September 2012 through August 2013.

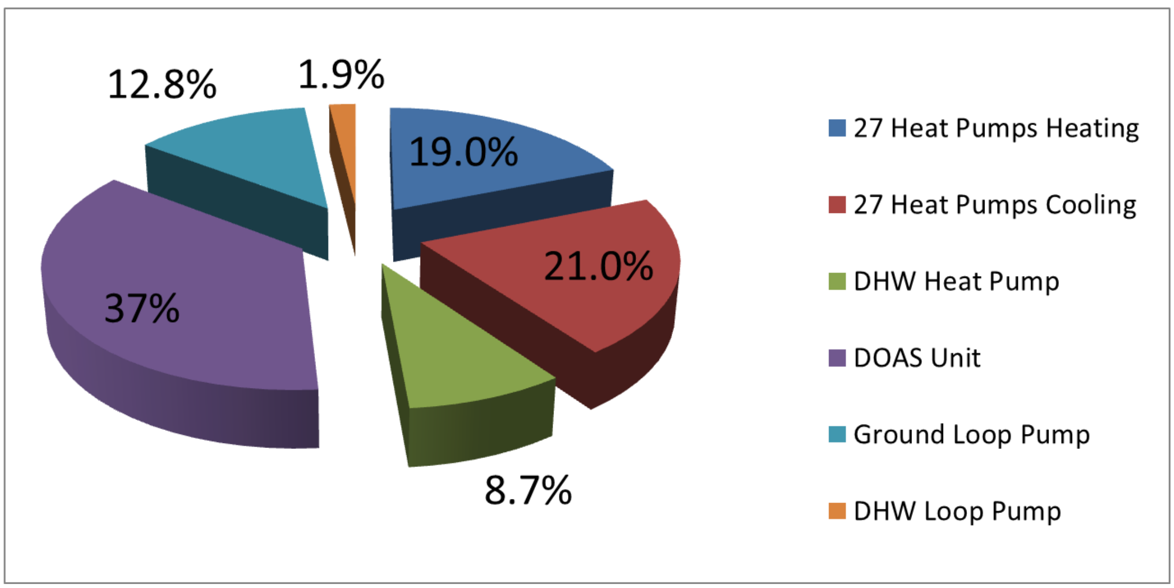

Fig. 28. Percentage of power used by major components and subsystems in the GSHP system from September 2013 through August 2014.

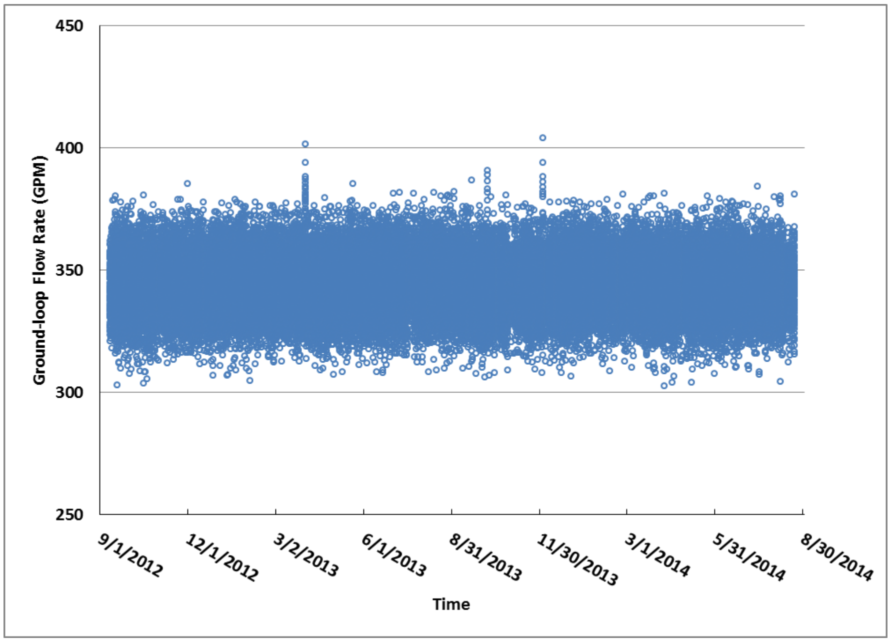

Fig. 29. Measured ground loop flow rate from September 2012 to August 2014.

2.3 OUTPUTS AND ENERGY EFFICIENCY OF WSHP UNITS AND DOAS

The heating and cooling outputs of each WAHP unit were not measured; instead, they are calculated in this study by multiplying the measured power consumption of each WAHP unit and the estimated coefficient of performance (COP) for heating mode or energy efficiency ratio (EER) for cooling mode at each time interval. Since the entering air temperature, the entering air flow rate, and the entering fluid flow rate of each WAHP unit varied within a narrow range and thus were considered as fixed values, the COPs or EERs of these WAHP units are treated as functions of the entering fluid temperature only, as expressed by Eqs. (2) and (3):

$$COP = a * EWT^2 + b * EWT + c \quad (2)$$

$$EER = a * EWT^2 + b * EWT + c \quad (3)$$

where

EWT is the heat pump entering fluid temperature, in °F;

a, b, and c are curve-fit coefficients for the correlations between COP/EER and EWT derived from the catalog data of each individual WAHP unit.

Based on the measured heat pump entering fluid temperature (ground loop supply fluid temperature) and the correlations described above for each individual WAHP unit, the COP or EER of the 27 WAHP units is calculated at every 15 min time interval. Then, the heating and cooling outputs of each WAHP unit are calculated by multiplying the measured power consumption of each WAHP unit in heating or cooling mode with the calculated COP or EER values at the same time.

The heating and cooling outputs of all the 27 WAHP units combined at each 15 min interval from Sept. 1, 2012, to Sept. 1, 2103, is shown in Fig. 30. As can be seen in this figure, the maximum heating output was 778 kBtu/h, and the maximum cooling output was 658 kBtu/h. Both values are less than the installed capacities of the 27 WAHP units, which have a combined capacity of 84 tons (1,008 kBtu/h). The annual cumulative heating output of the 27 WAHP units is 1.4 times higher than the annual cumulative cooling output during the full year from September 2012 through August 2013. This is not a surprise because the conditioned OA continuously supplied by the DOAS offset some cooling loads, but increased heating loads for the WAHP units.

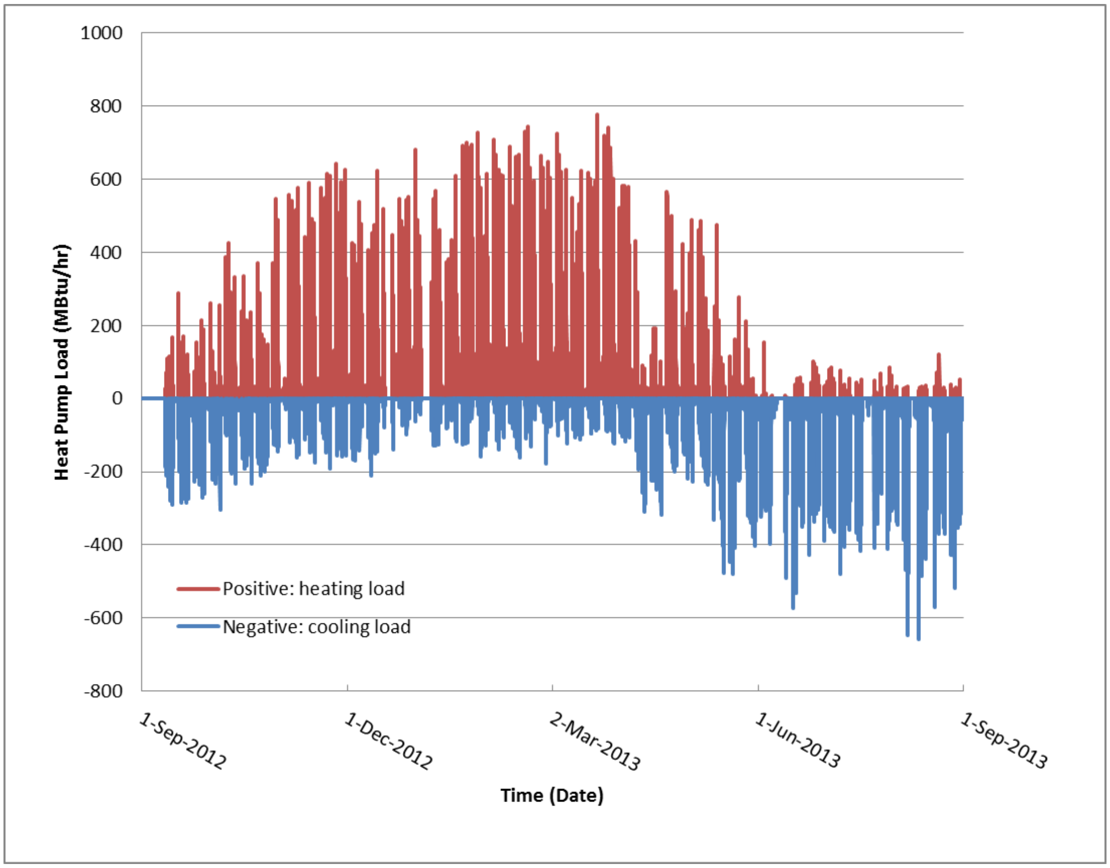

Fig. 30. Cooling and heating output of the 27 WAHP units at each 15 min interval from September 1, 2012, through September 1, 2013.

With the heating and cooling output and the associated power consumption of all the 27 WAHP units, the combined COP or EER for the 27 WAHP units can be estimated with Eqs. (4) and (5); they are shown in Figs. 31 and 32, respectively. The annual average of the combined COP and EER for the 27 WAHP units is 4.9 and 18.5, respectively.

$$COP\_HP = \frac{\sum_{i=1}^{27} QH\_HP(i)}{\sum_{i=1}^{27} PH\_HP(i)} \tag{4}$$

$$EER\_HP = \frac{\sum_{i=1}^{27} QC\_HP(i)}{\sum_{i=1}^{27} PC\_HP(i)} \tag{5}$$

where

COP_HP, *EER_HP* are the combined COP and EER for the 27 WAHP units;
QH_HP(i), *QC_HP(i)* are the i^{th} WAHP unit's heating and cooling output, respectively;

PH_HP(i), PC_HP(i) are the i[th] WAHP unit's power consumption for heating and cooling, respectively;

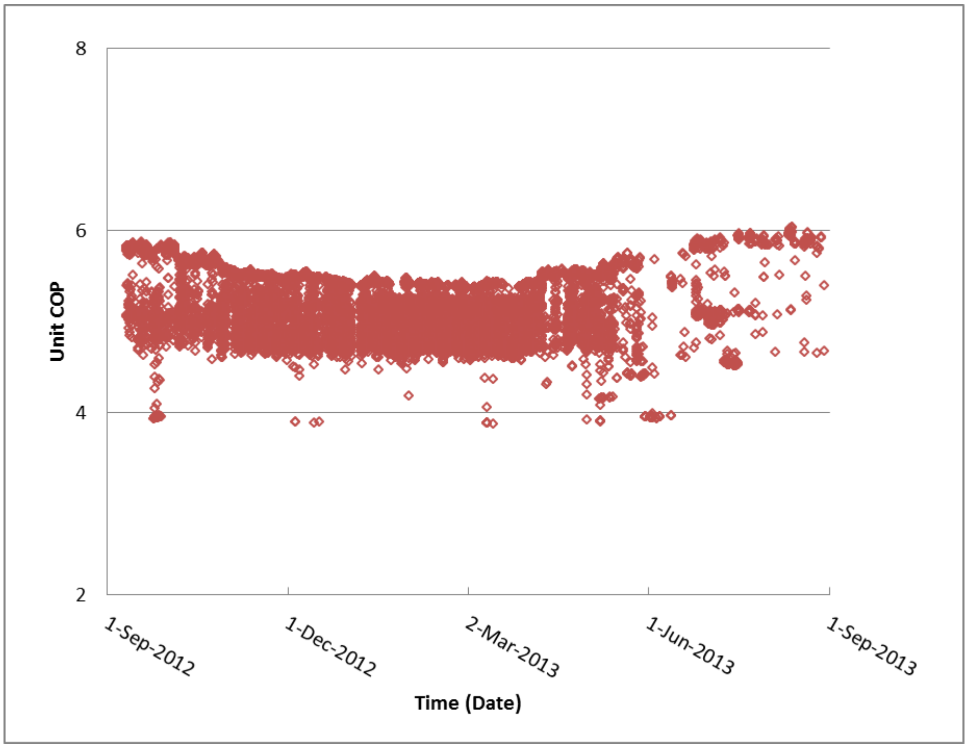

Fig. 31. Combined COP for the 27 WAHP units from September 2012 to August 2013.

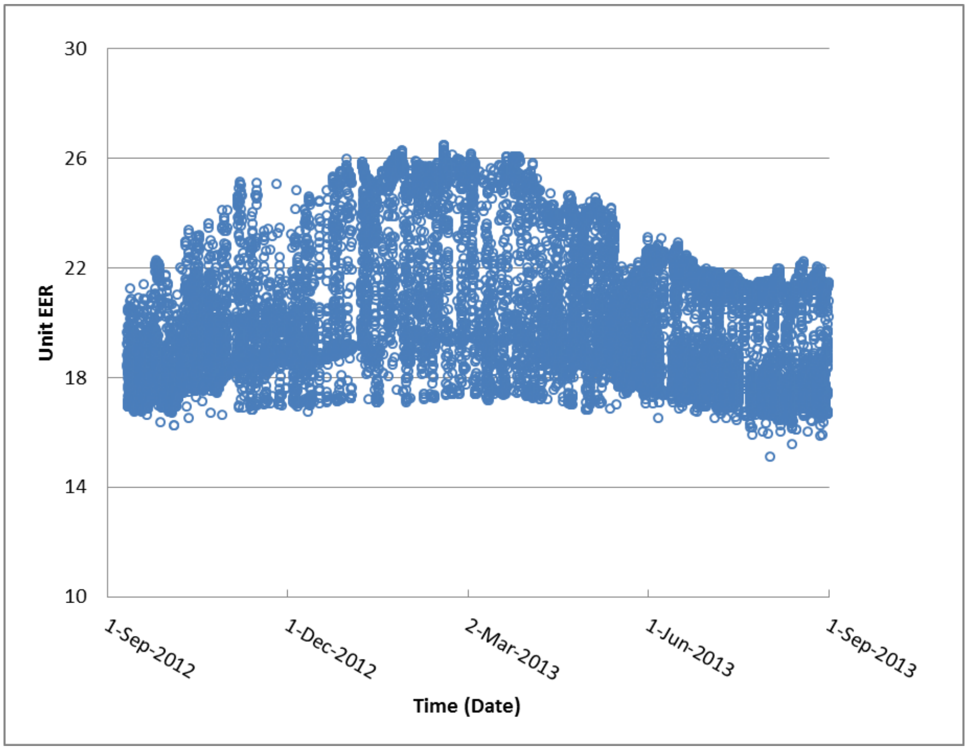

Fig. 32. Combined EER for the 27 WAHP units from September 2012 to August 2013.

The cooling and heating outputs of the DOAS are calculated based on the operation sequence of the DOAS and the ambient air conditions. As discussed in Sect. 1.2, when the DPT of OA leaving the DOAS is higher than 55°F, the DOAS turns on its refrigerant unit to cool the OA to 55°F. At other times when the DOAS is on, only the enthalpy wheel operates to pre-heat or pre-cool the OA by exchanging heat and moisture with the air exhausting from the building. Because there is not any measurement for the OA temperature leaving the DOAS, the following assumptions are made to calculate the DOAS heating and cooling outputs (including both the contributions from the enthalpy wheel and the refrigerant unit when it is turned on):

- The exhaust air (EA) temperature is 70°F with a 60% relative humidity.
- There is not any additional heating provided by the DOAS except through the enthalpy wheel (as discussed in Sect. 2.2.1, Fig. 16).
- The temperature and humidity of the OA leaving the enthalpy wheel are the average of the temperatures of the OA and the EA when they enter the enthalpy wheel.
- In addition to the sensible heat transferred from the EA to the OA, the latent cooling for dehumidifying the OA is also accounted for (since it otherwise will become the latent load for the WAHP).
- When the refrigerant unit is on, it cools and dehumidifies the OA to be saturated at 55°F.

The following procedures are followed to calculate the heating and cooling provided by the DOAS:

- If the OA dry bulb temperature (DBT) is equal to or lower than 55°F, it will exchange heat and moisture with the exhaust air through the enthalpy wheel. The sensible heat provided by the ERV (Q_{sh}) is calculated with Eq. (6), where T_{OA_Out} is the OA temperature leaving the enthalpy wheel.
- If the OA DBT is higher than 55°F, then:
  - If the OA DPT is equal to or lower than 55°F, it will exchange heat and moisture with the EA through the enthalpy wheel. The sensible heat (Q_{sh}), the sensible cooling (Q_{sc}), and the latent cooling (Q_{lc}) provided by the enthalpy wheel are calculated with Eqs. (6), (7), and (8), respectively, where T_{OA_Out} and W_{OA_Out} are the conditions of the OA leaving the enthalpy wheel.
  - If the OA DPT is higher than 55°F, it will exchange heat and moisture with the EA through the enthalpy wheel and then be further cooled and dehumidified by the refrigerant unit. The sensible and latent cooling provided by both the enthalpy wheel and the refrigerant unit are calculated with Eqs. (6), (7), and (8), respectively, where T_{OA_Out} and W_{OA_Out} are the conditions of the OA leaving the refrigerant unit.

When the fans of the DOAS run at full speed to supply the design air flow rate (8,010 ft^3/min), the power consumption of the DOAS is around 10 kWh. Therefore, when the measured hourly power consumption is larger than 10 kWh, the fans are considered to be running at full speed, and the design air flow rate is used in the calculation. If the power consumption is less than 10 kWh, the fans are considered to be operating at partial speed, and the air flow rate is adjusted using Eq. (9). The entering OA condition (T_{OA_In} and W_{OA_In}) is obtained from the weather data measured at a nearby airport (WBT 2014). The calculated cooling outputs of the DOAS during the year from September 2012 through August 2013 are shown in Fig. 33.

$$Q_{sh} = q\rho c_p(T_{OA\_In} - T_{OA\_Out}) \tag{6}$$

$$Q_{sc} = q\rho c_p(T_{OA\_Out} - T_{OA\_In}) \tag{7}$$

$$Q_{lc} = q\rho h_{fg}(W_{OA\_Out} - W_{OA\_In}) \tag{8}$$

$$q = q_{rate}\sqrt[3]{\frac{P}{P_{rate}}} \tag{9}$$

where

Q_{sh} is the sensible heat, in Btu/h;
Q_{sc} is the sensible cooling, in Btu/h;
Q_{lc} is the latent cooling, in Btu/h;
c_p is air specific heat, in Btu/lb·°F = 0.2388 Btu/lb·°F;
ρ is air density at standard conditions = 0.075lb/ft^3;
q is the estimated air flow rate, in ft^3/min;
q_{rate} is the design air flow rate, in ft^3/min;
h_{fg} is the latent heat of vaporization of water, in Btu/lb =1060 Btu/lb;
P is the measured power consumption of the fan, in kWh/h;
P_{rate} is the rated power consumption of the fan, in kWh/h.

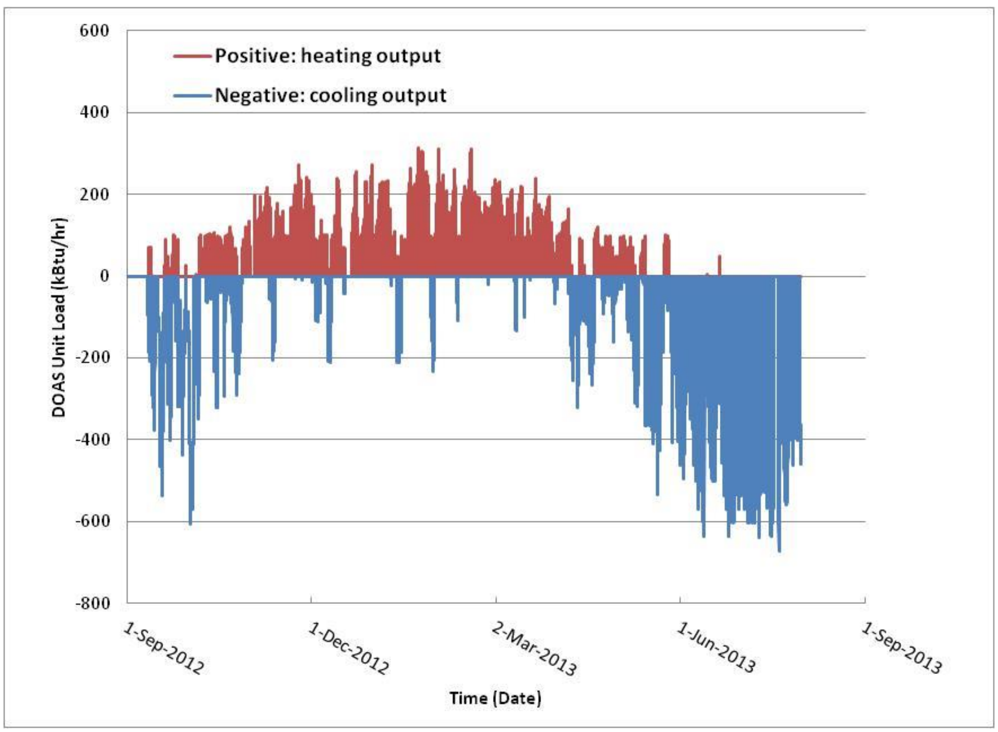

Fig. 33. Cooling and heating outputs of the DOAS from September 2012 through August 2013.

Based on the calculated cooling and heating outputs and the measured power consumption of the DOAS, the effective COP of the DOAS is calculated with Eq. (10) at each time interval. The calculated annual average COP of the DOAS is 6.6, which is lower than the catalog data provided by the manufacturer (COP=7.4). The difference is thought to be due to the continuous operation of the supply and return fans of the DOAS (even when the temperature differential between OA and EA are small).

$$COP = \frac{Q\_DOAS}{P\_DOAS} \tag{10}$$

where

COP is the coefficient of performance for the DOAS;
Q_DOAS is the annual cumulative output of the DOAS, including Q_{sh}, Q_{sc}, and Q_{lc};
P_DOAS is the annual cumulative power consumption of the DOAS.

2.4 ENERGY EFFICIENCY OF THE DEMONSTRATED GSHP SYSTEM

Based on the measured and calculated data discussed in the previous sections, the energy efficiency of the GSHP system can be evaluated. The annual total heating and cooling output of the entire GSHP system is the summation of the heating and/or cooling outputs of the DOAS, the 27 WAHP units, and the WWHP. Figure 34 shows the percentages of the heating and cooling outputs of each subsystem of the GSHP system. As shown in Fig. 34, the DOAS has largest cooling output for conditioning the OA. The monthly cooling and heating outputs of the entire GSHP system are shown in Fig. 35.

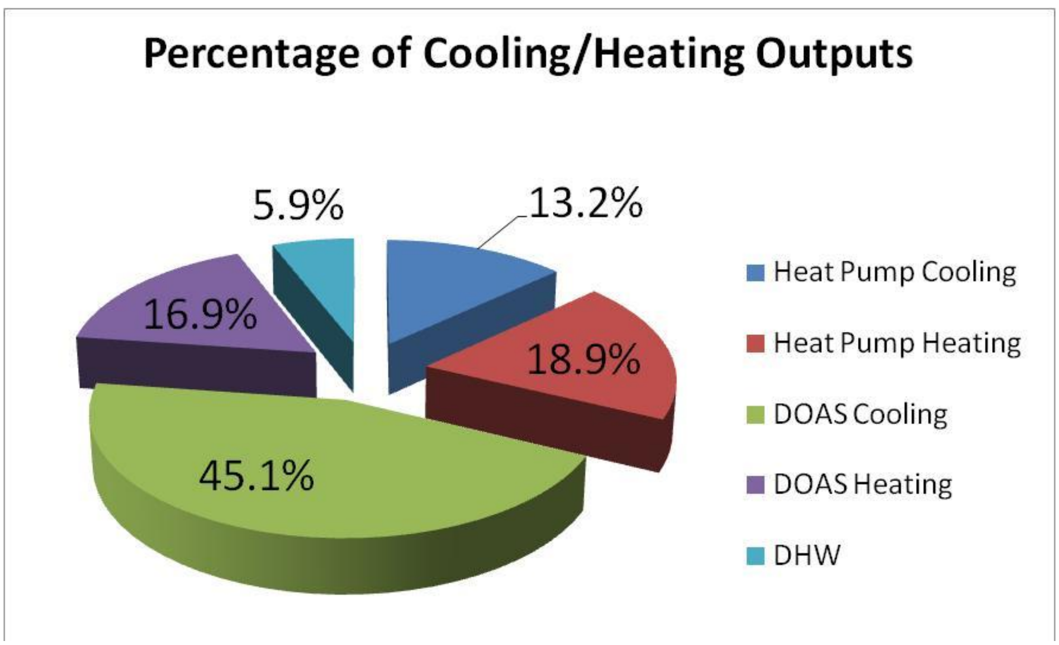

Fig. 34. Contributions from subsystems to the total system heating and cooling outputs.

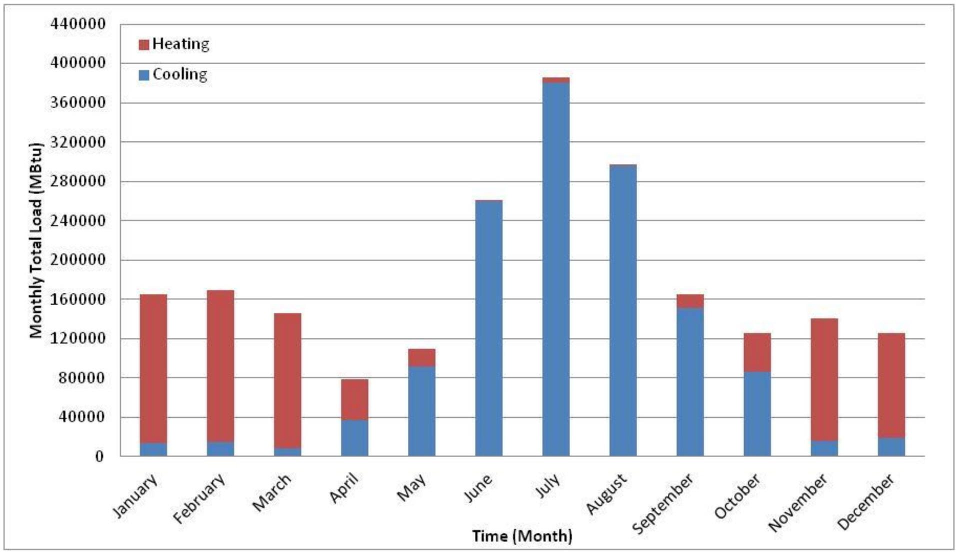

Fig. 35. Monthly distribution of the total system cooling and heating outputs.

Two terms are used to evaluate the energy efficiency of the demonstrated GSHP system: system cooling efficiency (evaluated with EER), system heating efficiency (evaluated with COP). Since each individual WAHP unit could run in heating or cooling mode at any given time, the pumping power associated with heating and cooling operations at each time interval is separated based on the ratio between the needed water flow for WAHP units running in cooling and heating modes, respectively, at each time interval. The total amount of water flow needed by WAHP units running in heating and cooling modes, respectively, is estimated with the rated water flow rate for each WAHP unit and its operation status. The measured power consumption of the DOAS is also divided into two parts—heating and cooling—separately based on the ratio of the calculated heating and cooling outputs of the DOAS. The system COP and EER are calculated using the following equations.

$$COP_{sys} = \frac{QH_{DOAS} + QH_{DHW} + QH_{WAHP}}{3.41(WH_{DOAS} + WH_{DHW} + WH_{WAHP} + WH_{GLP} + W_{DHWP})} \tag{11}$$

$$EER_{sys} = \frac{QC_{DOAS} + QC_{WAHP}}{WC_{DOAS} + WC_{WAHP} + WC_{GLP}} \tag{12}$$

where

- QH is the annual heating output of a subsystem (DOAS, DHW, or WAHP), in kBtu;
- QC is the annual cooling output of a subsystem (DOAS, DHW, or WAHP), in kBtu;
- WH is the annual power consumption for heating by a subsystem (DOAS, DHW, WAHP, ground loop pump, or the DHW loop pump), in kWh;
- WC is the annual power consumption for cooling by a subsystem (DOAS, WAHP, or ground loop pump), in kWh.

The calculated annual average COP and EER of the GSHP system are 4.5 and 17.5, respectively. These system efficiencies are only slightly lower than the annual average COP and EER of the combined 27 WAHP units (4.9 COP and 18.5 EER as shown in Figs. 31 and 32). It appears that the high efficiency of the DOAS (an overall effective COP of 6.6) offsets the impact of the pumping power on the system efficiencies.

3. ANNUAL ENERGY ANALYSIS

We conducted simulations with a calibrated computer model of the Wilder Grove facility to predict the performance of a baseline HVAC system for providing the same service as the GSHP system. The development of the computer model, its calibration against available measured data, and the baseline HVAC system are discussed in the following sections, followed by a comparison of the performance of the two systems for a full year.

3.1 BUILDING ENERGY MODEL

A computer model of the Wilder Grove facility was created using eQUEST, a widely used building energy simulation program. eQUEST is powered by the latest development of the DOE-2 program and has improved capability to simulate various GSHP systems (Liu and Göran 2008).

Figure 36 shows side and top views of the real Solid Waste Services Administration Building, and Fig. 37 shows a 3-dimensional rendering of the modeled building. The building geometry and thermal zoning were specified in the computer model based on the as-built drawings. Parameters of the demonstrated GSHP system (e.g., the location, capacity, and efficiency of each WAHP unit; the size and layout of the ground heat exchanger; the pumping configuration and associated control; as well as the design flow rate of the OA ventilation), and some building parameters (e.g., the construction of the roof, windows, and doors; the load intensities and the operation schedules for lighting, equipment, and occupancy) were obtained from the as-built design documents and specified in the computer model. The remaining building and HVAC related parameters were set to typical values originally and fine-tuned during the calibration process later.

Fig. 36. Photographs of the Solid Waste Services Administration Building.

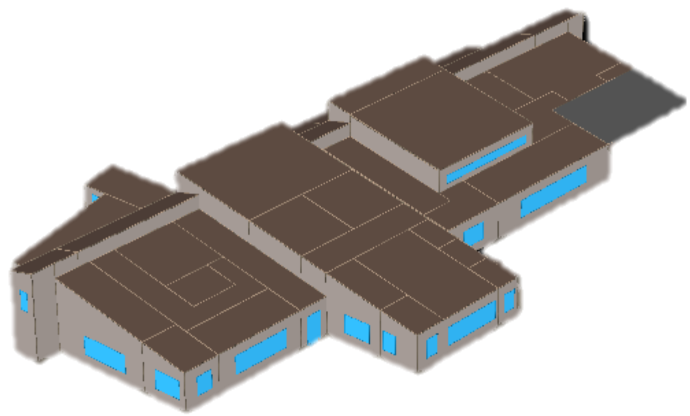

Fig. 37. Three-dimensional rendering of the simulated building.

As discussed in previous sections, the variable speed pump in the demonstrated GSHP system provides nearly constant water flow circulation; it appears independent to the operation of heat pump units. It is simulated in the computer model as a variable speed pump but without a two-way solenoid valve installed at any of the heat pump units. Therefore, it provides constant flow circulation in the water loop in the simulated GSHP system. Simulation with a normal variable-speed pump operation is performed later to evaluate the effects of pumping control on the operational efficiency and energy consumption of the GSHP system.

DOAS is modeled as a ground-coupled, cooling-only WAHP unit equipped with an enthalpy wheel. As with the real DOAS, there is no heating capacity in the simulated DOAS except the enthalpy wheel. Due to limitations of the current version of eQUEST, the simulated DOAS provides OA supply at a constant temperature and a constant flow rate. The OA supply temperature and the operation schedule of the DOAS are later adjusted in the calibration process to make the simulation-predicted building heating and cooling loads match the measured data.

Though DHW is provided by the WWHP in the demonstrated GSHP system, it is not simulated in the computer model because of the lack of information about the flow rate and temperature of the DHW system, and its relatively small contribution (less than 10% as shown in Fig 27) to the total power consumption of the entire GSHP system.

3.2 MODEL CALIBRATION

A two-step procedure is used in this study to calibrate the computer model. The first step is to ensure that the modeled building has the same demands for space heating and cooling as the real building. These demands include not only the heat removed or added to offset the heat gains or losses in the building to maintain room temperatures at their set points, but also the heating and cooling needed to condition the OA. Therefore, the combined heating and cooling outputs of the 27 WAHP units (for space conditioning) and the DOAS (for OA conditioning) described in previous sections are used as the target for the calibration. The second step is to ensure that the model-predicted power consumption of the GSHP system matches the measured data. Parameters of the computer model, which are not from direct measurements and thus have relatively high uncertainties, are adjusted during the calibration process to make the model-predicted results match the targets.

As previously stated, 12% of the data entries were missing during the 1 year period from September 2012 through August 2013. The exact reason for the missing data is unknown. It may be due to temporary power outages or some issues in data acquisition and/or transfer. To fill these missing data, the four parameter (4P) change-point model of the Inverse Modeling Tool (IMT) (Kissock et al. 2002) was used. Figure 38 shows the best-fit 4P models for the heating and cooling output as a function of OA temperatures based on the available measured data. With these correlations and the OA temperatures measured at a nearby airport, the missed data of the combined heating and cooling outputs are calculated. The resulting full-year cumulative heating and cooling outputs are 15% and 7%, respectively, higher than the sum of the available data during the year. The total of the full-year cumulative heating and cooling outputs is 12% higher than that calculated with only the available data.

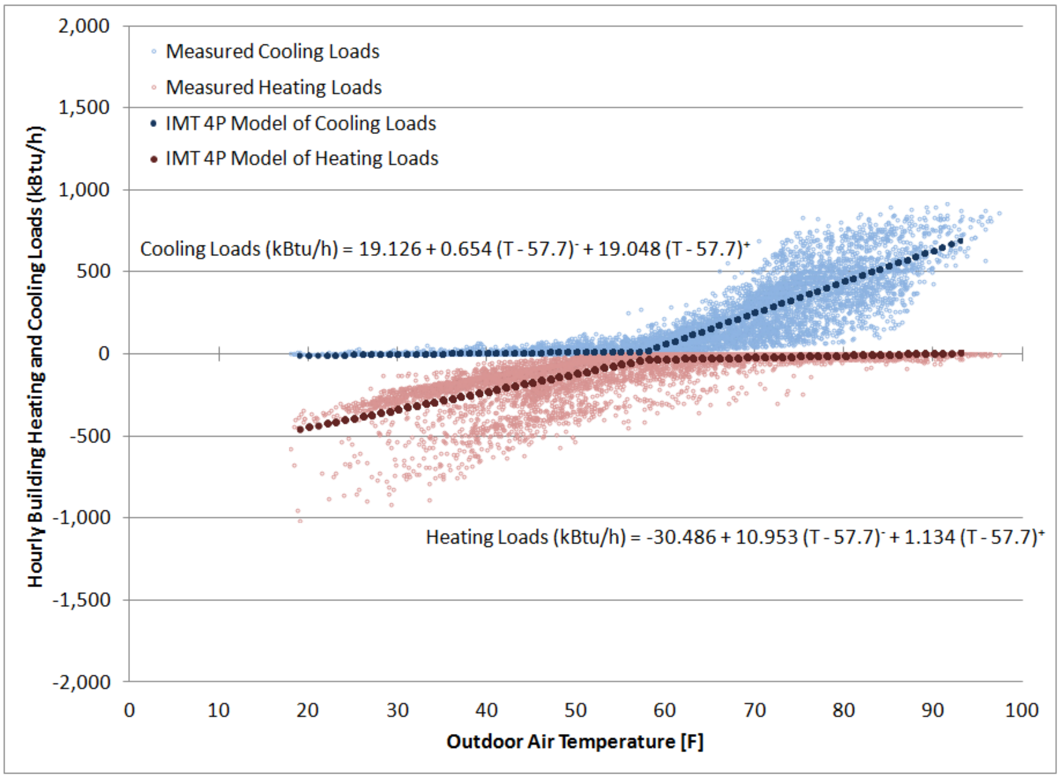

Fig. 38. Correlations between ambient temperatures and the heating and cooling outputs of the GSHP system.

The computer model was run with the actual weather data for years 2012 and 2013, respectively, and the simulation results for the two years were combined to have a data set in line with the measurement period. In the first step of the calibration process, a few parameters were adjusted, including room temperature set points and schedules, heat pump night cycle control, OA flow rate, building envelope constructions, window areas, and power density and schedules of internal heat gains, With these adjustments, the model-predicted monthly heating and cooling outputs of the GSHP system matched the calibration data (the extended full-year heating and cooling outputs) reasonably well, with a coefficient of variation of the root mean square error (CVRMS) of 10% and a normalized mean bias error (NMBE) of 4.6%. This calibration result meets the compliance requirements of ASHRAE Guideline 14 (ASHRAE 2002), which requires that "the computer model shall have an NMBE of 5% and a CVRMS of 15% relative to monthly calibration data."

As shown in Figs. 39 and 40, the model-predicted monthly building heating and cooling outputs followed the trends of the measured data closely. The model-predicted annual heating and cooling output is only 4.2% higher than that of the extended full-year data.

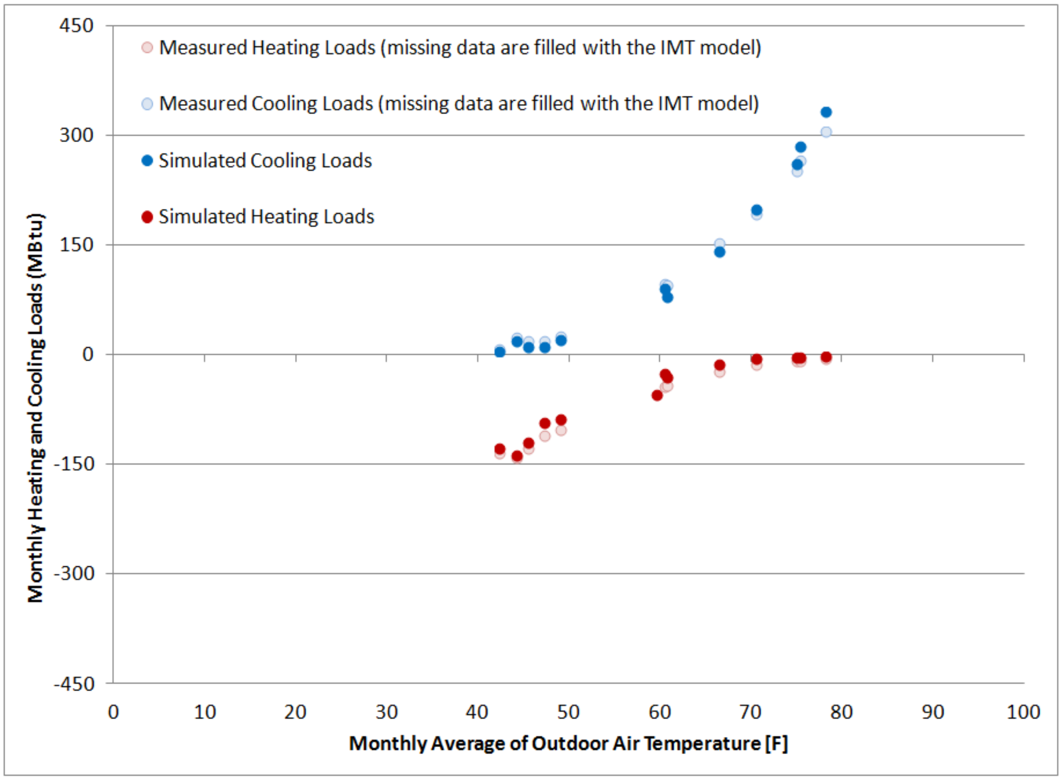

Fig. 39. Scatter plots comparing measured and model-predicted monthly heating and cooling outputs against monthly average of outdoor air temperatures.

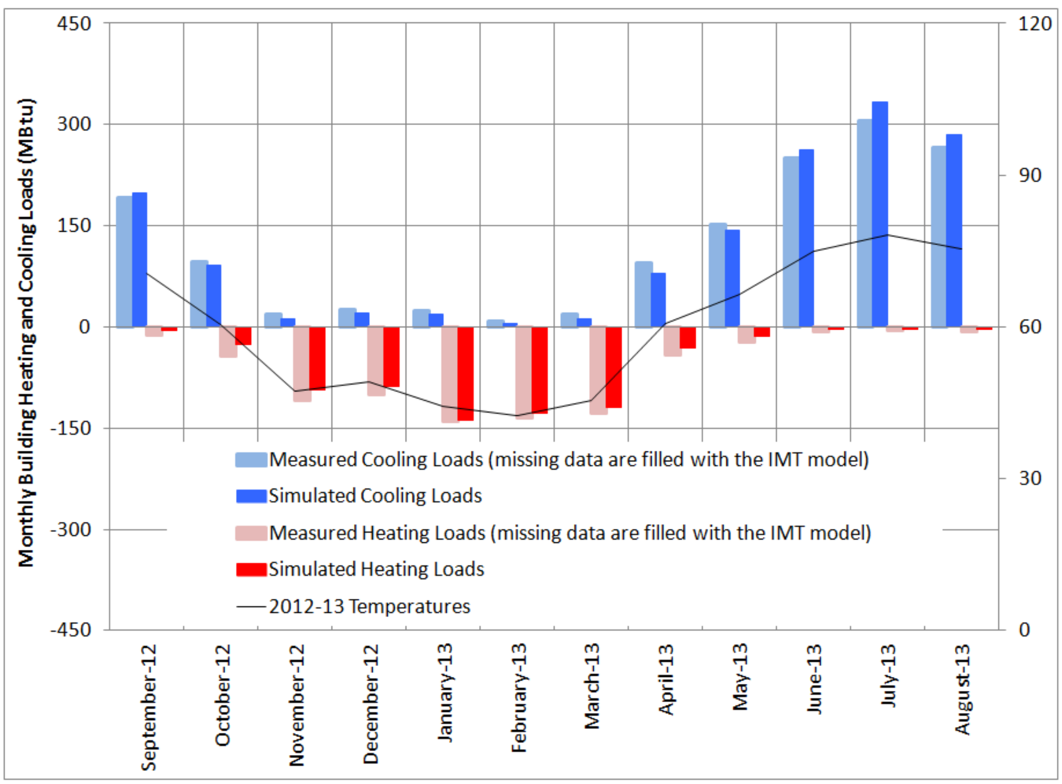

Fig. 40. Comparisons between model-predicted and measured monthly heating and cooling outputs.

The computer model was further calibrated in the second step against the measured power consumption data of the GSHP system (excluding the DHW-related power consumption) by adjusting the schedule and the OA supply temperature of the DOAS. As shown in Fig. 41, the model-predicted monthly power consumption of the GSHP system followed the trends of the measured data closely, except in April and May. As discussed in Sect. 2.2.1, the DOAS operation in April and May of 2013 was quite different from other months in the year; it consumed much less power in these two months. The model-predicted annual power consumption is only 2.3% higher than the measured data (with missing data being filled).

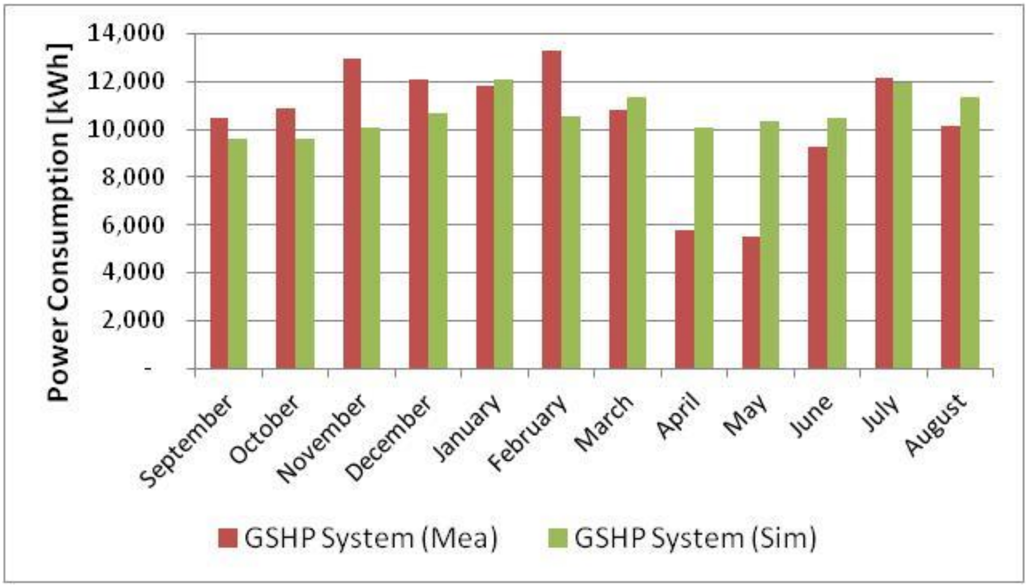

Fig. 41. Comparison of model-predicted and measured monthly GSHP system power consumption.

3.3 BASELINE HVAC SYSTEM

According to the *2003 Commercial Buildings Energy Consumption Survey* conducted by DOE's Energy Information Administration, the most typically used HVAC system in medium-sized office building is a multi-zone variable-air-volume (VAV) system (EIA 2003). The survey indicated that about half of the buildings use gas furnace heat at the main air handler with electric resistance reheat; and the rest half use hydronic heat for the main air handler and the reheat coils. Given the original goal of the building owner to build a high performance facility, the more energy efficient option—hydronic heat and reheat—was chosen as the baseline for this study. It is assumed that an air-cooled chiller and a natural gas boiler are used to provide chilled and hot water to the VAV system.

Figure 42 is a schematic chart of a typical VAV system, excerpted from the DOE-2 user manual (LBNL and JJH 2006). The simulated VAV system is enabled to provide air economizer operation, which supplies near-free cooling to the building with up to 100% OA when the ambient air is colder than 65ºF. The VAV system uses a central air-handling unit to cool the air and distribute it to each zone through high-speed ductwork in the building. The cooling air supply temperature of the VAV system can be reset based on outdoor air temperatures. Space heating is provided by hot water heating coils inside each zone, and there is also central heating in the air-handling unit of the VAV system. OA ventilation with the same design air flow as the real building is provided through the VAV system during the occupied time. There is not any ERV in the baseline system.

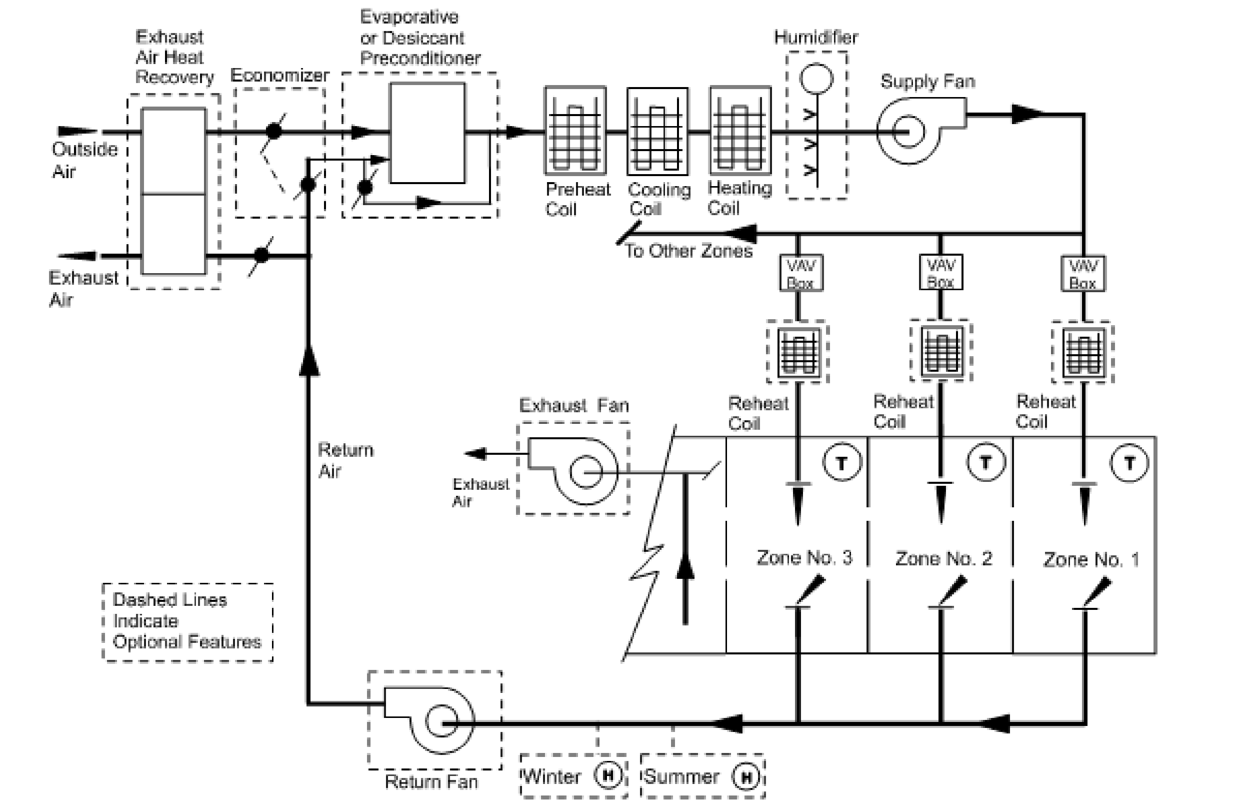

Fig. 42. A typical VAV system. *Source:* Lawrence Berkeley National Laboratory and James J. Hirsch & Associates. 2006. *DOE-2.2 Documentation Volume 3: Topics*. Available at http://doe2.com/download/DOE-22/DOE22Vol3-Topics.pdf.

Principal design parameters of the simulated baseline VAV system are summarized in Table 3. The major components of the VAV system are auto-sized by eQUEST according to the heating and cooling loads predicted by the calibrated building model.

Table 3. Principal design parameters of the simulated baseline VAV system

Component	Capacity	Efficiency
Air-cooled chiller	73 tons	9.5 EER (2.8 COP[a])
Natural gas boiler	1,600 kBtu/h	75% E_t [b]
Supply fan	22,253 ft^3/min, 3.5 in. TSP[c]	0.63 (total efficiency)
Return fan	22,253 ft^3/min, 1.2 in. TSP[c]	0.63 (total efficiency)
Chilled water pump	174 gal/min, 49.9 ft	0.77 mechanical efficiency and 0.865 motor efficiency
Hot water pump	35.7 gal/min, 43.9 ft	0.77 mechanical efficiency and 0.865 motor efficiency

[a] Minimum efficiency for air-cooled and electrically operated chiller (ASHRAE Standard 90.1- 2004).
[b] Minimum efficiency for gas-fired boilers with a heating capacity between 300 and 2,500 kBtu/h, which is rated by thermal efficiency (E_t) (ASHRAE Standard 90.1- 2004).
[c] TSP is the total static pressure, represented in inches of water column.

3.4 COMPARISON BETWEEN DEMONSTRATED GSHP SYSTEM AND BASELINE VAV SYSTEM

Using the calibrated computer model and weather data measured at a nearby airport, the full-year performance of the demonstrated GSHP system and the baseline VAV system were predicted. Simulation results show that both systems can maintain the indoor temperature of the building within the same range (70 to 75°F) when it is occupied.

The simulation-predicted annual consumption of electricity and natural gas by the two systems (including the energy consumption for conditioning the OA) is listed in Table 4. The equivalent site and source energy consumption and carbon emissions were also calculated; these are listed in the same table.

Table 4. Energy consumptions and carbon emissions of the as-built GSHP and the baseline VAV system

	Electric consumption (kWh)	Natural gas consumption (MBtu)	Site energy (MBtu)	Source energy (MBtu)	Carbon emissions (Mt)
Baseline VAV	151,720	487	1,004	2,314	153
GSHP	123,400	0	421	1,450	97
Savings	28,320	487	583	864	56
Percentage change	**19%**	**100%**	**58%**	**37%**	**36%**

All the conversion factors for calculating the source energy consumption and carbon emissions are listed in Table 5, which is adopted from a report published by the National Renewable Energy Laboratory (Deru and Torcellini 2007).

Table 5. Conversion factors for source energy consumption and carbon emissions in North Carolina

Source energy factor for electricity delivered to building	3.443	kWh/kWh
Source energy factor for natural gas delivered to building	1.092	Btu/Btu
Total emission factors for delivered electricity	1.74	lb/kWh
Precombustion emission factors for fuel delivered to building	27.8	lb/MBtu
Emission factors for on-site combustion in a commercial boiler	123	lb/MBtu

As shown in Table 4, **compared with the VAV system**, the **GSHP system completely eliminates natural gas consumption (an annual savings of 487 MBtu natural gas) and reduces electric energy consumption by 19% (an annual savings of 28,320 kWh electricity). The savings in electricity and natural gas means a 58% savings in site energy, a 37% savings in source energy, and a 36% reduction in equivalent CO_2 emissions.**

Results for the electric energy end use of the baseline VAV system and the as-built GSHP system are shown in Fig. 43. As Fig. 43 shows, although the GSHP system uses electric energy for space heating (while the baseline VAV system consumes natural gas for space heating), it uses much less electric energy for space cooling than the baseline VAV system. The electric energy uses of the VAV and the GSHP systems for indoor air circulation and OA ventilation are almost identical. The as-built GSHP system consumes nearly two times more electric energy for running the circulation pumps than the VAV system.

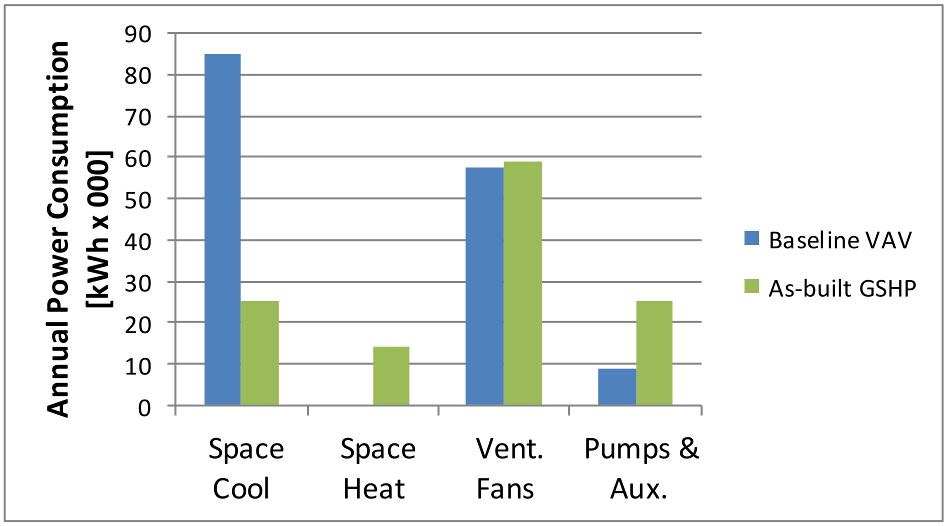

Fig. 43. Simulation results for electric energy end use of the baseline VAV system and the as-built GSHP system.

Additional simulation with a properly configured and controlled variable speed pump was performed. In this simulation, the flow rate in the water loop of the GSHP system varies in response to the operation of the WAHPs to maintain a constant pressure differential across the supply and return lines of the water loop.

The simulation results show that while the other end use energy consumption is almost identical, the variable speed pump reduces the pumping power consumption by 66% (Fig. 44), and it results in a 14% reduction in the total power consumption of the GSHP system. Also shown in Fig. 45, the variable speed pump reduces the pumping power fraction from 20% to only 8%.

The improved pumping efficiency increases the energy savings and carbon emission reductions compared with the baseline VAV system. As shown in Table 6, **the savings in site energy and source energy are increased to 64% and 46%, respectively, and the reduction of equivalent CO_2 emissions is increased to 45%.**

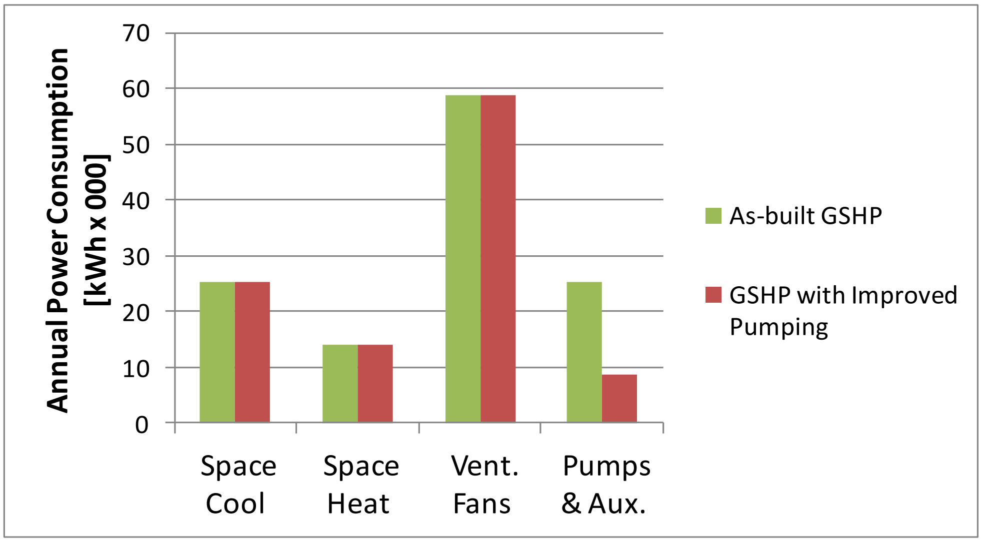

Fig. 44. Simulation results for electric energy end use of GSHP systems with and without a variable speed pump.

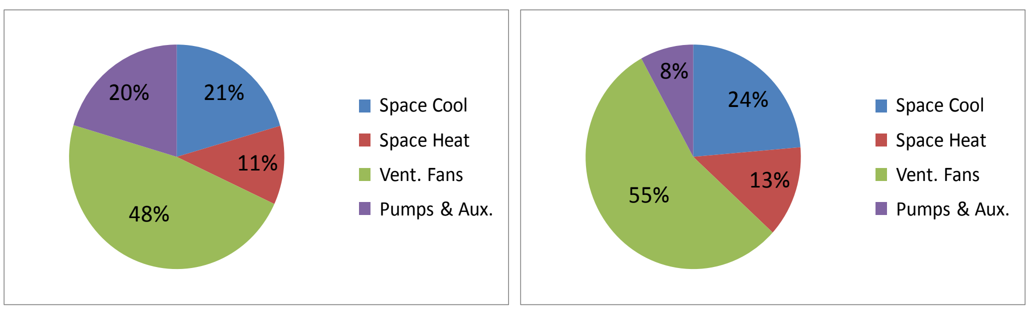

Fig. 45. Percentages of end use power consumption of the GSHP system resulting from (a) the as-built constant speed pump and (b) a properly controlled variable speed pump.

Table 6. Energy consumptions and carbon emissions of the baseline VAV system and the GSHP system with properly operated variable speed pump

	Electric consumption (kWh)	Natural gas consumption (MBtu)	Site energy (MBtu)	Source energy (MBtu)	Carbon emissions (Mt)
Baseline VAV	151,720	487	1,004	2,314	153
GSHP	106,530	0	364	1,252	84
Savings	45,190	487	641	1,062	69
Percentage change	**30%**	**100%**	**64%**	**46%**	**45%**

4. ANALYSIS OF COST-EFFECTIVENESS

The cost effectiveness of the demonstrated GSHP system was evaluated using a simple payback period, which is the ratio of the cost premium for the GSHP system to the annual operating cost savings it achieves.

4.1 COST PREMIUM

The GSHP system cost premium is the difference between the installed cost of the GSHP system and the baseline VAV system. Table 7 summarizes the itemized installed cost of the demonstrated GSHP system, which was provided by the grantee. The total installed cost of the GSHP system is $1,807,750. The estimated cost for the baseline VAV system is $840,000 (based on the 24,000 ft^2 total floor space and $35/ft^2 normalized cost for typical HVAC systems in 2010, which is the average of a few available cost data). The resulting cost premium of the GSHP system is $967,750.

Table 7. Itemized installed cost of the demonstrated GSHP system

Cost item	Total material costs	Total labor cost	Total raw cost
General conditions	$68,654	$18,029	$86,683
Architectural	–	–	–
Structural	–	–	–
Civil	–	–	–
Wells and loop piping	$120,327	$319,071	$439,398
Plumbing system	$63,040	$21,273	$84,313
Mechanical	$468,918	$375,228	$844,146
Electrical	$15,000	$33,776	$48,776
Instrumentation and controls	$226,156	$78,279	$304,435
Subtotal direct costs			**$1,807,750**

Given the 159.5 ton installed cooling capacity of the GSHP system and the 24,000 ft^2 floor space, the installed cost can be normalized as $11,298 per installed cooling ton, or $75.3/ft^2 of the conditioned floor space. The $11,298/ton cost is about 49% higher than the national average for the installed GSHP systems, which is about $7,571/ton (2010 dollars*) based on a study by Department of Defense (DOD 2007). The floor-area-based normalized cost of the demonstrated GSHP system, $75.3/ft^2, is also much higher than the average value reported in a recently completed study (Kavanaugh, Green, and Mescher 2012). The reported average cost is $20.75/ft^2, which includes $15.46/ft^2 for the HVAC system (inside the building) and $5.29/ft^2 for the ground loop.

To the best knowledge of the authors, the installed cost includes expenses unique to the ARRA-funded demonstration project that are not common in ordinary HVAC projects, including the additional management and reporting required by the ARRA grant; prevailing wage requirements for all the laborers involved in this project; and hardware, software, and manpower for continuous performance monitoring. In addition, as mentioned by the grantee, the ground loop and the WWHP for DHW were oversized to allow for future expansion of the facility.

* The conversion was made using the CPI [Consumer Price Index] Inflation Calculator of U.S. Bureau of Labor Statistics (available at http://data.bls.gov/cgi-bin/cpicalc.pl).

4.2 OPERATING COST SAVINGS

The operating costs of the GSHP system and the baseline VAV system are calculated based on the predicted annual electric and natural gas consumption and the local utility rates for electricity and natural gas. Figure 46 shows the historical price of natural gas in North Carolina from 2006 through 2014 (EIA 2014). As can be seen, the price of natural gas dropped sharply from its peak ($17.5/MBtu) in July 2008 to about $8/MBtu in recent years. The average price during the 1 year period encompassed in this study is $8.9225/MBtu. The price of electricity usually does not change much, and the average commercial electricity rate in Raleigh, $0.085/kWh*, is used in this analysis.

Based on the above energy prices and the predicted annual energy consumption (listed in Table 4), annual energy costs for operating the GSHP and the baseline VAV systems are calculated to be $10,489 and $17,237, respectively. The annual energy cost savings achieved by the GSHP system is $6,748, which is a 39% cost savings compared with the energy cost of the baseline VAV system.

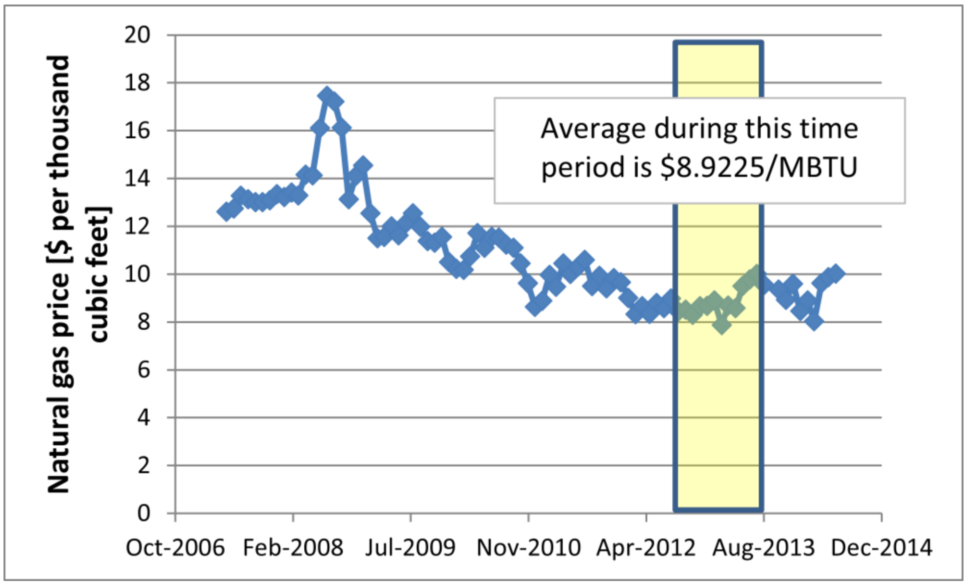

Fig. 46. Natural gas prices in North Carolina, 2006–2014.

4.3 SIMPLE PAYBACK

Based on the calculated cost premium ($967,750) and the predicted annual operating cost savings ($6,748) of the GSHP system, **the simple payback period for the GSHP system is 143 years.**

Figure 47 shows the relationship between the GSHP system cost and the resulting simple payback period for the same energy cost savings. The baseline VAV cost is $6,000/ton or $35/ft^2. The payback would be zero if the normalized cost of the GSHP system were the same as that of the baseline VAV system. When the cost of GSHP is increased, the payback period increases linearly. **With the achieved energy cost savings and the cost of the baseline VAV system, to make the payback period shorter than 10 years, the cost of GSHP system needs to be less than $6,500/ton or $38/ft^2.**

* Source: http://www.electricitylocal.com/states/north-carolina/raleigh/#ref.

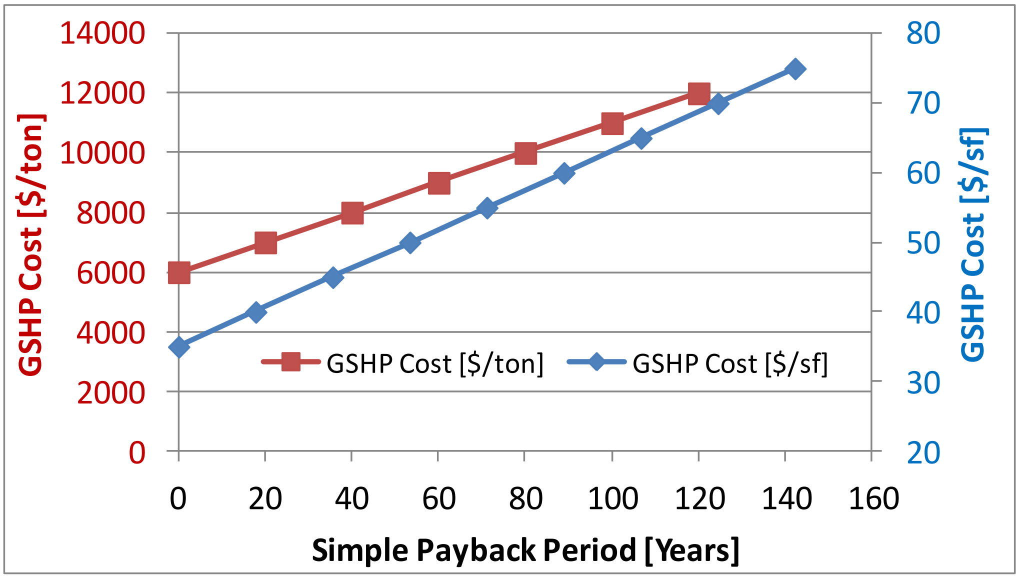

Fig. 47. Relationship between GSHP system cost and the resulting simple payback period.

As mentioned earlier, this ARRA-funded demonstration project has many costs that are not common in ordinary commercial GSHP projects, including the additional management and reporting required by the ARRA grant, prevailing wage requirements, and performance monitoring. In addition, the GHX and DHW system are oversized to allow for future expansion of the facility. As a result, this demonstration project has a longer payback period than would be expected for ordinary commercial GSHP projects in this region.

5. CONCLUSIONS

This section summarizes the conclusions drawn from this case study, including the performance of the GSHP system based on measured data, benefits achieved by the GSHP system compared with the baseline system, and the cost effectiveness of the demonstrated GSHP system. In addition, lessons learned regarding performance data collection and analysis, as well as areas that need further improvement, are summarized.

5.1 PERFORMANCE, ENERGY SAVINGS, COST EFFECTIVENESS

Based on our analysis of the measured and calculated data, the following conclusions can be drawn with regard to the performance of the demonstrated GSHP system:

- The ground loop supply fluid temperature was within a favorable range (61–77°F) for the operation of the WSHP units during the 1 year period encompassed in this study. Given the moderate imbalance between the annual cumulative heat rejection and extraction loads imposed to the GHX (with a ratio of 2.6:1), the narrow range of the ground loop supply temperatures indicate that the GHX is oversized.

- During the 1 year period from September 2012 through August 2013, the ground loop flow rate varied within only a narrow range (300–400 gal/min), and it was more than 300 gal/min even when no heat pump was in operation. There is no clear relationship between the ground loop flow rate and either the power consumption of the heat pump units or the ground loop temperature differential. This indicates that there must be some issues in the configuration and/or control of the pumping system.

The nearly constant flow rate is likely due to the high bypass flow in the pumping system, which may be due to missing or malfunctioning two-way solenoid valves in the heat pump unit. The nearly constant water flow rate was still under investigation by the grantee when this report is written. Due to continuous operation with a nearly constant water flow rate, the ground loop pump contributed 16.4% to the total measured power consumption of the entire GSHP system (including the DOAS and DHW). It was found that the total power consumption of the ground loop pumping decreased by 30% during the following year from September 2013 to August 2014 although there was not any significant change in the ground loop flow rate. The decreased power use is likely due to a lower pressure differential set point after August 2013.

- Given the favorable ground loop supply temperature, the heat pumps had high energy efficiency—the annual average COP and EER of the 27 WAHP units are 4.9 and 18.5, respectively. It is also noticed that the total installed capacity of the 27 WAHP units is 30–50% larger than the peak cooling and heating demand of the building. In addition, the annual cumulative heating output of the 27 WAHP units is 1.4 times higher than the annual cumulative cooling output during the 1 year period from September 2012 through August 2013. It is thought to be due to the contribution of the OA supplied from the DOAS, which offset some cooling loads but increased the heating load of the WAHP units.

- The annual average effective COP of the DOAS is 6.6. It is observed that the two 7.5 hp fans of the DOAS ran at full speed almost continuously during the year from September 2012 through August 2013. As a result, DOAS contributed the most (45.4%) to the annual total power consumption of the GSHP system. Performance data in the following year indicate a 37% reduction in the annual power consumption of the DOAS after changing the control to operate the DOAS only during the scheduled occupied time period. The annual power consumption of the entire GSHP system was also reduced by 20% after changing DOAS control strategy.

- The calculated annual average COP and EER of the GSHP system are 4.5 and 17.5, respectively. These system efficiencies are only slightly lower than the annual average COP and EER of the 27 WAHP units (4.9 COP and 18.5 EER). It appears that the high efficiency of the DOAS offset the impact of the pumping power on the system efficiencies.

Simulations of the demonstrated GSHP system and the baseline VAV system were performed using a calibrated computer model of the facility. The simulation results indicate that, compared with the VAV system, the GSHP system completely eliminates natural gas consumption (an annual savings of 487 MBtu natural gas) and reduces electric energy consumption by 19% (an annual savings of 28,320 kWh electricity). The savings in electricity and natural gas means a 58% savings in site energy, a 37% savings in source energy, and a 36% reduction in equivalent CO_2 emissions. Further simulation analysis indicates that if the ground loop pump can vary the system water flow based on the operation of the heat pump units (as would be expected for a variable speed pump), the savings in site energy and source energy would be increased to 64% and 46%, respectively; and the reduction of equivalent CO_2 emissions would be increased to 45%.

The installed cost of the demonstrated GSHP system is $1,807,750, which is normalized as $11,298 per installed cooling ton, or $75.3/ft^2 of the conditioned building floor space. The $11,298/ton cost is about 49% higher than the national average of the installed cost of GSHP systems, which is about $7,571/ton (2010 dollars) based on a study by Department of Defense (DOD 2007). The floor-area-based normalized cost of the demonstrated GSHP system, $75.3/ft^2, is also much higher than the average value reported in a recently completed study (Kavanaugh, Green, and Mescher 2012). The reported average cost is $20.75/ft^2, which includes $15.46/ft^2 for the HVAC system (inside the building) and $5.29/ft^2 for the ground loop. The installed cost includes expenses unique to the ARRA-funded demonstration project that are not common in ordinary HVAC projects, including the additional management and reporting required by the ARRA grant; prevailing wage requirements for all the laborers involved in this project; and hardware, software, and manpower for continuous performance monitoring. In addition, as mentioned by the grantee, the ground loop and the WWHP for DHW were oversized to allow for future expansion of the facility.

With the achieved energy cost savings and the typical cost of the baseline VAV system, to make the payback period shorter than 10 years, the cost of GSHP system needs to be less than \$6,500/ton or \$38/ft^2.

5.2 LESSONS LEARNED

5.2.1 Performance data collection and analysis

More data points, including the flow rate and temperature of the OA leaving the DOAS, are needed for analyzing the performance of the DOAS. The only available measured performance data for DOAS in this study is the total power consumption. Given the wide range of operating conditions and various controls of the DOAS, a few assumptions and approximations had to be made to estimate the heating and cooling outputs of the DOAS.

On the other hand, power consumption by each of the 27 WAHP units is measured in this project. It is an extra cost for the already expensive GSHP system. Since the performance of the WAHP units are usually well characterized with performance curves/maps in the manufacturer's catalog data, the power consumption of the WAHP units can be reasonably estimated with the measured ground loop supply temperatures and the performance curves/maps, as well as the air and water flow rates of the WAHP unit, which are usually constant and available from design documents. This "venture sensing" procedure can eliminate the needs and associated cost for measuring the power consumption at each WAHP unit.

5.2.2 System Design and Control

Variable speed pumps should not be oversized, and the bypass flow should be minimized so that the pump can ramp down to its minimum allowed flow rate. Otherwise, the variable speed pump may perform just like a large constant speed pump and waste lots of pumping energy. It is especially important for GSHP system since the excessive pumping power will increase the heat rejection load to the GHX, which may degrade the cooling efficiency of the GSHP system. For the demonstrated GSHP system, if the variable speed pump can properly adjust the system flow rate according to the operation of the heat pump units, it will reduce the pumping power consumption by 66% and result in a 14% reduction in the total power consumption of the GSHP system. The pumping power fraction could be reduced from 20% to only 8%.

Operation schedule and OA supply temperature can significantly impact the energy consumption of the DOAS system and the entire GSHP system, especially when the DOAS is also coupled with GHX. Although an enthalpy wheel can recover the otherwise wasted energy from the EA to preheat or precool the OA, it will consume significant fan power, especially when it runs all year long with constant air flow. The operation of DOAS will affect the heating and cooling loads of the heat pump units for space conditioning, as well as the heat rejection and extraction load of the GHX. It is thus necessary to optimize the operation and OA supply temperature while providing sufficient OA ventilation. The efficiency of the DOAS can be improved if the enthalpy wheel in the DOAS is bypassed when the enthalpy differential between the OA and EA is not large enough to offset the fan power consumed for pushing the air steams through the enthalpy wheel.

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APPENDIX A. SEQUENCE OF OPERATIONS FOR GROUND LOOP PUMP

(excerpted from the design document provided by the grantee)

A.1 GROUND COUPLED WATER LOOP

A. Start-up

1. Upon start-up, the Ground Loop Water Pumps will be off and the SV bypass valve will be closed. Vendor supplied water flow control valves on the heat pump units will be open. A start command issued through the BAS HMI or the RI will cause one of the ground loop pumps to start. Once pump status is obtained by sensing DP across the pump, the Ground Coupled Water Loop will operate.

B. Shut-Down

1. A shut-down command can be issued through the BAS HMI or the RI. System shut-down commands are manual. A shut-down command will cause the operating Ground Loop Water Pump to shut-down. The SV bypass valve will automatically close. The Heat Pump VC will then cause the water flow controls valves to open.

C. Pressure Control

1. *Each GLW Pump is controlled via its respective Variable Frequency Drive (VFD) controller. The BAS monitors and controls operation of both pumps. The pumps operate in a lead/lag configuration as described in the "Pump Sequencing" Section (F) below.*

2. *A differential pressure (DP) transmitter is installed across the supply and return piping of the Ground Water Loop system to sense the DP across the system. The system shall be configured to maintain system DP using a control setpoint (adjustable) of **50 psig**. As the load on the system increases, i.e. cause the DP to increase, the BAS will modulate the speed on the operating pump to ramp down in speed. The control loop for the pump will configured to prevent the pumps from operating below 20% speed. Operating the pumps below 20% will cause them to overheat.*

3. *As heating or cooling loads are reduced within the facility and the VC heat pump controls modulate their respective MAV to the closed position, the ground coupled water loop DP will increase. When the DP reaches 60 psig, the bypass SV will modulate open to maintain a maximum 60 psig DP. This valve is self-contained. The final DP will be set by the Test and Balance Contractor.*

APPENDIX B. GROUND LOOP PUMPING DESIGN

(excerpted from mechanical drawing M-451 of the demonstrated GSHP system)