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Case Study of the ARRA-Funded GSHP Demonstration at North Village Student Housing, Furman University

Mini Malhotra, PhD
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October 2015

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OAK RIDGE NATIONAL LABORATORY
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Energy and Transportation Science Division

CASE STUDY OF THE ARRA-FUNDED GSHP DEMONSTRATION AT NORTH VILLAGE STUDENT HOUSING, FURMAN UNIVERSITY

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ACRONYMS

ARRA	American Recovery and Reinvestment Act of 2009
ASHP	air source heat pump
CO_2e	equivalent CO_2 emissions
COP	coefficient of performance
$COPh_{eqp}$	heat pump efficiency in heating mode
$COPc_{eqp}$	heat pump efficiency in cooling mode
CMRMSE	coefficient of variation of root mean squared error
DOE	Department of Energy
DP	differential pressure
EER	energy efficiency rating
EIR	energy input ratio
FGL	flow rate
GHX	ground heat exchanger
gpm	gallon per minute
GSHP	ground source heat pump
HDPE	high-density polyethylene
MBBE	normalized mean bias error
NREL	National Renewable Energy Laboratory
OA	outdoor air
ORNL	Oak Ridge National Laboratory
QGL_c	heat rejected to ground loop
QGL_h	heat extracted from ground loop
QHP_c	heat pump cooling delivered
QHP_h	heat pump heating delivered
TD	temperature differential
TDBS	dry-bulb temperature of the air entering the heat pump
TGLS	ground-loop supply water temperature
TGLR	ground-loop return water temperature
THPR	heat pump return temperature
THPS	heat pump supply temperature
TWBS	wet-bulb temperature of the air entering the heat pump
W	GSHP system power consumption
WAHP	water-to-air heat pump
WHP_c	heat pump power consumption in cooling mode
WHP_h	heat pump power consumption in heating mode
WLP	ground loop pump power consumption

EXECUTIVE SUMMARY

With funding provided by the American Recovery and Reinvestment Act, 26 ground source heat pump (GSHP) projects were competitively selected in 2009 to demonstrate the benefits of GSHP systems and innovative technologies for cost reduction and/or performance improvement. One of the selected demonstration projects was proposed by Furman University for ten student housing buildings—the North Village located on the campus in Greeneville, South Carolina.

All ten buildings are identical in floor plan and construction. Each building is conditioned by an identical GSHP system consisting of 25 water-to-air heat pump (WAHP) units, a closed-loop vertical ground heat exchanger (GHX) installed under an adjacent parking lot, and two redundant 7.5 hp variable-speed pumps to circulate water through the GHX and the WAHPs.

The actual performance of the GSHP systems is analyzed with available measured data for 2014. The annual energy performance is compared with a baseline scenario in which the building is conditioned by air-source heat pumps (ASHPs) with the minimum allowed efficiencies specified in ASHRAE Standard 90.1-2013 (SEER 13 for cooling and 7.8 HSPF for heating) and supplemental electric heaters. The comparison is made in terms of energy savings, operating cost savings, cost-effectiveness, and environmental benefits. Finally, limitations in conducting this analysis are identified and recommendations for further improving the operational efficiency of the GSHP systems are made.

Energy Performance and Cost Effectiveness

The annual operational efficiencies, evaluated by the coefficient of performance (COP), of the GSHP systems ranged between 4.8 and 5.5 for cooling and 3.8 and 4.9 for heating. A comparison of the measured monthly heat pump energy use for six apartments in buildings B and I indicates that the new GSHP units in building B used less than 1/3 as much electricity as the existing ASHPs at building I. Accounting for the pumping energy associated with the six GSHP units, the energy savings achieved by the GSHP system was about 60% compared with the existing ASHPs.

Utility bills for the 10 buildings (including electricity consumption for space conditioning, lighting, appliance, and other loads) during the pre- and post-retrofit periods were analyzed. The weather-normalized monthly electricity use for all buildings combined for the year 2014 (post-retrofit) was about 36% lower than that during 2010 (pre-retrofit) when the outdoor air temperature was lower than 40°F. The electricity saving percentages were smaller (by 10–20%) when the outdoor air temperature was between 50 and 70°F, and it varied between 0 and 50% (which is thought to be due to varying occupancy levels and different thermostat settings, especially during the summer break) when the outdoor air temperature was higher than 70°F. Overall, the GSHP retrofit has resulted in a 27.3% reduction in the annual electricity use of the ten buildings. The achieved annual electricity savings was 715,384 kWh, which has a value of $65,815 at the local electricity rate.

Compared with the baseline system, the GSHP system achieved significant energy savings and CO_2 emission reductions. For example, the GSHP system in building B demonstrated 37.8% site and source energy savings and CO_2 emissions reductions compared with the baseline system. The reduced electricity consumption resulted in a $3,807 annual operating cost savings for the building. Table ES.1 provides a performance comparison between the two systems. Assuming the replacement cost of the baseline ASHP system would be $3,500 per unit[1] for 25 units (i.e., $87,500 per building), the cost premium for the GSHP system would be $319,915, which results in a simple payback period of 84 years for this GSHP retrofit.

[1] This excludes the cost of ductwork, assuming that the existing ductwork would be used.

Table ES.1. Performance comparison between the GSHP system and the baseline system (for one of the ten buildings)

	Baseline system	GSHP system	Savings
Site energy use (MMBtu)	109,570	68,190	37.8%
Source energy use (MMBtu)	377,250	234,778	37.8%
CO_2 emissions (lb)	190,652	118,651	37.8%
Total annual energy cost ($)	$10,080	$6,273	37.8%

Lessons Learned and Issues

- Compared with other GSHP systems that have been studied previously, the GSHP systems at Furman University, except at buildings A and K, used significantly less pumping energy; and their pumping power fractions (10.5–13%) were about half those of the GSHP systems studied previously. The better pumping performance was the result of the following good practices:
    - Locating the differential pressure sensor at the hydraulically most remote WAHP in the piping system
    - Using an auto-flow valve for each WAHP to maintain a constant flow rate at the WAHP
    - Minimizing bypass flow

- Patterns in the relationship between the flow rate and the temperature differential of the ground loop could be a very useful indicator for evaluating pumping performance. Ideally, the pattern should demonstrate a “U” shape, meaning the temperature differential is kept at the design value while the flow rate varies within a large range. For a GSHP system with excessive pumping at part-load conditions, the pattern will show a “V” shape with a substantially high minimum flow rate.

- Malfunctioning sensors can be identified by comparing the patterns in the relationships among different measurements with expected or historical patterns.

- The two-way solenoid valves at each heat pump should be closed when the solenoid is not energized. In this way, a malfunctioning valve can be easily identified (e.g., from a service call).

Recommendations for Further Improvements

- To more accurately analyze the performance of the GSHP system (e.g., accounting for simultaneous heating and cooling operation of the individual heat pumps in a building), the heating and cooling output and the associated power consumption of each heat pump should be measured. If it is too expensive to do so, at least the runtime of each heat pump in each operation mode should be recorded.

- Measured data indicate excessive pumping in the GSHP systems at buildings A and K. It is recommended that the 2-way solenoid valves of each heat pump in these buildings be checked. It is very likely that the excessive pumping is due to a few malfunctioning 2-way valves.

1. PROJECT OVERVIEW

North Village student housing at Furman University (Fig. 1) is an apartment complex of 11 three-story, 24-unit buildings at Furman University in Greeneville, South Carolina, constructed between 1997 and 2001. Fig. 2 shows the geographical location within the United States of the demonstration site, which has a humid subtropical climate with short, cool winters and hot, humid summers. In December 2009, this site was competitively selected by the US Department of Energy (DOE) for a demonstration project to replace the existing system (i.e., air source heat pumps [ASHPs]) in the buildings with ground source heat pump (GSHP) systems to provide space heating and space cooling. The goal of this demonstration project is to validate the technical and economic feasibility of the GSHP system for providing space conditioning in this region. Should the demonstration prove satisfactory and feasible, it will encourage similar GSHP applications, thus helping save energy and reduce carbon emissions.

Fig. 1. Aerial view of the GSHP demonstration site at North Village student housing at Furman University.

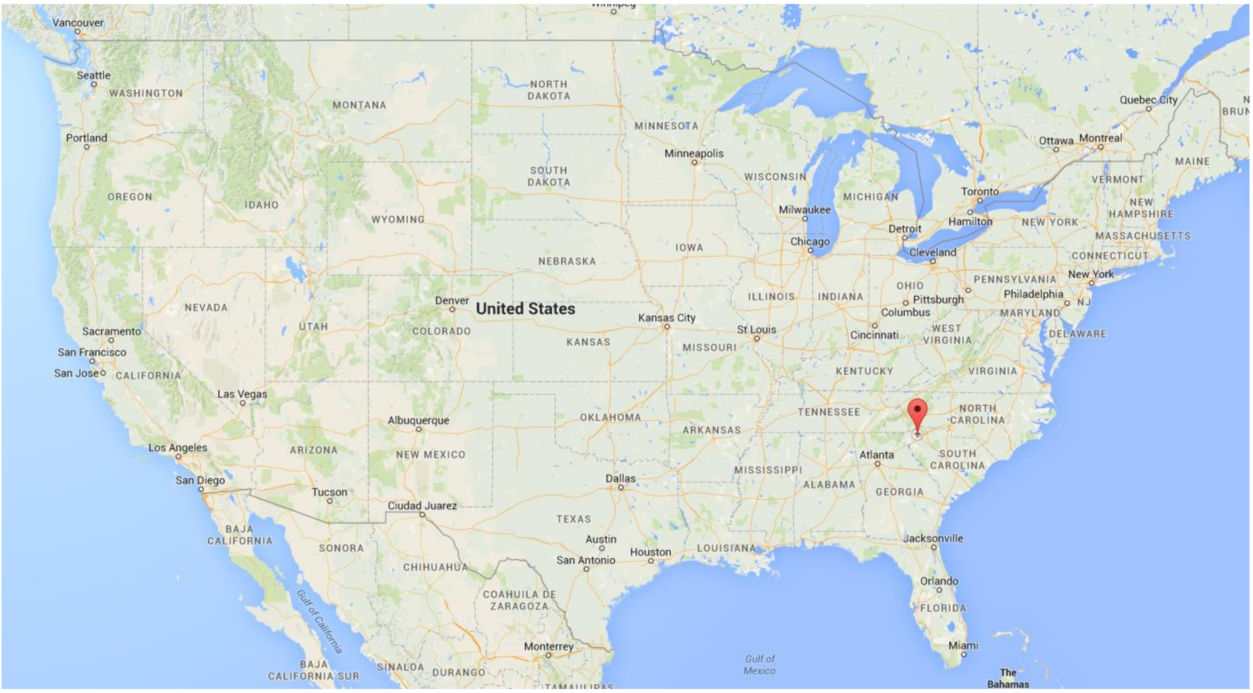


Fig. 2. Geographic location within the United States of the GSHP demonstration site.

Among the 11 buildings of North Village (named A through K), ten buildings were retrofitted with GSHP systems.[2] Construction of the GSHP systems for the different buildings took place in three phases from 2011 to 2013. Measured data for the GSHP systems became available starting in January 2012 as they became operational in different buildings. Based on measured data and other relevant information, this case study evaluated the performance metrics, including the energy efficiency of the overall GSHP systems, electricity end uses of all the major equipment of the GSHP systems, benefits achieved by the GSHP system (i.e., energy and cost savings, carbon emission reductions) compared with a baseline system, and the cost-effectiveness of the GSHP installation. This case study also identified areas for further improving the operational energy efficiency of the GSHP systems.

The buildings and the GSHP systems are described in the following sections.

1.1 BUILDING INFORMATION

The North Village is an apartment complex of 11 three-story buildings constructed between 1997 and 2001. Fig. 3 shows a view of buildings G, H, and I. All of the buildings have identical layouts and each consists of twenty-four 1,200 ft^2 apartments for 92 students, 2 laundry rooms, and mechanical and electrical rooms. The buildings are wood-frame structures with slab-on grade floors, vented attics, R-11 wall insulation, and R-30 ceiling insulation. The buildings house students during the academic year at nearly full occupancy, and six of the buildings are used for summer student housing at 80–90% occupancy. The remaining buildings are occupied during the summer for a few weeks per year. Originally, space conditioning in each apartment in all of the buildings was provided by an individual ASHP that used R-22 refrigerant and a supplemental electric resistance heater. The original systems were approaching end-of-life after 13–15 years of service.

[2] Building J could not be retrofitted within the project budget.

Fig. 3. Buildings G, H and I of North Village student housing at Furman University.

1.2 GSHP SYSTEM

In the ten retrofitted buildings, new GSHP systems were installed as the sole source of space conditioning in the apartments. The GSHP systems serving each of the ten buildings are identical. Fig. 4 shows a schematic of the GSHP system installed in one of the buildings, along with the data collections points, which are explained in Table 1. Each GSHP system consists of 25 McQuay water-to-air heat pumps (WAHPs), a 2.5 ton unit in each of the 24 apartments and a 1.5 ton unit for the laundry rooms. The combined nominal capacity of each GSHP system is 61.5 tons.

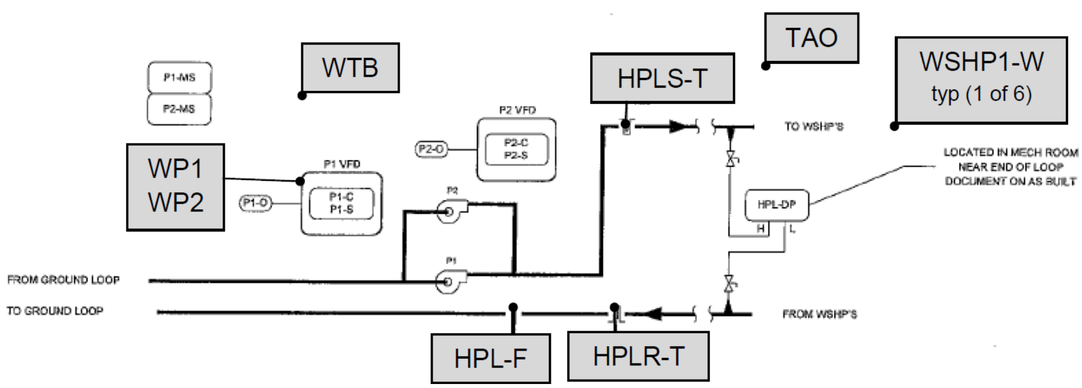


Fig. 4. GSHP system schematic and data collection points.

1.2.1 Ground Heat Exchanger

The ground heat exchanger (GHX) of each GSHP system consists of 20 vertical bores, each 500 ft deep. The wells are set up in a 4×5 grid on 20 ft centers. Each 6” diameter vertical bore contains a single U-shape HDPE pipe. In total, each GHX has 10,000 ft bores and 20,000 ft HDPE pipe. Based on the 61.5

ton installed heat pump capacity, the GHX is sized for 163 bore-ft/ton. Fig. 5 shows the layout of the bore fields. Each dot represents a bore for the associated building's GHX.

Fig. 5. Layout of the bore fields.

1.2.2 Variable-speed pumping

All the 25 WAHP units in a building are connected by a common two-pipe water loop, which has a central pumping station to circulate water through the GHX and each of the WAHPs. The central pumping station is located in a new pump room on the first floor of each apartment building. It consists of two redundant 7.5 hp variable-speed, pressure-controlled circulation pumps (Fig. 6, P-1 and P-2). The two pumps are normally operated in a lead-and-lag configuration (i.e., two pumps run alternately). However, if the lead pump runs at a speed greater than 95% of its full speed for 5 minutes, the lag pump will start.

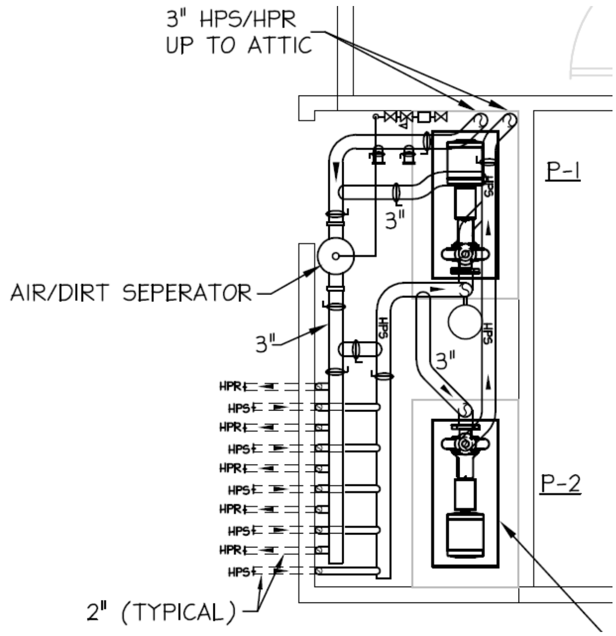


Fig. 6. Central pumping station.

Each WAHP has a 2-way valve that can block water flow from entering the heat pump if the heat pump is not called on by the thermostat. The pumping system was elaborately designed with the following features:

- A differential pressure (DP) sensor is installed at the hydraulically most remote WAHP in the hydronic piping system.

- An auto-flow valve is used at each WAHP, which automatically regulates its opening in response to the change in hydronic pressure imposed on it to keep a constant flow rate at the WAHP.

- Bypass flow is minimized by using a dedicated bypass pipe at the end of the riser of the water loop (as shown in Fig. 7) and a 2-way solenoid valve at each WAHP (i.e., system flow rate could go down to near zero when no WAHP is running).

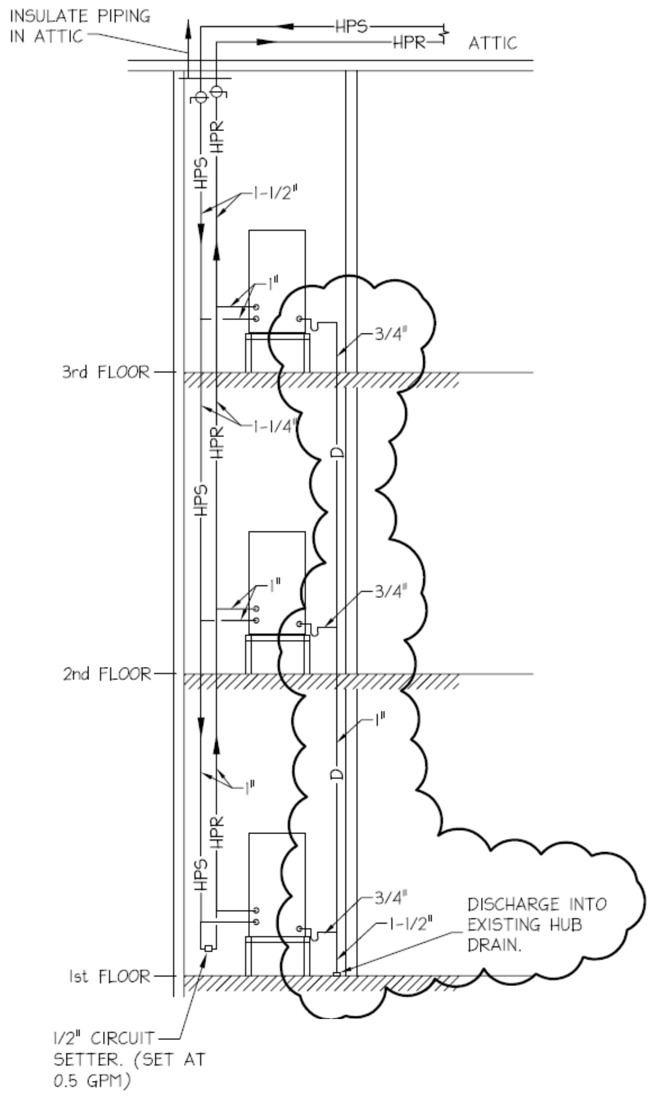


Fig. 7. Schematic of piping connection to WAHPs.

1.3 DATA COLLECTION

The university implemented a data acquisition system at each building to monitor the performance of the GSHP system. The monitoring system collects data at 15 minute intervals for the ground-loop supply and return water temperature and flow rate, pump power, total building electricity use, and outdoor air (OA) temperature. In addition, electricity consumption for 6 of the 25 heat pumps in building B was measured. Table 1 lists the collected data points. Fig. 4 shows the locations of these data points.

Table 1. GSHP system monitoring data points

No.	Data Point	Description	Units	Sensor
1	HPLS-T	Heat Pump Supply Temperature	deg F	Johnson Controls TE-6300
2	HPLR-T	Heat Pump Return Temperature	deg F	Johnson Controls TE-6300
3	HPL-F	Ground Loop Flow	GPM	Onicon (F-1210)
4	WTB	Total Building Energy	kWh	Veris H8026
5	WP1	P1 - Pump power	kW	ABB ACH550
6	WP2	P2 - Pump power	kW	ABB ACH550
7	TAO	Ambient Temperature	deg F	JCI-TE-6300

No.	Data Point	Description	Units	Sensor
1	WSHP1-W	WSHP-1 Energy	kWh	Veris Hawkeye 922
2	WSHP2-W	WSHP-2 Energy	kWh	Veris Hawkeye 922
3	WSHP3-W	WSHP-3 Energy	kWh	Veris Hawkeye 922
4	WSHP4-W	WSHP-4 Energy	kWh	Veris Hawkeye 922
5	WSHP5-W	WSHP-5 Energy	kWh	Veris Hawkeye 922
6	WSHP6-W	WSHP-6 Energy	kWh	Veris Hawkeye 922

2. ANALYSIS OF MEASURED PERFORMANCE OF GSHP SYSTEMS

Performance data of the GSHP systems in the ten buildings are available starting from as early as January 2012 (in Buildings A, B, and C) to as late as September/October 2013 (in Buildings G, H, I and K), as shown in Fig. 8. Note that during this measurement period, there are periods spanning from a month to over a year when some of the performance data were missing for at least one GSHP system. The performance of the GSHP system in each building was analyzed for the entire period for which a partial or full set of performance data was available. However, the comparison of the full-year performance of the systems among buildings was based only on the data for 2014, for which most buildings have a complete data set. For four buildings (A, B, F, and G), data were missing for up to 1 month during 2014.

For each building, the performance of the GSHP system was analyzed for (1) source-side operation, by examining the trend of the measured water temperature and flow rate in the ground loop and the measured pumping power, and calculating the heat transfer in the ground loop; (2) load-side operation, by computing the operational efficiency of the heat pump; and (3) overall GSHP system, by computing the operational efficiency of the GSHP system.

The GSHP system is of a distributed configuration consisting of 25 heat pumps on the load side. However, there was no measurement of the performance of each individual heat pump (except the power draws of six heat pumps in building B), and the only available measured data are for the source side of the system. Therefore, the performance of the heat pumps and of the overall GSHP system was calculated by approximating the individual 25 heat pumps in a building as one aggregated heat pump and using the measured source-side performance data and the performance curves published in the heat pump manufacturers' literature. This approximation neglects possible simultaneous heating and cooling operation of the 25 heat pumps. Since all the apartments are on the perimeter of the building, and their heating and cooling demands are mostly determined by the weather, simultaneous heating and cooling by individual heat pumps would not be significant.

In analyzing the measured data, faults and abnormalities in operation and potential improvements in GSHP system performance were identified. In particular, the patterns of circulation pump operation for the ten GSHP systems were depicted using the measured data and compared across all buildings. Since the GSHP system configuration in all buildings is identical, any deviation from the expected signature pattern indicated a potential issue with the control or operation of the system or the data acquisition system. Such issues were further investigated by on-site inspection, short-term measurements, and interviews with the building energy managers about known operational issues.

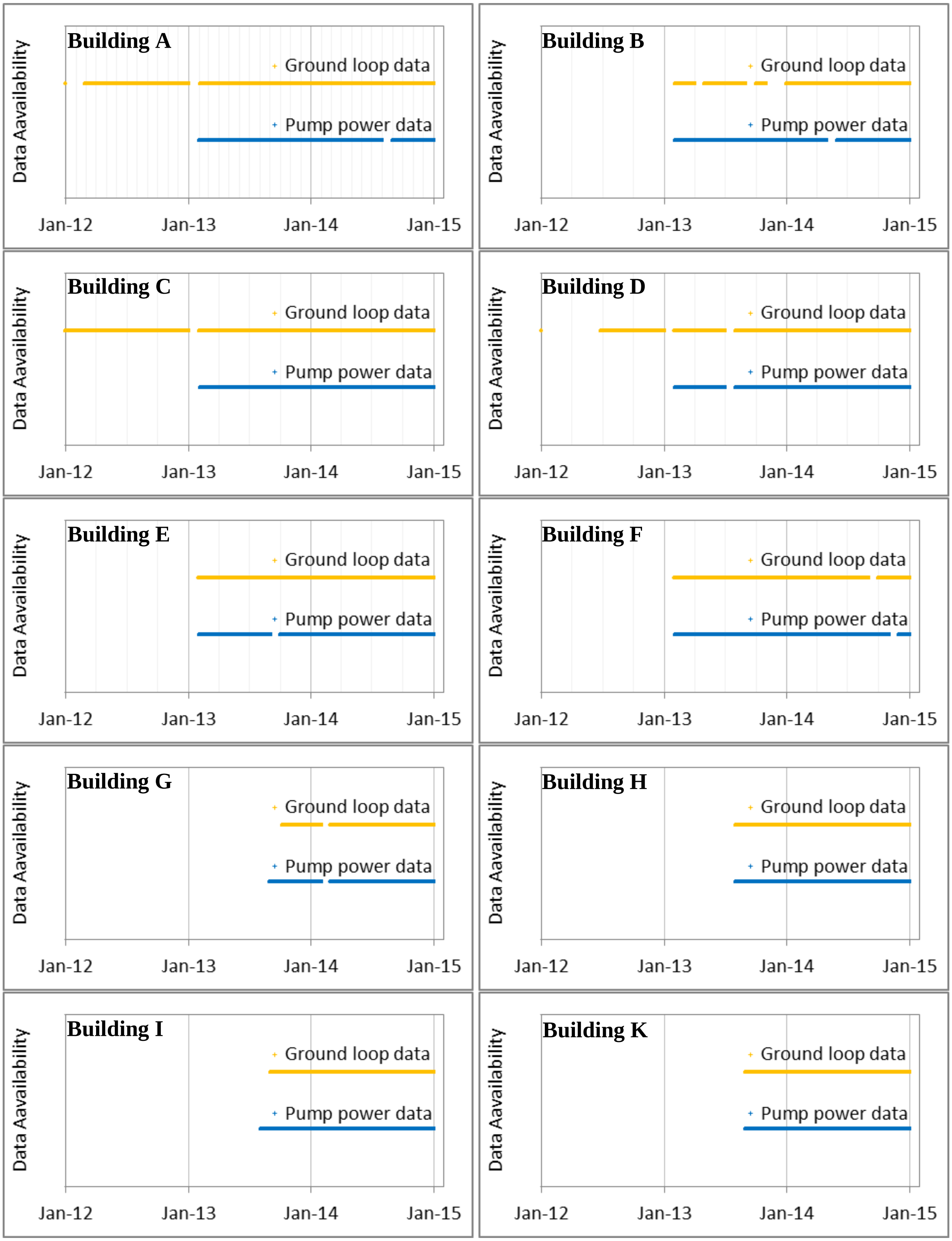


Fig. 8. Data availability.

2.1 SOURCE-SIDE OPERATION

2.1.1 Measured Water Temperature Trend and Flow Rate

For the ten buildings, the ground-loop supply and return water temperature and the coincidental OA temperature from January 2012 through January 2015 are shown in Fig. 9. The ground-loop supply water temperature, which is the temperature of the water entering the heat pump, was as low as 50.0–57.7°F in the winter and as high as 80.2–88.8°F in the summer. Compared with the OA temperature, the ground-loop supply water temperature was relatively stable: it was up to 40°F higher than the OA temperature on the coldest day of the winter and about 10–15°F lower than the OA temperature on the hottest day of the summer. Measured data also shows that the maximum ground-loop supply temperature in a few buildings increased slightly (by about 1–2°F) over the years.

Fig. 10 shows the temperature differential (TD) between the supply and return mains of the common water loop (referred as "ground loop" hereinafter). As shown in Fig. 10, the maximum TD of the ground loop is within 9.5–12.5°F in the summer and 6.4–9.8°F in the winter. Since each individual heat pump usually operates with a 10°F TD in cooling mode and a 6°F TD in heating mode, the measured TD indicates the ground loop may have had a slight underflow in some buildings.

Fig. 11 shows the measured ground-loop flow rate in the ten buildings. The minimum flow rates were as low as 0–0.4 gpm in buildings B and H, but they were higher than 15 gpm in buildings A, C (in 2012 and part of 2013), E, and K. On the other hand, the maximum flow rates of the GSHP systems were all near 100 gpm, except for a few outliers (i.e., a flow rate of around 120 gpm, which is the design maximum flow rate when all the 25 heat pumps are running). As shown in Fig. 12, the high minimum flow rates occurred even when the ground-loop TD was zero (i.e., when no heat pump was running). It was found that the high minimum flow in building C (~30 gpm) was due to several malfunctioning 2-way solenoid valves, which were kept in the open position even when the associated heat pumps were off. The minimum flow rate was reduced to 3 gpm after the solenoid valves were fixed. It is likely that buildings A, E, and K may have the same issue, but that has not been confirmed yet.

Fig. 12 shows scatter plots of the flow rate and TD of the ground loop in each building. In each plot, the pattern revealed by the measured data is also compared with the ideal pattern between the flow rate and TD (indicated by the dashed black lines). As can be seen in Fig. 12, the relationships between the measured flow rate and TD show two different patterns. For buildings A, E, and K, the pattern is more like a "V" shape; the pattern is close to a "U" shape in the other buildings, except building C, which shows both patterns because the malfunctioning solenoid valves were fixed in 2013. The pattern shown in Building B is the closest fit to the ideal pattern. The V shape shows a high minimum flow rate when the TD is zero and a linear increase in TD when the flow rate increases. In contrast, the U shape shows a near-zero minimum flow rate and the TD remaining nearly constant in both heating and cooling modes when the loop flow rate varies within a wide range (as indicated by the two vertical legs of the U shape). Fig. 12 illustrates that anomalous operation of the pumping system can be visually identified by the V-shaped pattern between the flow rate and the TD.

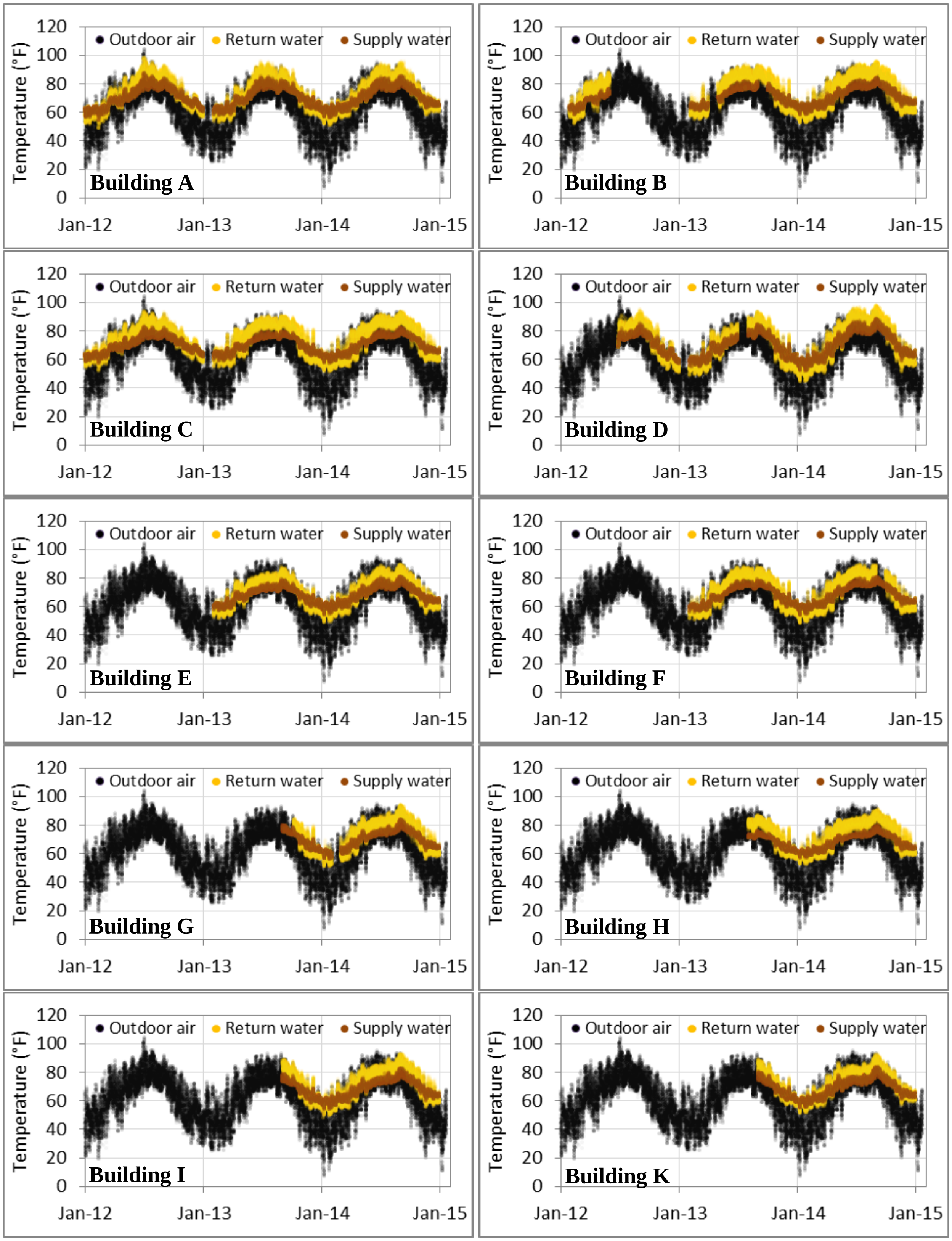


Fig. 9. Ground-loop supply and return water temperature.

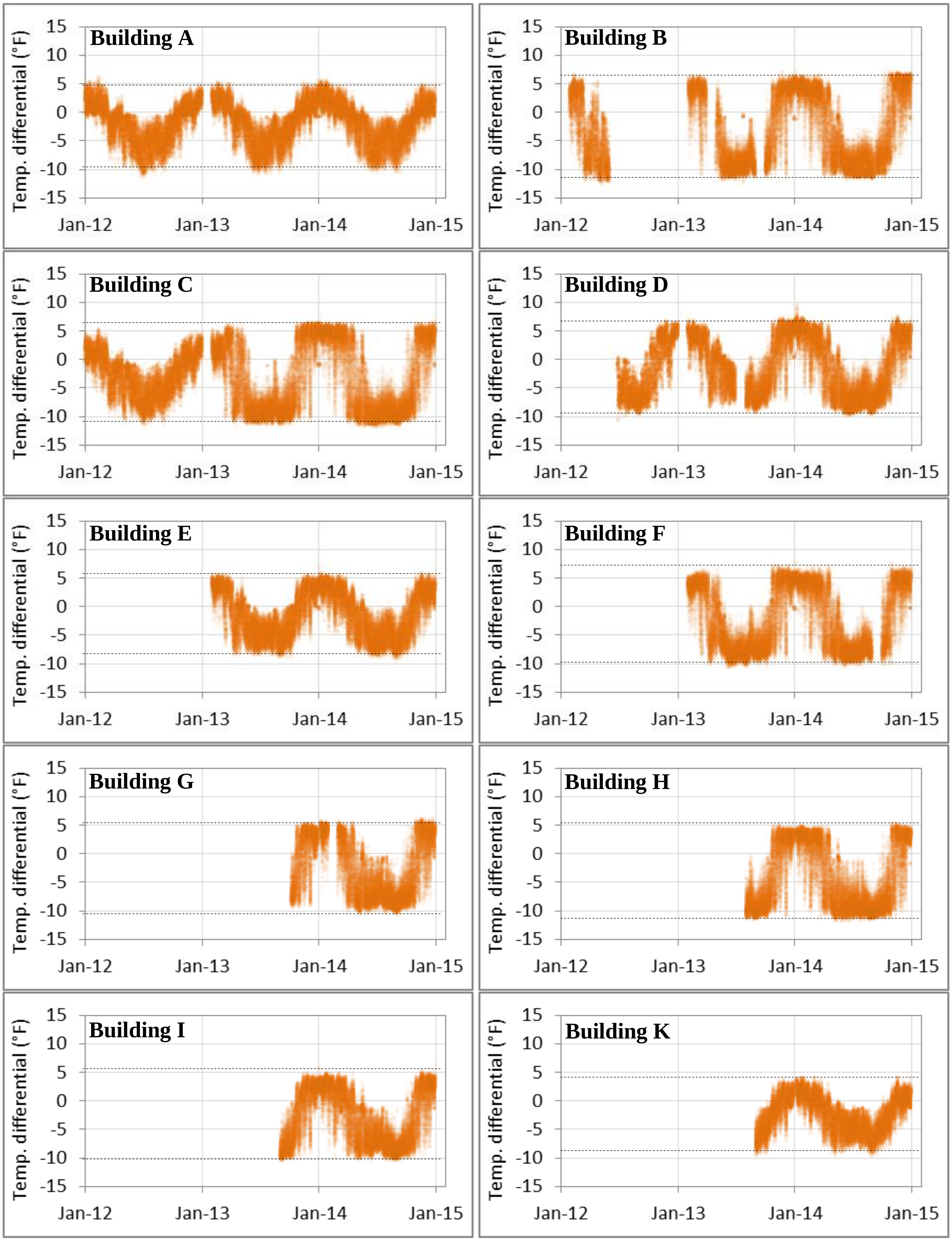


Fig. 10. Ground-loop temperature differential.

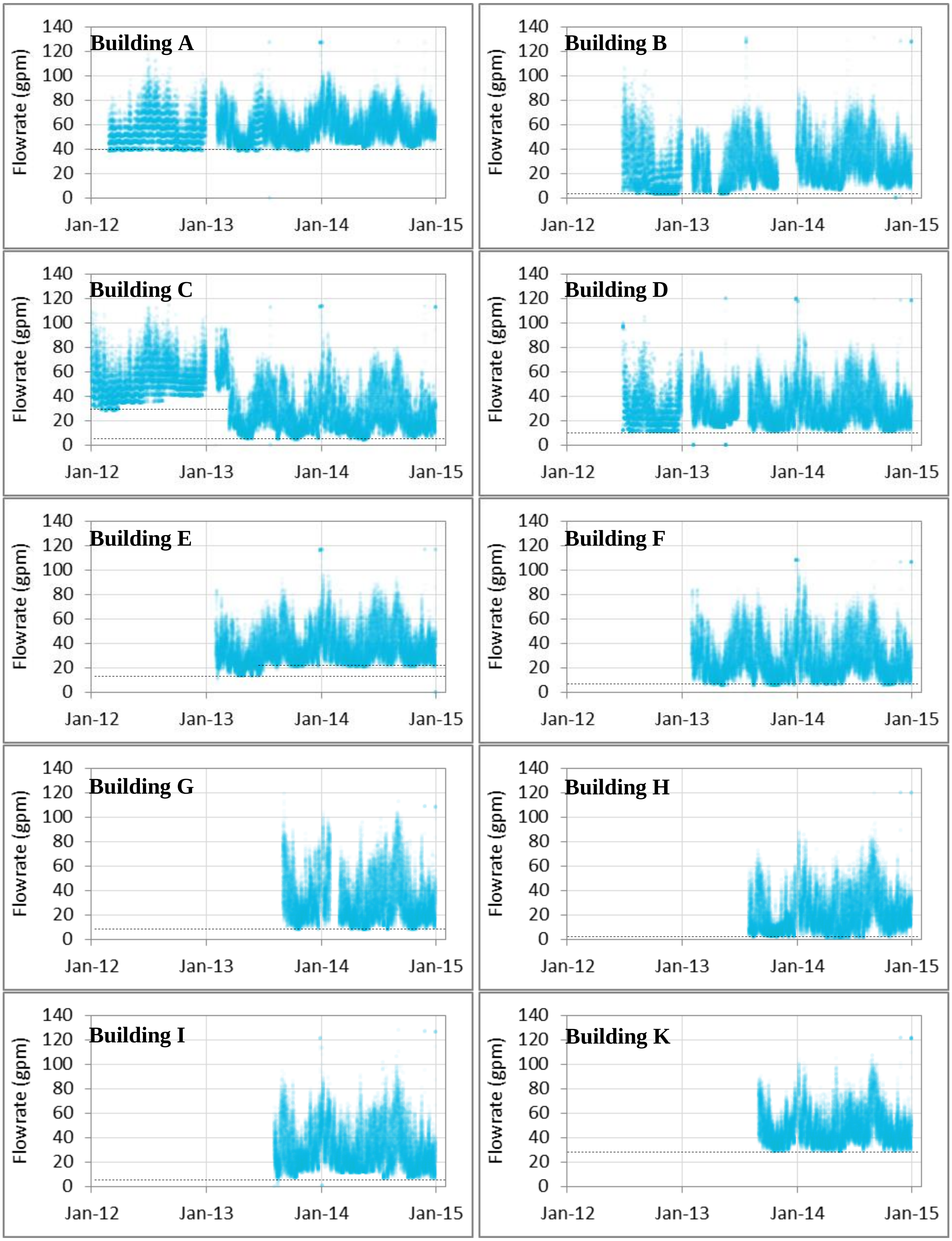


Fig. 11. Ground-loop water flow rate.

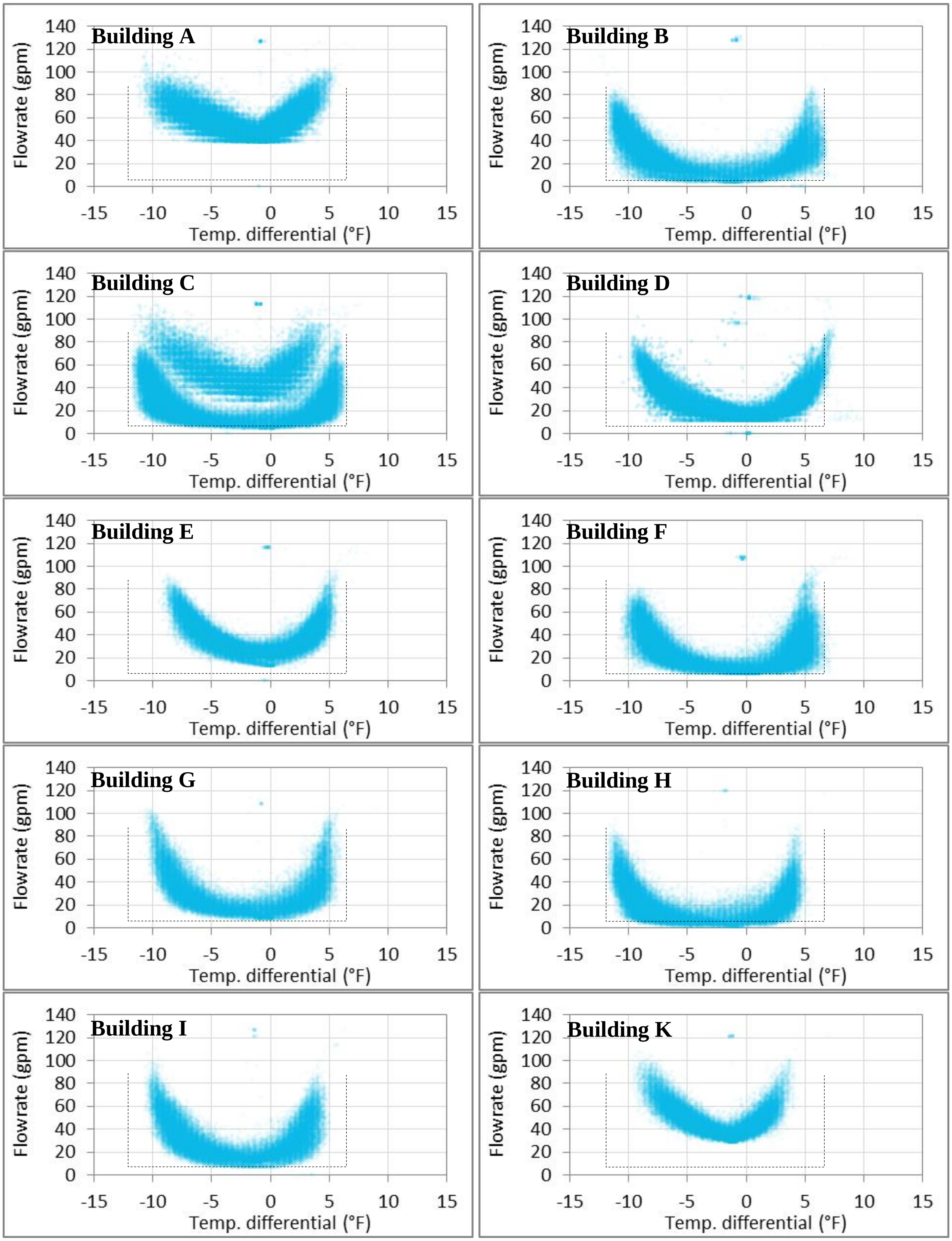


Fig. 12. Ground-loop flow rate versus temperature differential.

2.1.2 Pumping Power Consumption

The pumping power of each GSHP system is a measurement of the variable-speed driver of the pump. Fig. 13 shows the scatter plots of the measured loop flow rate and the pumping power at each building. In each plot, the trend line of the measured data is also shown along with an approximated correlation between the flow rate and the pump power. As can be seen in these plots, the actual flow–power relationship does not exactly follow the pump Affinity Law (i.e., the pump power will change by the cube of a change in the flow rate, expressed as $Power \propto flow^3$), and it is more close to $Power \propto flow^{2.5}$. It is because that the pump is controlled to maintain a constant pressure differential across the farthest heat pump in the hydronic piping system (i.e., the pressure drop at this part of the piping system is independent to the change of the flow rate). Building K shows several different patterns between the flow rate and the pumping power. This building is the latest being commissioned and it is likely that the different patterns are resulting from adjustments of valves in the hydronic system.

2.1.3 Heat Transfer in Ground Loop

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

$$QGL = k * FMW * \frac{(TMWS - TMWR)}{1000} \tag{1}$$

where

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

Fig. 14 shows the time series plots of the daily ground-loop heat transfer rate for each building during the monitoring period. The heat extraction rate during heating operation is shown with positive values (red bars), and the heat rejection rate during cooling operation is shown with negative values (blue bars). As shown in Fig. 14, the ground-loop heat transfer in each building has a similar pattern. In 2014, the maximum ground-loop heat extraction rate was about 4 MMBTU/day at all the buildings except buildings H, I, and K, where the maximum heat extraction rates were about 3 MMBTU/day. The maximum ground loop heat rejection rate was about −6 MMBTU/day at all buildings except buildings D, E, and F, where the maximum heat rejection rates were about −4 MMBTU/day. The maximum hourly heat extraction rate of the ground loop varied within 374–450 kBtu/h at different buildings. The maximum hourly heat rejection rates varied in a bigger range, within 400–737 kBtu/h at different buildings. During 2014, the annual heat extraction amounted to 84–245 MMBtu (lower in buildings G, H, I, and K), and the heat rejection amounted to 422–644 MMBtu at different buildings (lower in buildings D, E, and F). The ratio of annual heat rejection to annual heat extraction was 2.0-7.6 (highest in building K and lowest in buildings D and F). As discussed in previous Section 2.1.1, the GHX leaving water temperature in some buildings increased slightly (by about 1–2°F) over the years. It is likely due to the excessive heat rejection over heat extraction on an annual basis. It can also be observed from Fig. 14 that the ground-loop heat rejection rate at all buildings dropped during the summer break when those buildings were not fully occupied. It appears that buildings G, H, I, and K had smaller heat rejection rates before the summer break than other buildings, which had double peaks before and after the summer break.

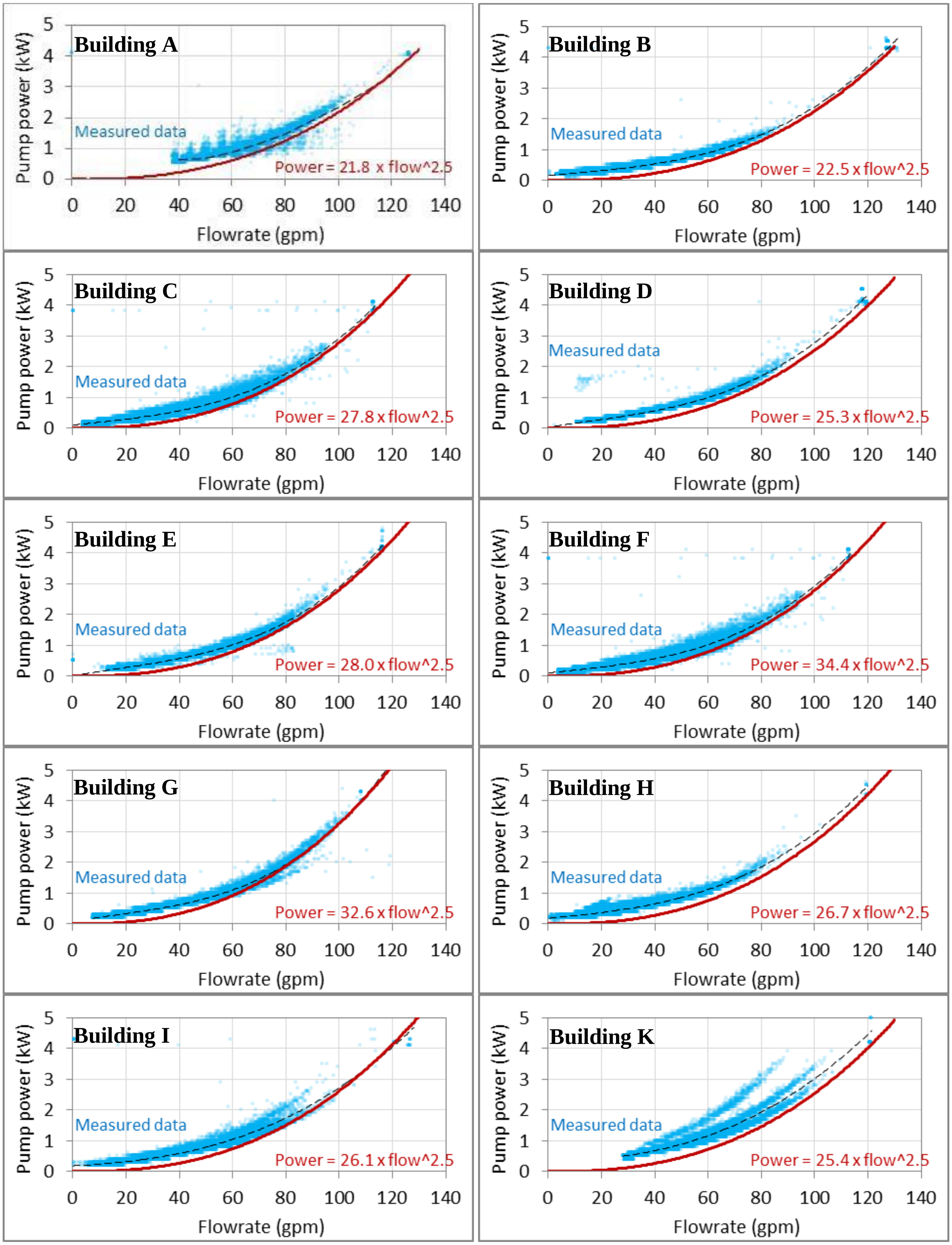


Fig. 13. Pump power consumption versus ground-loop water flow rate.

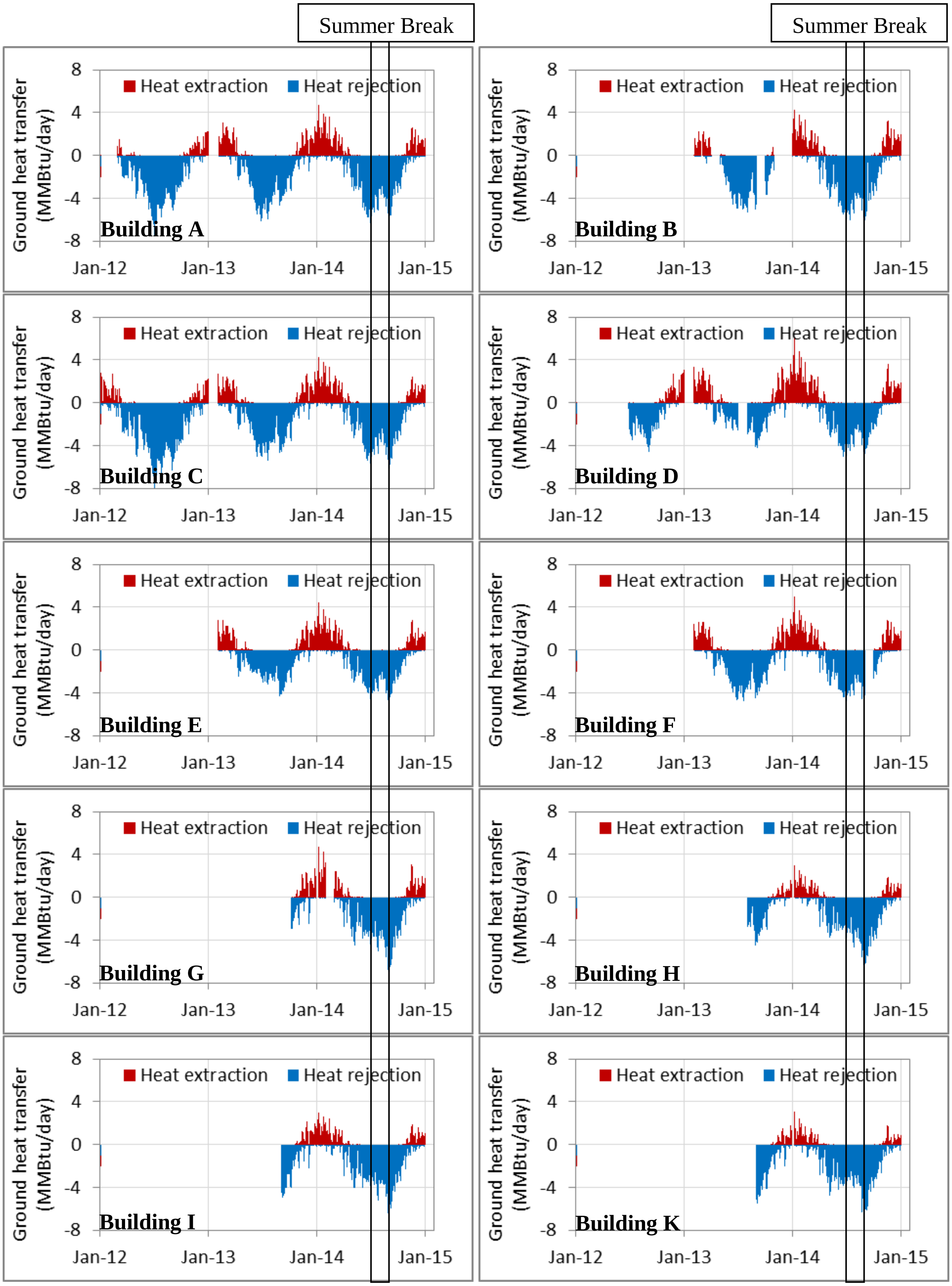


Fig. 14. Ground-loop daily heat transfer rate.

2.2 LOAD-SIDE OPERATION AND GSHP SYSTEM EFFICIENCY

The demonstrated GSHP system is in a distributed configuration consisting of 25 heat pumps. Unfortunately, there was no measurement of the heating/cooling output and associated power consumption of each individual heat pump (except for six heat pumps in building B). Given this limitation, the performance of the heat pumps and of the entire GSHP system was evaluated based on available measured data for the ground loop and the catalog data for the heat pump units. In this evaluation, the 25 individual heat pumps were approximated as one aggregated heat pump. This approximation neglects the simultaneous heating and cooling operation of various heat pumps and thus could under-estimate the total heating and cooling loads of the building and the associated heat pump power consumption. However, since simultaneous heating and cooling occurs mostly at shoulder seasons when the building heating or cooling loads are small, the error in the calculated annual building loads and associated power consumptions that results from neglecting the simultaneous heating and cooling would also be small.

With this approximation, the overall efficiency of the GSHP system was determined with the approach shown in Fig. 15. This approach first calculates the heating and cooling energy provided by the GSHP system, and the associated heat pump power consumption, with measured data and other available information, at each 15 minute time interval. From the measured ground-loop supply and return water temperature (TGLS and TGLR) and flow rate (FGL), heat extracted from or rejected to the ground loop (QGL_h or QGL_c) was calculated (as described in Section 2.1.3). The operational efficiency of the heat pump in heating and cooling modes ($COPh_{eqp}$ and $COPc_{eqp}$) was determined from the manufacturer's catalog data with the measured TGLS and typical values for entering air temperature (dry-bulb temperature (TDBS) for heating mode; and wet-bulb temperature (TWBS) for cooling mode). Then heat pump power consumption in heating mode (WHP_h) was calculated with QGL_h and $COPh_{eqp}$, and the power consumption in cooling mode (WHP_c) was calculated with QGL_c and $COPc_{eqp}$. Then, based on the heat balance of the vapor compression cycle of the heat pump, the heating delivered to the building (QHP_h) was calculated with QGL_h and WHP_h; and the cooling delivered to the building (QHP_c) was calculated with QGL_c and WHP_c.

Finally, the annual operational efficiencies of the GSHP system in the heating and cooling modes (COPhsys and COPcsys) were calculated as the ratios between the cumulative heating and cooling delivered and the cumulative power consumption of the heat pumps and the central circulation pump in heating and cooling modes, respectively, as expressed in Eqs. (2) and (3).

$$\mathrm{COPh}_{\mathrm{sys}} = \frac{\sum_{i=1}^{nh} \mathrm{QHP_h}(i)}{3.413 * \sum_{i=1}^{nh} [\mathrm{WHP_h}(i) + \mathrm{WLP_h}(i)]} \tag{2}$$

$$\mathrm{COPc}_{\mathrm{sys}} = \frac{\sum_{i=1}^{nc} \mathrm{QHP_c}(i)}{3.413 * \sum_{i=1}^{nc} [\mathrm{WHP_c}(i) + \mathrm{WLP_c}(i)]} \tag{3}$$

where WLP_h and WLP_c are the power consumption of the central variable-speed pump when the GSHP system runs in heating mode (i.e., ground loop extracts heat) and cooling mode (i.e., ground loop rejects heat), respectively; *i* is the counter of time intervals; *nc* and *nh* are the total number of time intervals in cooling and heating mode, respectively.

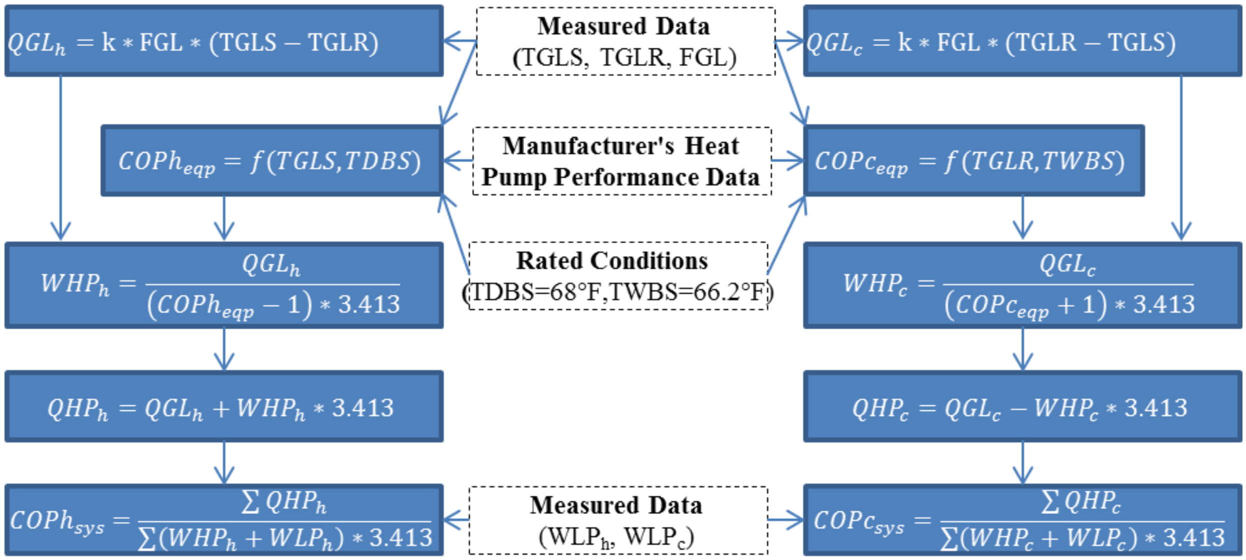


Fig. 15. Determining GSHP system efficiency.

2.2.1 Operational Efficiency of Heat Pump Equipment

The operational efficiency of the heat pump, indicated by the coefficient of performance (COP) for heating and cooling, was determined based on the heat pump manufacturer's performance data (plotted in Fig. 16), the measured ground-loop supply temperature, and the design water flow rate of the heat pump.

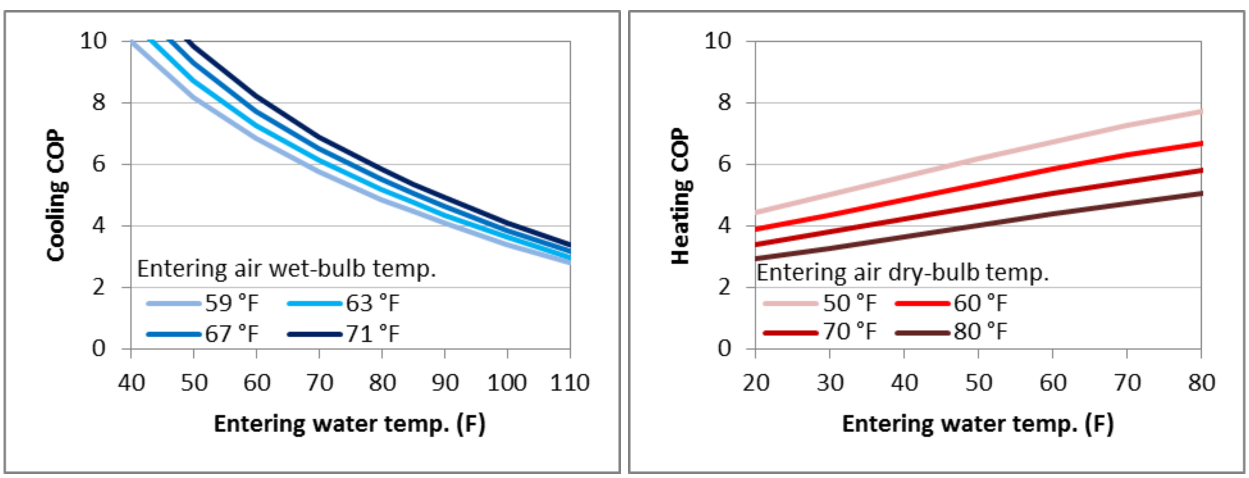


Fig. 16. Heat pump efficiency curves in heating and cooling modes.

Building energy simulation programs usually use the energy input ratio (EIR), which is the reverse of COP, for energy calculations. EIRs at conditions other than the rating condition can be determined based on the EIR at the rating condition and a correction factor, which is a function of the entering water temperature (*T1*) and the entering air temperature (*T2*), as expressed in Eq. (4). The coefficients (C_0 through C_5) in Eq. (4) for the heating and cooling modes are derived from the data shown in Fig. 16 and listed in Table 2.

$$EIR = EIR_{rated} * f(T_1, T_2) = EIR_{rated} * (C_0 + C_1 T_1 + C_2 {T_1}^2 + C_3 T_2 + C_4 {T_2}^2 + C_5 T_1 T_2) \quad (4)$$

where, T_1 is the measured ground-loop supply temperature (TGLS); and T_2 is 66.2°F for cooling mode and 70°F for heating mode.[3]

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

Coefficient	Cooling EIR	Heating EIR
C0	0.966869	0.622597
C1	0.016775	−0.008322
C2	0.000121	0.000063
C3	−0.023768	0.005693
C4	0.000202	0.000079
C5	−0.000254	−0.000085

Fig. 17 shows the heat pump heating and cooling efficiencies ($COPh_{eqp}$ and $COPc_{eqp}$) versus OA temperature at the ten buildings. For 2014, aggregating the heating or cooling delivered and the associated power consumed by the heat pump, the average COPs of the heat pump at the ten buildings were 5.1–5.4 for heating and 5.3–6.1 for cooling.

2.2.2 Heat Pump Power Consumption

Fig. 18 shows the scatter plots of the calculated heat pump power draws in heating and cooling modes (WHP_h and WHP_c) versus OA temperature. As shown in Figure 18, the heat pumps changed their operation mode when the OA temperature was at about 60°F. The heat pump power consumption in cooling mode increased almost linearly with the increase in OA temperature, a pattern that was reversed for heating operation. The peak heat pump power draw (in both heating and cooling modes) was 15–25 kW at different buildings. During 2014, the total heat pump power consumption was between 30,091 and 40,467 kWh per building.

2.2.3 Pump Power Fraction

Pumping performance is evaluated as the ratio of the pumping power consumption relative to the total GSHP system power consumption (referred as "pumping power fraction"). Fig. 19 shows the scatter plots of the pumping power fraction against OA temperatures. The pumping power fraction ranged between 8 and 60%, higher at moderate temperatures. Aggregated by month, the pumping power fraction ranged from as low as 6.2–13.4% in building B to as high as 13.4–35.5% in buildings A and K. Buildings A and K had a much higher minimum flow rate in the ground loop than did the other buildings.

2.2.4 Energy Delivered by Heat Pump

Fig. 20 shows that during 2014, the heat pumps delivered a total of 360–547 MMBtu of cooling and 104–305 MMBtu of heating per building.

[3] These values are the heat pump performance rating conditions for entering air in cooling and heating modes, respectively.

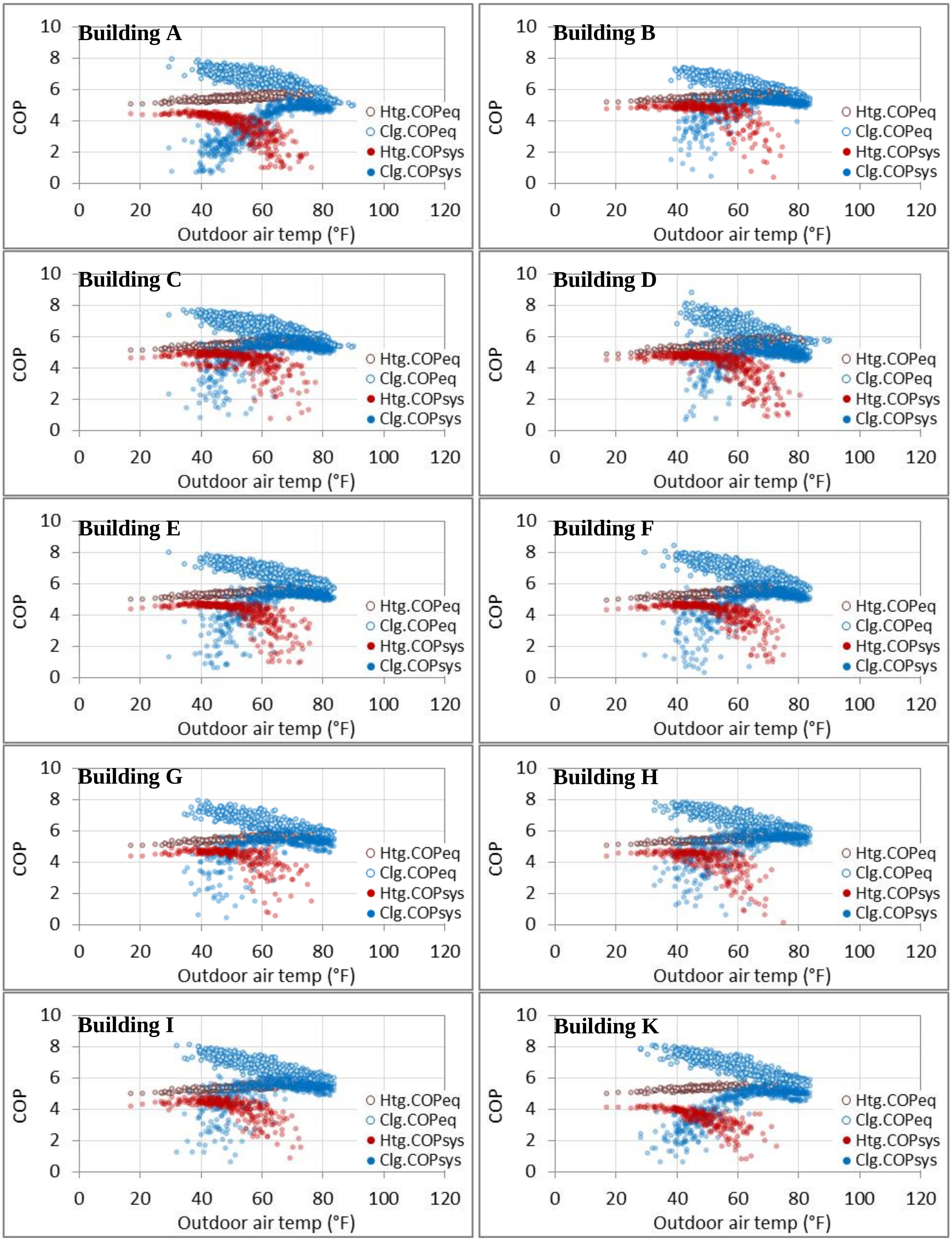


Fig. 17. Heat pump equipment and GSHP system COP in heating and cooling modes.

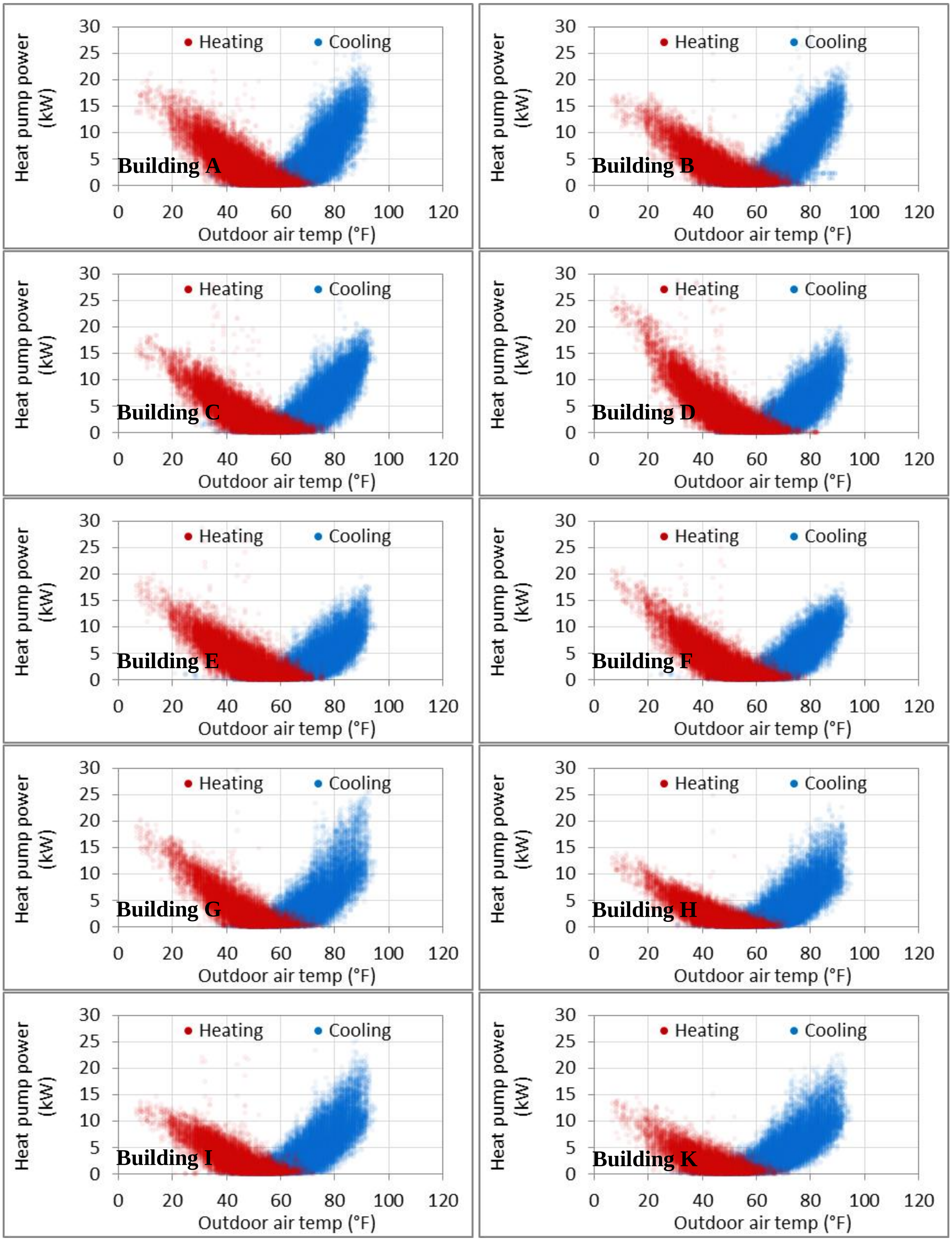


Fig. 18. Heat pump power consumption in heating and cooling modes.

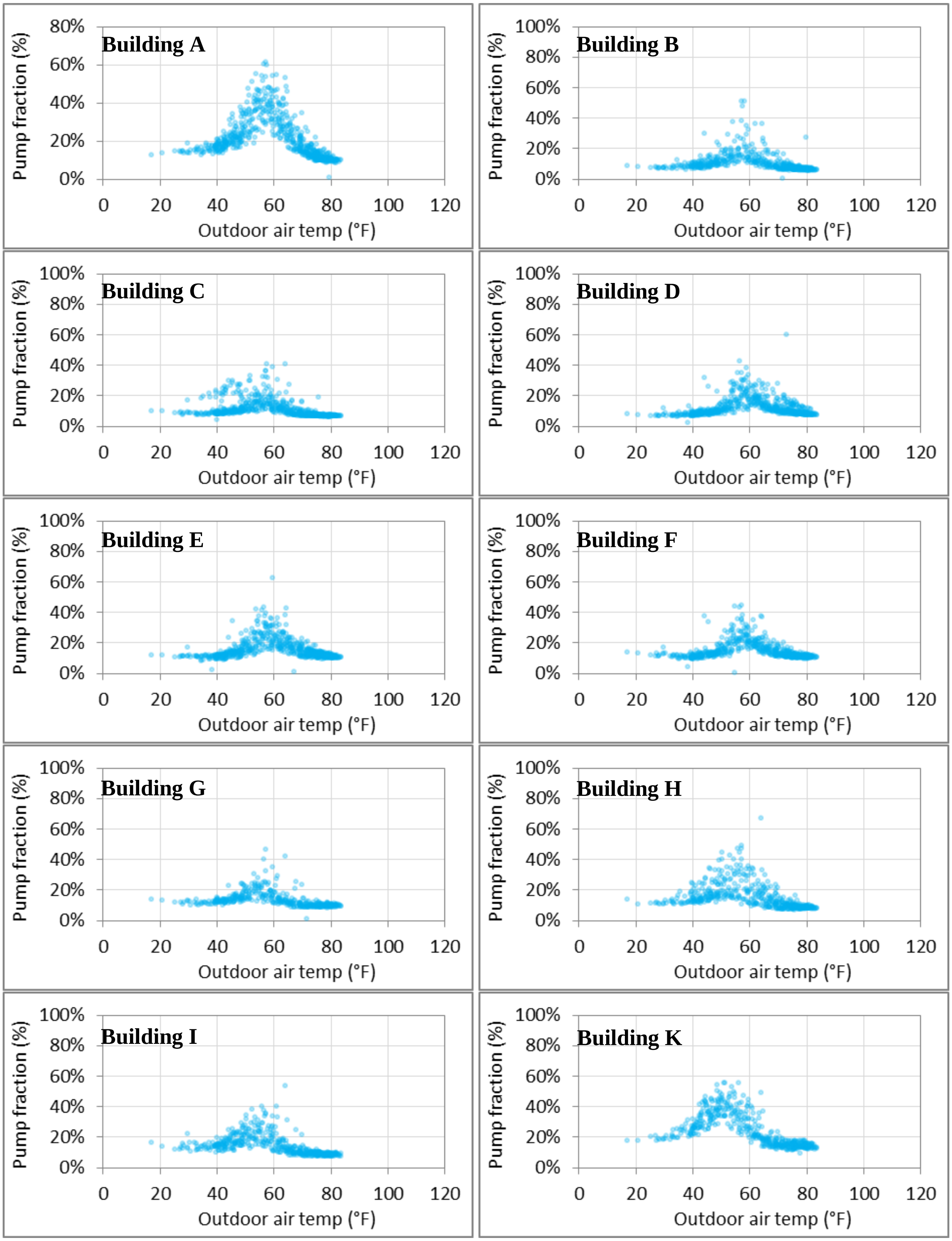


Fig. 19. Ground-loop pump power fraction versus outdoor air temperature.

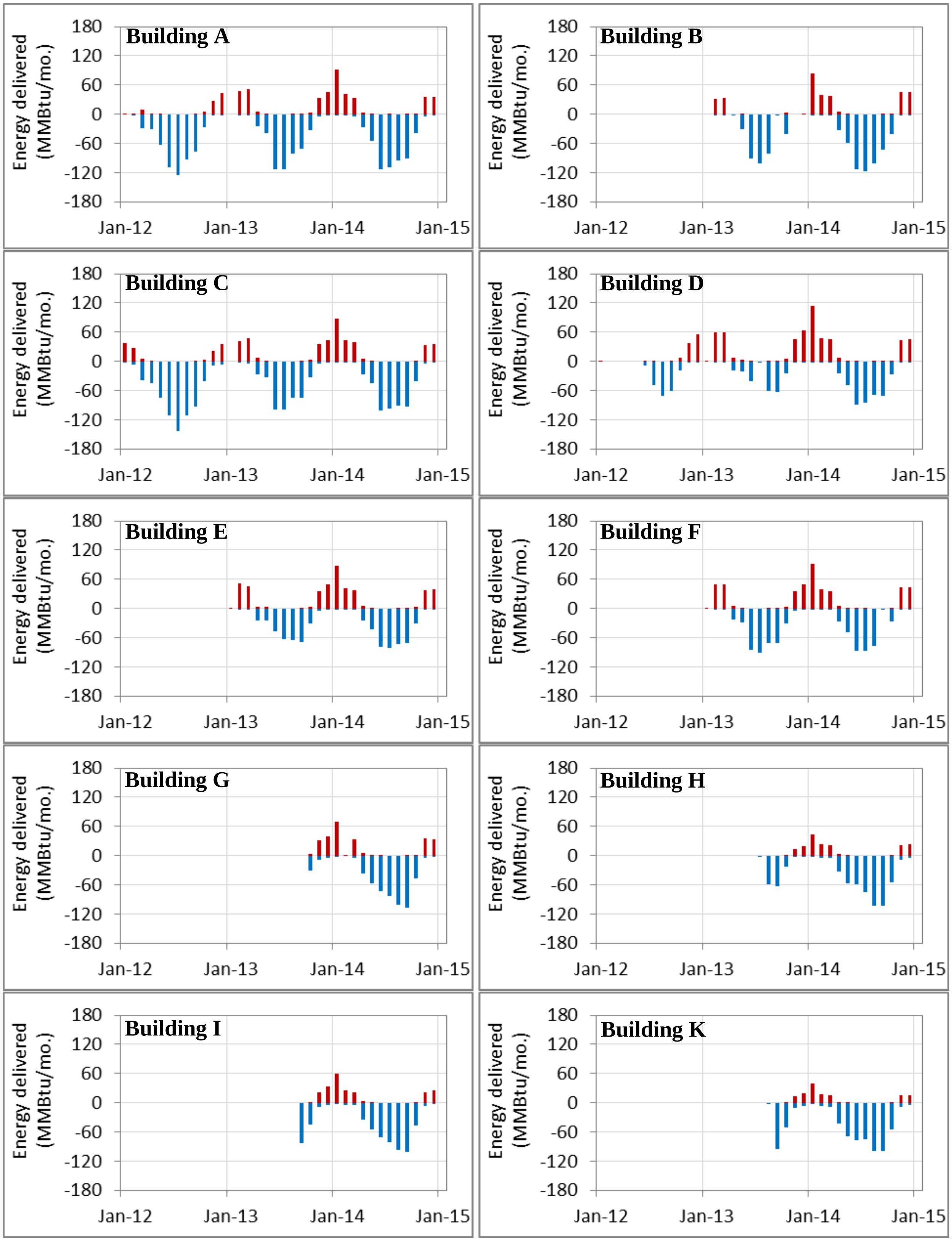


Fig. 20. Heating and cooling energy delivered by heat pump.

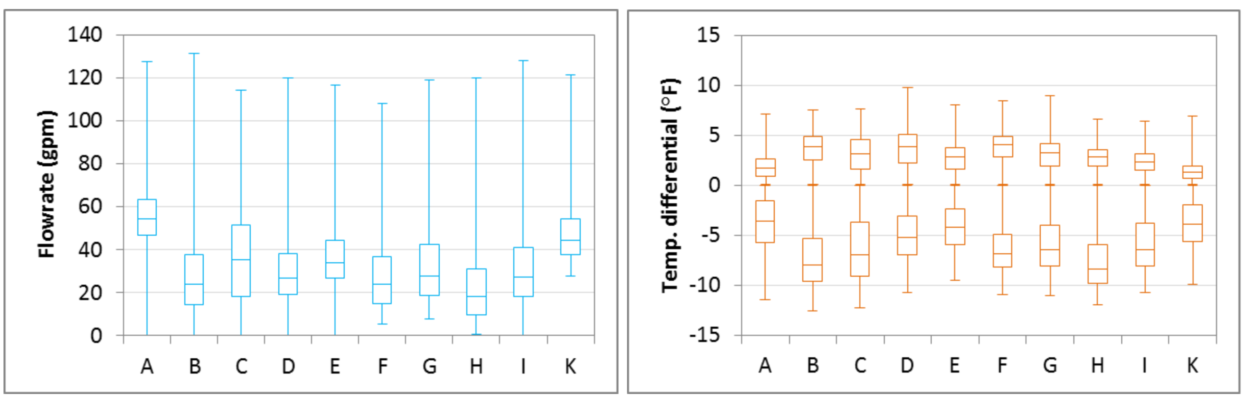
2.2.5 Overall Efficiency of GSHP System

Fig. 17 shows the overall efficiency of the GSHP system compared with the efficiency of the heat pumps themselves in both heating and cooling modes. As expected, the GSHP system efficiency was lower than the individual heat pump efficiency at moderate temperatures when building heating/cooling loads were small and the pumping energy use was high. The difference between the GSHP system efficiency and the heat pump efficiency diminished at temperatures with substantial heating/cooling loads (except in buildings A and K), which demonstrates low pumping energy consumptions (i.e., good pumping performance). The calculated average COP of the GSHP system during year 2014 ranged within 4.8 - 5.5 for cooling and 3.8 - 4.9 for heating.

2.3 SUMMARY

This section compares the performance among the GSHP systems at the 10 buildings in 2014, including the ground-loop flow rate and TD, pumping operation, and overall GSHP system COP.

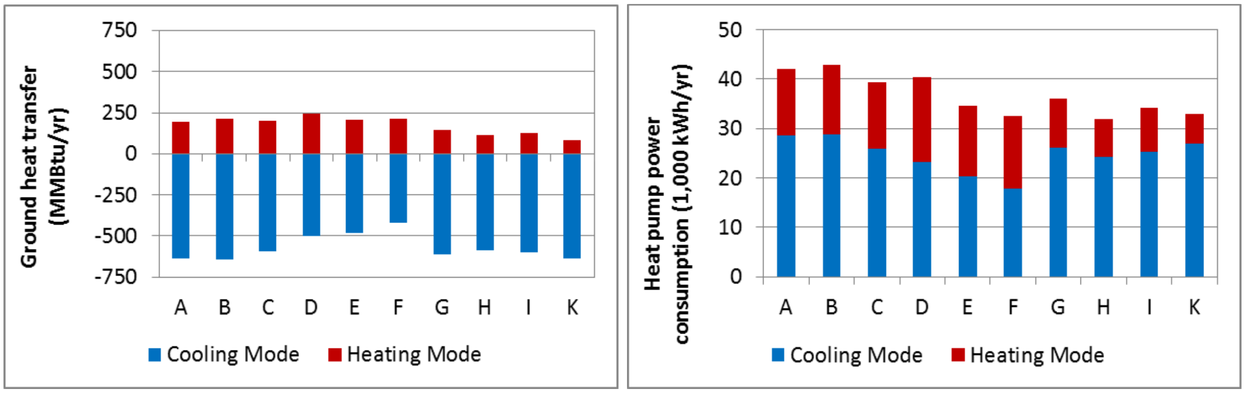
Fig. 21 compares the summary statistics of ground-loop TD and flow rate (i.e., minimum, first quartile, median, second quartile and maximum values) among buildings. Median flow rates were 18–35 gpm (except for 54 and 45 gpm in buildings A and K). Temperature differentials were smaller in heating mode (positive values) than in cooling mode (negative values). Median TDs were 5.3–8.4°F in cooling mode (less than 4°F in buildings A and K) and 2.4–4.1°F in heating mode (1.7 and 1.3°F in buildings A and K). The high flow rates and small TDs in the ground loops of buildings A and K indicate poor pumping performance at these building. It may be due to malfunctioning 2-way valves at some of the heat pump units; if so, the pumping performance can be improved by simply replacing those valves.



(a) (b)

Fig. 21. Comparison of ground-loop characteristics among buildings: (a) flow rate statistics and (b) temperature differential statistics.

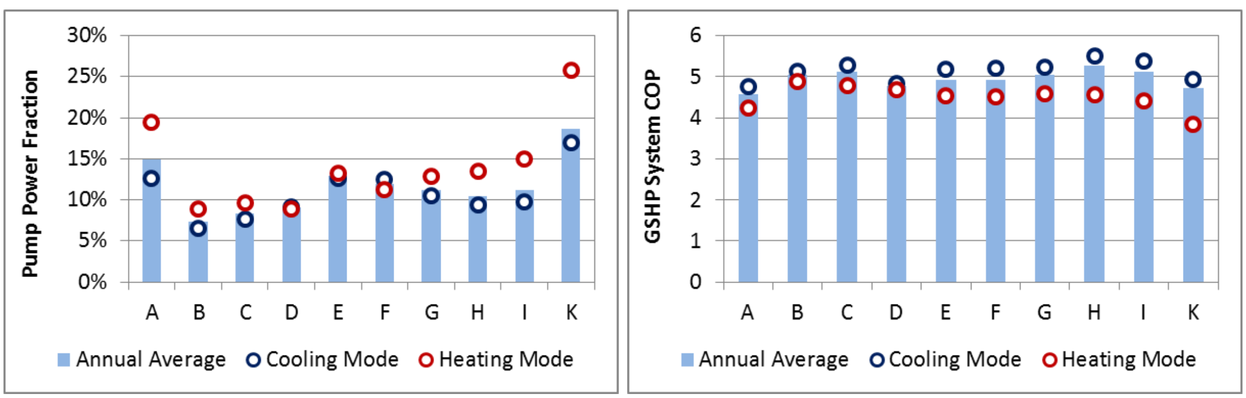
Fig. 22 compares the annual total values of ground heat transfer and heat pump power consumption among buildings. In all 10 buildings, the GSHP system rejected more heat to the ground than it extracted from the ground. The ratio between the heat rejection and heat extraction varied from 2.0 to 7.6 among the 10 buildings. Buildings D, E, and F had smaller ground heat transfer loads and heat pump power consumption totals than other buildings, which suggests a lower occupancy level or more energy-conscious thermostat settings in these buildings.



(a) **(b)**

Fig. 22. Comparison of (a) ground heat transfer and (b) heat pump power consumption among buildings.

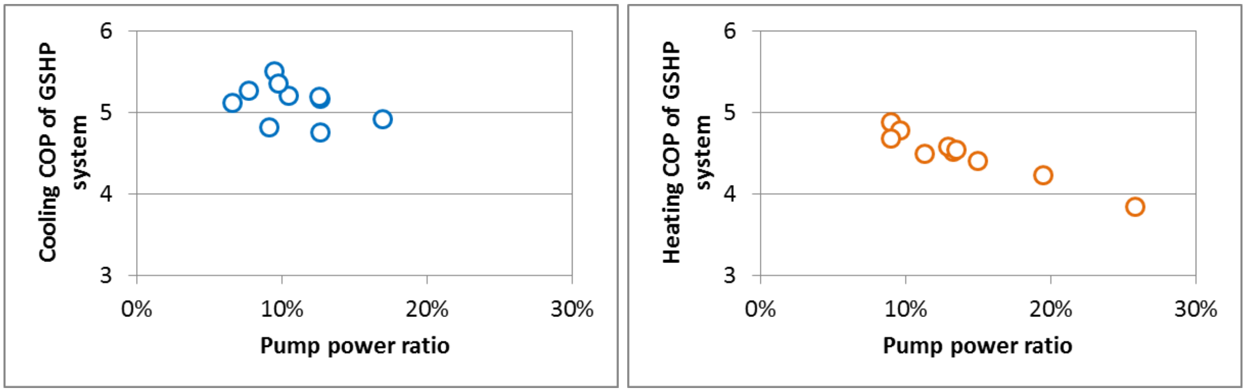
Fig. 23(a) compares the annual pumping power fraction among buildings. The pumping power fractions are averaged for the cooling and heating modes, as well as for the entire year. Most buildings had an annual average pumping power fraction between 10.5 and 13%. Buildings B, C, and D had lower pumping power fractions of 7.4, 8.4, and 9%, respectively. Buildings A and K had the highest pumping power fractions of 15 and 18.7%, respectively. In general, the pumping power fraction in heating mode was slightly higher than in the cooling mode. Fig. 23(b) compares the annual average COP among these GSHP systems. The annual GSHP system COPs ranged between 4.8 and 5.5 for cooling and 3.8 and 4.9 for heating. Accounting for both heating and cooling operations, the annual COPs of the GSHP systems in buildings B, C, H, and I exceeded 5.0.



(a) **(b)**

Fig. 23. Comparison of (a) pumping power fraction and (b) GSHP system COP among buildings.

Fig. 24 shows how the GSHP system COP relates to the pumping power fraction in cooling and heating modes. Apparently, the COP of the GSHP system shows a correlation with the pumping power fraction (i.e., the COP decreases with an increase in the pumping power faction) and such a relationship is more clearly demonstrated in heating mode.



(a) **(b)**

Fig. 24. GSHP system COP versus pump power ratio in (a) cooling mode and (b) heating mode.

3. COMPARISON BETWEEN PRE- AND POST-RETROFIT ENERGY USE

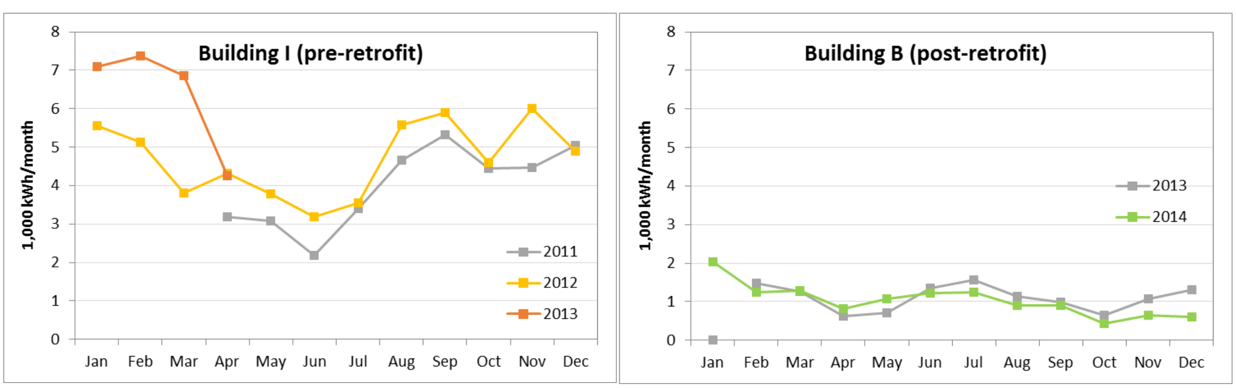
To determine the energy savings realized in Furman's North Village complex, pre- and post-retrofit energy use was analyzed. Two sources of data were used in this analysis:

1. After Building B was retrofitted with the new GSHP system, heat pump electricity use was measured at 6 of the 24 apartments of Building B. At approximately the same time, heat pump electricity use was also measured at 6 of the 24 apartments of Building I, which was still being served by the original ASHPs. Measured data from Building I could serve as a pre-retrofit baseline for Building B, since they have identical floor plans and a similar orientation.

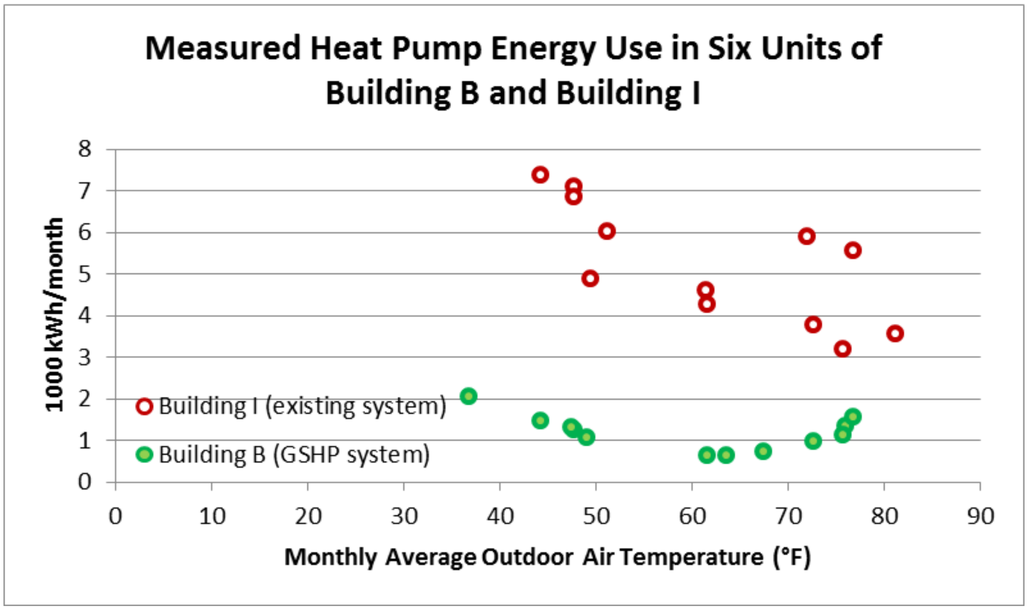
2. Utility bills for the different buildings of North Village are available starting during the months between October 1999 and August 2001 as they were completed. The GSHP system retrofit occurred between 2012 and February 2013 for buildings A through F, and between 2013 and September/October 2014 for the remaining four buildings. Therefore, for comparing pre- and post-retrofit utility bills, we consider 2002–2010 the pre-retrofit period and 2014 the post-retrofit period.

3.1 MEASURED HEAT PUMP ENERGY USE

Fig. 25 (a) compares the monthly heat pump power consumptions of six apartments in buildings B and I over multiple years when data are available. The two buildings are nearly identical but use different HVAC systems: building B uses the GSHP system, while building I uses the existing 13–15 year old ASHPs with supplemental electric heat. Fig. 25 (a) shows that the maximum monthly electricity use of the six ASHPs in building I was around 7,500 kWh and the minimum monthly electricity use was around 2,000 kWh. On the other hand, the maximum monthly electricity use of the six GSHP units in building B was 2,000 kWh. Fig. 25 (b) presented the weather-normalized comparison between the monthly heat pump power consumptions in the two buildings. Accounting for the pumping energy associated with the GSHP units, the GSHP system consumed 60% less electricity annually than the existing ASHPs.



(a)



(b)

Fig. 25. Measured heat pump energy use in six apartment units of buildings B and I: (a) monthly heat pump power consumption; (b) comparison of weather-normalized monthly heat pump power consumption.

3.2 UTILITY BILL ANALYSIS

Utility data for all the 10 buildings were obtained beginning in 2000. Fig. 26 shows the weather-normalized monthly electricity use for all buildings combined for 2014 (post-retrofit) and 2010 (pre-retrofit). Note that the electricity use includes all the electric power consumption in the building, not only for the HVAC system but also for lighting, appliances, and other uses. During the post-retrofit period, the electricity use in the 10 buildings was about 36% lower than electricity use during the pre-retrofit period, during months when the OA temperature was lower than 40°F. The electricity savings were smaller (10–20%) when the OA temperature was 50–70°F and varied between 0 and 50% when the OA temperature was higher than 70°F. The lower electricity use during warm-weather months in 2010 is thought to result from some buildings being closed then for summer break. Overall, the GSHP retrofit has resulted in a 27.3% reduction in the annual electricity use of the 10 buildings. The achieved electricity savings is 715,384 kWh/year, which has a value of $42,923 based on the average utility rate of $0.06/kWh provided by the university, or $65,815 if the average electricity rate in South Carolina of $0.092/kWh is applied.

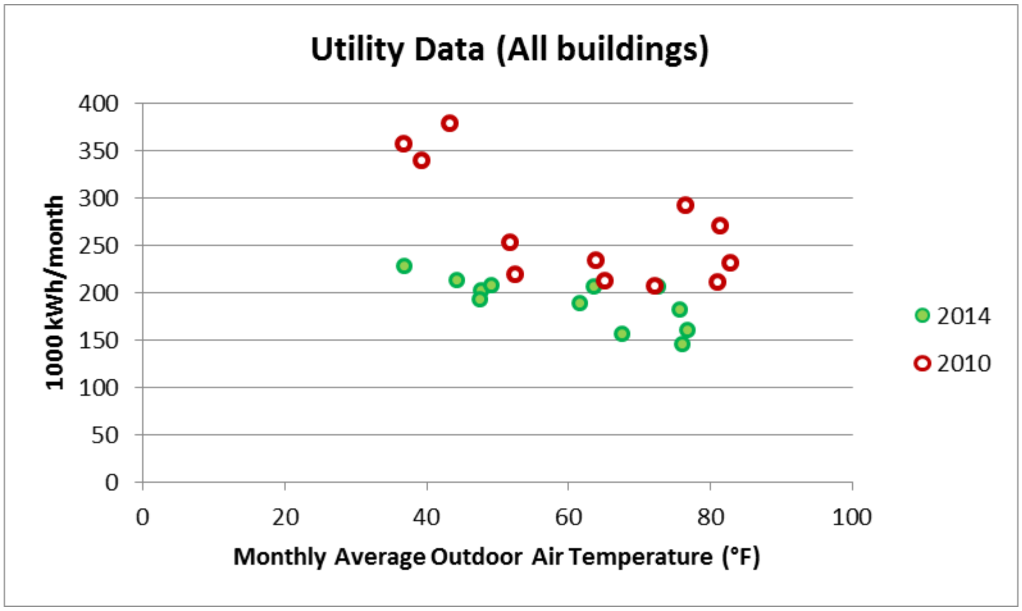


Fig. 26. Weather-normalized total electricity use for all buildings in 2010 and 2014.

Table 2 compares the combined monthly electricity use at the 10 buildings during the pre- and post-retrofit periods. Fig. 27 shows both the absolute values and the percentages of the electricity savings in each month. High savings occurred in January (36.5%), May (40.9%), and December (38.5%), when either the GSHP system replaced substantial resistance heat operation (in January and December) or the space-cooling load peaked (in May). Low savings occur during summer breaks (June through August) and the shoulder seasons when low occupancy or mild weather conditions resulted in both the ASHP and GSHP systems being a smaller fraction of the total building energy consumption.

Table 3. Combined monthly pre- and post-retrofit electricity use of the 10 buildings

Month	Avg Amb. TAO (F)	All Buildings Baseline (kWh)	GHP (kWh)	Savings (kWh)	Savings %
Jan	42.4	291,789	185,181	106,608	36.5%
Feb	45.1	247,950	194,102	53,848	21.7%
Mar	53.7	225,718	166,439	59,279	26.3%
Apr	62.4	194,654	142,494	52,159	26.8%
May	69.6	204,838	121,140	83,698	40.9%
Jun	76.8	141,063	132,562	8,501	6.0%
Jul	78.8	173,437	137,436	36,001	20.8%
Aug	78.4	197,929	157,221	40,707	20.6%
Sep	72.3	242,906	184,252	58,654	24.1%
Oct	62.2	227,320	163,378	63,942	28.1%
Nov	52.5	216,928	161,899	55,029	25.4%
Dec	44.7	251,680	154,721	96,958	38.5%
Annual		2,616,209	1,900,825	715,384	**27.3%**

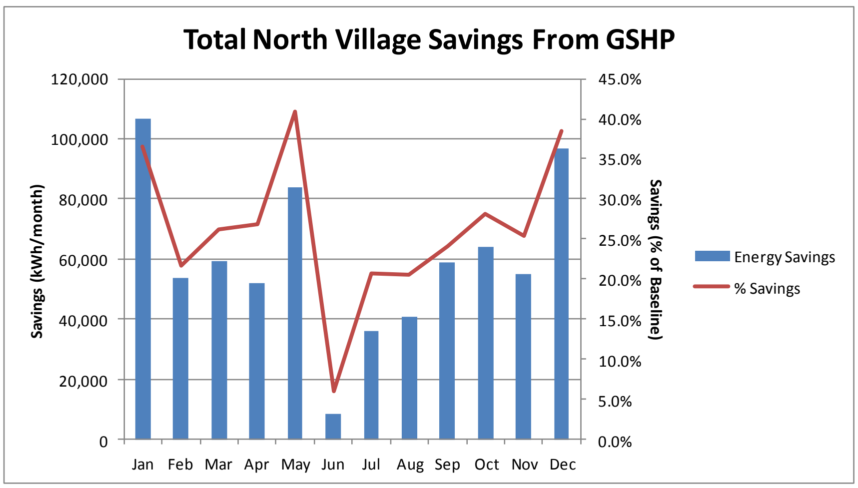


Fig. 27. Electricity savings in North Village after the GSHP retrofit.

4. COMPARISON OF GSHP SYSTEM WITH BASELINE SYSTEM

To determine energy savings and other benefits of the GSHP retrofit at Furman University, an energy simulation model was developed for Building B and the installed GSHP system, calibrated against measured data, and re-simulated with a baseline HVAC system. The measured performance of the GSHP system was then compared with the simulation-predicted baseline system performance to calculate the energy savings, operating cost savings, and emissions reduction benefits that could be achieved with respect to the baseline system. The following sections describe the development of the energy simulation model, its calibration against the available measured data, the baseline HVAC system, and the performance comparison between the GSHP and the baseline system.

4.1 ENERGY SIMULATION MODEL

An eQUEST energy model for Building B was developed based on the as-built design documents for the building and the GSHP system. eQUEST is a widely used building energy simulation program, which is powered by the latest development of the DOE-2 program and has improved capability to simulate various GSHP systems (Liu and Göran 2008).

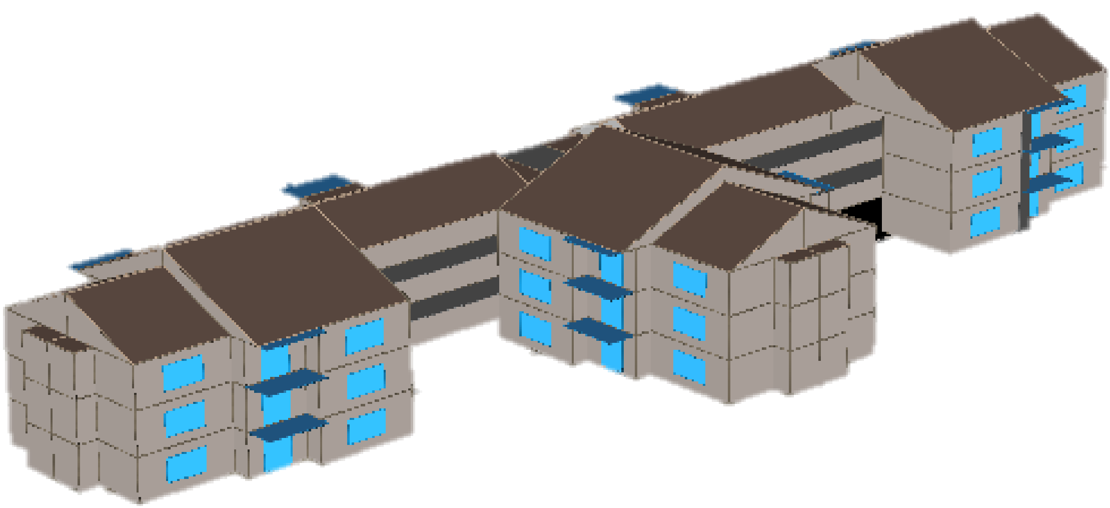
Fig. 28 shows the actual building and a 3-dimensional rendering of the eQUEST model of the building. The eQUEST model used the same geometry and thermal zoning as the real building. The construction characteristics specified in the model were obtained from the construction documents; they include R-11 wall insulation, R-30 ceiling insulation, and a 4 inch concrete slab. The window properties were specified assuming single-pane, clear windows with wood frames. The space conditions specified in the model were assumed to be typical for a multifamily residential building. The characteristics of the installed GSHP system, including the capacity and efficiency of the heat pump units, size and layout of the GHX, and pumping configuration and control were also specified in great detail according to the as-built design documents.

4.2 MODEL CALIBRATION

The objective of model calibration was to ensure that space heating and cooling loads predicted by the model match those calculated from the measured data, so that a valid comparison of the energy performance of the installed GSHP system could be made against a simulated baseline system. Actual weather data were obtained from Greenville Downtown Airport (9 miles southeast of the building) for the year 2014, for which the most-complete full-year measured data were available. However, space conditions in the apartments were unknown and uncertain owing to the diversity in the operation of different apartments in the building. Therefore, space conditions including lighting and equipment power density, schedules, and thermostat set points were fine-tuned during the calibration process. Since there were no data to indicate how each individual apartment was operated, it was assumed that all apartments had the same operating conditions, which represent the typical operation of the building.

First, the lighting and equipment loads and schedules were adjusted to match the monthly base load profiles. Then small adjustments to thermostat set points were made to match hourly and daily ground-loop loads and whole-building heating and cooling loads. For the year 2014, measured data for the entire month of May, a few hours in April and June, and 5 days in September were missing. Therefore, the calibration was performed against average daily values in each month. Adjustments were made until the results exceeded ASHRAE Guideline 14 criteria, which require <5% normalized mean biased error (NMBE) and <25% coefficient of variation of root mean squared error CVRMSE). The NMBE and CVRMSE of the calibrated model are listed in Table 4. Comparisons of measured and simulation-predicted heating, cooling, and ground-loop loads are shown in Fig. 29 and Fig. 30. As these figures show, a good match was achieved between the measured and simulation-predicted data.

(a)



(b)

Fig. 28. North Village building B: (a) actual building and (b) a 3-dimensional rendering of the eQUEST energy model of the building.

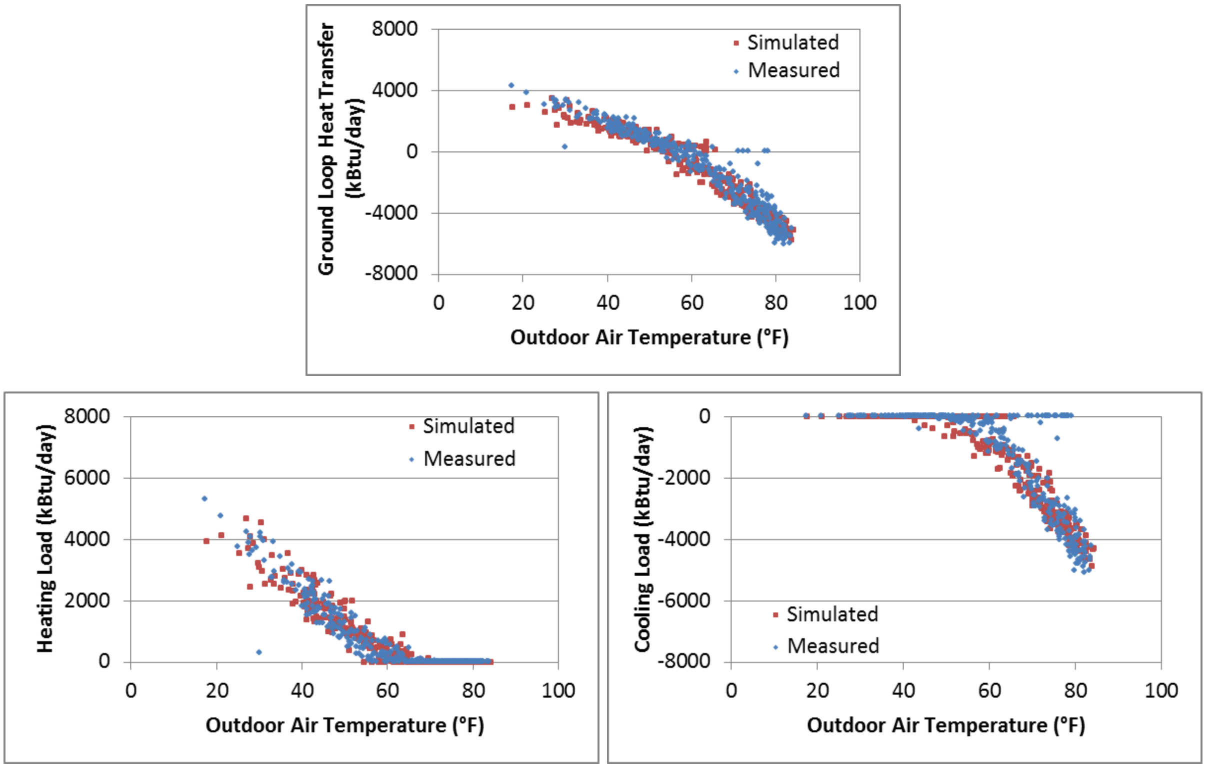


Fig. 29. Daily heating, cooling and ground-loop loads versus outdoor air temperature: predicted with the calibrated model vs. measured data.

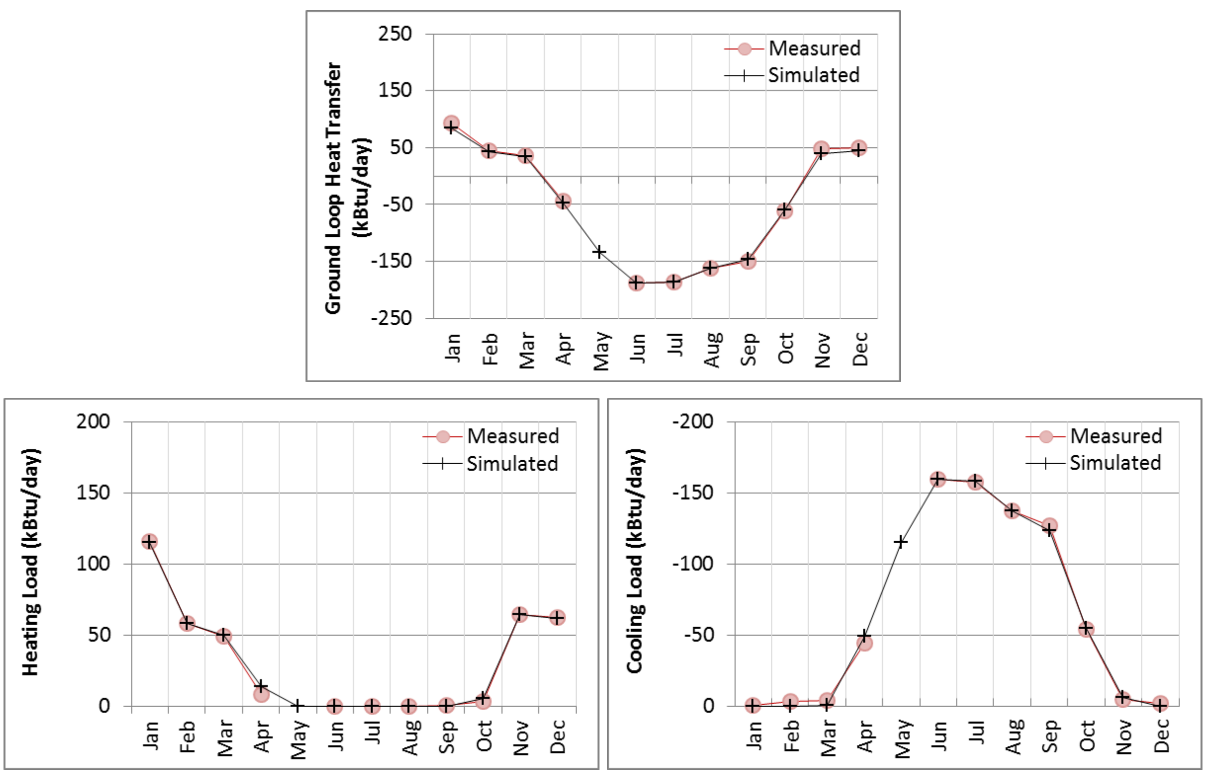


Fig. 30. Monthly heating, cooling and ground-loop loads: predicted with the calibrated model vs. measured data.

Table 4. CVRMSE and NMBE of the calibrated model

Parameter	NMBE	CVRMSE
QGL	4.7%	9.7%
QC	−0.8%	3.8%
QH	−1.9%	5.3%

QGL = ground-loop load; QC = cooling load; QH – heating load

4.3 COMPARISON WITH THE BASELINE SYSTEM

The energy consumption of the GSHP system was compared with that of a baseline system to determine the energy savings and environmental benefits. Before the GSHP retrofit, each apartment of building B and the laundry rooms were served by individual ASHPs, supplemented by electric resistance heat. The existing ductwork of the previous ASHP system was used for the new GSHP system. The calibrated energy model was modified to simulate the baseline system performance by replacing the GSHP system with individual ASHPs for each apartment and laundry room. The efficiencies of the baseline ASHP system were adopted from ASHRAE Standard 90.1-2013 (with SEER 13 cooling efficiency and 7.8 HSPF heating efficiency[4]). The eQUEST default performance curves for residential ASHP were used to model the baseline system.

The baseline system was simulated and the results were compared with the measured data for the GSHP system in building B. Fig. 31 shows the monthly electricity consumption of each end-use of the two systems. For the baseline system, electricity was consumed by the ASHP unit compressors for space heating or space cooling ("Space Heat" or "Space Cool"), the supplemental electric resistance heaters ("HP Supp"), and the ASHP unit supply fans for ventilation ("Vent Fans"). For the GSHP system, electricity was consumed by the GSHP unit compressors for space heating or space cooling, the GSHP unit supply fans, and the ground-loop circulation pump. As can be seen in Fig. 31, the GSHP system used about 50% less electricity in the heating season than the baseline system because the resistance heaters were not needed; and it used about 20–30% less electricity in the summer months. On an annual basis, the GSHP system reduced electricity use by 38% compared with the baseline system.

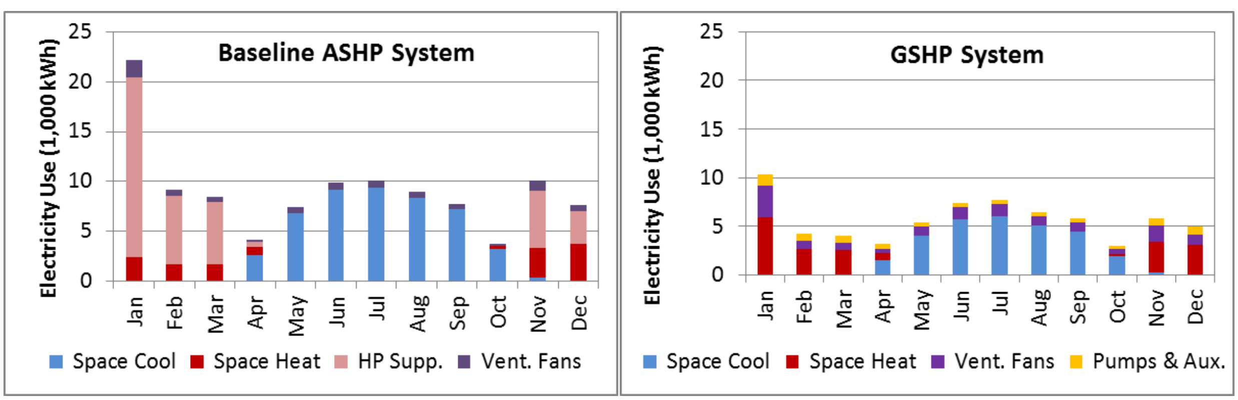


Fig. 31. Electricity use for the baseline system and the GSHP system.

The bar charts in Fig. 32 compare the site and source energy consumption, energy costs, and CO_2 emissions resulting from the two systems while satisfying the same demands for space heating and space cooling over a year. The source energy consumption and the equivalent CO_2 emissions (CO_2e) for the two systems were calculated using the source energy conversion factors and CO_2e emission factor for

[4] EIRclg = 0.2469, EIRhtg = 0.2665 (EIRclg = 3.412/1.063/SEER; EIRhtg = 1/0.481/HSPF; Source: http://esl.tamu.edu/docs/terp/2013/ESL-TR-13-04-01-DRAFT.pdf, see Figure 1 and 2 on p.3, FSEC linear model; also see footnote on p.2).

electricity in the region where the demonstration project is located. According to a report from National Renewable Energy Laboratory (NREL) (Deru and Torcellini 2007), the source-to-site energy conversion factor for electricity in the eastern region of the United States is 3.443 per unit of delivered electricity, and the CO_2e emission factor is 1.74 lb/kWh of delivered electricity. The energy cost for the two systems was calculated on the basis of a \$0.092/kWh price for electricity, as reported by the Energy information Administration for South Carolina.

As can be seen from Fig. 32, in each year, the GSHP system saved 41,380 kWh of site energy and 486,254 kBTU of source energy and reduced CO_2 emissions by 72,000 lb compared with the baseline system. Percentage-wise, the **GSHP system reduced site and source energy consumption and CO_2 emissions by 37.8% compared with the baseline system**. The resulting energy cost savings was \$3,807 per year.

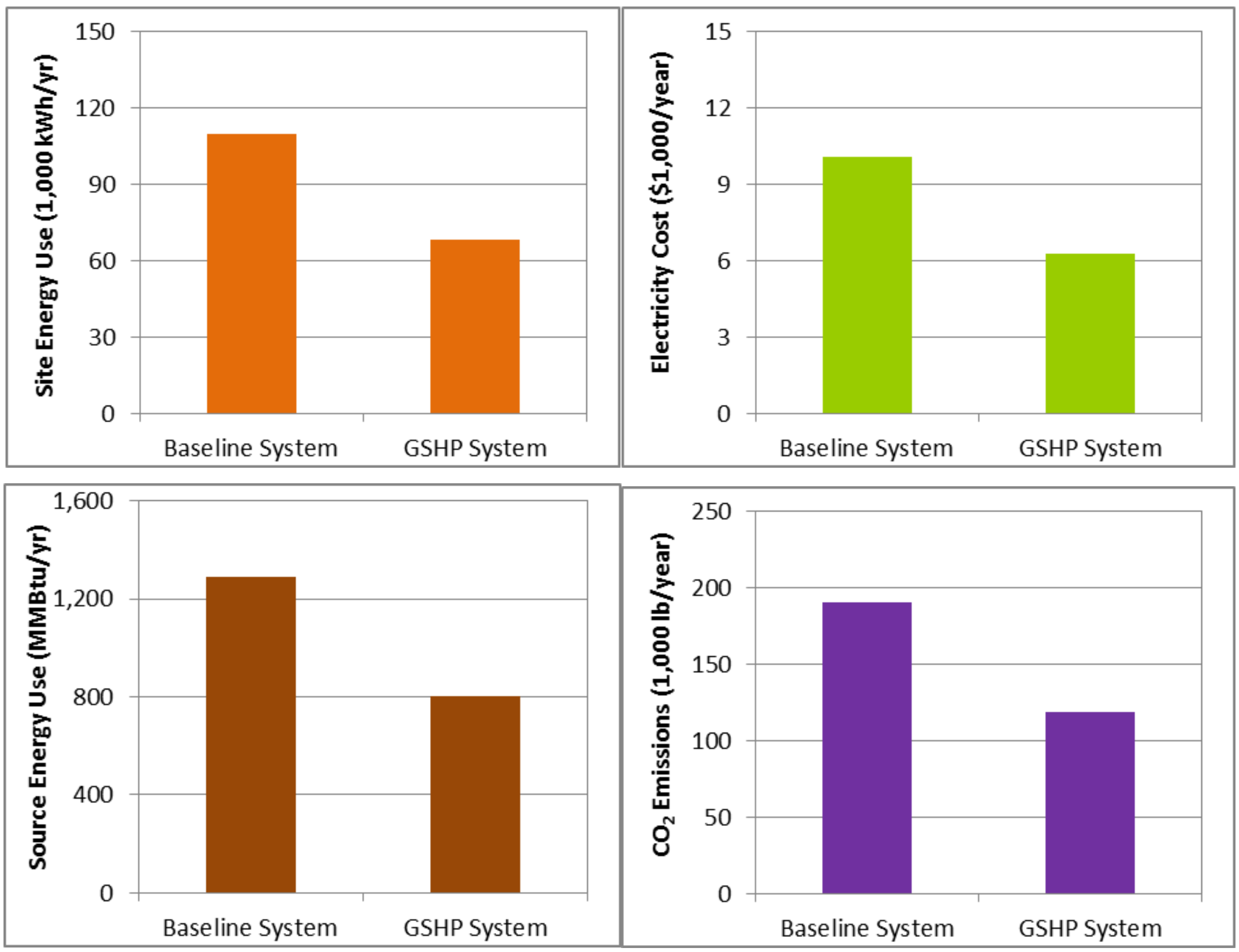


Fig. 32. Comparison of site and source energy use, energy cost and CO_2 emissions between the GSHP system and the baseline system in building B.

Based on the cost data provided by Furman University for retrofitting the ten buildings, the average cost of the GSHP retrofit per building was $407,415[5] (i.e., $6,625/ton of the demonstrated GSHP system). The cost breakdown is shown in Fig. 33. Since the GSHP system used the existing ductwork, there was no cost for ductwork installation. The average cost of well drilling (i.e., drilling vertical bores and installing the GHXs) per building was $178,273 (i.e., $17.8 per foot of bore depth), which contributed the most (43.8%) to the total cost, and the cost of the GSHP units ("WSHP" in Fig. 33) accounted for only 16% of the total cost, which was slightly more than the cost for piping installation.

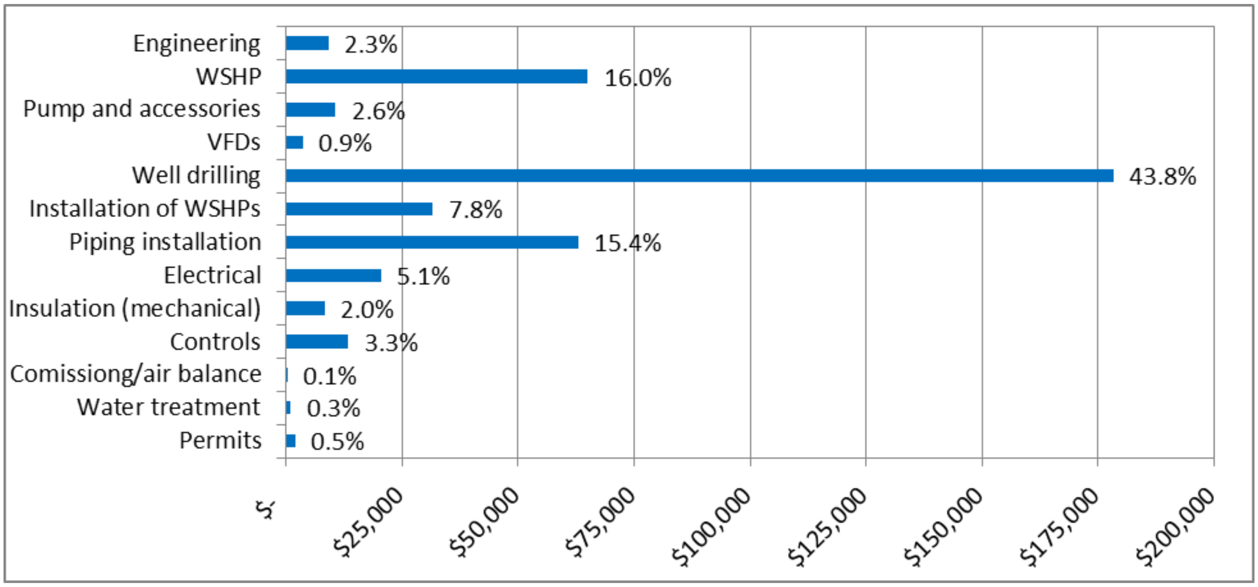


Fig. 33. GSHP system installation cost breakdown.

Assuming the replacement cost of the baseline ASHP system would have been $3,500 per unit[6] for 25 units (i.e., $87,500 per building), the cost premium for installing the GSHP system was $319,915. Based on this cost premium and the previously calculated annual energy cost savings, the simple payback period for this GSHP retrofit project is 84 years. If the drilling cost were $10 per linear foot of borehole, which is the national average, the simple payback would have been 63 years.

[5] Costs that are not expected to occur in a typical GSHP installation project are excluded, such as the American Recovery and Reinvestment Act–specific project management cost and the demolition and restoration costs for the parking lots where the GHXs were installed.
[6] This excludes the cost of ductwork, assuming that the existing ductwork would have been used.

5. CONCLUSION

This case study analyzes the energy performance of GSHP systems installed in 10 identical buildings of the North Village student housing complex at Furman University in Greeneville, South Carolina. The GSHP systems for each building are also identical. They are all in a distributed configuration, and each one consists of 25 WAHPs serving each of the 24 apartments and two laundry rooms, a vertical-bore closed-loop GHX with twenty 500 ft deep bores installed under the parking lot, and a two-pipe common loop with a central variable-speed pumping station to circulate water through the GHX and the WAHPs. The actual performance of these GSHP systems was analyzed based on available measured data for the year 2014. The annual energy consumption of one of the GSHP systems (at building B) was calculated based on the available measured data and other related information. It was compared with the performance of a baseline scenario—an ASHP system with the minimum allowed energy efficiencies specified in ASHRAE Standard 90.1-2013 and with supplemental electric resistance heat. This system is typical for student housing in the region. The comparison was made in terms of energy savings, operating cost savings, cost-effectiveness, and environmental benefits. The following sections summarize the results of the analysis, the lessons learned, and recommendations for improvement in the pumping control at several buildings and performance data collection.

5.1 ENERGY PERFORMANCE AND COST-EFFECTIVENESS

The ground-loop supply water temperatures at the ten buildings varied from 50.0–57.7°F in the heating season to 80.2–88.8°F in the cooling season. Compared with the coincidental OA temperature, the ground-loop supply water temperature was relatively stable; it was up to 40°F higher than the OA temperature on the coldest day of the year and about 10–15°F lower than the OA temperature on the hottest day of the year. In all 10 buildings, the GSHP system rejected more heat to the ground than it extracted from the ground. The ratio between heat rejection and heat extraction varied from 2.0 to 7.6 among the 10 buildings. Measured data from a few buildings that have been running the GSHP system for several years indicate that the maximum ground-loop supply temperature has increased by about 1–2°F over the years.

The maximum TD of the ground loop was within 9.5–12.5°F in the cooling season and 6.4–9.8°F in the heating season. Since each individual heat pump usually operates with a 10°F TD in cooling mode and a 6°F TD in heating mode, the measured maximum TD indicates that the ground loop was slightly underflow in some buildings (e.g., B and H). Median TDs were 5.3–8.4°F in cooling season (less than 4°F in buildings A and K) and 2.4–4.1°F in heating season (1.7 and 1.3°F in buildings A and K). Median flow rates in the ground loop were 18–35 gpm (exceptions were 54 and 45 gpm, respectively, in buildings A and K). The higher flow rates and smaller TDs at buildings A and K indicate excessive pumping.

Most buildings had an annual average pumping power fraction between 10.5 and 13%. Buildings B, C, and D had even lower pumping power fractions. Buildings A and K had the highest pumping power fractions of 15 and 18.7%, respectively. Further analysis indicates that the GSHP system COP decreases with an increase in the pumping power faction, especially in heating mode.

The annual operational efficiencies (i.e., COPs) of the GSHP systems ranged between 4.8 and 5.5 for cooling and 3.8 and 4.9 for heating. If heating and cooling operation are combined, the annual COPs of the GSHP systems in buildings B, C, H, and I exceeded 5.0.

Fig. 25, showing the monthly heat pump energy use for six apartments in buildings B and I, indicates that the new GSHP units in building B used less than 1/3 as much electricity as the existing ASHPs in building I. Accounting for the pumping energy associated with the six GSHP units, the energy savings achieved by the GSHP system was about 60% compared with the existing ASHPs.

Utility bills (including electricity consumption for HVAC, lighting, appliance, and other uses) of the 10 buildings during the pre- and post-retrofit periods were analyzed. The weather-normalized monthly electricity use for all buildings combined for 2014 (post-retrofit) was about 36% lower than the electricity use during 2010 (pre-retrofit) when the OA temperature was lower than 40°F. The electricity savings were smaller (by 10–20%) when the OA temperature was between 50 and 70°F, and they varied between 0 and 50% when the OA temperature was higher than 70°F. Overall, the GSHP retrofit has resulted in a 27.3% reduction in the annual electricity use of the 10 buildings. The achieved electricity savings is 715,384 kWh/year, or $65,815/year if the average electricity rate in South Carolina ($0.092/kWh) is applied.

Compared with the baseline system (i.e., a new SEER 13 ASHP with a supplemental electric resistance heater), the GSHP system achieved significant energy savings and CO_2 emission reductions. The GSHP system in building B demonstrated 37.8% site and source energy savings and CO_2 emission reductions compared with the baseline system. Table 5 provides a comparison between the two systems.

Table 5. Summary comparison between the GSHP system and the baseline system

	Baseline system	GSHP system	Savings
Site energy use (MMBtu)	109,570	68,190	37.8%
Source energy use (MMBtu)	377,250	234,778	37.8%
CO_2 emissions (lb)	190,652	118,651	37.8%
Total annual energy cost ($)	$10,080	$6,273	37.8%

Based on the local retail electricity rate of $0.092/kWh, the reduced electricity consumption resulted in a $3,807 operating cost savings per year for building B. Assuming the replacement cost of the baseline ASHP system would be $3,500 per unit[7] for 25 units (i.e., $87,500 per building), the cost premium for the GSHP system would be $319,915, which results in a simple payback period of 84 years for this GSHP retrofit.

5.2 LESSONS LEARNED

- Compared with other GSHP systems studied previously, the GSHP systems at Furman University (except those at buildings A and K) used significantly less pumping energy, and their pumping power fractions were about half as high as for the other GSHP systems studied previously. The better pumping performance resulted from following good practices:
  - Locating the DP sensor at the hydraulically most remote WAHP in the piping system
  - Using an auto-flow valve for each WAHP to maintain a constant flow rate at the WAHP
  - Minimizing bypass flow

- The patterns in the relationship between the flow rate and the TD of the ground loop could be an useful indicator for evaluating the pumping performance. Ideally, the pattern should be a “U” shape, which indicates that the TD is kept at the design value while the flow rate varies within a large range. For a GSHP system with excessive pumping at part-load conditions, the pattern will have a “V” shape and a substantially high minimum flow rate (as shown in Fig. 19 for buildings A and K).

[7] This excludes the cost of ductwork, assuming that the existing ductwork would be used.

- Malfunctioning sensors may be identified by comparing patterns in the relationships among different measurements with the expected or historical patterns. In the initial data analysis, it was found that the ground-loop flow rates in a few buildings were much lower for a certain time period, but the pumping power was about the same. Onsite inspection identified faulty flow meters in these buildings.

- The 2-way solenoid valves at each heat pump should be closed when the solenoid is not energized. Doing so allows a malfunctioning valve to be easily identified (e.g., from a service call).

5.3 RECOMMENDATIONS FOR FURTHER IMPROVEMENTS

- To more accurately analyze the performance of the GSHP system (e.g., accounting for simultaneous heating and cooling operation of the individual heat pumps in a building), the heating and cooling output and the associated power consumptions should be measured. If it is too expensive to do so, at least the runtime of each heat pump in each operation mode should be recorded.

- Measured data indicate excessive pumping in the GSHP systems at buildings A and K. It is recommended that the 2-way solenoid valves of each heat pump in these buildings be checked. It is likely that the excessive pumping is due to a few malfunctioning 2-way valves.

6. REFERENCES

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

Liu, X., and H. Göran. 2008. “Enhancement of an integrated simulation tool for ground source heat pump system design and energy analysis,” in Proceedings of the 10th International Thermal Energy Storage Conference, Stockton, NJ, June 2008.