

# Performance Analysis of Ground Source Heat Pump Demonstration Projects in the United States

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## Abstract

Twenty-six ground source heat pump (GSHP) projects were competitively selected by the US Department of Energy in 2009 and awarded grants funded by the American Recovery and Reinvestment Act to demonstrate the benefits of GSHP systems and innovative technologies for cost reduction and/or performance improvement. Since 2014, a series of in-depth case studies have been conducted for 10 of the demonstration projects. The GSHP systems studied serve various types of buildings across the United States. They utilize different ground sources and heat pump equipment, and they have different configurations (i.e., distributed or central). These case studies analyzed the measured performance data of the GSHP systems collected from the grantees using a combination of methods, including utility bill analysis, performance data visualization, profiling and benchmarking, and calibrated computer simulations. These case studies indicate the GSHP systems saved 27%–66% primary energy and reduced CO<sub>2</sub> emissions by 21%–66% compared with conventional heating, ventilation, and air conditioning (HVAC) systems. The operational efficiency of these GSHP systems can be further improved with better designs and controls.

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*Keywords:* commercial, ground source heat pump, performance, cost, case study

## 1. Introduction

High initial costs and a lack of public awareness about ground source heat pump (GSHP) technology are the two major barriers preventing rapid deployment of this energy-saving technology in the United States [1]. To tackle these barriers, 26 GSHP projects were competitively selected by the US Department of Energy in 2009 and awarded grants funded by the American Recovery and Reinvestment Act to demonstrate the benefits of GSHP systems and innovative technologies for cost reduction and/or performance improvement. Ten of these demonstration projects were selected for in-depth case studies, which cover a wide range of building sizes, principal activities, ground sources, heat pump equipment, and system configurations. Figure 1 shows the location of the 10 GSHP projects studied. This paper presents an overview of these GSHP systems, the

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methodology for performance analysis, evaluation of key performance metrics, and lessons learned from the case studies.



Fig. 1. Locations of the 10 GSHP demonstration projects studied.

## 2. Overview of GSHP systems studied

A brief overview of the 10 GSHP systems and the buildings they serve is provided below, and more information about these GSHP systems is listed in Table 1. Additional details are presented in a series of technical reports [2–11].

**Case #1** is located in Kalispell, Montana. A central GSHP system replaced the existing electric heaters in an existing 2,044 m<sup>2</sup> warehouse and a newly added 864 m<sup>2</sup> truck bay. Groundwater is used as the heat source for a 280 kW heating-only water-to-water heat pump (WWHP).

**Case #2** is located in Cedarville, Arkansas. A distributed GSHP system replaced the existing aging heating, ventilation, and air conditioning (HVAC) systems in a 6,000 m<sup>2</sup> high school building, and also provides space conditioning to a newly added 511 m<sup>2</sup> cafeteria and laboratories. This system uses closed-loop vertical bore ground heat exchangers (CLVB-GHXs) and 45 packaged water-to-air heat pump (WAHP) units, which have a combined cooling capacity of 812 kW. Most of the WAHP units provide dedicated humidity control in the spaces they serve.

**Case #3** is located in Raleigh, North Carolina. A distributed GSHP system serves a new 2,230 m<sup>2</sup> office building. This system uses CLVB-GHXs, 27 WAHP units (with a total 300 kW combined capacity) for space heating and cooling, a 175 kW WAHP unit in a dedicated outdoor air system (DOAS), and an 87.5 kW WWHP dedicated to supply domestic hot water (DHW).

**Case #4** is located in Albany, New York. A distributed GSHP system provides space conditioning and outdoor air ventilation in a new 17,187 m<sup>2</sup> university student housing building. This system consists of CLVB-GHXs, 188 WAHP units with a combined capacity of approximately 1,253 kW, four rooftop units that provide conditioned outdoor air to common areas, and 27 energy recovery ventilator units, each delivering partially conditioned outdoor air to five identical dwelling units.

**Case #5** is located in Rochester, Michigan. This ground source variable refrigerant flow (GS-VRF) system provides space heating and cooling to a new 15,979 m<sup>2</sup> institutional building. It uses CLVB-GHXs and 50 water-source variable refrigerant flow (WS-VRF) units with a combined cooling capacity of 1,120 kW. In addition, three WWHP units, each with 140 kW cooling capacity, are used to provide chilled water (CHW) and hot water (HW) to a DOAS.

**Case #6** is located in Butte, Montana. It consists of a 175 kW WWHP and a set of closed-loop high density polyethylene (HDPE) pipes immersed in a flooded mine. This system works in conjunction with the originally installed central heating and cooling system to condition an existing 5,200 m<sup>2</sup> research facility.

**Case #7** is located in Greenville, South Carolina. Ten identical distributed GSHP systems serve 10 identical student housing buildings on a university campus. Each GSHP system consists of 25 WAHP units and CLVB-GHXs installed under an adjacent parking lot. The combined capacity of the 10 GSHP systems is 2,153 kW.

**Case #8** is located in Denver, Colorado. A central GSHP system consists of seven 105 kW modular WWHP for producing CHW and HW, and another 19 kW WWHP dedicated to producing DHW. Recycled water from the city's municipal wastewater system is used as the heat sink and heat source for the WWHPs. The GSHP system serves a new 13,000 m<sup>2</sup> addition to the Denver Museum of Nature and Science.

**Case #9** is located in Lincoln, Nebraska. A 49,000 kW central GSHP system provides hot and chilled water to a newly constructed 25,000 m<sup>2</sup> adult detention facility. This GSHP system uses CLVB-GHXs, 25 modular WWHPs, and three central variable speed (VS) pumping stations to circulate water through the CLVB-GHXs and the primary CHW and HW loops.

**Case #10** is located in Muncie, Indiana. It is a part of a district central GSHP system serving a university campus, which has approximately 667,111 m<sup>2</sup> of floor space. The GSHP system studied consists of CLVB-GHXs, two 8,750 kW heat recovery (HR) chillers, and five central VS pumping stations to circulate water through the CLVB-GHXs and distribute CHW and HW throughout the campus.

Table 1. Characteristics of ground source heat pump (GSHP) systems studied

	Case #1	Case #2	Case #3	Case #4	Case #5
Configuration	Central	Distributed	Distributed	Distributed	Distributed
Heat pump	One 280 kW WWHP	45 WAHPs with 812 kW total capacity	28 WAHPs with 475 kW total capacity and one 87.5 kW WWHP for DHW	188 WAHPs and 4 ground source roof top units (1253 kW total capacity)	50 HR WS-VRF units with 1120 kW total capacity and three 140 kW WWHPs for DOAS
Ground source	Open loop with two shallow ground water wells (11 m deep)	CLVB-GHXs, 98 bores (11,948 m total bore length)	CLVB-GHXs, 60 bores (6,126 m total bore length)	CLVB-GHXs, 150 bores (20,574 m total bore length)	CLVB-GHXs, 256 bores (24,969 total bore length)
Pump and control	One 5.6 kW VS pump with fixed DP control	Two 18.65 kW VS pumps with fixed DP control	Two 14.9 kW VS pumps with fixed DP control	Three 14.9 kW VS pumps (primary) with fixed temperature control and two 29.8 kW VS pumps (secondary) with fixed DP control	Two 44.8 kW VS pumps to circulate water flow through all WS-VRF units at constant flow rate
Installed cost (\$/kW)	1,183	2,061	3,228	3,959	6,357
Installed cost (\$/m <sup>2</sup> )	192	254	811	333	613
	Case #6	Case #7	Case #8	Case #9	Case #10
Configuration	Central	Distributed	Central	Central	Central
Heat pump	One 175 kW WWHP	250 WAHPs with 2,153 kW total capacity	Seven 105 kW modular WWHP with 735 kW total capacity	Twenty-eight 175 kW modular WWHP with 4,900 kW total capacity	Two 8,750 kW HR chillers with 17,500 kW total capacity
Ground source	3,658 m HDPE pipe immersed in mine water	CLVB-GHXs, 200 bores (30,480 m total bore length)	Recycled municipal wastewater	CLVB-GHXs, 667 bores (60,990 m total bore length)	CLVB-GHXs, 1,803 bores (219,822 m total bore length)
Pump and control	Two 5.6 kW constant speed pump	Two 5.6 kW VS pumps in each of the 10 buildings with fixed DP control	Five central pumping stations each with two 11–15 kW pressure-controlled VS pumps	Three central pumping stations each with two 19–56 kW pressure-controlled VS pumps	Five central pumping stations each with two/three 93–261 kW pressure-controlled VS pumps
Installed cost (\$/kW)	Not available	1,893	7,203	608*	986*
Installed cost (\$/m <sup>2</sup> )	Not available	152	407	119*	Not available

Note:

CLVB-GHX = closed-loop vertical bore ground heat exchanger; DHW = domestic hot water; DP = differential pressure; VS = variable speed; WAHP = water-to-air heat pump; WWHP = water-to-water heat pump; WS-VRF = water source variable refrigerant flow

\* Includes only the installed costs of the equipment in the central energy plant; does not include the costs of water distribution pipeline and the HVAC equipment inside the buildings served by the energy plant.

### 3. Methodology for performance analysis

Performance analysis was based on design documents, measured data, utility bills, and other information collected from the grant recipients. It is more challenging to evaluate the performance of distributed GSHP systems than central GSHP systems because multiple (possibly hundreds) WAHPs are used, each of which can independently provide space heating or cooling to the zone(s) it serves at any given time. Consequently, it is necessary to determine the heating and cooling outputs and the associated power consumption for each WAHP.

However, the measured data for distributed GSHP systems typically only include temperature and flow rate in the ground loop, power draw (or pump speed) of circulation pump(s), and the runtime of each WAHP in heating and cooling modes, respectively. Except for case #3, power consumption of distributed WAHPs was not directly measured. As a result, the heating and cooling output and the associated power consumption of each WAHP was calculated based on manufacturer's published performance data for that particular WAHP using the measured runtime and the supply water temperature from the ground source. This calculation procedure was discussed in detail in a technical report [3], and the calculated data were partially validated by comparing the daily sum of the calculated heat rejection and heat extraction of each WAHP with the measured daily heat transfer load of the ground source. The discrepancy between the two sets of data is less than 20% for most days [3].

For central GSHP systems, heating and cooling outputs were calculated using the measured flow rate and temperature difference in the CHW and HW loops. The power consumption of WWHPs (or HR chillers) was either measured (cases #1, #8, and #10), or it was calculated based on the measured leaving water temperatures at the source and load sides of each WWHP and the published performance data of the WWHP (cases #6 and #9).

The measured performance data were analyzed to evaluate:

- the operational efficiency of the heat pump (WAHPs or WWHPs) and the overall GSHP system (including pumping power and energy consumption of supplemental heating and cooling, etc.),
- the performance of the pumping system,
- any faults or abnormalities in the system's operation, and
- potential improvements to the system.

For case studies of distributed GSHP systems, detailed hourly energy simulation models of the host building(s) were created and calibrated against available data for the building's heating and cooling loads. These models were used to predict the energy consumption of a baseline conventional HVAC system for satisfying the same heating and cooling demands of the host building(s). For central GSHP systems, generic models of conventional HVAC equipment (e.g., chiller, boiler, and cooling tower) were used to estimate the energy consumption of the conventional HVAC equipment for providing the measured heating and cooling outputs of the GSHP systems. The selected baseline HVAC system for each case study is the most commonly used HVAC system for the host building(s), and the efficiencies of the cooling and heating equipment used in the baseline systems are the minimum allowed by ASHRAE standard 90.1-2013 [12]. The measured or calculated power consumption of GSHP systems was compared with the simulation-predicted baseline HVAC system's energy consumption to determine the energy savings, operating cost savings, and carbon emission reduction resulting from the GSHP systems.

## 4. Results

### 4.1. Actual performance of GSHP systems studied

GSHP system performance was evaluated using a system coefficient of performance (SCOP) metric. SCOP is the ratio of the aggregated heating or cooling output of each WAHP or WWHP to all the power consumed for providing such output, including the power consumed by the WAHP or WWHP and the circulation pumps. For several central GSHP systems (#6, #9, and #10), only the pumping power in the energy plant (e.g., for circulating water through WWHPs and a ground source) was accounted for. The pumping power inside the buildings served by the energy plant was excluded because of a lack of data. In case #8, a cooling tower and a supplemental heating were used in the GSHP system and their contributions were accounted for in the SCOP calculation [9]. Because the performance data for DHW production were not available, the contribution of the WWHP dedicated for DHW were not included in the SCOP evaluation.

As Fig. 2 shows, SCOP of the GSHP systems studied varied from 2.5 to 4.35 for heating and from 3 to 5.3 for cooling. For cases #8 and #10, which provide both CHW and HW simultaneously, the effective SCOP—the ratio of the combined heating and cooling outputs of the WWHPs to all the power consumed by the GSHP system—was evaluated. The relatively low effective SCOP (3.7) for the HR chiller in case #10 is a result of the mismatch between the heating/cooling output of the chiller and the heating/cooling demand of the building, which is discussed in more detail below and in a technical report [11]. The relatively low effective SCOP (3.7) for the modular WWHPs in case #8 is a result of the supplemental heating used to boost the HW temperature and the use of cooling tower for pre-cooling operation [9]. Other factors contributing to the variation in SCOP values include the different supply water temperature from various ground sources (i.e., ground-coupled, groundwater, recycled municipal water, or mine water) and different pumping performance.

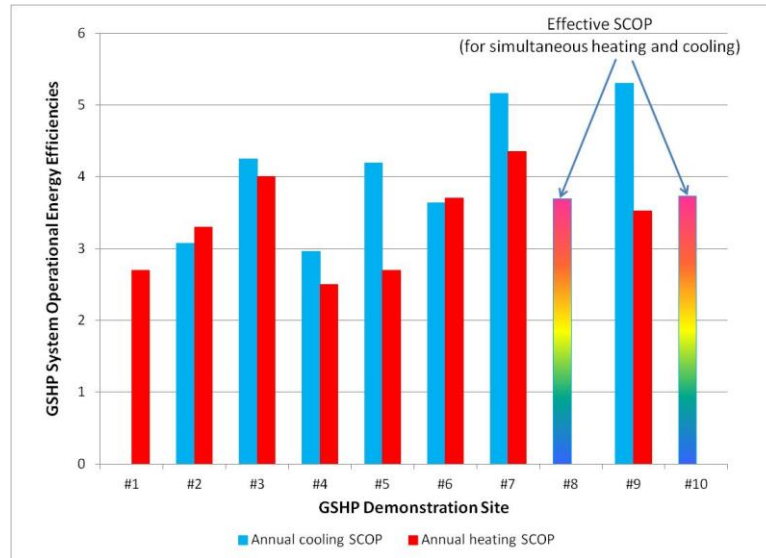


Fig. 2. Annual operational efficiency of the 10 ground source heat pump (GSHP) systems studied.

Figure 3 shows the percentage of pumping energy in the overall GSHP system energy use varied from 11% (case #9) to 45% (case #4). As introduced earlier, cases #6, #9, and #10 only accounts for a part of the pumping energy due to the lack of data. Different from other cases, case #8 involves cooling tower and boiler in addition to pumps and heat pumps. If only pumps and heat pumps are accounted for (to be consistent with other cases), the contribution of pumping energy in case #8 would be 16%. Considering the above factors, case #7 probably has the lowest pumping energy percentage (13%, which is about 1/3 less than that in other distributed GSHP systems). A case study [8] identified that the better pumping performance in case #7 resulted from following good practices: (1) locating the DP sensor at the hydraulically most remote WAHP in the piping system; (2) using an auto-flow valve for each WAHP to maintain a constant flow rate at the WAHP; and (3) minimizing bypass flow.

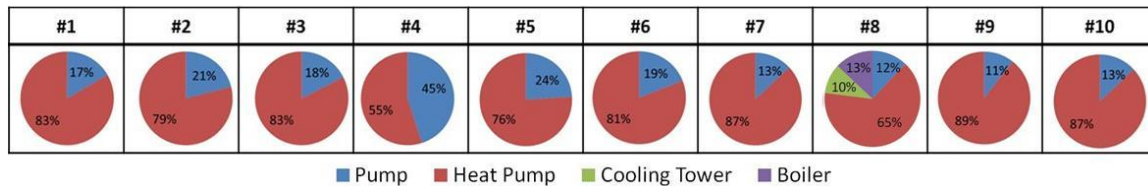


Fig. 3. Percentage of energy use for pumps and heat pumps in the overall ground source heat pump (GSHP) system energy use

Because of the inclusion of pumping energy use, the SCOP is lower than the efficiency of the WAHPs or WWHPs, especially during part-load conditions as shown in Fig. 4, which is an example of case #2 [3]. It indicates that reducing pumping energy in part-load conditions could improve the operational efficiency of a GSHP system.



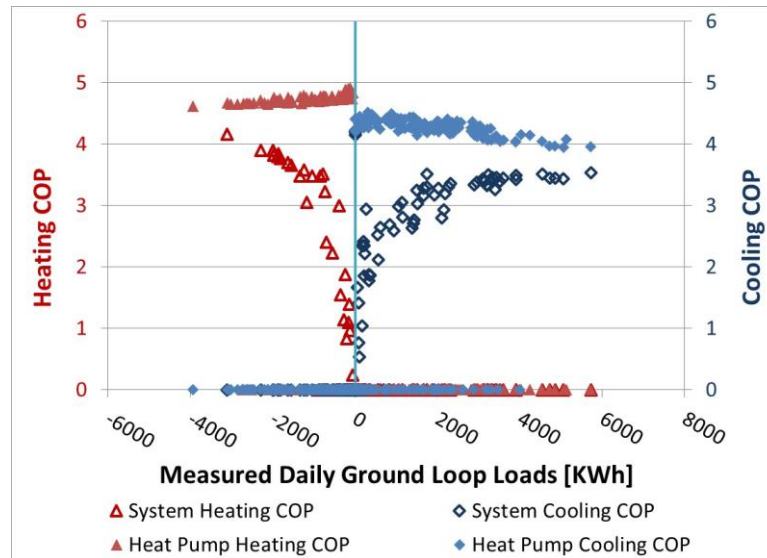


Fig. 4. Operational efficiency of ground source heat pump (GSHP) systems and heat pump equipment at various loading conditions.

Figure 5 shows that the GSHP systems saved 27%–66% in source energy compared with the baseline HVAC systems. Further, they reduced CO<sub>2</sub> emissions by 21%–66%. Operating costs were reduced by 18%–65%. The variation in energy savings and emission reductions is a result of the different performance between the GSHP systems and the baseline HVAC systems. The variation in energy cost savings is also affected by different energy prices at various locations. Descriptions of the baseline systems and their performance, as well as the energy prices used in each case study, are presented in [2–11].

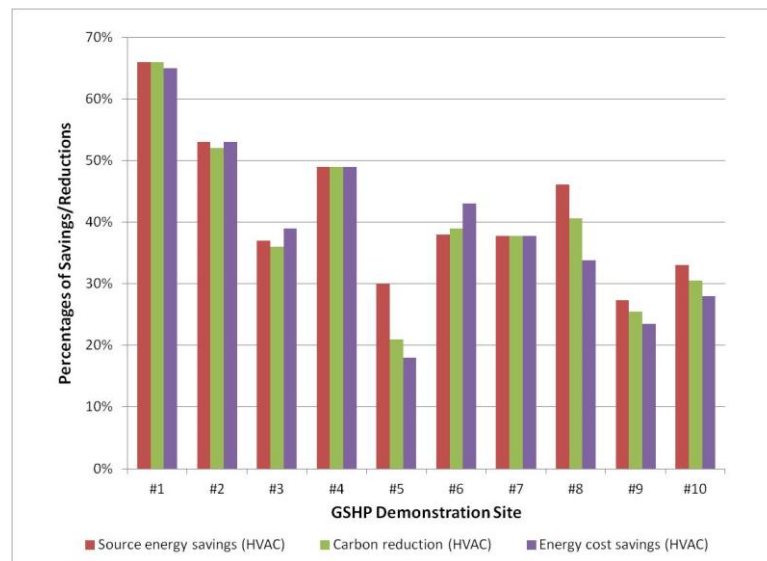


Fig. 5. Benefits of the 10 ground source heat pump (GSHP) systems studied.

Except for cases #6, #9, and #10, of which the cost data was either not available or only for the central energy plant, the installed costs of the other 7 GSHP systems varied widely from \$1,183 per kW to \$7,203 per kW. The lowest cost was case #1, which uses a shallow aquifer as its heat source, and the highest cost at case #8, which uses multiple heat sinks and heat sources for WWHPs, including a cooling tower, a boiler, and a 1 km long two-way pipeline to access recycled municipal wastewater. CLVB-GHXs were found to be oversized in a few projects (for future extension). In addition, extra engineering work was performed to implement newer GSHP technologies (e.g., the GS-VRF in case #5 and the recycled-water heat pump system in case #8). The costs of

these GSHP systems would have been lower should the CLVB-GHXs be properly sized for the installed capacity of the GSHP system, or the system design be simplified. The wide range in the installed cost is also due to the inconsistency of the category and coverage of the cost data reported by the grantees. In some cases, only the installed cost of major GSHP equipment was provided; while in other cases, the reported cost may also include the cost of other components, such as building energy manage system, ductwork, and pipeline.

#### 4.2. Characteristics of uncommon ground sources

Two uncommon heat sinks/sources were used in the demonstration projects—flooded mines (case #6) and recycled municipal wastewater (case #8). The supply water temperature for the heat exchanger immersed in mine water was stable at around 25°C during heating operation, but it increased sharply during cooling operation (Fig. 6a), which was the result of thermal stratification in the flooded mine shaft. However, if convection in the mine water can be induced with a small pump or if the mine water can be directly used in an open loop configuration, the temperature surge in cooling operation could probably be avoided [7]. As Fig. 6b shows, the temperature of the recycled municipal wastewater was relatively stable compared to the ambient temperature; therefore, the recycled municipal wastewater is more favorable for effective operation of the heat pump than the ambient air [9].

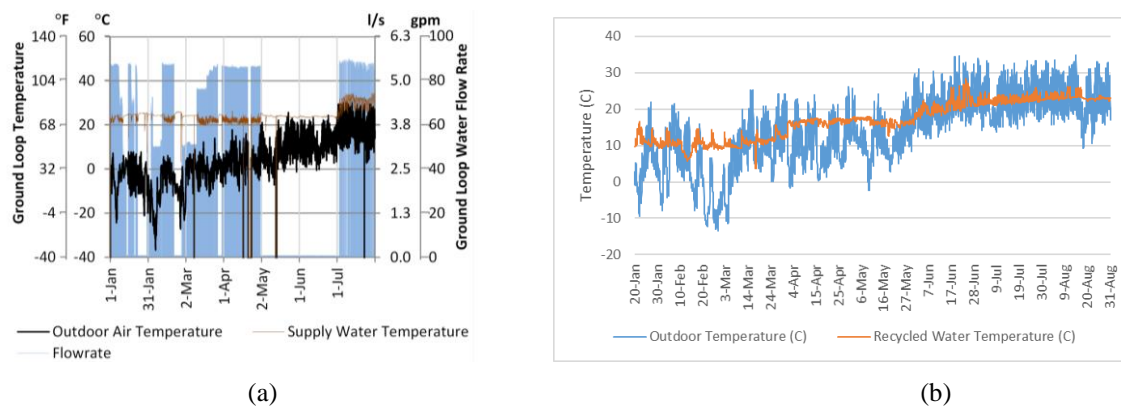


Fig. 6. Temperature of uncommon heat sources/sinks: (a) closed-loop mine water, (b) recycled municipal wastewater.

#### 4.3. Lessons learned

More energy savings would have been achieved if the design and control of the GSHP systems were improved. The suggested improvements are discussed below.

##### 4.3.1. Avoid excessive pumping

It is found in all the 10 case studies that excessive flow was pumped through the water loop of the GSHP systems. This excessive pumping is demonstrated by a V-shaped pattern between the system flow rate and the temperature differential (TD) across the supply and return mains of the water loop (Fig. 7a)—the smaller TD occurred when the system flow rate was lower than its maximum, which resulted from excessive circulation in the water loop. An ideal control would result in a U-shaped pattern between the TD and system flow rate. So, the TD is maintained at the design value when heat pumps are running (about 5.6°C in cooling mode and 3.3°C in heating mode), and the system flow rate is reduced to zero when all the heat pumps are off. Excessive pumping not only increases pumping energy use but also adds more heat rejection load to the ground source, which results in lower system efficiency and less energy savings.

The excessive pumping is caused by oversized pumps, and more often, suboptimal controls. It is a common practice to run VS pumps year-round even when all the heat pumps are off. The conventional control for VS pumps is to maintain a fixed differential pressure (DP) across the supply and return mains of the water loop (Fig. 7b). The set point of DP is usually determined at full load condition when all the heat pumps are running. However, when building heating and cooling loads become smaller, some heat pumps will be turned off and the water flow to these heat pumps are blocked off. In this case, the system flow rate is reduced, and the pressure

drop along the supply and return mains (“P\_M” in Fig. 7b) becomes smaller. However, since the DP is maintained at a fixed value, more water flow than necessary is pumped through the heat pumps that are running, which increases the pressure drop across the heat pumps (“P\_HP” in Fig. 7b) and results in smaller TD. A recent study indicates the pumping energy can be reduced by more than 50% with an innovative flow-demand-based pumping control [13].

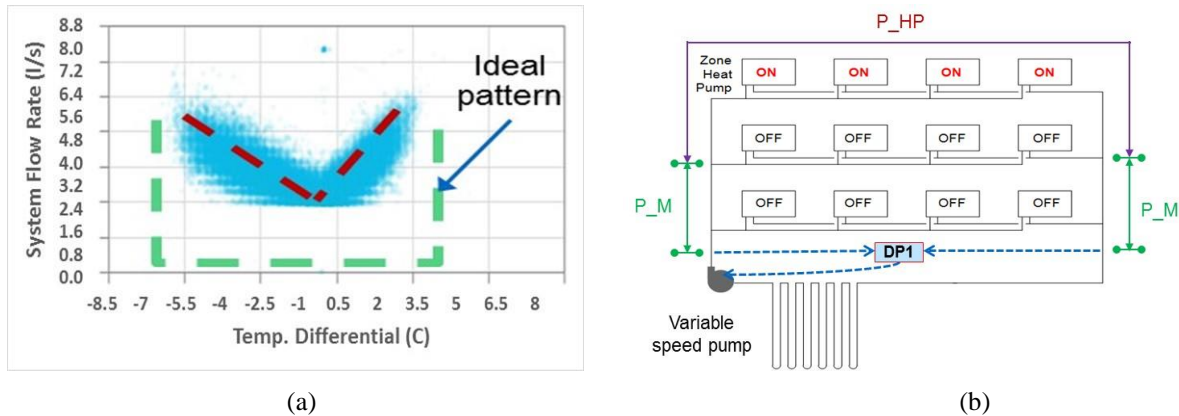


Fig. 7. Pumping performance resulting from conventional pumping control: (a) the pattern between system flow rate and water loop temperature differential and (b) the typical pumping configuration and control of distributed ground source heat pump (GSHP) systems.

#### 4.3.2. Modulate fan power

The nearly continuous operation of the constant-speed fans of the GS-VRF system in case #5 contributed 33% in the annual power consumption of the entire GS-VRF system, which offset the heat recovery benefit offered by the VRF technology. This resulted in an annual SCOP value similar to that of conventional GSHP systems (Fig. 2). Because of the unique load matching capability of the VRF indoor units, their fans ran much longer than conventional WAHPs would for conditioning the same space. A preliminary analysis indicated that 62% of the fan energy can be avoided by modulating the fan speed to match the heating/cooling outputs of the VRF indoor units [6].

#### 4.3.3. Balance supply and demand

The HR chiller used in case #10 could operate very efficiently if the produced hot and chilled water is fully utilized to satisfy the cooling and heating demands. However, the current configuration and control of the chillers resulted in overproduction of hot or chilled water. This is wasteful and has resulted in excessive chiller energy use and unexpectedly high ground loop temperatures. As a result, the operational efficiency of the GSHP system is lower than expected. The operational efficiency could be increased by operating the two chillers separately—dedicating one to producing CHW and the other to operating as a heat pump that only produces HW. In this case, there would be no need to dump excess CHW or HW into the ground. In addition, the ground loop loads would be more balanced because more heat would be extracted from the ground to satisfy the heating demand [11].

#### 4.3.4. Reset the hot water temperature

The hot water temperature produced by a WWHP is limited (e.g., less than 54°C), which constrains WWHP's operation. Applying outdoor air reset control to decrease the hot water supply temperature set point at part-load conditions can increase WWHP's operation and reduce or eliminate the need of supplemental heating (to boost the hot water supply temperature). In addition, lowering the hot water supply temperature from 49°C to 43°C would result in a ~0.5 SCOP improvement (a 9%–14% increase). However, since the hot water supply temperature affects the heat transfer performance of the terminal units, the outdoor air reset control should consider both the system efficiency and the space heating performance.

#### 4.3.5. Monitor system performance to maximize energy savings

Continuous monitoring and analysis of system performance metrics (i.e., SCOP and TD) can help identify abnormal operations and improve operational efficiency of GSHP systems. For example, an unrealistic pumping



control strategy—maintaining the supply water temperature from the ground source at a fixed value by adjusting the pump speed—was identified in case #4 by examining TD of the ground loop. This poor pumping control resulted in continuous operation of the VS pumps at their maximum speed almost all year long and the pumps consumed about the same amount of energy as all the WAHPs combined. After identifying this control issue, a simple repair to the pumping control reduced pumping power by 93%, which resulted in a 15% reduction in building energy use (Fig. 8a–b).

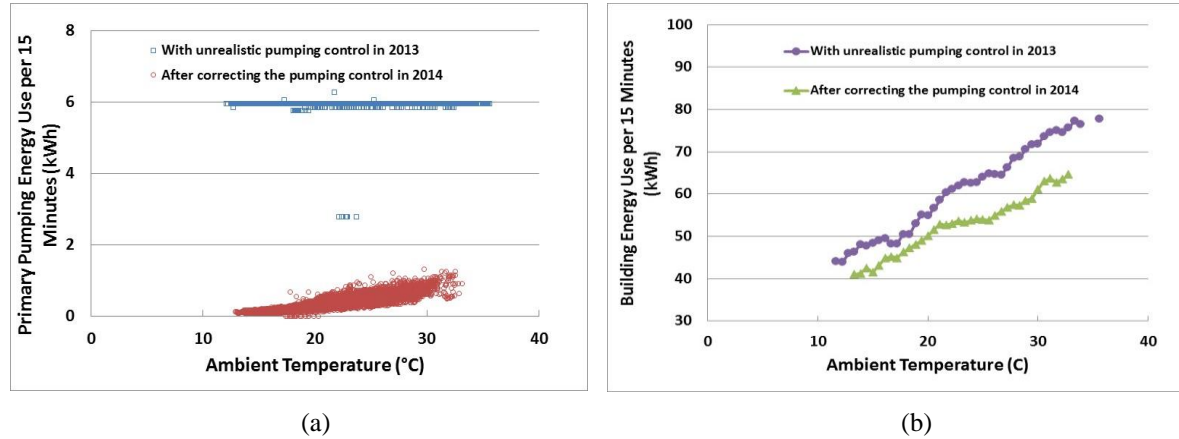


Fig. 8. An example showing impacts of pumping control: (a) pumping energy consumption, and (b) building energy consumption.

## 5. Conclusions

Results of ten case studies indicate the demonstrated GSHP systems consumed 27%–66% less source energy for conditioning buildings compared with conventional HVAC systems. Further, they reduced CO<sub>2</sub> emission in the range of 21%–66%. Operating costs were also reduced by 18%–65%. However, the installed costs are still higher than conventional HVAC systems. It is due, in part, to oversized CLVB-GHXs and overcomplicated system designs.

The operational efficiency of GSHP systems can be further improved with better system designs and controls. Excessive pumping could be reduced by improving pumping controls, and fan speeds of the VRF indoor units should be modulated to match their varying heating or cooling output to reduce fan energy. Balancing the heating/cooling output and heating/cooling demand is the key to realizing the benefits of HR chillers. Optimizing hot water temperature in the system design and operation can help maximize the use of GSHP for space heating and reduce or eliminate supplemental heating for boosting the hot water temperature. Finally, continuous performance monitoring and analysis are very helpful to identify abnormal operations and improve performance of GSHP systems.

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