

[Related Commercial Resources](#)

CHAPTER 35. GROUND-SOURCE HEAT PUMPS AND GEOTHERMAL ENERGY

THE use of ground thermal resources can be subdivided into three general categories: ground-source heat pump applications (generally $<90^{\circ}\text{F}$, which require a heat pump to provide useful energy), low- ($>90^{\circ}\text{F}$ and $<195^{\circ}\text{F}$) and intermediate-temperature ($>195^{\circ}\text{F}$ and $<300^{\circ}\text{F}$) geothermal direct-use applications, and high-temperature ($>300^{\circ}\text{F}$) geothermal electric power production. This chapter covers only ground-source heat pumps and direct-use geothermal energy systems with low and intermediate temperature. Design aspects of the building heat pump loop may be found in [Chapter 9 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#).

1. GROUND-SOURCE HEAT PUMPS

Ground-source heat pumps (GSHPs) were originally developed to heat and cool residential buildings but are also now widely applied in the commercial sector, with a primary goal of improving energy performance over conventional systems. Many installation recommendations and design guides appropriate to residential design must be amended for large buildings. In large buildings, GSHPs save not only energy but also water because they often displace cooling towers for cooling. Kavanaugh (1991) and Oklahoma State University (1988a, 1988b) discuss design and installation of ground-source heat pumps in more detail, but their focus is primarily residential and light commercial applications. Kavanaugh and Rafferty (2014) provide a more complete overview of the design of commercial and institution scale ground-source heat pump systems. For comprehensive additional coverage of commercial and institutional design and construction of ground-source heat pump systems, see ANSI/CSA/IGSHPA *Standard* C448-16.

1.1 TERMINOLOGY

The term **ground-source heat pump (GSHP)** is used for a variety of systems that use the ground, groundwater, or surface water as a heat source and sink. The general terms include **ground-coupled (GCHP)**, **groundwater (GWHP)**, and **surface-water (SWHP) heat pumps**. Many parallel terms exist (e.g., **geothermal heat pumps [GHPs]**, **geo-exchange**, and **ground-source [GS] systems**) and are used to meet a variety of marketing or institutional needs (Kavanaugh 1992). See [Chapter 9 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#) for a discussion of the merits of various other non-ground heat sources/sinks.

This chapter focuses primarily on the ground heat exchanger portion of GSHP systems, although the heat pump units used in these systems are unique to GSHP technology as well. GSHP systems typically use extended-range water-source heat pump units, in most cases of water-to-air configuration. Extended-range units are specifically designed for operation at entering water temperatures between 23°F in heating mode and 104°F in cooling mode. Units not meeting extended-range criteria are not suitable for use in GSHP systems (except for some groundwater heat pump systems). Some applications (e.g., groundwater loops, deep-surface-water loops, interior core zones of ground-coupled loops when perimeter zones require heating) include a free-cooling mode when water-loop temperatures fall near or below 55°F . This is typically accomplished by inserting a water coil in the return air stream before the refrigerant coil.

Ground-Coupled Heat Pump Systems

The GCHP is a subset of the GSHP and is often called a closed-loop heat pump. A GCHP system consists of a reversible vapor compression cycle that is linked to a closed ground heat exchanger (also called a **ground loop**) buried in soil ([Figure 1](#)). The most widely used unit is a water-to-air heat pump, which circulates water or a water/antifreeze solution through a liquid-to-refrigerant heat exchanger and a buried thermoplastic piping network. Heat pump units often include desuperheater heat exchangers (shown on the left in [Figure 1](#)). These devices use hot refrigerant at the compressor outlet to heat domestic hot water. A second type of GCHP is the **direct-exchange configuration (DXGCHP)**, which circulates the refrigerant directly (rather than a secondary heat transfer fluid) in a network of buried copper piping. DXGCHP systems are also referred to as direct geoexchange (DGX), or direct exchange geothermal systems. They are further subdivided into applications such as DGX-to-air or DGX-to-water, depending on the medium used to exchange heat with the building.

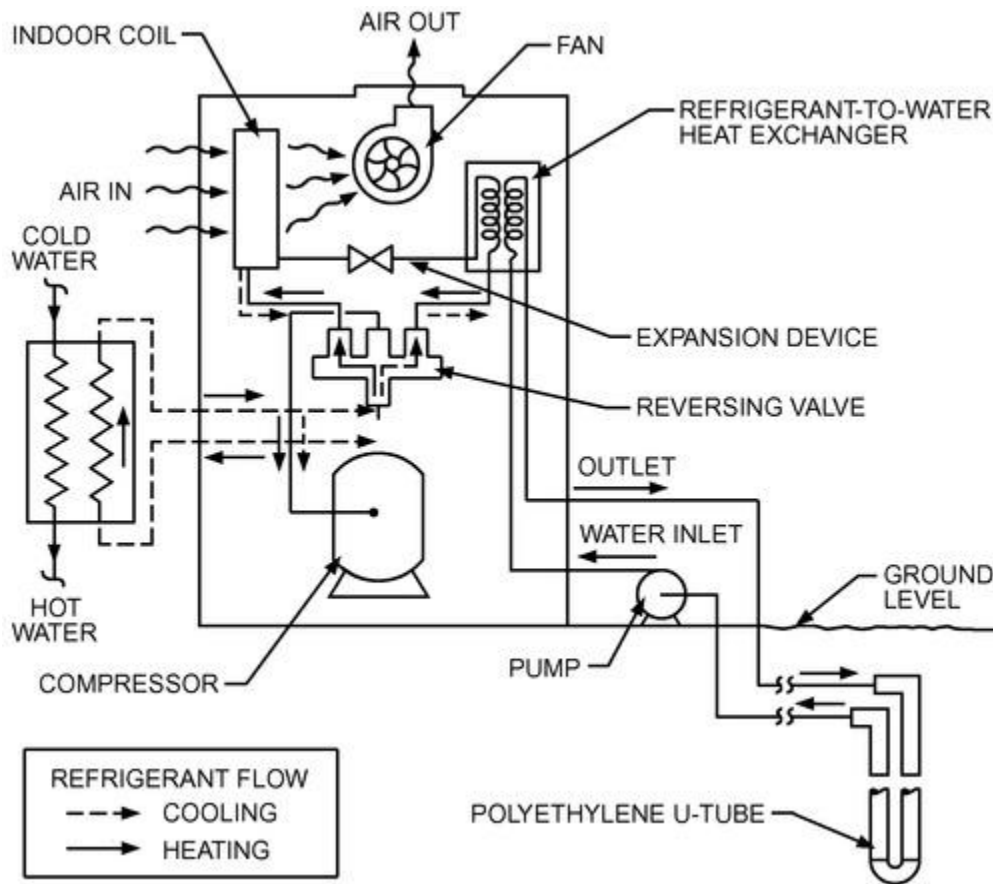


Figure 1. Vertical Closed-Loop Ground-Coupled Heat Pump System (Kavanaugh 1985)

The GCHP is further subdivided by whether its ground heat exchanger design is vertical or horizontal. **Vertical GCHPs** (Figure 2) generally consist of two small-diameter, high-density polyethylene (HDPE) tubes placed in a vertical borehole that is subsequently filled with a solid medium. The tubes are thermally fused at the bottom of the bore to a close return U-bend. Vertical tubes range from 0.75 to 1.5 in. nominal diameter. Bore depths normally range from 50 to 400 ft depending on local drilling conditions and available equipment, but can go to 600 ft or more if procedures for deep boreholes are followed (see the section on Pump and Piping System Options). Boreholes are typically 4 to 6 in. in diameter.

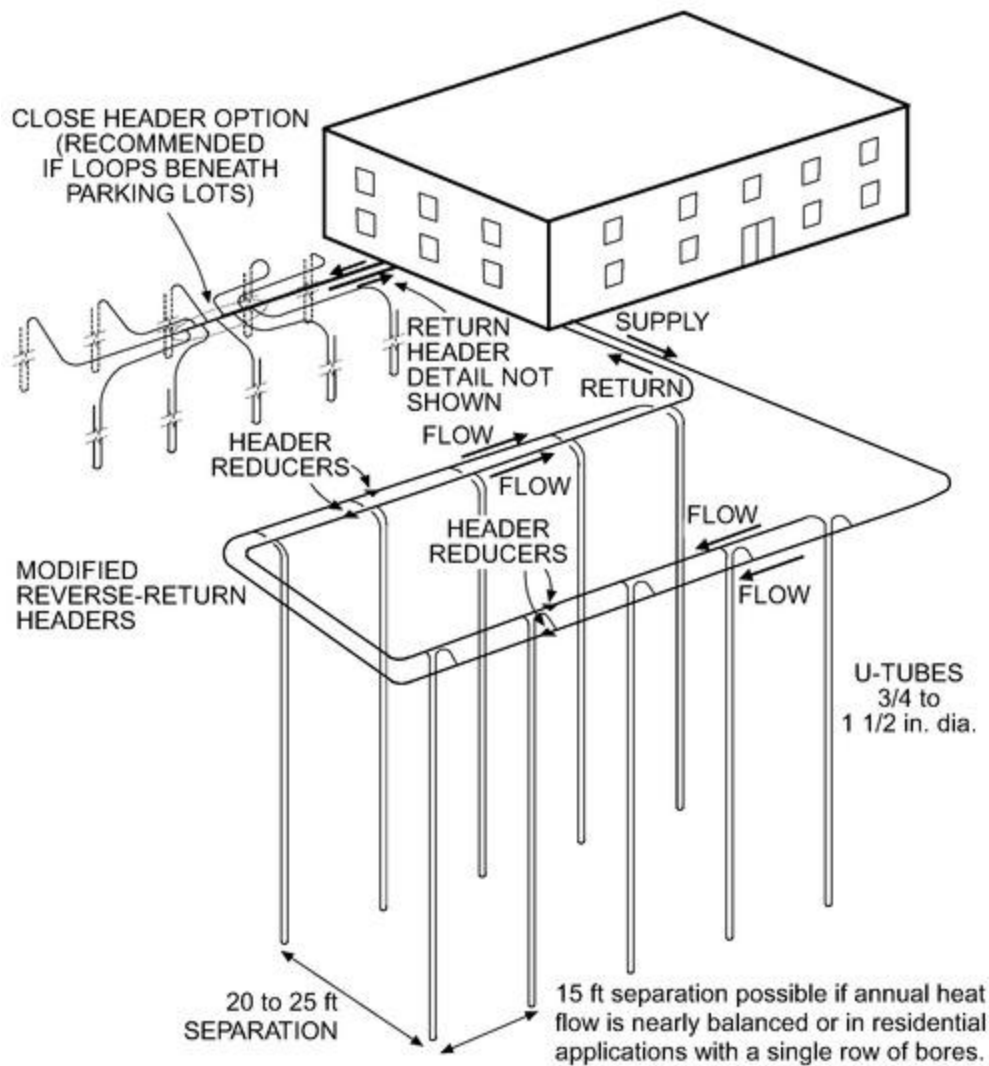


Figure 2. Vertical Ground-Coupled Heat Pump Piping

To reduce thermal interference between individual bores, a minimum borehole separation distance of 20 ft is recommended when loops are placed in a grid pattern. This distance may be reduced when bores are placed in a single row, the annual ground load is balanced (i.e., energy released in the ground is approximately equal to the energy extracted on an annual basis), or water movement or evaporation and subsequent recharge mitigates the effect of heat build-up in the loop field.

Advantages of the vertical GCHP are that it (1) requires relatively small plots of ground, (2) is in contact with soil that varies very little in temperature and thermal properties, (3) requires the smallest amount of pipe and pumping energy, and (4) can yield the most efficient GCHP system performance. Disadvantages are (1) typically higher cost because expensive equipment is needed to drill the borehole and (2) the limited availability of contractors to perform such work.

Horizontal GCHPs (Figure 3) include single-pipe, multiple-pipe, spiral (see Figure 27), and horizontally bored layouts. Single-pipe horizontal GCHPs are placed in narrow trenches at least 4 ft deep. These designs require the greatest amount of ground area. Multiple pipes (usually two, four, or six), placed in a single trench, can reduce the amount of required ground area. Trench length is reduced with multiple-pipe GCHPs, but total pipe length must be increased to overcome thermal interference from adjacent pipes. The spiral coil further reduces required ground area. These horizontal ground heat exchangers are made by stretching small-diameter polyethylene tubing from the tight coil in which it is shipped into an extended coil that can be placed vertically in a narrow trench or laid flat at the bottom of a wide trench. Recommended trench lengths are much shorter than those of single-pipe horizontal GCHPs, but pipe lengths must be much longer to achieve equivalent thermal performance. When horizontally bored loops are grouted and placed in the deep earth, as shown in the bottom of Figure 3, design lengths are near those for vertical systems, because annual temperature and moisture content variations approach deep-earth values.

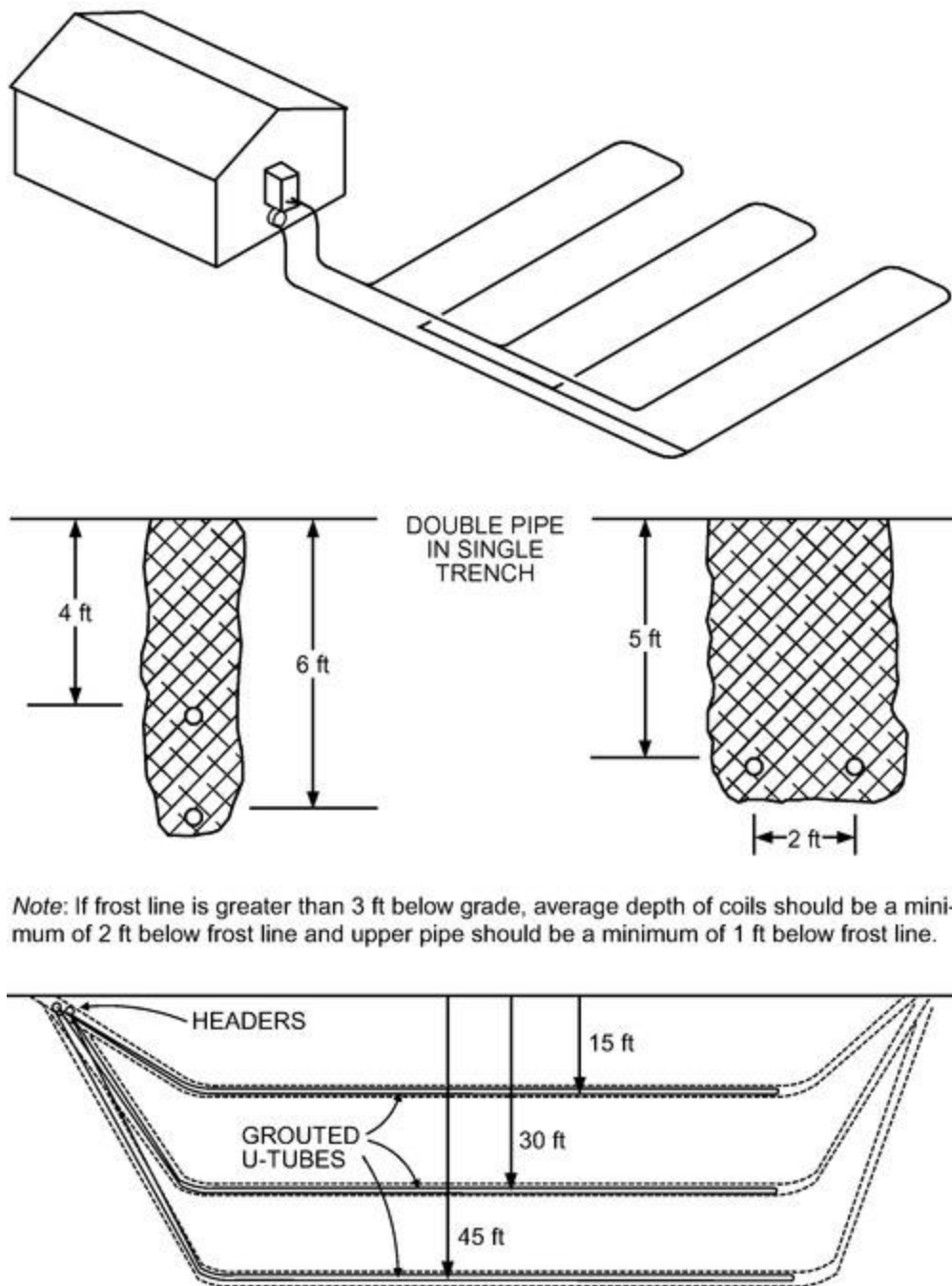


Figure 3. Trenched Horizontal (top) and Horizontally Bored (bottom) Ground-Coupled Heat Pump Piping

Advantages of horizontal GCHPs are that (1) they are typically less expensive than vertical GCHPs because relatively low-cost installation equipment is widely available, (2) many residential applications have adequate ground area, and (3) trained equipment operators are more widely available. Disadvantages include (1) a larger ground area requirement; (2) greater adverse variations in performance because ground temperatures and thermal properties fluctuate with season, rainfall, and burial depth; (3) slightly higher pumping-energy requirements; and (4) lower system efficiencies. Oklahoma State University (1988a, 1988b), Remund and Carda (2014), and Svec (1990) discuss design and installation of horizontal GCHPs.

Hybrid systems are a variation of ground-coupled systems in which a smaller ground heat exchanger is used, augmented in cooling mode by a fluid cooler or a cooling tower. This approach can have merit in large cooling-dominated applications. The ground heat exchanger is sized to meet the heating requirements. The downsized loop is used in conjunction with the fluid cooler or cooling tower with an isolation heat exchanger to meet the heat rejection load. Using the cooler reduces the capital cost of the ground heat exchanger in such applications, but somewhat increases maintenance requirements. For heavily heating-dominant applications, a downsized loop also can be augmented with an auxiliary heat source such as electric resistance, solar collectors, or fossil fuel.

Groundwater Heat Pump (GWHP) Systems

The second subset of GSHPs is groundwater heat pumps (Figure 4). Until the development of GCHPs, they were the most widely used type of GSHP. In the commercial sector, GWHPs can be an attractive alternative because large

quantities of water can be delivered from and returned to relatively inexpensive wells that require very little ground area. Whereas the cost per unit capacity of the ground heat exchanger is relatively constant for GCHPs, the cost per unit capacity of a well water system is much lower for a large GWHP system. A pair of high-volume wells can serve an entire building. Properly designed groundwater loops with correctly developed water wells require no more maintenance than conventional air and water central HVAC. When groundwater is injected back into the aquifer by a second well, net water use is zero.

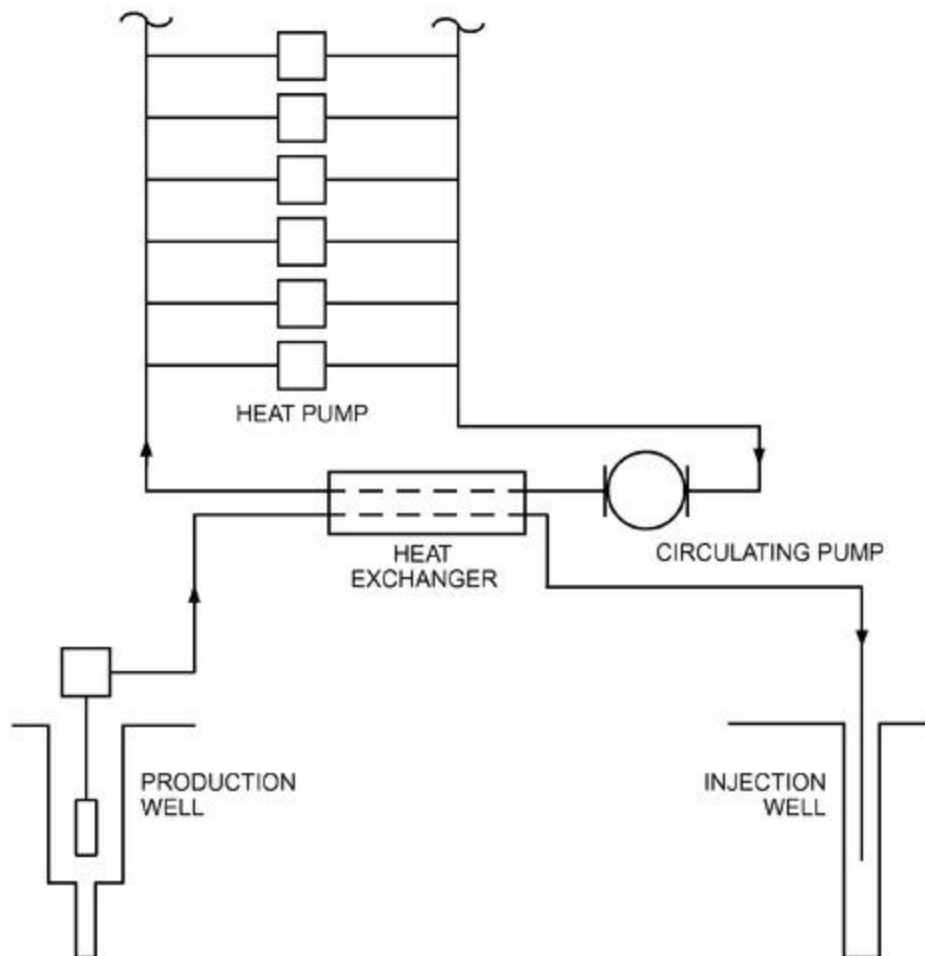


Figure 4. Unitary Groundwater Heat Pump System

One widely used design places a central water-to-water heat exchanger between the groundwater and a closed water loop, which is connected to water-to-air heat pumps in the building. A second possibility is to circulate groundwater through a heat recovery chiller (isolated with a heat exchanger), and to heat and cool the building with a distributed hydronic loop.

Both types and other variations may be suited for direct preconditioning in much of the United States. Groundwater below 60°F can be circulated directly through hydronic coils in series or in parallel with heat pumps. The cool groundwater can displace a large amount of energy that would otherwise have to be generated by mechanical refrigeration.

Advantages of GWHPs under suitable conditions are (1) they cost less than GCHP equipment, (2) the space required for the water well is very compact, (3) water well contractors are widely available, and (4) the technology has been used for decades in some of the largest commercial systems.

Disadvantages are that (1) local environmental regulations may be restrictive, (2) water availability may be limited, (3) fouling precautions may be necessary if groundwater is used directly in the heat pumps and water quality is poor, and (4) pumping energy may be high if the system is poorly designed or draws from a deep aquifer.

Surface Water Heat Pump Systems

Surface water heat pumps are included as a subset of GSHPs because of the similarities in applications and installation methods. SWHPs can be either closed-loop systems similar to GCHPs or open-loop systems similar to GWHPs. However, the thermal characteristics of surface water bodies are quite different than those of the ground or groundwater. Some unique applications are possible, though special precautions may be warranted.

Closed-loop SWHPs (Figures 5 and 36) consist of water-to-air or water-to-water heat pumps connected to a piping network (also called a **surface water loop**) placed in a lake, river, or other open body of water. A pump circulates water or a water/antifreeze solution through the heat pump water-to-refrigerant heat exchanger and the submerged

piping loop, which transfers heat to or from the body of water. The recommended piping material is thermally fused HDPE tubing with ultraviolet (UV) radiation protection.

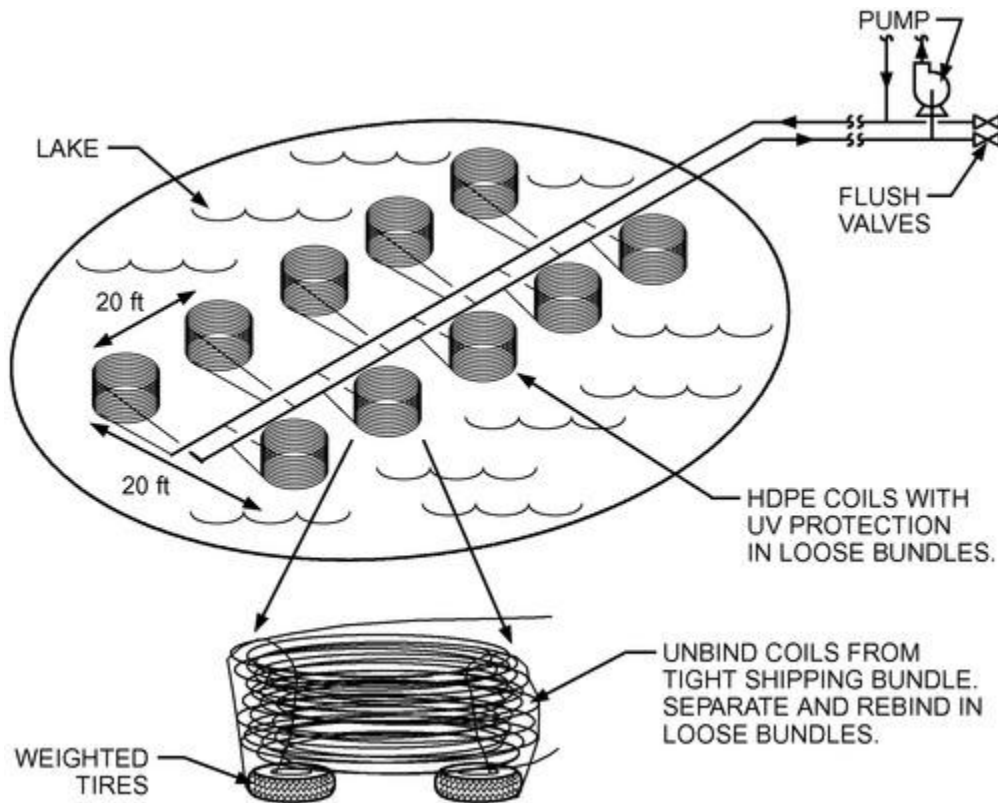


Figure 5. Lake Loop Piping

Advantages of closed-loop SWHPs are (1) relatively low cost (compared to GCHPs) because of reduced excavation costs, (2) low pumping-energy requirements, (3) low maintenance requirements, and (4) low operating cost. Disadvantages are (1) the possibility of coil damage in public lakes and (2) wide variation in water temperature with outdoor conditions if lakes are small and/or shallow. Such variation in water temperature would cause undesirable variations in efficiency and capacity, though not as severe as with air-source heat pumps.

Open-loop SWHPs can use surface water bodies the way cooling towers are used, but without the need for fan energy or frequent maintenance. In warm climates, lakes can also serve as heat sources during winter heating mode, but in colder climates where water temperatures drop below 45°F, closed-loop systems are the only viable option for heating.

Lake water can be pumped directly to water-to-air or water-to-water heat pumps or through an intermediate heat exchanger that is connected to the units with a closed piping loop. Direct systems tend to be smaller, having only a few heat pumps. In deep lakes (40 ft or more), there is often enough thermal stratification throughout the year that direct cooling or precooling is possible. Water can be pumped from the bottom of deep lakes through a coil in the return air duct. Total cooling is possible if water is 50°F or below. Precooling is possible with warmer water, which can then be circulated through the heat pump units. Large-scale cooling-only systems have been deployed successfully in some locations, including Cornell University and the city of Toronto (Cornell University 2006; Enwave [no date]).

SWHP Description and Performance. A SWHP system was installed in the resource center for a community college in Alabama in 2016. The upper two levels are a library, offices, a computer center, and study rooms. The lower level is a conference center with a multi-purpose room, two conference rooms, an office, and a kitchen.

Cooling and heating are provided by twenty water-to-air heat pumps (WAHPs) totaling 47 tons, three outside air WAHPs totaling 23 tons, and two 2 ton} ductless mini-split units (for a data center). The WAHPs are connected to a SWHP coil of sixty 500 ft 1 in. nominal high-density polyethylene (HDPE) coils as shown in [Figure 6](#). [Figure 7](#) shows the energy use and billed demand for the 20,000 ft² all-electric building, and [Figure 8](#) plots the inlet and outlet temperatures of the lake heat exchanger when local outdoor air temperatures were near the average for late August.

Installation Cost. The system mechanical cost was \$609,000 (\$30.45/ft² [Table 1](#)). This included \$90,000 (\$1286/ton) for 14.8% of the total for the lake heat exchanger. The cost for the twenty water-to-air heat pumps, three water-to-air outdoor air units, and two ductless mini-splits was \$151,000, or 24.8% of the total. Other significant costs were \$110,000 (18.0%) for ductwork, \$81,000 (13.3%) for controls, and \$76,000 (12.5%) for interior piping.

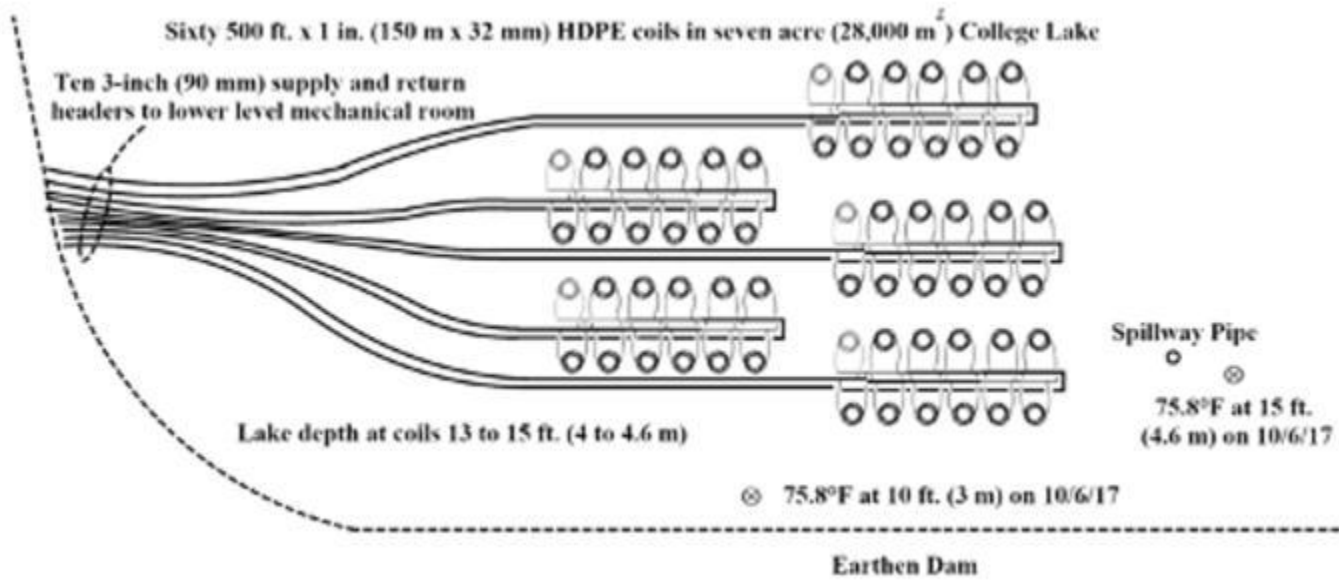


Figure 6. Lake Heat Exchanger and Nearby Temperatures

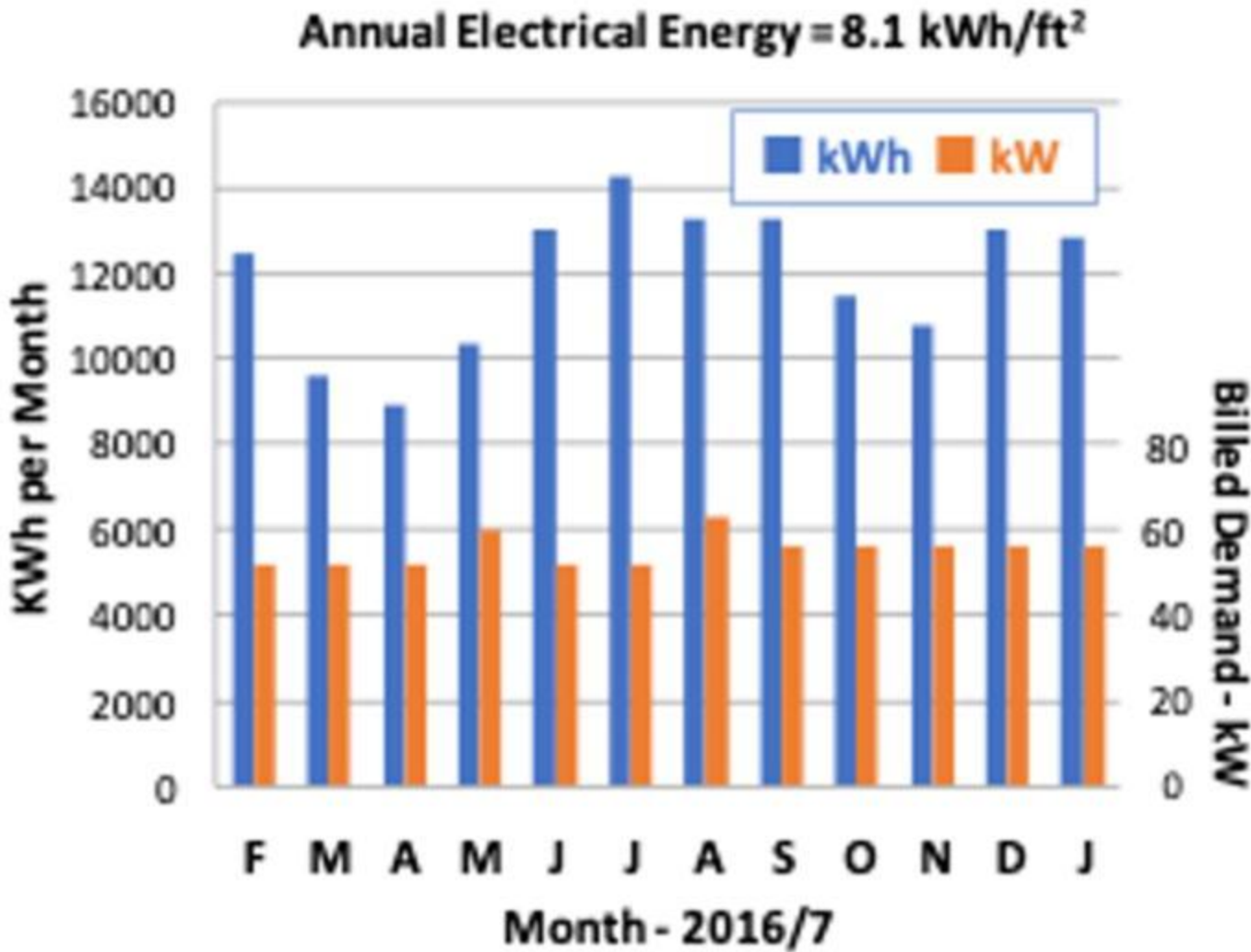


Figure 7. Monthly Energy Consumption and Billed Demand

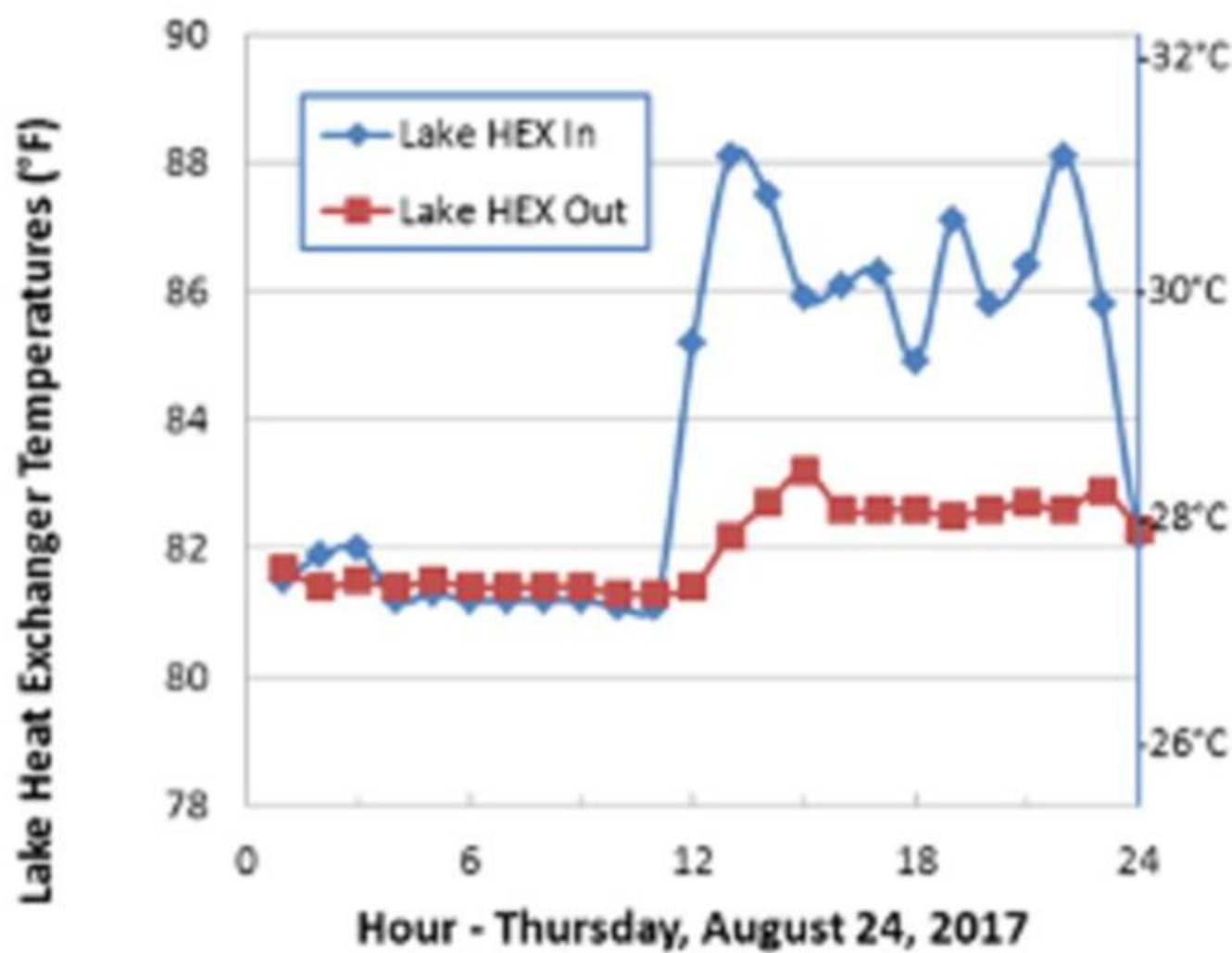


Figure 8. Summer Lake Loop Liquid Temperatures

1.2 GENERAL INFORMATION

Site Characterization

Site characteristics influence the type of GSHP system most suitable for a particular location. Site characterization is the evaluation of a site’s geology and hydrogeology with respect to its effect on GSHP system design. Important issues include presence or absence of water, depth to water, water (or soil/rock) temperature, groundwater quality, available land area depth to rock, rock type, and the nature and thickness of unconsolidated materials overlying the rock. Information about the nature of water resources at the site helps to determine whether an open-loop system may be possible. Depth to water affects pumping energy for an open-loop system and possibly the type of rig used for drilling closed-loop boreholes. Groundwater temperature in most locations is the same as the undisturbed ground temperature. These temperatures are key inputs to the design of GSHP systems. Available land area will impact the decision for vertical versus horizontal ground coupling, as well as the potential depth of vertical ground coupling. The types of soil and rock allow a preliminary evaluation of the range of thermal conductivity/diffusivity that might be expected. The thickness and nature of the unconsolidated (soil, gravel, sand, clay, etc.) materials overlying the rock affect whether casing is required in the upper portion of boreholes for closed-loop systems, a factor that increases drilling cost.

Table 1 SWHP System Installation Costs

Item	Cost, \$	% of Total
Lake loop	90,000	14.8
WAHPs (20), OA units (3), mini-splits (2)	151,000	24.8
Pumps (2) and accessories	20,500	3.4
Ventilation accessories	29,500	4.8
Hydronic (interior) piping	76,000	12.5
Ductwork mechanical room fabrication and installation	35,00	5.7
Insulation	40,000	6.6

Controls	81,000	13.3
Misc. and equipment	11,000	1.8
Total	609,000	100
Cost per unit floor area, \$/ft ²		30.45
Cost per unit floor conditioned area, \$/ft ²		35.41
Cost per unit cooling capacity, \$/ton		8700
Lake coil cost per unit cooling capacity, \$/ton		1286

After the GSHP system type has been decided, specific details about the subsurface materials' (rock/soil) thermal conductivity and diffusivity, water well static and pumping levels, drawdown, etc., are necessary to design the system. There are many sources for gathering site characterization information: geologic and hydrologic maps, state geology and water regulatory agencies, the U.S. Geological Survey (USGS 2000), and geotechnical studies of the site. Among the best sources of information are completion reports for nearby water wells. These reports are filed by the driller upon completion of a water well and provide a great deal of information of interest for both open- and closed-loop designs. The most thorough versions of well completion reports (level of detail varies by state) cover all of the issues of interest to GSHP designers. Information about access to and interpretation of these reports and other sources of information for site characterization is included in Rafferty (2000a) and Sachs (2002).

Once the type of system has been selected, more site-specific tests (e.g., ground thermal properties test for GCHP, well flow test for GWHP) can be used to determine the parameters necessary for system design. In many areas, ground heat exchangers are regulated by the state or other jurisdictions and under the jurisdiction of a state water rights authority, department of natural resources or environmental quality, or possibly a federal agency such as the U.S. Army Corps of Engineers. The regulation scope may include any type of ground-coupled system. The engineer or designer should be aware of and versed in regulatory issues affecting the project site. More recently, the U.S. Department of Homeland Security has required that certain activities, especially those close to drinking water sources, be excluded or stringently regulated. For security reasons, resource protection zones may not be found in public records archives. The simplest solution may be to contact the permitting authority.

Commissioning GSHP Systems

The design phase of GSHP commissioning requires a thorough site survey and characterization, accurate load modeling, and ensuring that the design chosen (and its documentation) meets the design intent.

The construction phase is dominated by observation of installation and verification of preliminary checks and tests. It also involves planning, training development, and other activities to help future building operators understand the HVAC system.

The acceptance phase starts with functional tests and verification of all test results. It continues with full documentation: completing the commission report to include records of design changes and all as-built plans and documents, and completing the operations and maintenance manual and system manual. Finally, after system testing and balancing is complete, the owner's operating staff are trained. The acceptance phase ends at substantial completion, at which date the warranty period begins.

[Table 2](#) provides information on tasks and participants involved in the GSHP commissioning process. Additional details on this topic, along with preventive maintenance and troubleshooting information, are included in Caneta Research (2001). Also, per ANSI/CSA/IGSHPA *Standard* C448-16, the contractor must provide the owner with a written maintenance procedure.

Codes and Standards

Current *Uniform Code* and *International Code* revisions now address ground-source heat pump systems. The *Uniform Code* now contains an independent volume, the *Uniform Solar Electric and Hydronic Code* (IAPMO 2021), which discusses ground-source piping for GSHP systems in Chapters 4 (Hydronics) and 7 (Geothermal Energy Systems). The *International Code's* Chapter 12 (Hydronic Piping) covers GSHP piping and systems; ground-source specific information is included in the last section (1210). In addition to standards issued by the International Ground Source Heat Pump Association (IGSHPA 2017), the Canadian Standards Association, in conjunction with U.S. industry and professional organizations, has released a binational standard, ANSI/CSA/IGSHPA *Standard* C448-16, which covers most forms of open- and closed-loop GSHP and GWHP systems.

1.3 GROUND-COUPLED HEAT PUMP SYSTEMS USING WATER-BASED HEAT TRANSFER FLUIDS

Ground-coupled heat pumps commonly use a secondary water-based heat transfer fluid to extract/reject heat from/to the ground. The fluid exchanges heat with the refrigerant in the heat pump and circulates through the ground in buried

thermoplastic tubing. This section discusses how to design different configurations of the ground heat exchanger, considering building loads and the related zone heat pump operations.

Vertical Design

This section provides an overview of a suggested design procedure for vertical, ground-coupled systems; related information and equations are discussed in more detail in Kavanaugh and Rafferty (2014). Several public software programs are available for performing the repetitive computations necessary for system optimization. Shonder et al. (1999, 2000) tested the accuracy of these programs, and agreement was attained with several programs in subsequent evaluations.

Table 2 Example of GSHP Commissioning Process for Mechanical Design

System	Function	Performed By	Witnessed By
Heat pump piping	Pressure test, clean, and fill	Contractor	A/E
Ground source piping	Pressure test, clean, fill, and purge air; check for compliance with ICC (2012) sections 1207 and 1208	Contractor Contractor	A/E —
Pumps	Inspect, test, and start up	Contractor	—
Heat recovery unit	Inspect, test, and start up; provide clean set of filters, staff instruction	Manufacturer Contractor Manufacturer	CA — CA/owner
Heat pump units	Inspect, test, and start up; provide clean filters, staff instruction	Manufacturer Contractor Manufacturer	— — CA/owner
Chemical treatment	Flushing and cleaning, chemical treatment, staff instruction	Contractor Contractor/manufacturer Manufacturer	A/E and CA — CA/owner
Balancing	Balancing, spot checking, follow-up site visits	TAB contractor TAB contractor TAB contractor	— A/E and CA CA
Controls	Installation/commissioning, staff instruction, performance testing, seasonal testing	Contractor CA CA CA	— CA/owner — —

Source: Caneta (2001).

CA = Commissioning authority

A/E = Architect/engineer

TAB = Testing, adjusting, and balancing

Table 3 Thermal Properties of Selected Soils, Rocks, and Bore Grouts/Fills

	Dry Density, [*] lb/ft ³	Conductivity, [*] Btu/h·ft·°F	Diffusivity, ft ² /day
Soils			
Heavy clay, 15% water	120	0.8 to 1.1	0.45 to 0.65
5% water	120	0.6 to 0.8	0.5 to 0.65
Light clay, 15% water	80	0.4 to 0.6	0.35 to 0.5
5% water	80	0.3 to 0.5	0.35 to 0.6
Heavy sand, 15% water	120	1.6 to 2.2	0.9 to 1.2
5% water	120	1.2 to 1.9	1.0 to 1.5
Light sand, 15% water	80	0.6 to 1.2	0.5 to 1.0
5% water	80	0.5 to 1.1	0.6 to 1.3
Rocks			
Granite	165	1.3 to 2.1	0.9 to 1.4

Limestone	150 to 175	1.4 to 2.2	0.9 to 1.4
Sandstone		1.2 to 2.0	0.7 to 1.2
Shale, wet	160 to 170	0.8 to 1.4	0.7 to 0.9
dry		0.6 to 1.2	0.6 to 0.8
Grouts/Backfills		Liquid [*] Density, lb/ft ³	Conductivity, [*] Btu/h·ft·°F
Bentonite (20 to 30% solids)		9.32 to 9.81	0.42 to 0.45
10-25% bentonite/20-50% SiO ₂ sand/35-55% mix water		11.27 to 13.51	0.57 to 0.95
8-12% bentonite/55-65% SiO ₂ sand/28-34% mix water		14.39 to 14.93	1.00 to 1.20
Low-density bentonite/graphite (plus additives) [*]		10.00 to 12.00	0.79 to 1.60
Neat cement (not recommended)		10.40 to 14.80	0.88 to 1.60
30% concrete/70% SiO ₂ sand (plus plasticizer)		13.80 to 16.00	0.40 to 0.45

^{*} Intermediate densities and thermal conductivities can be obtained by mixing silica sand and graphite in different proportions. Contact grout manufacturer for additional information on thermal properties and density of various grout silica sand/graphite mixtures.

A more recent publication (Kavanaugh 2008) updates the design recommendations for GCHP systems as follows:

1. Calculate peak zone cooling and heating loads, and estimate off-peak loads.
2. Estimate annual heat rejection into and absorption from the ground heat exchanger to account for potential ground temperature change.
3. Select preliminary loop operating temperatures and flow rate to begin optimization of first cost and efficiency (selecting temperatures near normal ground temperature results in high efficiencies but larger and more costly ground heat exchangers).
4. Select heat pumps to meet cooling and heating loads, and locate units to ensure accessibility for maintenance and to minimize duct cost and fan power and noise.
5. Arrange heat pump into ground heat exchanger circuits to minimize system cost, pump energy and electrical demand.
6. Conduct site survey to determine ground thermal properties and drilling conditions (see following recommendations).
7. Determine and evaluate possible loop field arrangements that are likely to be optimum for the building and site (bore depth, separation distance, completion methods, annulus grout/fill, and header arrangements); include subheader circuits (typically 5 to 15 U-tubes on each) with isolation valves to allow air and debris flushing of sections of loop field through a set of full-port purge valves.
8. Determine optimum ground heat exchanger dimensions with [Equations \(7\)](#) and [\(8\)](#) or software; one or more alternatives (depth, number of bores, grout/fill material, etc.) that provide equivalent performance may yield more competitive bids.
9. Iterate to determine optimum operating temperatures, flows, loop field arrangement, depth, bores, grout/fill materials, etc.
10. Lay out interior piping and compute head loss through critical path.
11. Select pumps and control method, determine system efficiency, and consider modifying water distribution system if pump demand exceeds 8% of the system total demand or air distribution system if fan demand exceeds 12% of the system total.

Deliverables from this process that are necessary to adequately describe a GCHP installation include, as a minimum,

- Heat pump specifications at rated conditions
- Pump(s) specifications, expansion tank size, and air separator
- Fluid specifications: system volume, inhibitors, antifreeze concentration (if required), water quality, etc.

- Design operating conditions: entering and leaving ground heat exchanger temperatures, return air temperatures (including wet bulb in cooling), airflow rates, and liquid flow rates
- Pipe header details with ground heat exchanger layout, including pipe diameters, spacing, and clearance from building and utilities
- Bore depth and approximate bore diameter
- Piping material specifications, and visual inspection and pressure testing requirements
- Grout/fill specifications: thermal conductivity and acceptable placement methods to eliminate voids
- Purge provisions and flow requirements to ensure removal of air and debris without reinjecting air when switching to adjacent subheader circuits
- Instructions on connecting to building loop(s) and coordinating building and ground heat exchanger flushing
- If applicable, a drilling report from the thermal properties test borehole that includes the type of equipment used (rig, bit, etc.), drilling fluid (air, foam, drilling mud), depth of hole, description of drilled soil or rock, time needed to drill the borehole, any special conditions encountered.
- Sequence of operation for controls

Thermal Property Testing. In the design of vertical GCHPs, accurate knowledge of soil/rock formation thermal properties is critical. These properties can be estimated in the field by installing a loop of approximately the same size and depth as the heat exchangers planned for the site. The test loop location should be chosen with care, and designed to be used for the eventual full borefield, especially if a GSHP is a likely final system choice (this may require the test loop to meet all local ground heat exchanger standards). Heat is added in a water loop at a constant rate, and data are collected as shown in [Figure 9](#). Inverse methods are applied to find thermal conductivity, diffusivity, and temperature of the formation. These methods are based on either the line source (Gehlin 1998; Mogensen 1983; Witte et al. 2002), the cylindrical heat source (Ingersoll and Zobel 1954), or a numerical algorithm (Austin et al. 2000; Shonder and Beck 1999; Spitler et al. 1999). More than one of these methods should be applied, when possible, to enhance reported accuracy. Recommended test specifications are as follows (Kavanaugh 2000, 2001):

- Accuracy of temperature measurement and recording devices should be $\pm 0.5^\circ\text{F}$.
- Flow rates should be sufficient to provide a differential loop temperature of 6 to 12°F . This is the temperature differential for an actual heat pump system.
- A waiting period of five days is suggested for low-conductivity soils ($k < 1.0 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$) after the ground heat exchanger has been installed and grouted (or filled) before the thermal conductivity test is initiated. A delay of three days is recommended for higher-conductivity formations ($k > 1.0 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$).
- The initial ground temperature measurement should be made at the end of the waiting period by directly inserting a probe inside a liquid-filled ground heat exchanger at three locations, representing the average, or by temperature measurement as liquid exits the loop during the period immediately after start-up.
- Data collection should be at least once every 10 min.
- All aboveground piping should be insulated with a minimum of 0.5 in. closed-cell insulation or equivalent. Test rigs should be enclosed in a sealed cabinet that is insulated with a minimum of 1.0 in. fiberglass insulation or equivalent.
- If retesting a bore is necessary, loop temperature should be allowed to return to within 0.5°F of the pretest initial ground temperature. This typically requires a 10- to 12-day delay in mid- to high-conductivity formations and 14 days in low-conductivity formations if a complete 48 h test has been conducted. Waiting periods can be proportionally reduced if tests are shorter.

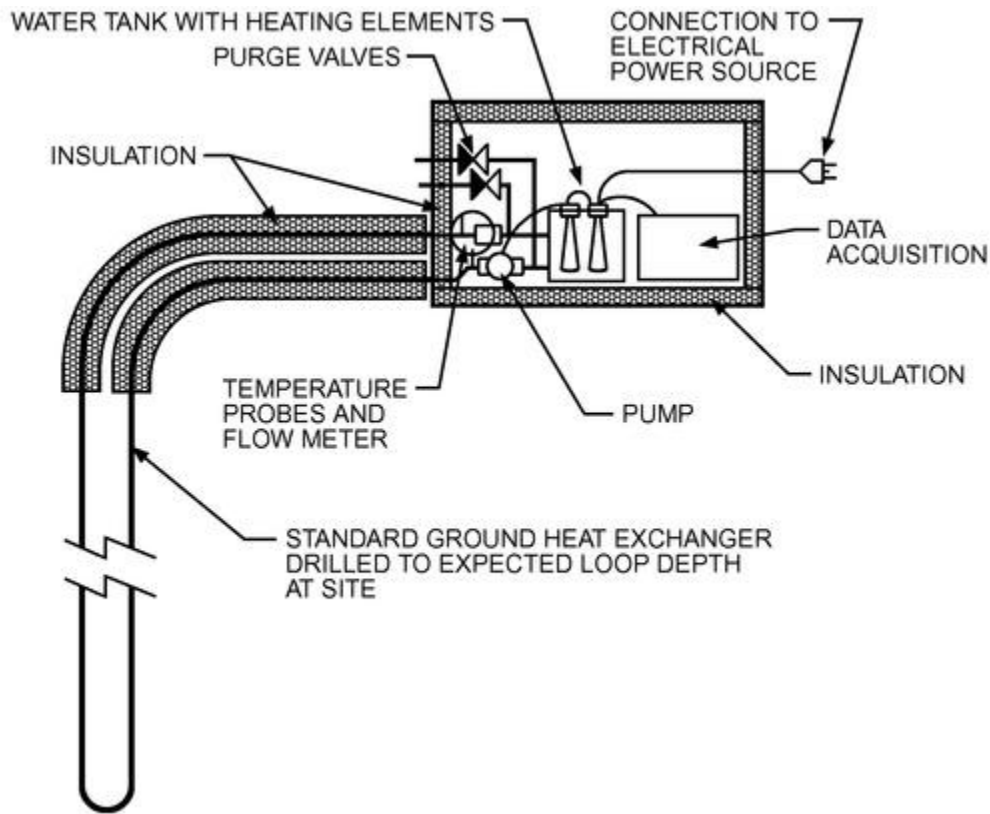


Figure 9. Thermal Properties Test Apparatus

Thermal Property Test Example Calculation. There are several methods to interpret a thermal property test. This example uses a graphical method based on the line-source equation (Ingersoll et al. 1954) to determine the effective ground thermal conductivity and the cylindrical heat source equation to obtain the effective bore thermal resistance.

Assuming that heat injection q° is constant, it can be shown that the mean fluid temperature in the borehole as a function of time $T_f(t)$ is given by

$$T_f(t) = T_g + \frac{q}{L_b} R_b + \frac{q/L_b}{4\pi k_g} \left[\ln \left(\frac{4\alpha_g t}{r_b^2} \right) - \gamma \right] \quad (1)$$

where the term in brackets is the line source approximation to radial heat transfer in the ground, α_g is the ground thermal diffusivity ($\alpha_g = k_g/\rho c_p$), γ is Euler's constant ($= 0.5772$), and

$$k_g = \frac{q}{4\pi L_b \times m} = \frac{2602 \text{ W} \times 3.41 \text{ Btu/Wh}}{4\pi \times 244 \text{ ft} \times 2.4827^\circ\text{F/h}} = 1.17 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

[Equation \(1\)](#) can be rearranged to give

$$T_f(t) = b + m \times \ln(t) \quad \text{with } m = (q/L_b)/4\pi k_g \quad (2)$$

In essence, the graphical method consists of plotting the time evolution of the mean fluid temperature during the TP test to get the slope m° and evaluate k_g with [Equation \(2\)](#). [Figure 10](#) is a semi-log plot of average fluid temperature versus time. Note that the early time test data should not be considered because of transient conditions in the borehole. In the case of [Figure 10](#), the first four hours of testing were not used. As shown in [Figure 10](#), the slope m° of temperature versus $\ln(\text{time})$ is 2.4827. For this example, the input power is 2606 W and bore length L_b is 244 ft.

Thus,

$$k_g = \frac{3.41 \text{ Btu/Wh} \times \text{W/W}}{4\pi L_b \times m} = \frac{3.41 \times 2602 \text{ W}}{4\pi \times 244 \text{ ft} \times 2.4827} = 1.17 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$$

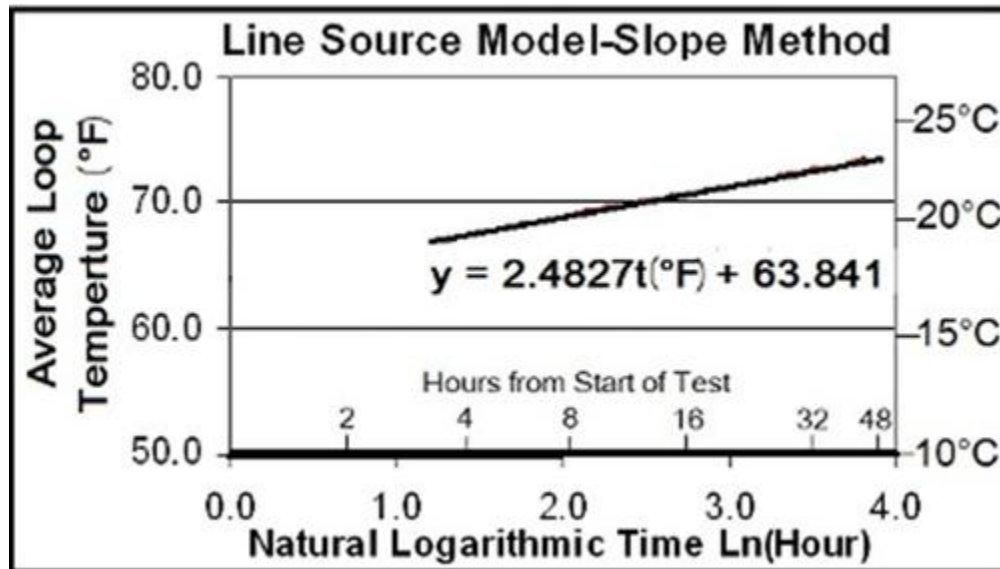


Figure 10. Example Thermal Property Test Results

This value represents the effective (or average) ground thermal conductivity over the borehole length. A third thermal property, thermal diffusivity ($\alpha_g = k_g/\rho c_p$) is determined from estimated values for density ρ and specific heat c_p , from drilling log data and values from sources with more detail than [Table 3](#) such as Kavanaugh and Rafferty (2014).

Borehole Thermal Resistance. The borehole thermal resistance R_b (h·ft·°F/Btu) is the combined effects of the internal convective resistance, the U-tube walls, the grout or fill in the annular region, and contact resistances. Contact resistances are minimal unless the annulus is not completely grouted or filled, or the grout remains rigid when the tubing shrinks. There is a degree of uncertainty in mathematically determining borehole resistance since the exact location of the heat exchanger tube in the annulus varies. However, it is possible to estimate the borehole thermal resistance from a TP test. [Equation \(1\)](#) can be rearranged to give the effective borehole thermal resistance:

$$R_b = \frac{T_f(t) - T_g}{q/L_b} - \frac{1}{4\pi k_g} \left[\ln \left(\frac{4\alpha_g t}{r_b^2} \right) - \gamma \right] \quad (3)$$

where the first and second terms on the right represent the total thermal resistance from fluid to ground and the ground thermal resistance R_g (Kavanaugh 2010). A slightly more accurate representation of R_g is given by the cylindrical heat source solution (Ingersoll et al. 1954) where $R_g = G_{Fo}/k_g$.

The thermal resistance of the ground is a function of time t from the instantiation of a constant heat rate, bore diameter d_b , and thermal properties ($\alpha_g = k_g/\rho c_p$) expressed in terms of a Fourier number ($Fo = 4\alpha_g t/d_b^2$). The Fourier number is used to find the G-factor as described by Carslaw and Jaeger (1947).

A graphical correlation of Fourier number vs. G-factor for a constant heat rate from a cylinder to the ground is provided in [Figure 12](#).

Borehole Resistance Example Calculation. The value for the thermal property test shown in [Equation \(1\)](#) and [Figure 10](#) are used to demonstrate the computation of borehole resistance. Although this example only computes borehole resistance at 48 h, it should be calculated at several times during the test. A well-conducted test will result in consistent values for borehole resistance at any time 8 to 12 h after the start of the test. Additional values that would be provided from a TP test include

T_g	=	56°F
T_f (48 h)	=	72.7°F
d_b	=	0.5 ft
ρ	=	130 lb/ft ³
c_p	=	0.24 Btu/lb·°F

Thus,

$$\alpha_g = k_g / \rho c_p = 1.17 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} / (130 \text{ lb/ft}^3 \times 0.24 \text{ Btu/lb} \cdot ^\circ\text{F}) \\ = 0.0375 \text{ ft}^2/\text{h}$$

$$\text{Fo} = 4\alpha_g t / d_b^2 = 4 \times 0.0375 \text{ ft}^2/\text{h} \times 48 \text{ h} / (0.5 \text{ ft})^2 = 28.8$$

From [Figure 12](#),

$$G_{\text{Fo}} \text{ (for Fo} = 28.8) \approx 0.34 \text{ and}$$

$$R_g = G_{\text{Fo}} / k_g = 0.34 / 1.17 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} = 0.291 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

[Equation \(4\)](#) is used to compute borehole resistance:

$$R_b = \frac{(T_f - T_g)L_b}{q} = \frac{(72.7 - 56^\circ\text{F})244 \text{ ft}}{3.41 \text{ Btu/Wh} \times 2606 \text{ W}} \\ = 0.458 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu} \quad (4)$$

Ground Heat Exchanger Sizing. This is perhaps the most critical step in the design of a vertical GCHP. Ground-loop design methods must proceed with limited information; a major missing component is long-term, field-monitored data, which are needed to further validate the design method to address effects of water movement and long-term heat storage more fully. The conservative designer can assume no benefit from water movement; designers who assume maximum benefit must ignore annual imbalances in heat rejection and absorption.

Two design methods are presented in the following section. Both methods have been implemented in design software tools, and are based on the assumption that heat transfer in the ground is governed by conduction only. The **concentric cylinder source** method is based on the solution of the equation for heat transfer from a cylinder buried in the earth. This equation was developed and evaluated by Carslaw and Jaeger (1947) and was suggested by Ingersoll and Zobel (1954) as an appropriate method of sizing ground heat exchangers. Kavanaugh (1985) adjusted the method to account for the U-bend arrangement and hourly heat rate variations. Alternative design methods are described by Eskilson (1987), Morrison (1997), Philippe et al. (2010), Spitler (2000), and Spitler et al. (2000). A second method, attributed to Eskilson, is presented after this first method. Finally, a review of vertical borehole ground heat exchanger design methods has been presented by Spitler and Bernier (2016).

Vertical Ground Heat Exchanger Sizing using the Concentric Cylinder Source Method. The method of Ingersoll and Zobel (1954), based on the following steady-state heat transfer equation, can be used to size vertical ground heat exchangers:

$$q = \frac{L(t_g - t_w)}{R_{ov}} \quad (5)$$

where

q	=	heat transfer rate, Btu/h
L	=	required total bore length, ft
t_g	=	ground temperature, $^\circ\text{F}$
t_w	=	liquid temperature, $^\circ\text{F}$
R_{ov}	=	overall resistance of ground and bore, $\text{ft} \cdot \text{h} \cdot ^\circ\text{F}/\text{Btu}$

The heat rate delivered to the ground in the cooling mode by the condenser includes the heat of the heat pump and auxiliary equipment. Thus, q_{cond} can be calculated to be

$$q_{\text{cond}}/q_{lc} = \frac{\text{EER} + 3.412}{\text{EER}} \quad (6)$$

where

EER	=	cooling energy efficiency ratio, Btu/h·W
q_{cond}	=	heat pump condenser heat rejection rate to ground, Btu/h
q_{lc}	=	building design cooling block load, Btu/h
q_{lh}	=	building design heating block load, Btu/h

However, the heat of the heat pump and auxiliary equipment in heating mode is delivered to the building. Thus the heat removed from the ground by the evaporator is

$$q_{\text{evap}}/q_{lh} = \frac{\text{COP} - 1}{\text{COP}} \quad (7)$$

where q_{evap} is the heat pump evaporator heat extraction rate from ground, Btu/h; q_{lh} is the building design heating block load, Btu/h; and COP is the heating coefficient of performance, W/W. The design (e.g., peak) block load is the average building load during the block (e.g., a specified time period, usually a few hours) on the design day (e.g., worst-case weather and occupancy conditions). Ground heat exchanger design also requires calculation of the monthly part-load factors (PLFs), which are the actual monthly loads divided by the monthly load if the building operated continuously at the design block load. Both the design block load and PLFs can be computed using building simulations or design guidelines (see ACCA [2008, 2016] and [Chapters 17 and 18 of the 2021 ASHRAE Handbook—Fundamentals](#)).

Table 4 Summary of Potential Completion Methods for Different Geological Regime Types

Geological Regime Type	Grout			Backfill with Cuttings or Sand/Gravel Mix	Two-Fill with	
	$0.4 < k \leq 0.8$ Btu/h·ft·°F	$0.8 < k \leq 1.2$ Btu/h·ft·°F	$k > 1.2$ Btu/h·ft·°F		Cuttings or Sand/Gravel Mix Below Aquifers	Other* Below Aquifers
Clay or low-permeability rock,						
no aquifer	—	Yes	Yes	—	Yes	Yes
single-aquifer	—	Yes	Yes	—	—	Yes
multiple-aquifer	Yes	Yes	Yes	Yes	Yes	Yes
Permeable rock,						
no shallow aquifers	—	Yes	Yes	Yes	Yes	Yes
single-aquifer	—	Yes	Yes	Yes	Yes	Yes
multiple-aquifers	—	Yes	Yes	Yes	—	—
Karst terrains with secondary permeability	—	Yes	Yes	Yes	—	—
Fractured terrains with secondary permeability	—	Yes	Yes	Yes	Yes	Yes

Yes = Recommended potentially viable backfill methods

* Use of backfill material that has thermal conductivity of $k \geq 1.4$ Btu/h·ft·°F

[Equation \(4\)](#) can be rearranged to solve for the required bore length L . The steady-state [Equation \(4\)](#) is modified to represent the variable heat rate of a ground heat exchanger by dividing the heat transfer q into a series of constant-heat-rate pulses. The thermal resistance R_{ov} is divided into contributions from the ground and borehole. The effective thermal resistance of the ground per unit length is calculated as a function of time corresponding to the time span over which a particular heat pulse occurs (annual R_{ga} , monthly R_{gm} , or peak short-term R_{gst}); the effective resistance is different from a steady-state resistance in that it accounts for the transient heat flow in the ground. The borehole thermal resistance R_b includes the thermal resistance of the pipe wall and interfaces between the pipe and fluid and the pipe and the ground. The resulting equation takes the following form for the required length to satisfy cooling loads:

$$L_c = \frac{q_a R_{ga} + (q_{lc} - 3.41 W_c)(R_b + \text{PLF}_m R_{gm} + F_{sc} R_{gst})}{t_g - \frac{t_{wt} + t_{wo}}{2} - t_p} \quad (8)$$

The required length for heating is

$$L_h = \frac{q_a R_{ga} + (q_{lh} - 3.41 W_h)(R_b + \text{PLF}_m R_{gm} + F_{sc} R_{gst})}{t_g - \frac{t_{wt} + t_{wo}}{2} - t_p} \quad (9)$$

where

- F_{sc} = short-circuit heat loss factor
 L_c = required total bore length for cooling, ft
 L_h = required total bore length for heating, ft

PLF_m	=	part-load factor during design month
q_a	=	net annual average heat transfer to ground, Btu/h
R_{ga}	=	effective thermal resistance of ground (annual pulse), $\text{ft}\cdot\text{h}\cdot^\circ\text{F}/\text{Btu}$
R_{gst}	=	effective thermal resistance of ground (peak short term) 1 to 6 h recommended, $\text{ft}\cdot\text{h}\cdot^\circ\text{F}/\text{Btu}$
R_{gm}	=	effective thermal resistance of ground (monthly pulse), $\text{ft}\cdot\text{h}\cdot^\circ\text{F}/\text{Btu}$
R_b	=	borehole thermal resistance, $\text{ft}\cdot\text{h}\cdot^\circ\text{F}/\text{Btu}$
t_g	=	undisturbed ground temperature, $^\circ\text{F}$
t_p	=	temperature penalty for interference of adjacent bores, $^\circ\text{F}$
t_{wi}	=	liquid temperature at heat pump inlet, $^\circ\text{F}$
t_{wo}	=	liquid temperature at heat pump outlet, $^\circ\text{F}$
W_c	=	system power input at design cooling load, W
W_h	=	system power input at design heating load, W

Note: Heat transfer rate, building loads, and temperature penalties are positive for heating and negative for cooling.

[Equations \(7\)](#) and [\(8\)](#) consider three different pulses of heat to account for long-term heat imbalances, average monthly heat rates during the design month, and maximum heat rates for a short-term period during a design day. This period could be as short as 1 h, but a 4 to 6 h block is recommended.

The required total bore length is the larger of the two lengths L_c and L_h calculated with [Equations \(7\)](#) and [\(8\)](#). The heat exchanger will be oversized during the season with shorter calculated L ; the resulting increase in efficiency lowers operating costs for that season. However, oversizing the heat exchangers increases first costs, so designers may consider using the shorter calculated L , and supplementing the GSHP system with season-specific equipment (e.g., a cooling tower for cooling, a boiler for heating) to address loads for the season with the longer L . See the section on Hybrid System Design for more information about these configurations.

Thermal resistance of the ground is calculated from ground properties, pipe dimensions, grout/fill thermal conductivity, and operating periods of the representative heat rate pulses. [Table 3](#) lists typical thermal properties for soils and fills for the annular region of the boreholes. Type of fill material depends on thermal, regulatory, and economic considerations. Historically, a relatively low-thermal-conductivity bentonite grout common in the water well industry had been used. More recently, thermally enhanced grouts have been developed to supplement or replace conventional grouts. Thermally enhanced grout has three primary components:

- **Bentonite** provides sealing properties to the mixture and suspends the thermal additive in the bore column to provide uniform heat transfer from top to bottom.
- **Thermal additive** (either silica sand or graphite) improves overall grout thermal conductivity (TC) and subsequent heat transfer capabilities.
- **Mix water** amounts are specified by the manufacturer to ensure that the grout will perform as advertised.

As with any engineered product, the components must be mixed according to manufacturer specifications to meet the minimum thermal performance and permeability requirements for a given project.

In some cases, such as when voids, fissures, or caverns are present, drill cuttings or manufactured sand/gravel mixes have been placed instead of bentonite or thermally enhanced grout. Note that placing such fill material from the surface may cause the borehole to bridge, leaving voids or ungrouted sections of the borehole. Also, thermal performance of drill cuttings or manufactured sand/gravel mix is subject to factors such as final placement density and height of the static water table, and is thus difficult to quantify. Because thermal properties of the fill material are critical to overall system performance, use of a tremie pipe to inject grout from the bottom upward is recommended. Nutter et al. (2001) contains a detailed evaluation of potential fills and grouts for vertical boreholes. Also, Jenkins (2009), Sachs (2002), and Skouby (2011) have additional recommendations regarding grout and grout placement.

[Table 4](#) summarizes potential completion methods for various geological conditions. "Two-fill" refers to the practice of placing a low-permeability material in the upper part of the hole and/or at intervals where needed to separate individual aquifers, and a more thermally advantageous material in the remaining intervals. When backfill completion methods are allowed in lieu of pressure-tremie grouting, the designer should be aware that thermal properties and subsequent system performance is subject to final backfill density and the location/height of the static water table, as previously mentioned.

Borehole thermal resistance, from the fluid to the borehole wall, considers the effects of pipe resistance R_p and bore annulus grout resistance R_{grt} :

$$R_b = R_p + R_{grt} \quad (10)$$

Pipe resistance includes the fluid's convective film resistance and the conductive resistance of the pipe walls. Contact resistances are negligible compared to the high resistance of plastic pipe walls and annular grouts. For a single U-tube (two tubes) the pipe resistance is

$$R_p = (R_{film} + R_{tube})/2 = [1/(\pi d_i h_{conv}) + \ln(d_o/d_i)/2\pi k_p]/2 \quad (11)$$

where h_{conv} is the convection coefficient inside the pipes, k_p is pipe thermal conductivity, d_o is the tube outer diameter, and d_i is the tube inner diameter.

A correlation for the grout's thermal resistance has been developed using shape factor correlations (Remund 1999):

$$R_{grt} = \left[\beta_0 \left(\frac{d_b}{d_o} \right)^{\beta_1} \times k_{grt} \right]^{-1} \quad (12)$$

where k_{grt} is the grout's thermal conductivity and d_b is bore diameter. Coefficients β_0 and β_1 in [Equation \(8\)](#) have been developed for three locations of the tubes, as shown in [Figure 11](#): centered in the bore and in contact each other (A), centered and spaced evenly in the bore (B), and centered and in contact with the bore wall (C). However, the most likely location of the U-tubes is BC, and coefficients for this location are unavailable. A similar but slightly more detailed solution was developed by Hellström (1991) and applied to a few design and simulation tools (Liu 2008; Philippe et. al. 2010). More recently, Javed and Spitler (2017) examined the accuracy of various methods to calculate borehole thermal resistance.

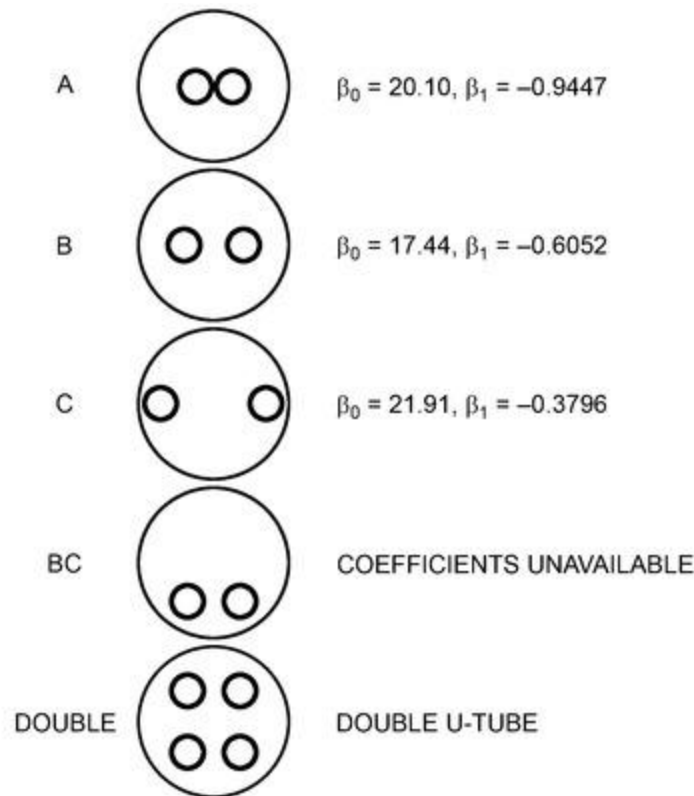


Figure 11. Coefficients for [Equation \(8\)](#)

Because locations of U-tubes cannot be determined even when spacers are installed, exact computation of bore thermal resistance values is somewhat uncertain. It is possible to apply the results from thermal property tests to calculate the bore thermal resistance if the U-tube dimensions, grout conductivity, and borehole diameter are known (Kavanaugh 2010). Thermal property tests were conducted at 15 installations where these values were known and the bore resistance was calculated. The field calculated bore resistances best matched the values computed with [Equations \(9\) to \(11\)](#) using

- Location C at 4 (27%) of the sites
- An average of location B and location C at 5 (33%) of the sites
- Location B at 5 (33%) of the sites, and
- Location A at 1 (7%) site

[Table 5](#) provides the bore resistances computed using [Equations \(9\) to \(11\)](#) for three different grout conductivities, three different fluid flow regimes (Reynolds number = 2000 [laminar], 4000 [transition], and 10,000 [fully turbulent]), three different U-tubes sizes, and three different bore diameters for locations B and C. Resistance is also computed for

a double U-tube in a bore. These values were calculated with a value of k_p equal to 0.24 Btu/h·ft·°F. Designers can choose to use the values of location B (conservative), BC (average), C (risky), or Double.

Table 5 Thermal Resistance of Bores R_b for Locations B, C, and Double

Tube Diameter and Dimension	Tube Location	Bore Diameter, in.	Thermal Distance of Bore, h·ft·°F/Btu								
			Fluid Reynolds Number = 2,000			Fluid Reynolds Number = 4,000			Fluid Reynolds Number = 10,000		
			Grout Conductivity, Btu/h·ft·°F			Grout Conductivity, Btu/h·ft·°F			Grout Conductivity, Btu/h·ft·°F		
			0.40	0.80	1.20	0.40	0.80	1.20	0.40	0.80	1.20
3/4 in. DR 11 HDPE U-Tube	B	4	0.47	0.30	0.25	0.40	0.24	0.19	0.39	0.23	0.18
		5	0.51	0.33	0.27	0.45	0.26	0.20	0.44	0.26	0.20
	C	4	0.33	0.24	0.20	0.27	0.17	0.14	0.26	0.17	0.14
		5	0.35	0.27	0.21	0.29	0.18	0.15	0.28	0.18	0.14
	Double	5	0.28	0.17	0.14	0.25	0.14	0.11	0.24	0.14	0.11
1 in. DR 11 HDPE U-Tube	B	4	0.42	0.28	0.24	0.36	0.22	0.17	0.35	0.21	0.17
		5	0.46	0.30	0.25	0.40	0.24	0.19	0.39	0.23	0.18
		6	0.50	0.32	0.26	0.44	0.26	0.20	0.43	0.25	0.19
	C	4	0.32	0.23	0.20	0.25	0.17	0.14	0.25	0.18	0.13
		5	0.33	0.24	0.21	0.27	0.17	0.14	0.26	0.17	0.14
		6	0.35	0.24	0.21	0.28	0.18	0.15	0.28	0.17	0.14
	Double	5	0.26	0.17	0.13	0.23	0.13	0.10	0.23	0.13	0.10
		6	0.27	0.17	0.14	0.24	0.14	0.11	0.24	0.14	0.10
	B	5	0.42	0.28	0.23	0.36	0.22	0.18	0.35	0.21	0.17
		6	0.45	0.29	0.24	0.39	0.23	0.18	0.38	0.23	0.18
1 1/4 in. DR 11 HDPE U-Tube	C	5	0.31	0.22	0.20	0.26	0.17	0.14	0.25	0.16	0.13
		6	0.32	0.23	0.20	0.26	0.17	0.14	0.26	0.16	0.13
	Double	6	0.25	0.16	0.13	0.23	0.13	0.10	0.22	0.13	0.10
	B	5	0.42	0.28	0.23	0.36	0.22	0.18	0.35	0.21	0.17
		6	0.45	0.29	0.24	0.39	0.23	0.18	0.38	0.23	0.18

The most difficult parameters to evaluate in [Equations \(7\)](#) and [\(8\)](#) are the equivalent thermal resistances of the ground. The solutions of Carslaw and Jaeger (1947) require that the time of operation, bore diameter, and thermal diffusivity of the ground be related in the dimensionless Fourier number (Fo):

$$Fo = \frac{4\alpha_g \tau}{d_b^2} \quad (13)$$

where

α_g = thermal diffusivity of ground, ft²/day

τ = time of operation, days

d_b = bore diameter, ft

The method may be modified to allow calculation of equivalent thermal resistances for varying heat pulses. A system can be modeled by three heat pulses: a 10 year (3650 day) pulse of q_a , a 1 month (30 day) pulse of q_m , and a 4 h (0.167 day) pulse of q_{st} . Three times are defined as

$$\tau_1 = 3650 \text{ days}$$

$$\tau_2 = 3650 + 30 = 3680 \text{ days}$$

$$\tau_f = 3650 + 30 + 0.167 = 3680.167 \text{ days}$$

The Fourier number is then computed with the following values:

$$Fo_f = 4\alpha\tau_f/d_b^2$$
$$Fo_1 = 4\alpha(\tau_f - \tau_1)/d_b^2$$
$$Fo_2 = 4\alpha(\tau_f - \tau_2)/d_b^2$$

An intermediate step in computing the ground’s thermal resistance using the methods of Ingersoll and Zobel (1954) is to identify a G-factor, which is determined from [Figure 12](#) for each Fourier value. The algorithm proposed by Cooper (1976) provides an alternative to using [Figure 11](#).

$$R_{ga} = (G_{Fo_f} - G_{Fo_1})/k_g$$

(14)

$$R_{gm} = (G_{Fo_1} - G_{Fo_2})/k_g$$

(15)

$$R_{gm} = G_{Fo_2}/k_g$$

(16)

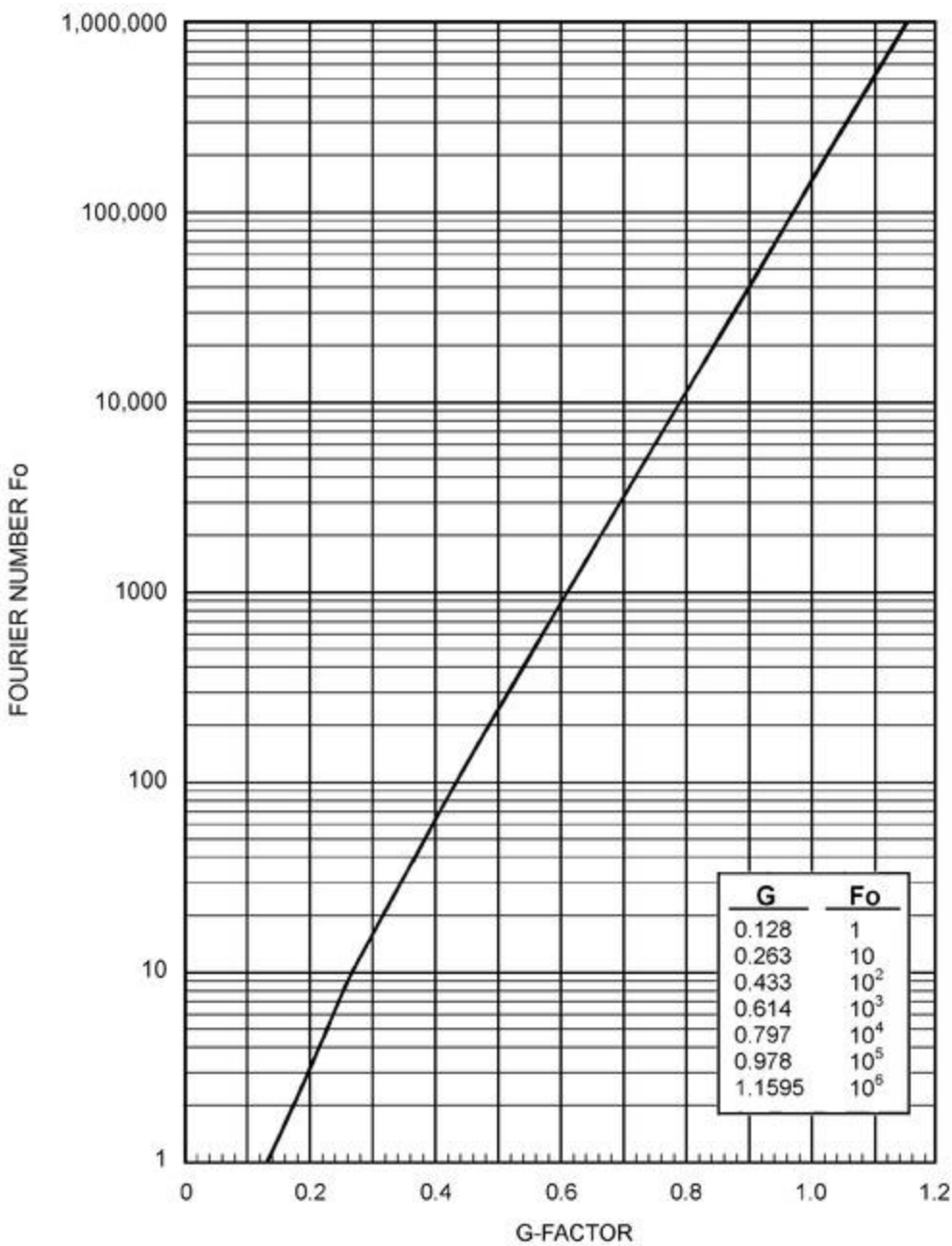


Figure 12. Fourier/G-Factor Graph for Ground Thermal Resistance (Kavanaugh and Rafferty 2014)

Correlations for the values of R_{ga} , R_{gm} , and R_{gst} were presented by Philippe et al. (2010) for a wide range of conditions.

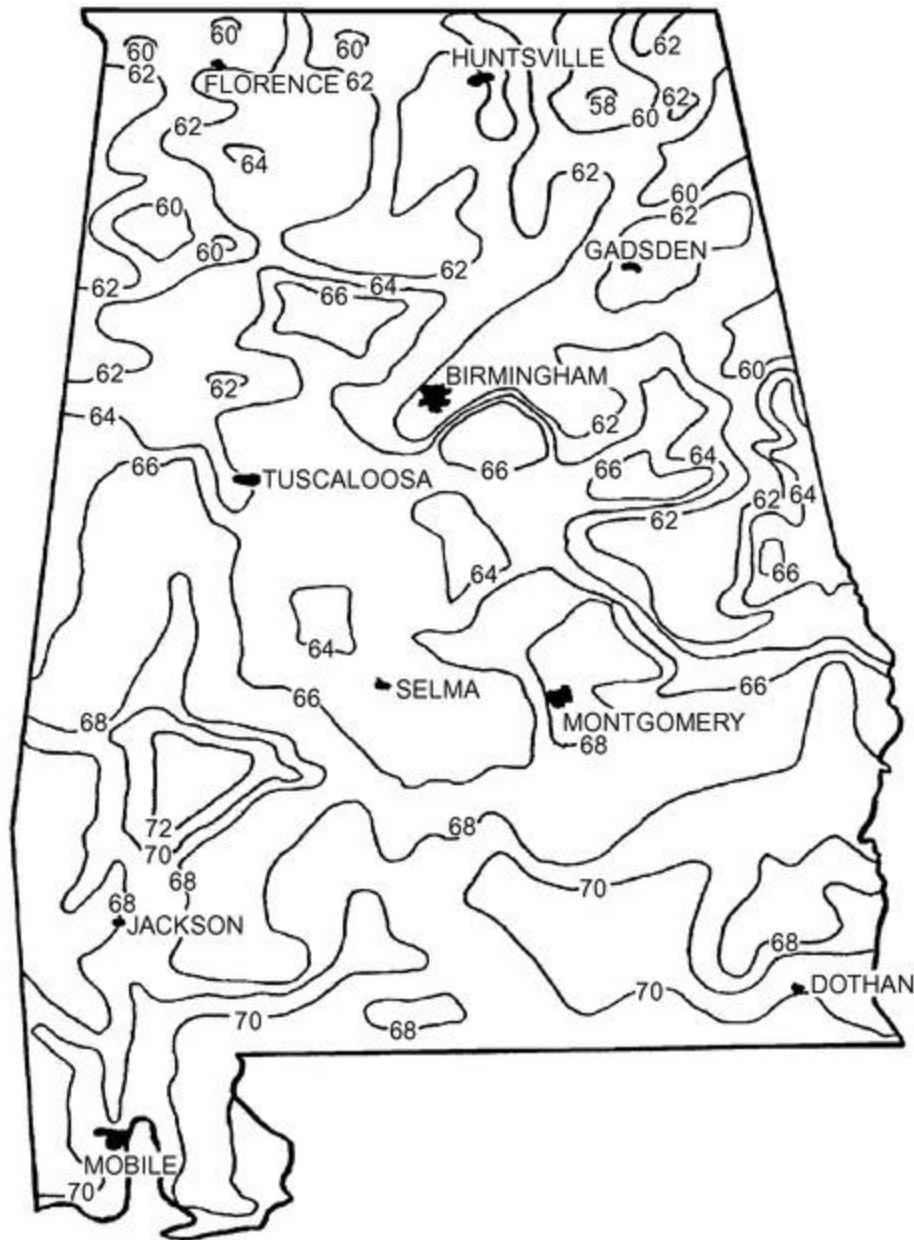


Figure 13. Water and Ground Temperatures in Alabama at 50 to 100 ft Depth (Chandler 1987)

Ranges of the ground thermal conductivity k_g are given in [Table 5](#). State geological surveys are a good source of soil and rock data. However, geotechnical site surveys are highly recommended to determine load soil, rock types, and drilling conditions.

Performance degrades somewhat because of short-circuiting heat losses between the upward- and downward-flowing legs of a conventional U-bend loop. This degradation can be accounted for by introducing the short-circuit heat loss factor F_{sc} in [Equations \(7\)](#) and [\(8\)](#), in the following table. Normally U-tubes are piped in parallel to the supply and return headers. Occasionally, when bore depths are shallow, two or three loops can be piped in series. In these cases, short-circuit heat loss is reduced; thus, the values for F_{sc} are smaller than that for a single bore piped in parallel. Alternatively, F_{sc} can be set to 1.0 and thermal short-circuiting can be included in [Equations \(7\)](#) and [\(8\)](#) by replacing R_b with an effective borehole thermal resistance R_b^* (Claesson and Hellström 2011). As noted by Javed and Spitler (2016), R_b^* values start to be significantly different from R_b for long boreholes and low flow rates ([Table 6](#)).

The remaining terms in [Equations \(7\)](#) and [\(8\)](#) are temperatures. The local deep-ground temperature t_g can best be obtained from local water well logs and geological surveys. A second, less accurate source is a temperature contour map, similar to [Figure 13](#), prepared by state geological surveys. A third source, which can yield ground temperatures within 4°F, is a map with contours, such as [Figure 14](#). Comparing [Figures 13](#) and [14](#) indicates the complex variations that would not be accounted for without detailed contour maps.

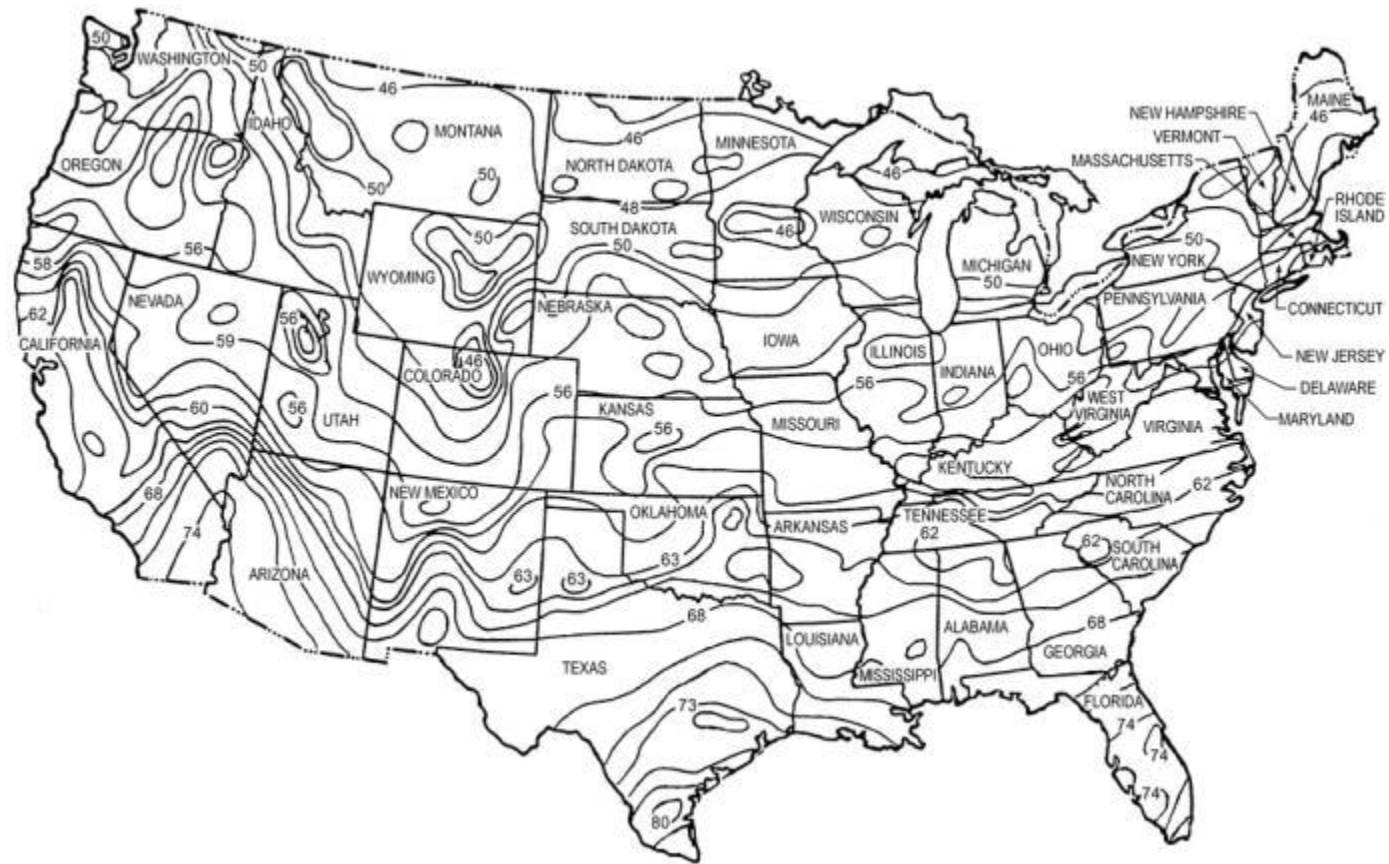


Figure 14. Approximate Groundwater Temperature (°F) in the Continental United States

Selecting the temperature t_{wi} of water entering the unit is critical in the design process. Choosing a value close to ground temperature results in higher system efficiency, but makes the required ground coil length very long and thus unreasonably expensive. Choosing a value far from t_g allows selection of a small, inexpensive ground coil, but the system’s heat pumps will have greatly reduced capacity and increased electric demand. Selecting t_{wi} to be 20 to 30°F higher than t_g in cooling and 10 to 20°F lower than t_g in heating is a good compromise between first cost and efficiency in many regions of the United States. The value for t_{wo} can be selected by adding/subtracting the temperature change through the heat pump with t_{wi} , where the temperature change is based on the flow rate and heat capacity of the water and the heat rejected/absorbed by the heat pump.

A final temperature to consider is the temperature penalty t_p , which is added to the undisturbed earth temperature to represent the build-up or reduction of thermal energy around each borehole over a period of forecast years. If annual cooling and heating ground loads are balanced, the temperature penalty will be zero. The t_p increases the bore length (L_c , L_h) required to achieve desired performance, and t_p increases for closely spaced boreholes; therefore, the designer must select a reasonable separation distance to balance required land area and bore length. The minimum recommended vertical bore separation distance is 20 ft.

Table 6 Short-Circuiting Heat Loss Factor

Bores per Loop	F_{sc}	
	2 gpm/ton	3 gpm/ton
1	1.06	1.04
2	1.03	1.02
3	1.02	1.01

The temperature penalty approximation method presented here is adapted from Kavanaugh and Rafferty (2014). The net annual heat transfer into and out of the ground q_a is a key factor. At the initial design phase, q_a can be computed using estimated **equivalent full-load hours (EFLHs)**, which are equal to the annual load divided by the heat pump capacity; final designs should use a more thorough analysis of site loads. In the EFLH method, the ground thermal load at full heat pump capacity is multiplied by the estimated EFLH values corresponding to the location, building type, and internal loads (Table 5), and these values are summed and divided by 8760 h to determine q_a :

(17)

$$q_a = \{ \text{capacity}[(\text{EER}_c + 3.412)/\text{EER}_c] \times \text{EFLH}_c + \text{capacity} \times [(\text{COP} - 1)/\text{COP}] \times \text{EFLH}_h \} / 8760 \text{ h}$$

where capacity is nominal heat pump capacity, which is positive for heating and negative for cooling, in Btu/h.

To the extent that annual loads are proportional to peak loads, the equivalent full-load hours method provides a simple estimate of annual loads from peak loads. The EFLHs in [Table 7](#) provide a quick means to estimate annual loads needed to size ground heat exchangers at the initial feasibility study phase of a project; the final design should use a more thorough analysis of the site loads. Because EFLHs vary with changes in both annual and peak loads, not all building parameters' effects are included in EFLHs. For instance, building operating hours change annual loads by increasing the amount of time that internal gains are at elevated levels, but they do not change the peak load. Occupancy hours can add load without increasing the installed capacity, thereby changing the EFLHs. Furthermore, changes in other parameters, such as internal gains, do not necessarily scale with system capacity in the same proportion as annual load, again leading to changing EFLHs. Potential users of EFLHs must understand these sources of variability to use them effectively (Carlson 2001).

Adjacent boreholes thermally interfere with each other, effectively restricting the volume of soil available to diffuse heat from/to the bore. Consider an internal bore surrounded by adjacent bores on all four sides ([Figure 15](#)). Assuming equal heat exchange from each bore, an adiabatic symmetry boundary exists at half the separation distance between each bore, $S_{bore}/2$. Net annual energy that would otherwise be diffused beyond the boundary (if the bore were not surrounded by any other bore) is stored in/extracted from and, over time, results in the temperature penalty. The temperature penalty for an internal bore is computed by dividing the stored energy by the heat capacity of the soil within the rectangular prism symmetry boundary:

$$t_{p,int} = \frac{Q_{stored}}{\rho c_p S_{bore}^2 L} \quad (18)$$

where

Q_{stored} = energy stored (or extracted from) within adiabatic symmetry boundary, Btu

$\rho c_p = ka$ = soil volumetric capacity, Btu/ft³·°F

S_{bore} = bore separation distance, ft

L = total bore length (either L_c or L_h), ft

Table 7 Equivalent Full-Load Hours (EFLH) for Typical Occupancy with Constant-Temperature Set Points

Location	EFLH Occupancy							
	School		Office		Retail		Hospital	
	Heating	Cooling	Heating	Cooling	Heating	Cooling	Heating	Cooling
Atlanta, GA	290-200	690-830	690-480	1080-1360	600-380	1380-1860	430-160	2010-2850
Baltimore, MD	460-320	500-610	890-720	690-1080	770-570	880-1480	590-300	1340-2340
Bismarck, ND	500-460	150-250	990-950	250-540	900-810	340-780	730-530	540-1290
Boston, MA	520-450	300-510	1000-960	450-970	870-760	610-1380	680-420	1020-2330
Charleston, WV	440-310	430-570	840-770	620-1140	730-620	820-1600	550-320	1260-2560
Charlotte, NC	320-200	650-730	780-530	1060-1340	670-420	1350-1830	490-180	1990-2820
Chicago, IL	470-390	280-410	920-820	420-780	810-670	550-1090	640-400	870-1780
Dallas, TX	200-120	830-890	520-340	1350-1580	440-280	1660-2090	310-100	2320-3100
Detroit, MI	480-400	230-360	1020-970	390-820	900-790	530-1170	710-460	870-1950
Fairbanks, AK	630-560	26-54	1170-1050	64-200	1090-930	110-320	930-690	210-600
Great Falls, MT	430-360	130-220	890-820	210-490	800-680	290-710	640-420	500-1210
Hilo, HI	1-0	1360-1390	23-13	2440-2580	14-8	2990-3370	0-0	4060-4910
Houston, TX	130-90	940-1000	350-250	1550-1770	300-190	1870-2290	200-70	2540-3320
Indianapolis, IN	480-400	380-560	920-840	560-1000	820-690	730-1410	640-390	1120-2250
Los Angeles, CA	160-80	780-910	580-370	1280-1670	440-250	1740-2350	180-20	2740-3770
Louisville, KY	430-290	550-670	830-710	770-1250	720-570	1000-1720	550-300	1480-2690
Madison, WI	470-390	210-310	900-840	320-640	800-700	420-900	640-440	680-1490
Memphis, TN	240-170	700-830	600-420	1090-1350	510-330	1350-1780	370-140	1910-2680
Miami, FL	12-6	1260-1300	46-34	1980-2150	37-25	2350-2740	12-1	3110-3890

Minneapolis, MN	500-420	200-300	950-860	320-610	860-720	430-870	700-470	680-1420
Montgomery, AL	180-120	840-910	470-330	1260-1510	400-250	1550-1990	260-90	2170-2950
Nashville, TN	320-250	570-740	680-590	830-1280	590-470	1030-1710	450-240	1490-2620
New Orleans, LA	110-67	920-990	320-230	1500-1720	260-160	1820-2240	160-46	2500-3280
New York, NY	440-350	360-550	870-790	540-1040	760-630	720-1480	590-330	1160-2440
Omaha, NE	400-330	310-440	800-720	480-820	720-600	610-1130	570-360	920-1780
Phoenix, AZ	110-65	950-1020	290-210	1340-1610	250-170	1630-2090	140-34	2220-3040
Pittsburgh, PA	500-470	300-530	950-910	440-920	840-750	600-1310	650-420	960-2160
Portland, ME	480-400	190-300	980-880	310-630	870-710	410-900	690-420	700-1520
Richmond, VA	410-270	630-730	820-660	880-1310	710-520	1110-1770	530-250	1650-2760
Sacramento, CA	360-220	680-850	990-640	1080-1430	830-480	1460-2020	540-120	2250-3180
Salt Lake City, UT	540-520	410-710	1060-1040	510-1090	930-830	660-1520	720-440	1060-2470
Seattle, WA	650-460	260-460	1370-1270	440-1200	1170-960	710-1860	850-360	1340-3270
St. Louis, MO	400-280	460-550	800-710	680-1100	700-570	850-1500	550-320	1260-2330
Tampa, FL	58-35	1050-1110	190-140	1800-2000	160-100	2170-2580	90-22	2910-3710
Tulsa, OK	300-240	580-770	620-560	830-1300	540-450	1030-1730	410-220	1470-2630

Source: Carlson (2001).

Notes:

- 1. The ranges in values are from internal gains at 0.6 and 2.5 W/ft².
- 2. Operating with large temperature setbacks during unoccupied periods (effectively turning off the system) reduces heating EFLHs by 20% and cooling EFLHs by 5%.

Equations relating EFLH to Heating and Cooling Degree Days allowing calculation of EFLH for locations other than those listed here can be found in Carlson (2001).

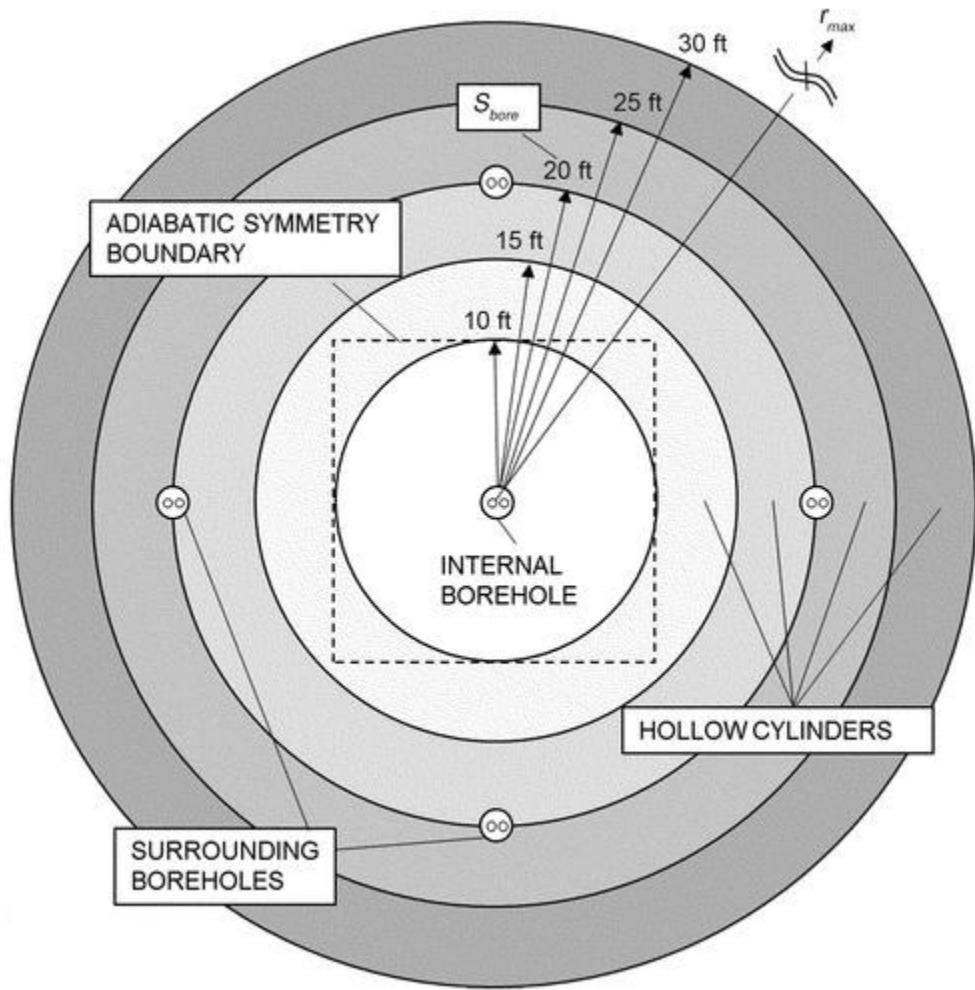


Figure 15. Representative Soil Cylinders and Adiabatic Symmetry Boundary for Heat Storage

An initial guess value is required for L and can be found using [Equation \(7\)](#) or [\(8\)](#) with a reasonable value for t_p (e.g. -10 to 10°F). Values for L and t_p are iterated to find the final solution. The energy that would have otherwise been stored in imaginary hollow cylinders of soil beyond the symmetry boundary is

$$Q_{\text{stored}} = \sum_{r=S_{\text{bore}}/2}^{T_{\text{max}}} \rho c_p \pi L (r_o^2 - r_i^2) \times \Delta t_r \quad (19)$$

where

- r_o = outer radius of hollow soil cylinder, ft
- r_i = inner radius of hollow soil cylinder, ft
- r_{max} = maximum considered radius
- Δt_r = soil temperature change at average radius $r = (r_o + r_i)/2$, $^\circ\text{F}$

Note that, because there is some overlap of the innermost hollow cylinder with the symmetry boundary, the impact of the overlap is neglected. The value for r_{max} is increased until the temperature rise in the outermost cylinder is negligible ($<0.5^\circ\text{F}$); beyond this distance, the storage effect is offset with evaporative cooling and moisture recharge mechanisms. Porous soil with high moisture content may require $r_{\text{max}} = S_{\text{bore}}$, whereas low-porosity soil may require as much as $r_{\text{max}} = 5 \times S_{\text{bore}}$ (Kavanaugh and Rafferty 2014). For a nominal configuration ([Figure 15](#)) with S_{bore} of 20 ft, four hollow cylinders with a 5 ft width, $r_o - r_i$, gives sufficiently resolved results; other separation distances may require different numbers and width of cylinders. The average soil temperature change for each cylinder is computed at the average radius, $r = (r_o + r_i)/2$, using the line source method (Carslaw and Jaeger 1959; Ingersoll and Zobel 1954; Kavanaugh and Rafferty 2014):

$$\Delta t_r = \frac{q_a I(X)}{2\pi k_g L} \quad (20)$$

where the $I(X)$ function is formed from the exponential integral; an approximation with less than 1% error for $X < 0.7$ is

$$I(X) = -\frac{1}{2} Ei(-X^2) \approx \ln\left(\frac{1}{X}\right) + \frac{X^2}{2} - \frac{X^4}{8} - \frac{\gamma}{2} \quad (21)$$

where γ is Euler's constant (0.57722...), Ei is the exponential integral, and the X term is

$$X = \frac{1}{2\sqrt{Fo}} = \frac{r}{2\sqrt{\alpha_v \tau}} \quad (22)$$

where

- r = $(r_o + r_i)/2$, average radius of soil cylinder, ft
- τ = time duration, days

The value for τ is the designer's choice and can be based on expected groundwater movement; τ_2 (from the preceding Fourier number calculations, value is usually about 10 years) can be used for minimal groundwater movement and vertical percolation of water through the borefield, whereas 365 days can be used for more substantial water movement. Finally, t_p is calculated by prorating $t_{p,int}$ based on the number of bores with a particular adjacency: interior, side, corner, midrow, and end ([Figure 16](#)), as well as accounting for heat diffusion at the bottom of the borefield:

$$t_p = \frac{N_{\text{int}} + 0.75N_{\text{side}} + 0.5N_{\text{corner}} + 0.5N_{\text{midrow}} + 0.25N_{\text{end}}}{(N_{\text{int}} + N_{\text{side}} + N_{\text{corner}} + N_{\text{midrow}} + N_{\text{end}}) + C_{f\text{Horiz}}} \times t_{p,int} \quad (23)$$

where

- N_{int} = number of interior boreholes, surrounded by four other bores
- N_{side} = number of side boreholes, surrounded by three other bores
- N_{corner} = number of corner boreholes, surrounded by two other bores
- N_{midrow} = number of boreholes in middle of row, surrounded by two other bores (only for borefield with a single row)
- N_{end} = number of end boreholes, surrounded by one other bore (only for borefield with a single row)

C_{fHoriz} = bottom diffusion factor

The bottom diffusion factor is the ratio of surface area of the sides and bottom of the borefield to the surface area of the sides:

$$C_{fHoriz} = \frac{[2L_{bore}(W_{field} + L_{field})] + W_{field}L_{field}}{2L_{bore}(W_{field} + L_{field})} \quad (24)$$

where the individual bore length L_{bore} is found by dividing L by the number of bores. The borefield length and width are

$$L_{field} = S_{bore}(N_{long} - 1) \quad (25)$$

$$W_{field} = S_{bore}(N_{wide} - 1) \quad (26)$$

where the borefield has N_{long} bores in the length direction and N_{wide} bores in the width direction (Figure 12).

Temperature Penalty Uncertainty. Calculating the temperature penalty is one of the more uncertain parts of heat exchanger length selection. Temperature penalties computed using the concentric cylinder source method presented here (Kavanaugh and Rafferty 2014) differ significantly from those obtained using the g -function approach (see the section on Alternative Sizing Method, after Example 1), as discussed by Bernier et al. (2008). The g -function method accounts for the ground heat conduction with more rigor; it includes both radial and axial heat transfer, rather than only radial heat transfer, and computes the temperature penalty from interfering boreholes at the borehole wall, rather than using the average soil temperature change. Both approaches only consider conductive heat transfer and therefore represent worst-case scenarios, where the actual temperature change is usually mitigated by groundwater recharge (vertical flow), groundwater movement (horizontal flow), and evaporation (and condensation) of water in the soil. Further research is needed to understand which ground heat exchanger sizing method best captures the temperature penalty related to long-term operation in applied systems. Note that, despite the uncertainty in long-term temperature penalty, the concentric cylinder source method has been used in many successful installations of GSHP systems.

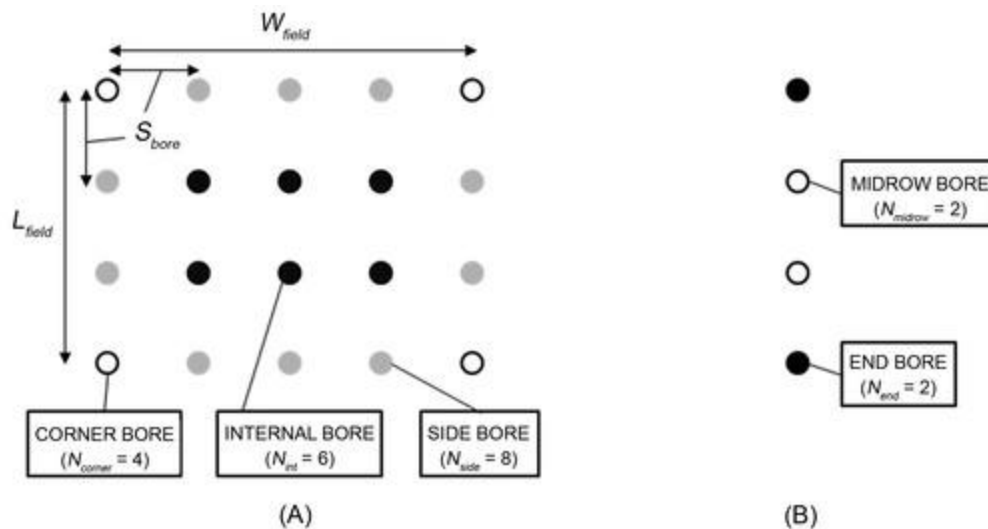


Figure 16. Borefield with (A) 20 Boreholes, $N_{wide} = 5$, $N_{long} = 4$, and (B) 4 Boreholes, $N_{wide} = 1$, $N_{long} = 4$ (i.e., Single Row)

Groundwater movement strongly affects long-term temperature change in a densely packed bore field (Chiasson et al. 2000a). A related factor is the evaporative cooling effect experienced with heat addition to the ground. Although thermal conductivity is somewhat reduced with lower moisture content (see Table 3), the net effect is beneficial in porous soils when water movement recharges the ground to original moisture levels. A similar effect may be experienced in cold climates when soil moisture freezes and the heat of solidification mitigates excessive temperature decline. Because these effects have not been thoroughly studied, the design engineer must establish a range of design lengths between one based on minimal groundwater movement, as in very tight clay soils with poor percolation rates, and a second based on the higher rates characteristic of porous aquifers.

Kavanaugh and Kavanaugh (2012a, 2012b) examined ground heat exchanger performance in 40 commercial buildings with vertical ground heat exchangers and between 5 and 25 years of operation. They calculated maximum approach temperature (difference between average loop temperatures t_{wi} and t_{wo} and initial ground temperature t_g) for all of the buildings; higher approach temperatures as years of operation increased would indicate an increase in ground temperature and raise concern about the expected life of ground heat exchangers with imbalanced cooling loads compared to heating loads.

In fact, the data suggested that older GSHP systems had lower approach temperatures. Results were not adjusted for many important factors such as vertical bore length, ground thermal properties, and vertical bore separation distance. Newer systems tended to have slightly shorter ground heat exchangers, but this was offset somewhat by the older systems' tendency to have smaller vertical bore separation distances and lower-conductivity grout and fill. Of the loops with the largest approach, three of the newer systems had vertical bore lengths less than 120 ft/ton. Two systems with long loops but large approach temperatures had low thermal conductivity grout (0.38 Btu/h·ft·°F) 15 ft bore separation, and indoor air temperatures below 70°F.

The study's data set is small, and significant long-term temperature change cannot be excluded at this point. Although much more field study is desirable, the absence of any significant trend of increased ground temperature (noted by elevation of maximum approach temperature) with increased years of GSHP operation suggests that long-term ground temperature change is not prevalent in properly designed GSHPs.

Results from this project cannot be applied to long-term temperature decline in which the amount of heat removed from the ground in heating far exceeds the heat rejected in cooling. In cold climates the heat capacity available at the freeze point of water is significant, but the impact on grout thermal and physical properties also needs further field study.

Example 1. Size a vertical ground-coupled heat pump system for a six-zone classroom addition in Atlanta, GA. The addition has a peak cooling (block) load q_{lc} of 247,000 Btu/h and a peak heating (block) load q_{lh} of 160,000 Btu/h. The design monthly part-load factor PLF_m is 0.28.

Ground temperature $t_g = 65^\circ\text{F}$

Ground conductivity k_g and diffusivity $a_g = 1.4 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$, $1.0 \text{ ft}^2/\text{day}$

Bore fill conductivity $k_{grt} = 1.0 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$

Vertical U-tube = 1.0 in. nominal, DR11, HDPE, 5 in. borehole diameter

2×10 grid (20 vertical bores) with $S_{bore} = 20 \text{ ft}$ separation

Bores per loop = 1, flow is 3 gpm/ton, so $F_{sc} = 1.04$

Reynolds number = 4000 (transition flow)

Heat pump inlet and outlet temperatures t_{wi} and $t_{wo} = 85$ and 95°F

Heat pump capacity = 247,000 Btu/h (maximum of peak block cooling and heating loads)

Heat pump cooling and heating efficiency (EER, COP) = 14.9 Btu/Wh, 4.4

10 year (3650 day), 1 month (30 day), and 4 h (0.167 day) heat pulse analysis:

$EFLH_c = 760 \text{ h}$, $EFLH_h = 245 \text{ h}$ ([Table 7](#))

$$q_{cond} = (q_{lc} + 3.41W_c) = q_{lc} \times \frac{EER + 3.412}{EER}$$

$$= -247,000 \text{ Btu/h} \times \frac{14.9 + 3.412}{14.9}$$

$$= -303,600 \text{ Btu/h}$$

$$q_{evap} = (q_{lh} - 3.41W_h) = q_{lh} \times \frac{COP - 1}{COP} = 160,000 \text{ Btu/h} \times \frac{4.4 - 1}{4.4}$$

$$= 123,600 \text{ Btu/h}$$

$$q_a = \frac{\text{capacity} \left(\frac{EER + 3.412}{EER} \right) \times EFLH_c + \text{capacity} \left(\frac{COP - 1}{COP} \right) \times EFLH_h}{8760 \text{ h}}$$

$$= \frac{-247,000 \left(\frac{14.9 + 3.412}{14.9} \right) \times 760 + 247,000 \left(\frac{4.4 - 1}{4.4} \right) \times 245}{8760}$$

$$= -21,000 \text{ Btu/h}$$

$$Fo_f = (4 \times 1.0 \text{ ft}^2/\text{day} \times 3680.167 \text{ days}) / (5 \text{ in.}/12 \text{ in./ft})^2 = 84,800;$$

from Fig. 8, $G_{Fof} = 0.96$

$$Fo_1 = [4 \times 1.0 \text{ ft}^2/\text{day} \times (3680.167 - 3650)] / (5 \text{ in.}/12 \text{ in./ft})^2 = 695;$$

from Fig. 8, $G_{Fol} = 0.58$

$$Fo_2 = [4 \times 1.0 \text{ ft}^2/\text{day} \times (3680.167 - 3680)] / (5 \text{ in.}/12 \text{ in./ft})^2 = 3.85;$$

from Fig. 8, $G_{Fo2} = 0.21$

$$R_{ga} = (0.96 - 0.58) / 1.4 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} = 0.271 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{gm} = (0.58 - 0.21) / 1.4 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} = 0.264 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{gst} = 0.21 / 1.4 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F} = 0.15 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_b = 0.185 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu} \text{ (average of Locations B and C, } k_{grt} = 1.0 \text{ Btu/h} \cdot \text{ft} \cdot ^\circ\text{F, transition flow for Re} = 4000).$$

The required total bore length, assuming no long-term ground temperature change ($t_p = 0$) caused by moisture evaporation and subsequent groundwater recharge through porous soil is

$$L_c = \frac{(-21,000 \times 0.271) - 303,600(0.185 + 0.28 \times 0.264 + 1.04 \times 0.15)}{65 - \frac{85 + 95}{2} - (-0)}$$

$$= 5267 \text{ ft} = 5267 \text{ ft}/20 \text{ bores} = 263 \text{ ft/bore}$$

For nonporous soils, a temperature penalty is computed. The annual heat imbalance q_a is applied for the 10 years plus one month time duration (3680 days). For the internal bore temperature penalty, the average temperature change for four (the temperature change of a fifth cylinder is less than 0.5°F , so it is not considered) 5 ft wide hollow cylinders extending beyond the symmetry boundary are computed, beginning with the inner hollow cylinder ($r_i = S_{bore}/2 = 20/2 \text{ ft} = 10 \text{ ft}$, $r_o = r_i + 5 \text{ ft} = 15 \text{ ft}$, and $r = [r_o + r_i]/2 = [15 + 10]/2 = 12.5 \text{ ft}$):

For $r = 12.5 \text{ ft}$,

$$X = \frac{r}{2\sqrt{\alpha_g \tau}} = \frac{12.5}{2\sqrt{10 \times 3680}} = 0.103$$

$$I(X) = \ln\left(\frac{1}{X}\right) + \frac{X^2}{2} - \frac{X^4}{8} - \frac{\gamma}{2}$$

$$= \ln\left(\frac{1}{0.103}\right) + \frac{0.103^2}{2} - \frac{0.103^4}{8} - \frac{0.5772}{2} = 1.989$$

Repeating for $r = 17.5$ ft: $X = 0.144$, $I(X) = 1.658$.

Repeating for $r = 22.5$ ft: $X = 0.186$, $I(X) = 1.413$.

Repeating for $r = 27.5$ ft: $X = 0.227$, $I(X) = 1.221$.

For the first iteration, a small value of -1°F is used as an initial guess, because the building is slightly cooling dominated (recall that negative values are used for cooling loads):

$$L_c = \frac{(-21,000 \times 0.271) - 303,600(0.185 + 0.28 \times 0.264 + 1.04 \times 0.15)}{65 - \frac{85 + 98}{2} - (-1)}$$

$$= 5487 \text{ ft, or } L_{\text{bore}} = 5487 \text{ ft}/20 \text{ bores} = 274 \text{ ft/bore}$$

So the average temperature changes of the hollow cylinders are, for the hollow cylinder with $r = 12.5$ ft,

$$\Delta t_r = \frac{q_a I(X)}{2\pi k_g L_c} = \frac{-21,000 \times 1.989}{2\pi \times 1 \times 5487} = 0.87^\circ\text{F}$$

Repeating for $r = 17.5$ ft: $\Delta t_r = -0.72^\circ\text{F}$.

Repeating for $r = 22.5$ ft: $\Delta t_r = -0.62^\circ\text{F}$.

Repeating for $r = 27.5$ ft: $\Delta t_r = -0.53^\circ\text{F}$.

The volumetric heat capacity is $\rho c_p = k_g/\alpha_g = 1.4 \text{ Btu}/\text{h}\cdot\text{ft}\cdot^\circ\text{F}/(1.0 \text{ ft}^2/\text{day}) \times 24 \text{ h/day} = 33.6 \text{ Btu}/\text{ft}^3\cdot^\circ\text{F}$, and the energy stored in the hollow cylinders is

$r_o \ r_i$	$\rho c_p L_c \pi (r_o^2 - r_i^2) \Delta t_r$	Q_{stored}
10 ft, 15 ft	$33.6 \times 5487 \pi (15^2 - 10^2)(-0.87)$	$-62.7 \times 10^6 \text{ Btu}$
15 ft, 20 ft	$33.6 \times 5487 \pi (20^2 - 15^2)(-0.72)$	$-73.1 \times 10^6 \text{ Btu}$
20 ft, 25 ft	$33.6 \times 5487 \pi (25^2 - 20^2)(-0.62)$	$-80.1 \times 10^6 \text{ Btu}$
25 ft, 30 ft	$33.6 \times 5487 \pi (30^2 - 25^2)(-0.53)$	$-84.6 \times 10^6 \text{ Btu}$
		$-301 \times 10^6 \text{ Btu}$

$$t_{p,int} = \frac{Q_{stored}}{\rho c_p S_{bore}^2 L_c} = \frac{-301 \times 10^6}{33.6 \times 20^2 \times 5487} = -4.1^\circ\text{F}$$

$$L_{field} = S_{bore}(N_{long} - 1) = 20 \text{ ft} (10 - 1) = 180 \text{ ft}$$

$$W_{field} = S_{bore}(N_{wide} - 1) = 20 \text{ ft} (2 - 1) = 20 \text{ ft}$$

$$C_{fHoriz} = \frac{[2L_{bore}(W_{field} + L_{field})] + W_{field} \times L_{field}}{2L_{bore}(W_{field} + L_{field})}$$

$$= \frac{[2 \times 275(20 + 180)] + 20 \times 180}{2 \times 275(20 + 180)} = 1.033$$

$$t_p = \frac{N_{int} + 0.75N_{side} + 0.5N_{corner}}{(N_{int} + N_{side} + N_{corner}) \times C_{fHoriz}} \times t_{p,int}$$

$$= \frac{0 + 0.75 \times 16 + 0.5 \times 4}{(0 + 16 + 4)1.033} \times -4.1 = -2.76^\circ\text{F}$$

For the second iteration, this t_p is substituted back into the equation for L_c , which yields 5921 ft, and a new corresponding t_p is calculated, -2.57°F . The solution is fully converged after four iterations with

$$L_c = 5875 \text{ ft} \quad L_{bore} = 294 \text{ ft} \quad t_p = -2.59^\circ\text{F}$$

Repeat the process using [Equation \(8\)](#) to find the bore length for heating L_h . The design bore length is the larger value of L_c and L_h .

Alternative Sizing Method. The traditional concentric cylinder source design method can be solved using a relatively simple procedure, because the effective ground thermal resistances R_{ga} , R_{gm} , and R_{gst} are calculated independent of borehole length. In contrast, the ***g*-function** method, discussed here and used in some software design tools, is more complex and requires a more involved iteration process to evaluate the design length. The benefit of using the *g*-function method is that it accounts for both radial and axial conduction, and effectively applies the long-term temperature penalty to the borehole wall (rather than using the average soil temperature change).

Thermal response factors, also known as ***g*-functions** (not to be confused with the *G*-factor) may be used as an alternative to calculate the ground thermal resistance required in [Equations \(7\)](#) and [\(8\)](#). The concept was first introduced by Eskilson (1987) and extended to short-time steps by Yavuzturk and Spitler (1999). The *g*-functions give a relation between the heat extracted (or rejected) from the ground per unit borehole length q_L and the borehole wall temperature T_b . The borehole wall temperature is given by

$$T_b = T_g - \frac{q_L}{2\pi k_g} g(t/t_s, r_b/H, B/H) \quad (27)$$

where *g* represents the *g*-function. As shown in [Equation \(26\)](#), the *g*-function depends on three non-dimensional parameters: B/H , the ratio of the borehole spacing over the borehole length; r_b/H , the ratio of the borehole radius over the borehole length; and t/t_s , a nondimensional time where t_s is a characteristic time ($= H^2/9\alpha_g$). Typical *g*-functions curves are presented in [Figure 17](#) for a 3×2 bore field.

The *g*-function curves are presented graphically in [Figure 17](#) as a function of $\ln(t/t_s)$ for six bore field spacings (B/H) and for a value of $r_b/H = 0.0005$. The curve for $B/H = \infty$ corresponds to the *g*-function of a single borehole. One of the major advantages of these nondimensional curves is that they apply to any 3×2 bore field.

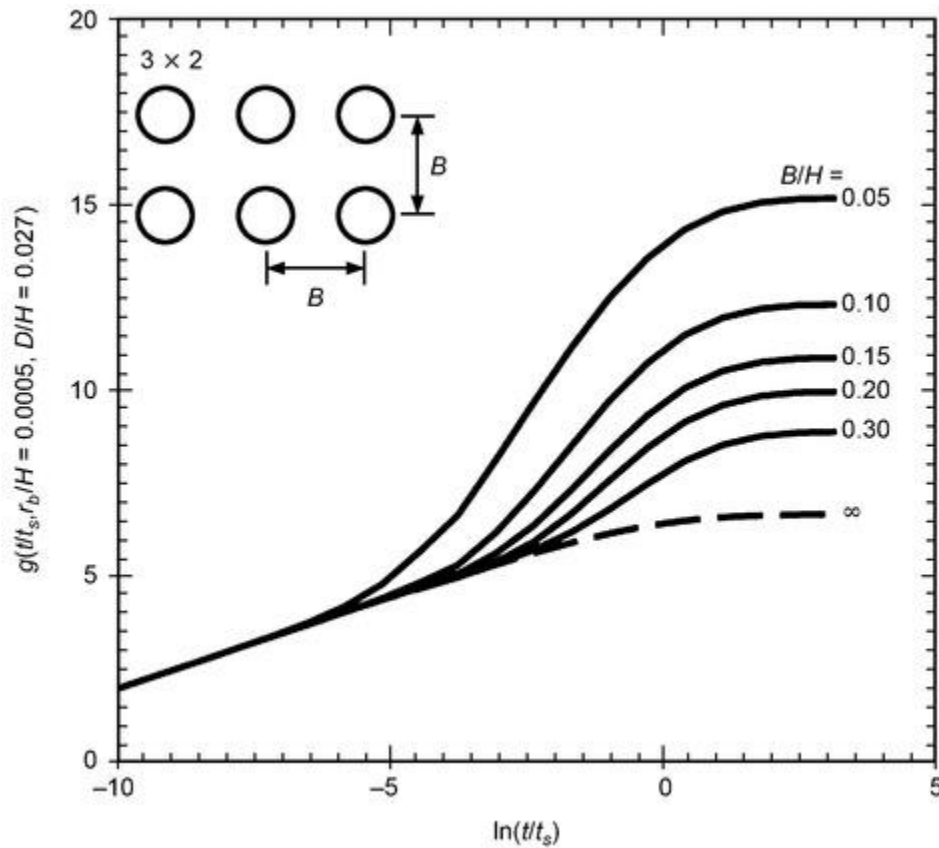


Figure 17. Typical g -Function Curves for 3×2 Bore Field

Eskilson (1987) provides g -function curves for a number of bore field geometries. Design software tools that use the g -function concept have a relatively large data set of g -function curves to choose from. Eskilson (1987) calculated g -functions using two-dimensional transient finite-difference equations on a radial-axial coordinate system for a single borehole in homogeneous ground. The temperature fields from a single borehole were superimposed in space to obtain the response from a borehole field with a certain configuration.

Example 2. Boreholes in a 3×2 bore field have the following characteristics: $r_b = 2$ in., $H = 328$ ft, and $B = 16.4$ ft. The undisturbed ground temperature is 59°F , the thermal conductivity is 0.87 Btu/h \cdot ft \cdot $^\circ\text{F}$, and the thermal diffusivity is 1.04 ft²/day. What is the resulting borehole wall temperature after 10 years of heat extraction at an average rate of 5.2 Btu/h per foot of bore?

Evaluation of the three non-dimensional parameters lead to: $r_b/H = 0.0005$, $B/H = 0.05$, $t_s = 31.5$ years and $\ln(t/t_s) = -1.15$. According to [Figure 13](#), the resulting g -function is 12.3 . Using [Equation \(26\)](#), the borehole wall temperature is then equal to 47.3°F .

The g -functions can be used to determine the design length of a bore field. One possible approach is to use [Equations \(7\)](#) and [\(8\)](#) but with two modifications. First, when g -functions are used, thermal interference among boreholes is implicitly accounted for and t_p can be eliminated. Second, the values of R_{ga} , R_{gm} , and R_{gst} are now based on g -functions. Hence, [Equations \(27\)](#) to [\(29\)](#) take the following forms:

$$R_{ga} = \frac{g(t_f) - g(t_f - t_1)}{2\pi k_g} \quad (28)$$

$$R_{gm} = \frac{g(t_f - t_1) - g(t_f - t_2)}{2\pi k_g} \quad (29)$$

$$R_{gst} = \frac{g(t_f - t_2)}{2\pi k_g} \quad (30)$$

where $g(t_s - t_y)$ is the g -function evaluated at $\ln[(t_s - t_y)/t_s]$ for a given bore field and B/H ratio. Note that determining L (i.e., $n_b \times H$, where n_b is the number of boreholes) is an iterative process because R_{ga} , R_{gm} , and R_{gst} depend on H , which is unknown beforehand. Thus, software tools are often required to accomplish this task. Because g -functions account for 3D heat transfer in the borefield, they are considered to be more accurate than the G -factors,

which derive from a radial-only heat transfer model. Borehole thermal capacity can be accounted for by using the short-time-step g -functions (Yavuzturk and Spitler 1999).

Example 3. A building has a cooling block load of 15 tons with a corresponding value of $q_{cond} = -225,200$ Btu/h. The annual ground imbalance $q_a = -10,236$ Btu/h, $PLF_m = 0.30$, and $F_{SC} = 1.0$. The 3×2 bore field has the following characteristics: $r_b = 2$ in., $B = 16.4$ ft, and $R_b = 0.173$ h · ft · °F/Btu. The undisturbed ground temperature is 50°F, the thermal conductivity is 1.93 Btu/h · ft · °F, and the thermal diffusivity is 1.04 ft²/day $\times 10^{-6}$. Calculate the equivalent thermal resistances R_{ga} , R_{gm} , and R_{gst} for three consecutive heat pulses of 10 years, 1 month, and 6 hours and the total required length if the maximum mean fluid temperature in the borehole is to be kept below 95°F.

From the problem statement, $t_f = 3680.25$ days, $t_2 = 3680$ days, and $t_1 = 3650$ days. After iterations, this leads to $g(t_f) = 12.34$, $g(t_f - t_1) = 3.99$, and $g(t_f - t_2) = 1.55$; and $R_{ga} = 0.689$ h · ft · °F/Btu, $R_{gm} = 0.201$ h · ft · °F/Btu, and $R_{gst} = 0.128$ h · ft · °F/Btu; and

$$L_c = \frac{-10,236 \times 0.689 - 225,200(0.173 + 0.3 \times 0.201 + 0.128)}{50 - 95} = 1965 \text{ ft}$$

Thus, 321 ft per bore is required with a borehole spacing of 16.4 ft. This represents a length of 131 ft per ton.

Simulation of Ground Heat Exchangers

After the design length has been determined, it is often necessary to evaluate the outlet fluid temperature of a bore field as a function of time, generally on an hourly basis, and estimate the annual heat pump energy consumption. Energy simulation can be used to compute this temperature (they can also be used iteratively to assist in sizing the ground heat exchanger). Some energy simulation programs use the duct ground storage (DST) model introduced by Hellström (1989) to evaluate the outlet fluid temperature of a bore field as a function of time. Yavuzturk and Spitler (1999) describe the calculation method behind the DST model.

The DST model calculates the transient thermal process in densely packed borehole fields. The boreholes are assumed to be evenly distributed within a cylindrical storage region in the ground. Although the DST model was originally intended to simulate borehole thermal energy storage (BTES) systems, it has been used to simulate ground source heat pump systems.

Other energy simulation programs have a g -function-based routine to evaluate the outlet fluid temperature of a bore field as a function of time (Fisher et al. 2006; Liu 2008). The following analysis is intended to give only the salient features of an hourly simulation based on g -functions. As an example, assuming that $F_{SC} = 1$ and that the borehole length and the inlet fluid temperature are known, and that the heat transfer rates for three consecutive time intervals (0 to t_1 , t_1 to t_2 , and t_2 to t_3) are given by Q_1 , Q_2 , and Q_3 , then, using temporal superposition, the mean fluid temperature at the end of the third time interval is given by

$$T_m = T_g - \left[\frac{Q_1[g(t_3-0) - g(t_3-t_1)] + Q_2[g(t_3-t_1) - g(t_3-t_2)] + Q_3g(t_3-t_2)}{2\pi k_g L} + \frac{Q_3 R_b}{L} \right] \quad (31)$$

with $T_m = (T_{wi} + T_{wo})/2$.

Based on the work of Yavuzturk and Spitler (1999), Equation (30) can be generalized for n time steps as follows:

$$T_m = T_g - \sum_{i=1}^n \frac{(Q_i - Q_{i-1})}{2\pi k_g L} g\left(\frac{t_n - t_{i-1}}{t_s}, \frac{r_b}{H}, \frac{B}{H}\right) - \frac{Q_n R_b}{L} \quad (32)$$

Solving Equation (31) can be computationally intensive if the number of time steps is large, because there is no recurrence in the summation term. In other words, the calculations performed at time step $n - 1$ cannot be used at time step n , and the ground loop loading history must be updated at each time step. Load aggregation is typically used to reduce the number of terms in the summation without sacrificing accuracy. It is based on the fact that recent ground loads have a more significant effect on the current mean fluid temperature than distant ground loads. For example, in the case of hourly simulations, the determination of T_m at the end of a year would require a summation of 8760 hourly terms according to Equation (31). One possible alternative is to aggregate (i.e., average) the ground loads of the first 8000 hours, then aggregate the next 730 hours and keep intact the last 30 hours. The summation term would then be reduced to 32 terms. Other aggregation schemes have been proposed by Bernier et al. (2004), Liu (2005), and Yavuzturk and Spitler (1999).

Hybrid System Design

The design methods described previously size the ground loop for the larger of the heating or cooling loads, including a temperature penalty for the amount of imbalance (which can be large in severe climates). An alternative approach for imbalanced buildings is to partially balance the load on the ground, both at peak and annual scale, by adding a supplemental device to help meet the larger of the two peak loads. This is a hybrid ground-source (or **hybrid ground-coupled**) system. Hybrids can provide several benefits for buildings with a load imbalance. The biggest economic effect is in decreasing the ground heat exchanger size/cost. First-cost savings have been reported of 6 to 16% of total HVAC system cost, with little consequence reported on operating cost (Hackel and Pertzborn 2011; Singh and Foster 1998) because the HVAC systems operates for the vast majority of the year at a fraction of peak design. More balanced loads resulting from hybrids can reduce the long-term ground temperature penalty associated with multiyear operation.

In most U.S. commercial buildings, the cooling load is dominant both annually and at the peak because of high internal loads, ventilation heat recovery, and good building envelopes. Heat from compressors, pumps, and fans also plays a factor; in heating mode, this heat is delivered to the building, so less heat is required from the ground. As a result, achieving annual thermal balance requires heat pumps in a ground-source system to operate in heating mode 1.6 to 1.8 h for every hour in cooling.

The ideal configuration of the ground heat exchanger and supplemental cooling device in a hybrid depends on many factors, such as climate, building peak load, and building annual loads. Carefully analyze which approach may work best for a specific building. One common configuration for cooling-dominated systems, a series hybrid, is shown in [Figure 18A](#). This approach could also be taken with a closed-circuit cooling tower (i.e., fluid cooler) downstream of the ground heat exchanger (GHX). In general, it is most effective to place the lower-temperature heat sink downstream; an energy model can help determine which order most often results in this scenario throughout the year. As a rule of thumb, in drier climates with warmer ground (e.g., desert southwestern United States) the tower is almost always the lower-temperature sink, whereas in humid climates with moderate-temperature ground (e.g. southeastern United States), the ground is often the lower-temperature sink. The hybrid can also be configured in parallel, as shown in [Figure 18B](#), which is especially desirable if the ground heat exchanger is small in comparison to the building peak cooling load (a series system in this example would require a more complex partial GHX bypass). In either case, there are two guidelines for design and operation of hybrid systems:

- A valve can be used to bypass the ground heat exchanger when the system is balanced; a dead band of 55 to 75°F can be used for this purpose. This valve can be three-way as shown, or two-way where appropriate.
- Cooling towers are optimal when they are oversized, use a variable-speed fan, and minimize fan speed across cells.

Control of a cooling-dominated hybrid depends on the configuration. If equipment is placed as shown in [Figure 18A](#), the temperature downstream of the tower can be used to control the use and speed of the tower based on a high limit (some additional savings are possible if the tower is controlled by the Δt between entering fluid and ambient wet-bulb temperatures, though this method depends on a difficult measurement of wet bulb). If the tower is located upstream of the ground heat exchanger, the temperature exiting the tower and the ground heat exchanger should both be used in tower control (to ensure the ground is not being cooled). For parallel configurations ([Figure 18B](#)), one practical tower control sequence bases tower operation on a calculation of the average of fluid temperature entering the heat pumps over the previous week (Xu 2007). Xu also suggests a strategy for controlling the parallel three-way valve.

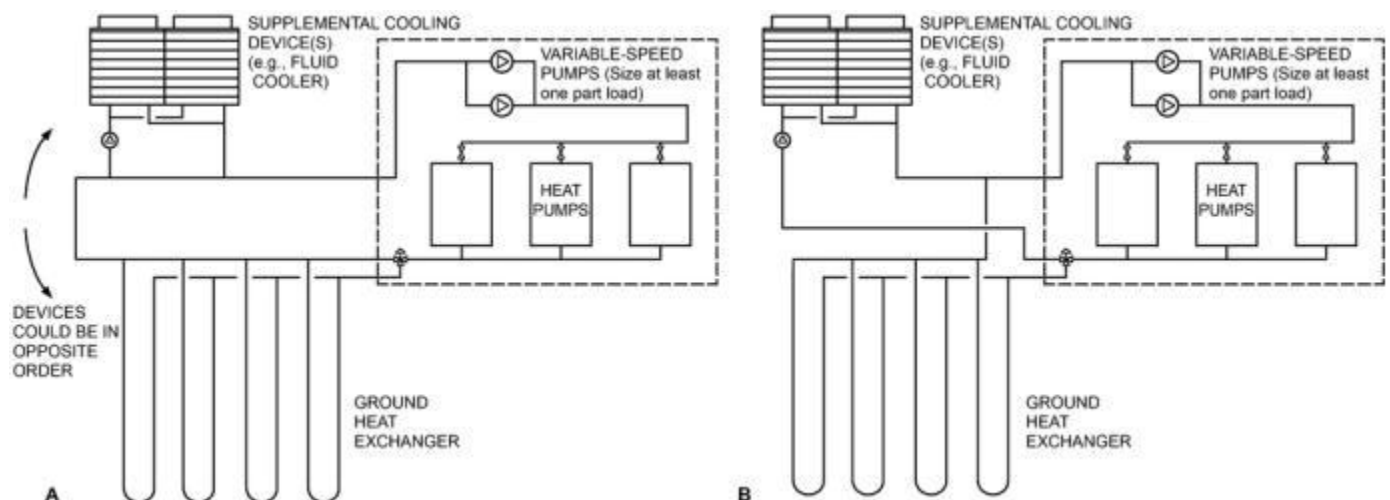


Figure 18. Hybrid System Configuration Options, (A) Series and (B) Parallel

For hybrid systems that require cooling towers, design also needs to consider water efficiency. Proper controls, as discussed previously, are a good start in minimizing water usage. Sequences can include a stage of cooling in which the spray pump is off (for when ambient temperatures are moderate). If the tower is run to precool the ground, this should be done carefully (Pertzborn et al. 2012) to avoid overusing the tower, which could result in significant energy and water penalties. Finally, operators should still follow the fundamental guidance for efficient tower operation (see [Chapter 40 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#)).

A heating-dominated hybrid with a boiler instead of a cooling tower can use a series configuration, with the boiler downstream of the loop (because of the boiler's high temperature output). The boiler is ideally controlled based on the temperature leaving the heat pumps.

Sizing hybrid components is a bit more complex than standard systems. For cooling dominated hybrids, Kavanaugh and Rafferty (2014) suggest that heat exchanger length for heating L_h be determined using [Equation \(8\)](#) with heating-mode loop temperatures t_{wi} and t_{wo} as low as possible to minimize L_c . A tower with an isolation heat exchanger is sized to meet the capacity difference between the required cooling length L_c from [Equation \(7\)](#) and the heating length L_h . Kavanaugh (1998) revised this method to include an additional iteration to size the ground heat exchanger only after estimating the annual heat rejection from the tower: $q_{tower}(\text{rated gpm}) = \text{gpm}_{system}(L_c - L_h)/L_c$, where L_c is calculated from [Equation \(5\)](#) but based on reduced EFLH_c to account for tower operation rejecting an estimated amount of the annual load. The strategy suggests eliminating long-term ground temperature change with additional tower operation.

A more detailed study (Hackel et al. 2009) included assumptions about typical installation and operating costs to demonstrate an optimized design strategy for cooling dominated hybrids. Based on life-cycle cost, this approach was roughly attractive whenever the peak heating load was less than 80% of the cooling load; savings increased logarithmically as the ratio decreased below 80%. A variety of cases were modeled, and the simplified best-fit regression for the hybrid ground heat exchanger length L_{hyb} in a cooling-dominated scenario was found to be proportional to heating load:

$$L_{hyb} = C_1 \times q_h / (t_g - t_{wo}) \quad (33)$$

where $C_1 = 254 \text{ ft} \cdot \text{h} \cdot ^\circ\text{F}/\text{kBtu}$, at $k = 1.4 \text{ Btu}/\text{ft} \cdot \text{h} \cdot ^\circ\text{F}$. For other ground conductivities, the change in ground heat exchanger size is approximately inversely proportional to the change in conductivity. In choosing t_{wo} , Hackel et al. (2009) also suggest in cooler climates it is often economical to include antifreeze in the system and allow the entering fluid temperature to drop to 35°F or lower. The supplemental cooling device (closed-circuit tower) should then be sized to meet the fraction of the cooling load that this smaller hybridized ground heat exchanger cannot. The study suggests that the tower should even be oversized slightly and its fan put on variable-speed control, to achieve optimal performance. Furthermore, tower sizing should be completed using the local peak wet-bulb and design entering fluid temperature for the hybrid.

This basic strategy of sizing hybrids was found to be valid for a wide range of economic scenarios. This economically justifies the general concept laid out in Kavanaugh (1998); however, results showed that it is generally not economically optimal to balance the load on the ground, and some increase in ground temperature can be accepted. Preliminary research showed that running the tower at night or in winter before there was significant cooling load (precooling the ground) to balance the ground load is not always necessary, and if done without care (i.e., in-depth energy analysis) could possibly lead to increased energy consumption (Pertzborn et al. 2012).

Regardless of the calculation method used, detailed building load calculation is critical when sizing and configuring a hybrid, to determine the impacts of heating and cooling loads across time scales from peak to annual. Further refinement of hybrid sizing (and control) can be done through energy simulation software, including some of the design software mentioned earlier, as well as a tool created as a result of ASHRAE research project RP-1384 (Hackel et al. 2009). Building energy simulation can estimate loads at every hour of the year, establishing better understanding of annual ground loads as well as the load sharing between ground heat exchanger and supplemental device.

Heating-Dominated Hybrids. These hybrids are needed in only very cold climates. In heating-dominated buildings in such climates (i.e., with approximately twice as much annual heating load as cooling load), however, a heating-dominated hybrid could decrease the size of the ground heat exchanger. Optimally, the ground heat exchanger is sized to meet the cooling load and a supplemental device meets the remaining heating load through a supplemental heat coil on the ground-coupled fluid loop. Coil energy could be supplied by gas boiler, solar panels, electric resistance, or another source. Several things need to be considered if using this approach:

- The potentially high temperatures of the heating coil are not covered by typical ground heat exchanger warranties.
- Controls must be maintained correctly. Place boiler in series, downstream of ground heat exchanger, with control of boiler and ground heat exchanger coordinated so that heat is never rejected to the ground while the boiler operates (very little of such heat would be recovered).
- The heat pumps operate whenever heating is needed, even though a high-temperature coil is also operating, reducing efficiency.

An alternative strategy for heating dominated buildings that poses some advantages is to simply remove some loads in the building from the ground-coupled loop and serve them by boilers, solar panels, or some other source which provides high enough temperatures to heat without added heat pump energy. Baseboard, unit heaters, and preheat coils can be good applications for this approach. This may alleviate some of the complications with boiler hybrids discussed previously.

Pump and Piping System Options

Loop design can have a substantial effect on both pumping power requirements and system installed cost. A GSHP survey (Caneta Research 1995a, 1995b) reported that installed pumping power varied from 0.04 to 0.21 hp/ton of heat pump power. This represents 4 to 21% of the total demand of typical GSHP systems and up to 50% of the total energy for some pump control schemes. [Table 8](#) gives a recommended set of guidelines for minimizing the power of closed-loop GSHPs and maximizing system efficiency.

Good grades ([Table 8](#)) can be obtained by minimizing extensive piping arrangements with long interior and exterior piping runs, high-head-loss fittings, valves, and control devices. Designers must compare the costs and advantages of large central piping loops and larger pumps with those of multiple smaller loops and smaller pumps. Pumping rates greater than 3.0 gpm/ton in closed-loop systems result in marginal equipment capacity gains in modern water-to-air heat pumps, and typically decrease overall system efficiency.

Kavanaugh et al. (2002) found the total cost of vertical GCHP ground-loop systems (including headers) ranged from \$6.00 to \$25.60 per foot of vertical bore. The cost of headers is a significant portion of the total and in many cases exceeded the cost of the vertical bore. The savings in vertical loop costs because of central systems' load diversity often is not warranted because of the increased cost of large-diameter piping networks connecting equipment inside the building and below-grade circuits connecting exterior ground heat exchangers.

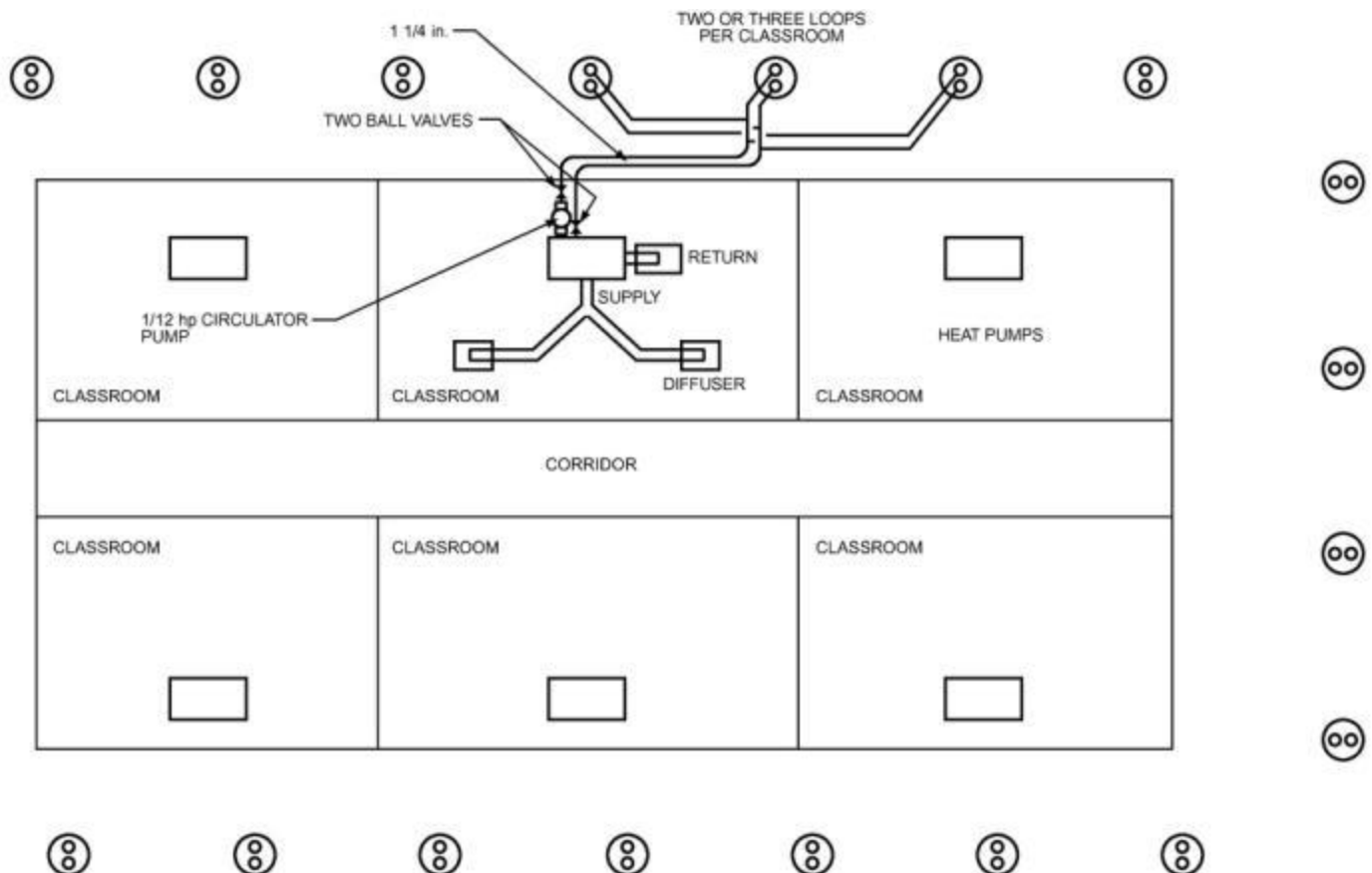


Figure 19. Unitary GCHP Loops with On/Off Circulator Pumps

In low-rise buildings with large footprints, such as a school, multiple unitary loop systems ([Figure 19](#)) are an effective option to offset the high cost of central interior piping and ground-loop header costs. Although the total length of vertical bore for unitary systems is greater than for central loop systems, the high cost of interior piping, exterior headers, and valve vaults often offsets the bore cost savings. Additionally, pump demand is substantially reduced in the unitary system because of the short header runs, so low-wattage on/off circulator pumps are suggested.

A compromise in applications with significant load diversity is to group ground heat exchangers into multiple smaller subcentral loops in different areas of the building ([Figure 20](#)). Subcentral loops can be served by on/off circulator pumps located on each heat pump if a check valve is installed on each heat pump to prevent reverse water circulation through idle units.

Table 8 Guidelines for Pump Power for GSHP Ground Heat Exchangers

Installed Pump Power		
hp _{pump} /100 tons	Grade	Head at 3 gpm/ton, ft of water
< 5	A	< 46
5 to 7.5	B	46 to 69
7.5 to 10	C	69 to 92
10 to 15	D	92 to 138
> 15	F	> 138

Source: Kavanaugh and Rafferty (2014).

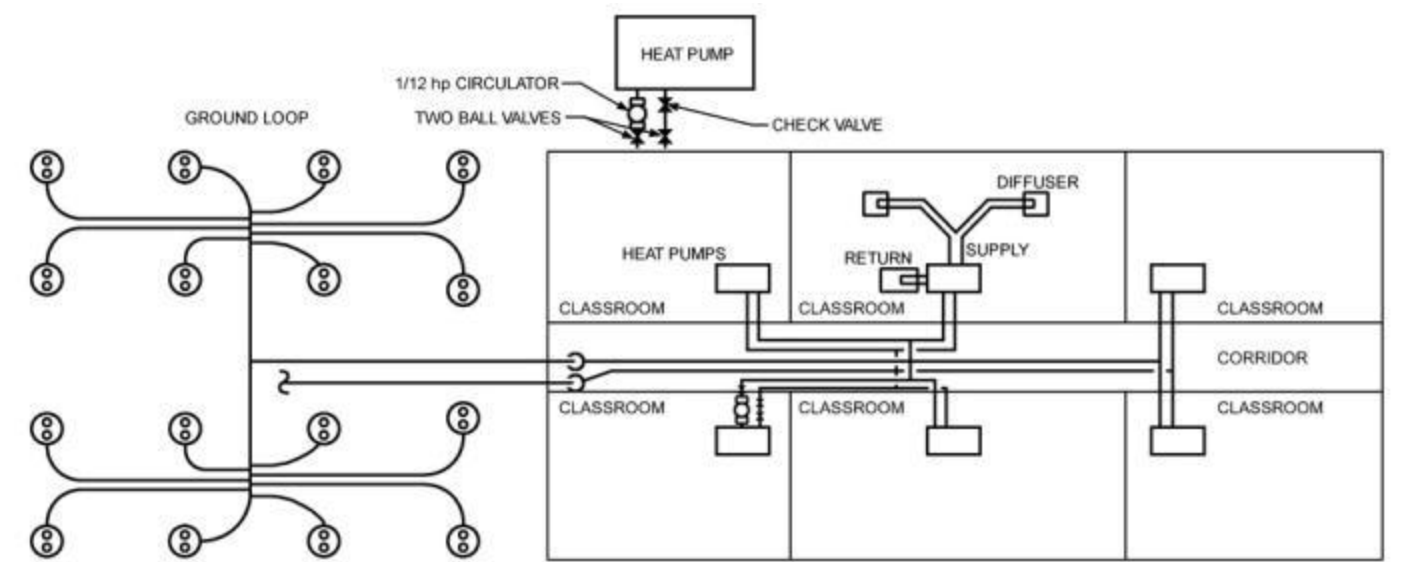


Figure 20. Subcentral GCHP Loop with On/Off Circulator Pumps

Figure 21 is an example of a central loop that can effectively reduce the cost of the required vertical bore in buildings with higher load diversities. The central ground heat exchanger consists of several subheader sets, each having 6 to 20 vertical U-tube heat exchangers. The subheaders are gathered into a valve manifold located either near the center of the loop field in a below-grade vault or in the building equipment room. Each subheader set has isolation valves for independent purging of air and debris. Interior piping is similar to conventional water-source heat pump systems in which interior piping is routed to individual water-to-air heat pumps in each zone and/or heat pump water heaters and water-to-water heat pumps.

Variable-speed drives (VSDs) are recommended for central loop systems because they offer substantial energy savings compared to primary/secondary pump schemes in GSHP applications. However, in buildings with primary occupancy of less than 60 h per week, measures should be incorporated to turn off the main VSD pump and provide some alternative means of pumping water to critical building zones during low-occupancy or unoccupied periods.

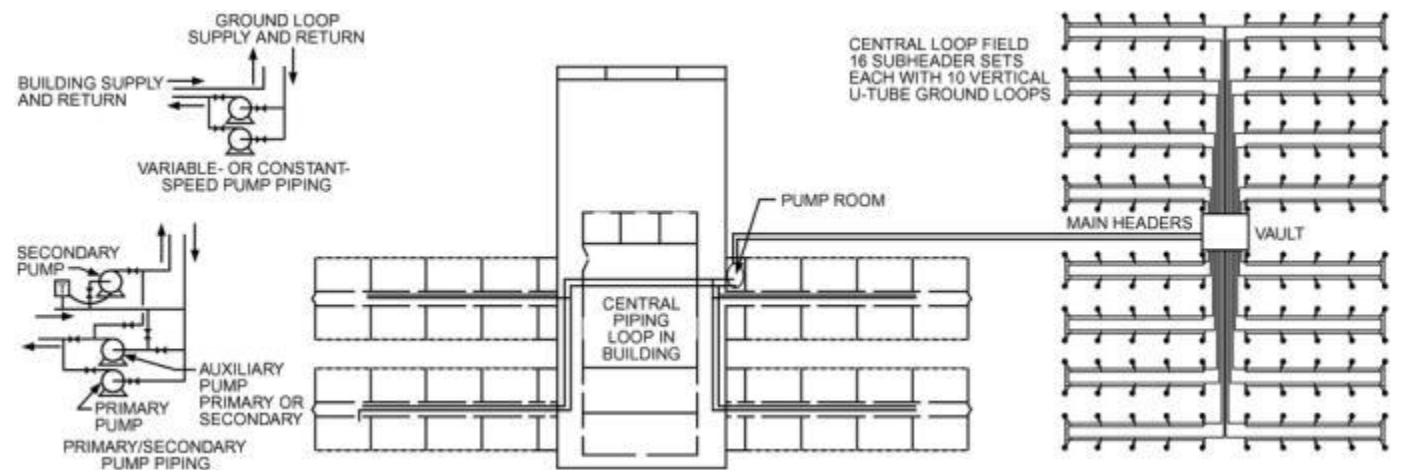


Figure 21. Central Loop GCHP

Several projects examined installation costs of nonresidential ground source heat pumps beginning in the mid 1990s: a large survey from an ASHRAE-sponsored research project (Caneta 1995) and a condensed report (Caneta 1998) studied systems located in colder climates, and Zimmerman (2000) looked at costs in several Tennessee Valley GSHP schools (Figure 22). Kavanaugh et al. (2012) also studied costs for the complete system and ground loop (Figure 23). Table 9 compiles the average, maximum, and minimum costs for these studies.

Table 9 Average Costs for Three GSHP Systems

	Caneta (1995)	Zimmerman (2000)	Kavanaugh et al. (2012)
System average, \$/ft ²	9.07	13.08	20.75
Maximum, \$/ft ²	14.34	17.41	26.10
Minimum, \$/ft ²	2.67	9.10	13.34
Ground loop average, \$/ft ²	3.49	3.76	5.29
Maximum, \$/ft ²	7.38	5.80	8.89
Minimum, \$/ft ²	0.60	1.93	3.35
Percent of total system cost	38.5	30.1	25.5

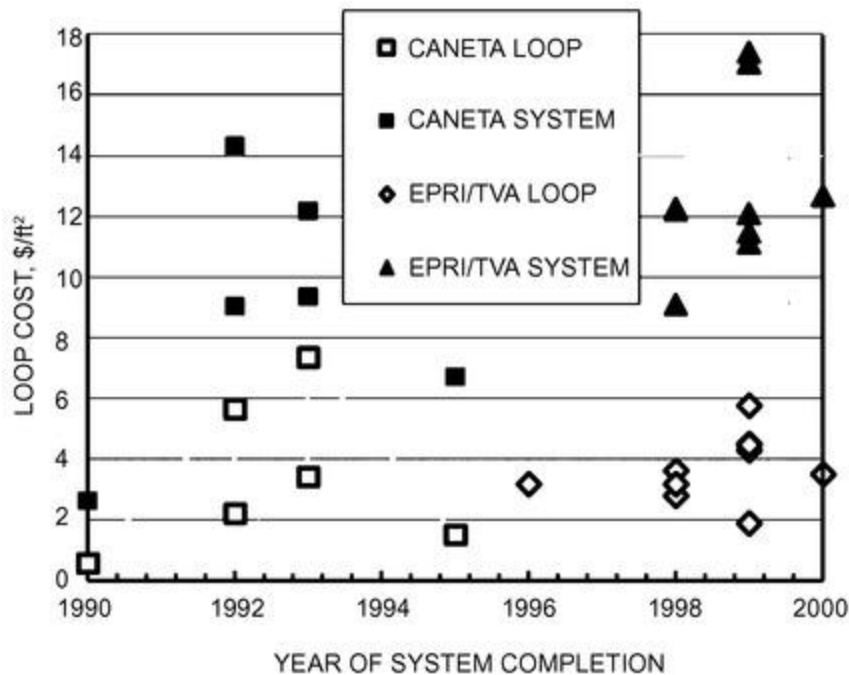


Figure 22. GSHP System and Loop Cost (Caneta 1995; Zimmerman 2000)

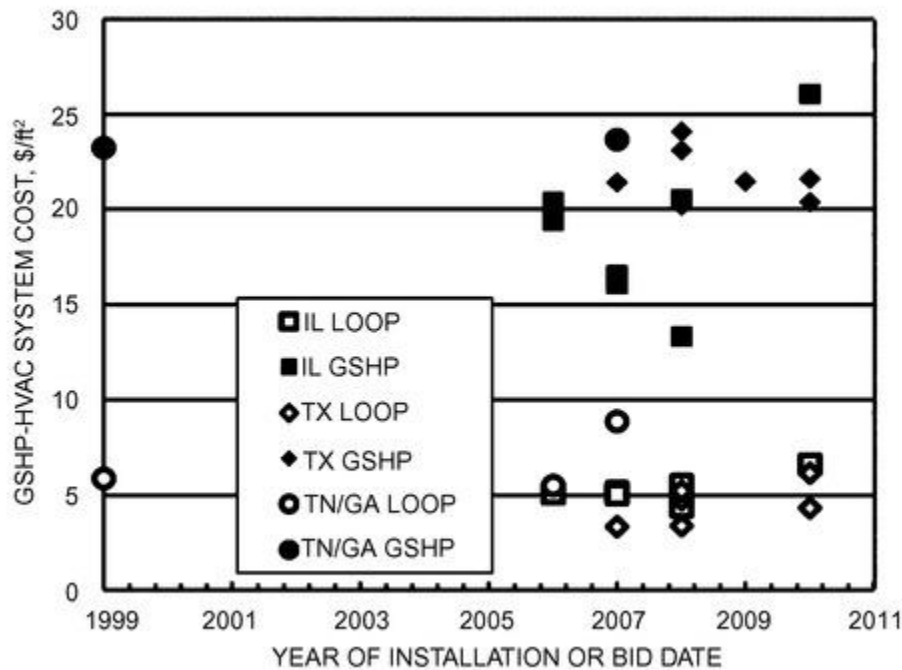


Figure 23. GSHP System and Ground Loop Cost Based on Building Floor Area

As [Table 9](#) shows, the percentage of ground loop costs to total GSHP system cost declined from 38.5% in 1995 to 25.5% in 2011. Results indicate there was a 177% increase in HVAC component costs compared to a 52% increase in ground loop costs during this 16 year period. The TX Loop (cost for ground loop installed in Texas) costs are an example of how multiple unitary loops can be very cost effective.

[Figure 20](#) shows the results of a cost comparison of a GSHP system with two four-pipe water-cooled chilled/hot-water systems for a 72,000 ft² school in Birmingham, Alabama with a 150 ton cooling requirement. The installation cost of the GSHP system was 11.7% lower than that of the constant-air-volume (CAV) system and 31.9% less than the variable-air-volume (VAV) system. Installation costs were based on 2014 data (Means 2014) but assumed the ground loop cost was \$15/ft of vertical ground heat exchanger. Costs include controls for components (thermostats, VAV actuators, sensors, etc.) but not building automation systems (BAS) or energy recovery units (ERUs).

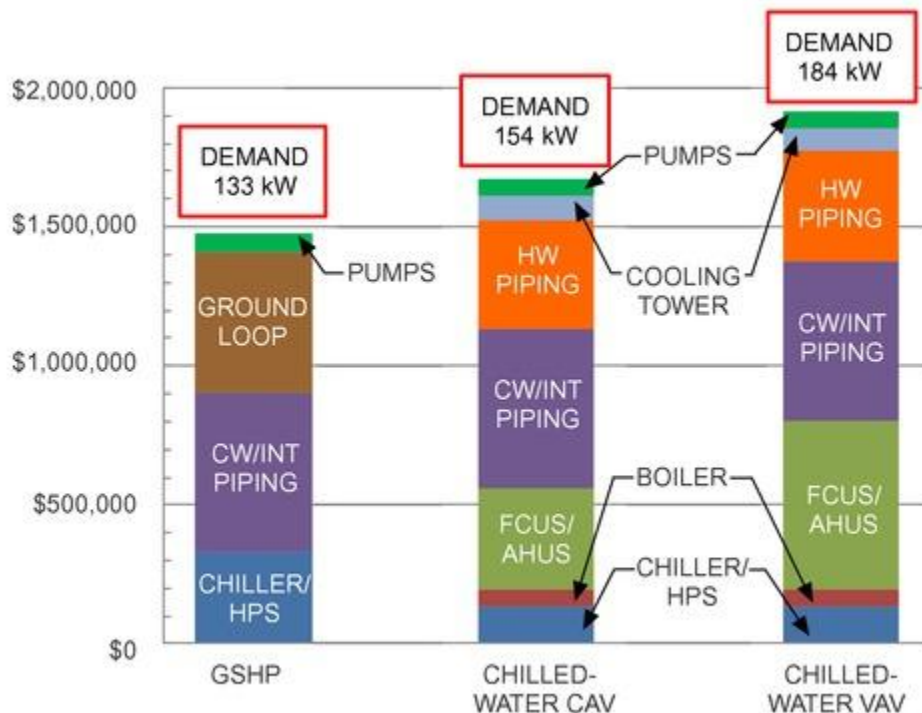


Figure 24. Project Installation Cost Comparison of 150 ton GSHP with Four-Pipe Systems

Additional cost data for regions across the United States are available in Battocletti and Glassley (2013).

GSHP Piping Materials. It is an industry standard that the buried ground-loop piping materials be fusion welded. For this reason, the current IGSHPA design and installation standards lists only high-density polyethylene (HDPE), and most recently an exception for cross-linked polyethylene (PEX-a) as approved materials for buried pipe. (IGSHPA 2011). ANSI/CSA/IGSHPA *Standard* C448-16 only discusses HDPE and PEX-a as approved materials for closed-loop buried pipe. In most cases, this pipe is buried without insulation.

Distribution systems in a building need to be carefully chosen and not solely driven by cost or ease of installation. As with any system, piping materials must be compatible with system design temperatures and with the fluids conveyed. Closed-loop systems typically use potable water and may also include antifreeze and/or water treatment chemicals. With newer refrigerants, care is needed to ensure the piping materials are compatible with the oils used in the refrigeration system and that equipment seals are compatible with the heat transfer fluids conveyed; for instance, polyolester (POE) oil used with R-410A is not compatible with PVC pipe. Consult chemical resistance charts provided by materials manufacturers.

Whether designing a new system or retrofitting an existing one, it is the design engineer's responsibility to size and select the proper piping materials for each application and to select only those materials allowed by code (ICC 2012). In large building systems, the aboveground piping specifications commonly include steel, iron, copper or PVC. Where the hydronic mains are 4 in. or larger, it is often more cost effective to specify steel pipe. Specifying steel pipe may require some type of water treatment to inhibit general corrosion and dielectric isolation when connected to ground-source heat pumps.

Water Quality, Heat Transfer Fluids, and Water Treatment. When engineers introduce dissimilar metals into closed-loop hydronic piping systems, improperly address water treatment requirements for their designs, or are not properly informed of the local groundwater regulations, water quality problems may result. Poor water quality contributes to a decline in mechanical system performance, increased maintenance, and can reduce the useful lifetime of mechanical system components.

The standard working fluid in small residential closed-loop piping systems is either potable water or, in colder regions, an antifreeze solution. Where required, the antifreeze solution is introduced after the ground loop is properly pressure tested, flushed, and purged of debris and air. (See the section on Antifreeze Requirements for further details on antifreeze solutions.) There has been little problem with or concern about water quality in these GSHP systems, because piping materials have historically been HDPE, copper, or stainless steel hose kits for final connection at the heat pumps.

Quality of potable water can vary, depending upon the source, and rules on its use vary from state to state. The chemistry of this water is one of the contributors to corrosion and water quality problems in closed-loop piping systems, so engineers need to be precise with hydronic piping and water treatment system specifications. [Chapter 50](#) offers some guidance on understanding the consequences of water quality on both open- and closed-loop hydronic systems.

The type of chemical used for water treatment in closed-loop systems is based on several criteria, including the materials being protected, the quality of the water in the piping system, local regulations, and cost. These systems are typically specified to include a rigorous process for cleaning the distribution piping, flushing the piping systems of air and debris, and then adjusting the water quality of the final local water to meet the long-term performance requirements of the building systems. Chemicals used to adjust water quality for these systems, including corrosion inhibitors, are often not acceptable for use when the circulating fluid is also connected to a ground loop. This is not a problem unless there is a pipe failure or problem with the ground loop, but a potential leak is a concern of the regulatory agencies that protect groundwater in each state.

Flush and Purge. To ensure that a ground-source heat pump system provides trouble-free operation, all ground-loop systems must be properly flushed and purged prior to connection to the building piping system (IGSHPA 2011). The current standard of care is defined by IGSHPA as providing a minimum velocity of 2 ft/s { through each piping system (but not in excess of the maximum flow velocity recommended by the pipe and fittings manufacturer) to remove air from the system. Flushing and purging each supply and return circuit in the forward and reverse directions for long enough to remove all debris and air from the system is also recommended. To attain proper flow velocity, appropriately sized and located purge ports and valves should be included in the mechanical design, preferably in a location where manipulation of the valve(s) allows independent flush and purge of the building and/or the ground loop. Special care should be taken to ensure that air is not pumped from the building into the ground loop.

Recommendations for Good GSHP Piping System Design.

- Before beginning design, consult the local regulatory agency for guidance on requirements related to the ground-loop portion. In many locations, this may be the Departments of Health or Environmental Services. Drilling for a vertical closed-loop system may not be allowed. Local groundwater conditions may limit the use of certain working fluids due to the sensitive nature of the resource and the concern for contamination.
- Conform to applicable codes on adding non-potable chemicals to building mechanical piping systems and for any discharge to public sewage systems. Good piping designs minimize the need for chemical treatment and any potential impact to the environment.
- Note that corrosion occurs at some level in all hydronic systems. The best GSHP distribution system from a corrosion standpoint is one that includes only HDPE. Breaking down the system into the subcentral or individual loops may accommodate the use of only HDPE in large buildings.

- Review pressure ratings of all piping materials specified for the system design temperatures.
- Minimize the number of fittings, valves, and specialties where practical. Fittings, valves, specialties, and pumps perform best when stainless steel or bronze fitted.
- Even in a closed-loop piping system, it is possible for air to be introduced by expansion and contraction of the pipe and occasional addition of make-up water. For suspended gases, air separators installed upstream of the pumps with an expansion tank can provide deaeration. For dissolved gases, particularly oxygen, the problem can only be addressed with chemical treatment.
- It is very important that all ground-loop and building systems are properly flushed and purged of debris. Not only can debris clog equipment, but biological material contributes to a drop in water pH as it decays.
- Take a water sample of the local potable water supply if it is planned to be the hydronic system's working fluid. If dissimilar metals are used and/or the local water quality is poor, a water-treatment specialist can help select a water treatment regimen for the system. Alert the owner to the requirement for periodic monitoring of water quality.
- If a chemical pot feeder is added into the system as a place to introduce the chemicals for the hydronic system, include an integral filter. The filter will help to maintain water quality and remove residual debris that may break loose after the system has been properly flushed and purged.
- If propylene glycol is to be used as an antifreeze, note that concentrations less than 20% may promote bacterial growth. Check this percentage with the supplier (NGWA 2010).

Pressure Considerations in Deeper Vertical Boreholes

In deep vertical boreholes, it is especially important for the designer to be aware of conditions that may lead to pipe failure. The hydrostatic pressure exerted on both the inner and outer wall of the U-bend piping during installation should be considered.

Internal Pressure Considerations. Pressure is exerted on the inner pipe wall during installation but before grout placement, because the pipe is filled with water to offset its buoyancy in a water- or drilling-fluid-filled bore. [Equation \(33\)](#) may be used to determine the **internal working pressure (IWP)** during vertical loop installation. (Note that the depth to the static water table, which is site specific, must be known.)

$$\text{IWP} = 0.433 \times \text{Depth} \quad (34)$$

where

IWP = internal working pressure, psi

Depth = depth to static water table, ft

The internal pressure caused by water column height is offset in the portion of the loop that is installed below the static water table.

Once the internal working pressure is known, the designer should ensure that the **internal pressure rating (IPR)** of the pipe is not exceeded. The IPR of HDPE is a function of pipe material type (cell classification), wall thickness (DR), and temperature. The IPR for HDPE 3408/3608 and 4710 at 73.4°F for DR-11 and DR-9 are shown in [Table 12](#); compensating factors to account for other temperatures are shown in [Table 11](#). To determine the IPR at the pipe's actual working temperature, simply multiply the IPR at 73.4°F by the appropriate temperature compensating multiplier.

External Pressure Considerations. Pressure is exerted on the outer pipe wall by the liquid grout slurry as it is pumped into the bore. As a liquid, grout rests against the outer pipe wall, exerting pressure until it hardens (sets). In general, the maximum working time for most grouts is 30 to 45 min, depending on factors such as makeup water temperature and chemistry, borehole temperature, etc. After this amount of time, the grout begins to set into its permanent state as a semirigid plug. Note that grouting is not required in some jurisdictions and not addressed in others; this has given rise to substituting a manufactured fill material, which may be acceptable in some cases.

After it sets, grout can partially support its own weight. It still exerts some pressure on the outer pipe wall, but the amount is far less compared to when in its liquid state. The amount of pressure exerted by the liquid grout column increases with depth, and is at maximum at the bottom of the bore. [Equation \(34\)](#) calculates the pressure exerted on the outer pipe wall after grout placement and before setting. The external pressure is due to density differences between the liquid grout column on the outer and the fluid (water) on the inner pipe wall:

$$\text{EWP} = 0.052(\rho_{\text{grout}} - \rho_{\text{water}}) \times \text{Depth} \quad (35)$$

where

EWP = external working pressure, psia

ρ_{water} = density of internal fluid, which is typically water during grout installation, lb/gal

ρ_{grout} = density of the external grouting fluid, lb/gal

Depth = borehole depth, ft

Once the external working pressure is known, the designer should ensure that the **external pressure rating (EPR)** of the pipe is not exceeded. The EPR of HDPE is a function of pipe material type (cell classification), wall thickness (DR), temperature, percent deflection (pipe ovality), and duration of exposure to external pressures. The EPR for HDPE 3408/3608 and 4710 at 73.4°F for DR-11 and DR-9 are shown in [Table 12](#), and compensating factors to account for other temperatures are shown in [Table 11](#). Compensation factors to account for pipe ovality are shown in [Figure 25](#) and [Table 13](#), and those to account for duration of sustained pressure on the outer wall of the HDPE pipe are shown in [Table 14](#). Note that the 1 h correction factor is appropriate for bentonite-based grouts, whose working time is generally 30 to 45 minutes. Cement-based grouting materials usually require use of 10 or 24 h compensation factors. Contact the manufacturer for more information.

Table 10 Internal Pressure Rating (IPR) for HDPE

HDPE at 73.4°F	psig (ft of water)	
	DR-11	DR-9
3408/3608	160 (368)	200 (461)
4710	200 (461)	250 (576)

Source: PPI (2018).

Table 11 Temperature Compensating Multipliers for HDPE

Pipe Temperature, °F	Compensating Multiplier
-20	2.54
-10	2.36
0	2.18
10	2.00
20	1.81
30	1.65
40	1.49
50	1.32
60	1.18
73.4	1.00
80	0.93
90	0.82
100	0.73
110	0.64
120	0.58
130	0.50
140	0.43

Source: PPI (2018).

Table 12 External Pressure Rating (EPR) for HDPE*

HDPE at 73.4°F	psig (ft of water)	
	DR-11	DR-9
3408/3608	185.7 (427.9)	362.6 (835.5)
4710	195.7 (450.9)	382.2 (880.6)

Source: PPI (2018).

* Based on the 1 h apparent modulus rating for HDPE, assuming a safety factor $N_s = 1.0$.

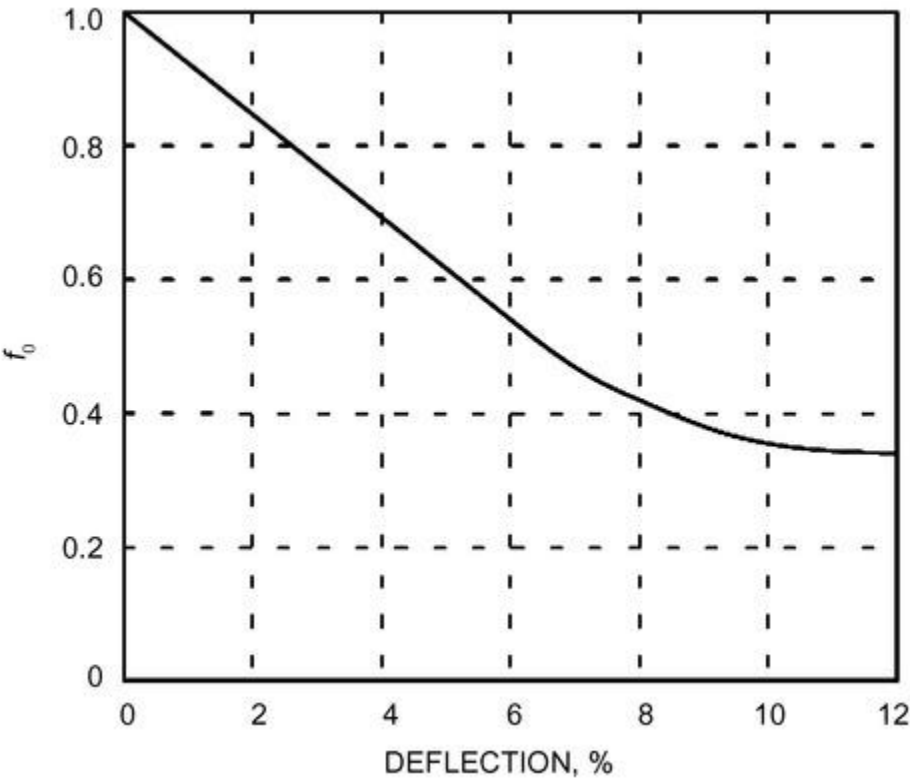


Figure 25. Ovality Compensation Factors for HDPE (PPI 2018)

Table 13 Safe Deflection Limits for Pressurized Pipe

Dimension Ratio (DR)	Safe Deflection, %
32.5	7.5
26	7.5
21	7.5
17	6.0
13.5	6.0
11	5.0
9	4.0
7.3	3.0

Source: PPI (2018).

To determine the EPR at the actual working temperature of the pipe for a given ovality, multiply the EPR at 73.4°F by the appropriate temperature, ovality, and sustained pressure duration compensating multipliers. Additionally, the percent deflection must be lower than the limits specified in [Table 13](#).

Table 14 Sustained External Pressure Duration Compensation Factors for HDPE

Duration	PE3408/3608	PE4710
0.5 h	1.054	1.051
1 h	1.000	1.000
2 h	0.959	0.949
10 h	0.838	0.833
12 h	0.811	0.808
24 h	0.770	0.769
100 h	0.703	0.705
1000 h	0.595	0.590
1 yr	0.514	0.513

10 yr	0.432	0.436
50 yr	0.378	0.372
100 yr	0.365	0.359

Source: PPI (2018).

As shown in [Figure 25](#), HDPE EPR is very sensitive to ovality, which is determined by calculating the percentage reduction in pipe diameter along the deformed section of pipe (when applicable):

$$\text{Ovality, \%} = \frac{\text{OD}_{nom} - \text{OD}_{min}}{\text{OD}_{nom}} \times 100 \quad (36)$$

where

OD_{nom} = nominal U-bend pipe diameter, in.

OD_{min} = U-bend pipe diameter along deformed section, when applicable, in.

Note that any section of pipe that exhibits ovality greater than the recommended limit shown in [Table 11](#) should be removed from the system and discarded.

If exceeding the U-bend EPR becomes a concern, there are three primary ways to minimize the potential for issues to occur:

- Use a heavier pipe wall thickness. Remember that thicker-walled pipe is more expensive and may not be as readily available; its use also effectively reduces the inside pipe diameter, which increases system head loss and associated pumping power requirements
- Pressurize the U-bend from the surface before pumping grout into the bore to counteract external pressures during grout placement. Do not pressurize the loop above its IPR. Additionally, if the bore is completely dry, adding pressure to the loop from the surface will not be an option. Without water in the bore to counteract the pressure applied to the inside pipe wall, the U-bend's internal pressure rating will quickly be exceeded (burst pressure).
- Reduce grout density by using graphite in place of silica sand (without sacrificing thermal performance). Densities of graphite-based mixes are typically low enough that bore collapse should no longer be a concern for common bore installation depths. Contact the grout manufacturer for additional information.

Example 4. Calculate the external pressure rating of HDPE 4710 and the external hydrostatic pressure exerted on the outer pipe wall during grout placement. Assume the pipe will be DR-11 and is perfectly round (0% ovality), that a bentonite-based grouting material is used in the bore annulus, and that the 1 h apparent modulus is appropriate.

Ground temperature, $t_g = 60^\circ\text{F}$

Grout density, $\rho_{grout} = 15.1 \text{ lb/gal}$

Internal fluid (water) density, $\rho_{water} = 8.34 \text{ lb/gal}$

Borehole depth = 500 ft

Bore fill conductivity $k_{grt} = 1.0 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$

Percent deflection (pipe ovality) compensation factor = 1.00

Temperature compensation factor = 1.18

$$\text{EPR} = (195.7 \text{ psi})(1.18) = 230.9 \text{ psi}$$

$$\text{EWP} = (0.052)(15.1 \text{ lb/gal} - 8.34 \text{ lb/gal})(500 \text{ ft}) = 175.76 \text{ psi}$$

The external pressure rating (EPR) exceeds the external working pressure (EWP) of the piping system during grout placement (assuming no safety factor, $N_s = 1.00$).

Table 15 Rating Conditions for Water-to-Air Heat Pumps for Total Cooling (TC, Btu/h), Energy Efficiency Ratio (EER, Btu/W · h), Heating Capacity (HC, Btu/h) and Coefficient of Performance (COP, W/W)

Entering Liquid and Air	WLHP Water Loop	GWHP Ground-water	GLHP Ground Loop	GLHP-PL (Part-Load)
ELT (sink, cooling)	86°F	59°F	77°F	68°F

ELT (source, heating)	68°F	50°F	32°F	41°F
EAT (db/wb, cooling)	80.6/66.2°F	80.6/66.2°F	80.6/66.2°F	80.6/66.2°F
EAT (heating)	68°F	68°F	68°F	68°F

Source: ANSI/ARI/ASHRAE/ISO Standard 13256-1.

Required fan power to overcome external static pressure (ESP) and pump power to circulate liquid for piping loop not included in calculation of TC, EER, HC, and COP.

Effect of GSHP Equipment Selection on Heat Exchanger Design

The ground heat exchanger must absorb the heat of compression and heat from auxiliary equipment (e.g., fans, pumps) in cooling mode. In heating mode, heat from auxiliary equipment reduces the amount of heat required from the ground. Therefore, the cooling- and heating-mode power values W_c and W_h in [Equations \(7\)](#) and [\(8\)](#) must include the auxiliary input.

Rated values for GSHPs published in compliance with ANSI/ARI/ASHRAE/ISO Standard 13256-1 do not include the auxiliary power required to circulate air and water through the distribution systems. Furthermore, the auxiliary power required to distribute chilled air (and water) can have a substantial negative effect on the equipment's cooling capacity.

[Table 15](#) summarizes the air and water temperatures used to generate the rated performance of water-to-air heat pumps. [Table 16](#) summarizes the conditions for ANSI/ARI/ASHRAE/ISO Standard 13256-2, which rates water-to-water heat pumps.

Actual heat pump performance can be substantially different from rated conditions. Designers must convert rated performance to design conditions by accounting for the effect of auxiliary power input and for design ELTs and EWTs. When water-to-water heat pumps or chillers are used in this application, corrections should include power for the pump(s) of the source/sink loop and the chilled-/hot-water loop, and power for fans in the air distribution system. [Table 17](#) demonstrates the difference between rated GSHP efficiency and actual system efficiencies for various options when the effect of auxiliary components is considered. Note that using a high-static-pressure air handler for air distribution significantly reduces cooling efficiency. In heating, 120°F hot water also lowers heating COP compared to direct condensation and hydronic systems (e.g., in-floor heating) that use lower-temperature water.

Table 16 Rating Conditions for Water-to-Water Heat Pumps for Total Cooling (TC, Btu/h), Energy Efficiency Ratio (EER, Btu/W · h), Heating Capacity (HC, Btu/h) and Coefficient of Performance (COP, W/W)

Entering Liquid and Air	WLHP (Water Loop)	GWHP (Ground-water)	GLHP (Ground Loop)	GLHP-PL (Part-Load)
ELT (sink)	86°F	59°F	77°F	68°F
ELT (source)	68°F	50°F	32°F	41°F
ELT (building)	53.6°F	53.6°F	53.6°F	53.6°F
ELT (building)	104°F	104°F	104°F	104°F

Source: ANSI/ARI/ASHRAE/ISO Standard 13256-2.

Pump power to circulate liquid for source/sink and building loops not included in calculation of TC, EER, HC, and COP.

Horizontal and Shallow Vertical System Design

The buried pipe of a closed-loop GSHP may theoretically produce a change in temperature in the ground up to 16 ft away. For all practical purposes, however, the ground temperature is essentially unchanged beyond about 3 ft from the pipe loop. For that reason, the pipe can be buried relatively near the ground surface and still benefit from the moderating temperatures that the earth provides. Because the ground temperature may fluctuate, an antifreeze solution must be used in most heating-dominated regions. The critical design aspect of horizontal applications is to have enough buried pipe loop in the available land area to serve the equipment. The design guidelines for residential horizontal loop installations can be found in IGSHPA (2009).

Limitations on selecting a horizontal loop design include the following:

- The minimum land area needed for most nonspiral horizontal loop designs for an average house is about 0.5 acre. Horizontal systems are not feasible for most urban houses, which are commonly built on smaller lots.
- The larger length of pipe buried relatively near the surface is more susceptible to being cut during excavations for other utilities.
- Soil moisture content must be properly accounted for in computing the required ground heat exchanger length, especially in sandy soils or on hilltops that may dry out in summer.

- Rocks and other obstructions near the surface may make excavation with a backhoe or trencher impractical.
- Multiple pipes are often placed in a single trench to reduce the land area needed for horizontal loop applications. Some common multiple-pipe arrangements are shown in [Figure 26](#). When pipes are placed at two depths, the bottom row is placed first, and then the trench is partially backfilled before the upper row is put in place. Rarely are more than two layers of pipe used in a single trench because of the extra time needed for the partial backfilling. Higher pipe densities in the trench provide diminishing returns because thermal interference between multiple pipes reduces the heat transfer effectiveness of each pipe. The most common multiple-pipe applications are the two-pipe arrangement used with chain trenchers and the four- or six-pipe arrangements placed in trenches made with a wide backhoe bucket.

Table 17 Rated Efficiency, Component Power, and Corrected System Efficiency for Various GSHP Equipment Options (86°F ELT Cooling/50°F ELT Heating)

GSHP Cooling Equipment and System Description	Rated EER, Btu/Wh	Evap. Type	Fan Power, kW	Cond. Pump, kW	CW Pump, kW	Parasitic Heat, kBtu/h	System EER, Btu/Wh
4 ton WAHP, 75/63°F EAT	16.5	45°F DX	0.63	0.21	—	−2.1 (4.4%)	13.8
10 ton WWHP, 4000 cfm/4 in. of water AHU	13.6	45°F CW	3.36	1.07	1.07	−15.1 (12.6%)	7.9
10 ton WWHP, four 1000 cfm/1 in. of water FCUs	13.6	45°F CW	1.80	1.07	1.07	−9.8 (8.0%)	9.4
500 ton chiller, 4 in. of water AHUs, 2 in. of water return fans, series FPVAV	24.0 0.5 kW/ton	45°F CW	314	27	36	−1190 (20%)	7.4
500 ton chiller, 200 to 1000 cfm/1 in. of water FCUs	24.0 0.5 kW/ton	45°F CW	90	27	36	−430 (7.2%)	12.8
GSHP Heating Equipment and System Description	Rated COP, Btu/Wh	Cond. Type	Fan Power, kW	Cond. Pump, kW	CW Pump, kW	Parasitic Heat, kBtu/h	System COP
4 ton WAHP, 70°F EAT	4.7	Dir.	0.63	0.21	—	+2.1 (4.4%)	4.0
10 ton WWHP, 4000 cfm/4 in. of water AHU	3.8	120°F HW	3.36	1.07	1.07	+14.4 (12.0%)	2.5
10 ton WWHP, four 1000 cfm/1 in. of water FCUs	3.8	120°F HW	1.57	1.07	1.07	+8.3 (6.9%)	2.8
10 ton WWHP, 4000 cfm/4 in. of water AHU	3.8	100°F HW	3.36	1.07	1.07	+14.4 (12.0%)	3.1
10 ton WWHP, in-floor heat	3.8	100°F HW	0	1.07	1.07	+3.7 (3.1%)	3.7

An overlapping spiral configuration ([Figure 27](#)) has also been used with some success. However, it requires special attention during backfilling to ensure that soil fills all the pockets formed by the overlapping pipe. Large quantities of water must be added to compact the soil around the overlapping pipes. The backfilling must be performed in stages to guarantee complete filling around the pipes and good soil contact. The high pipe density (up to 10 ft of pipe per linear foot of trench) may cause problems in prolonged extreme weather conditions, either from soil drying during cooling or from freezing during heating. This spiral design has been used in vertical trenches cut with a chain trencher as well as in laying the coil flat on the bottom of a large pit excavated with a bulldozer. Installations using the horizontal spiral coil on the bottom of a pit have generally performed better than those with spiral coils that were stood upright in a vertical trench.

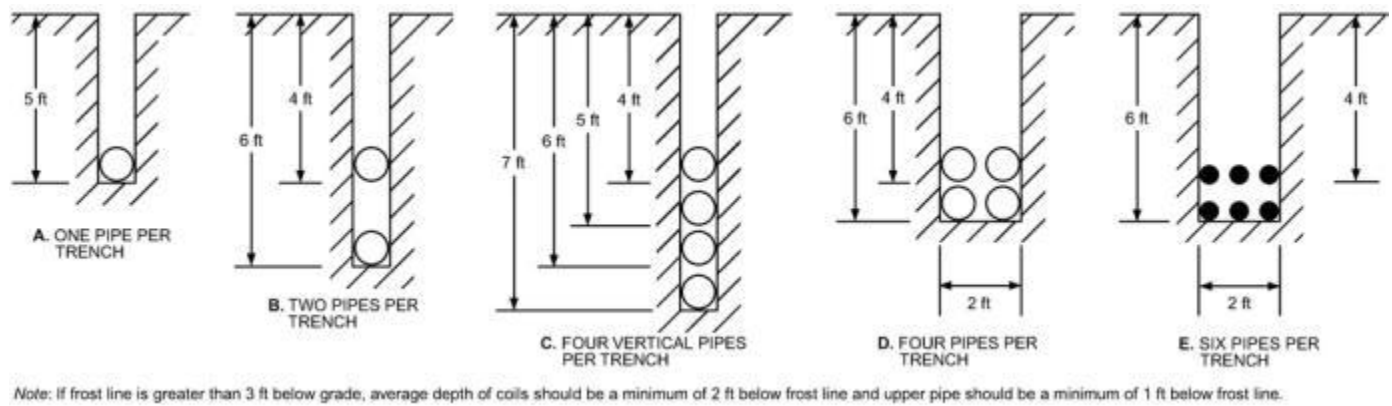


Figure 26. Horizontal Ground Heat Exchanger Configurations

The extra time needed to backfill and the extra pipe length required make spiral configurations nearly as expensive to install as straight pipe configurations. However, the reduced land area needed for the more compact design may allow use on smaller residential lots that are too small for conventional horizontal-pipe ground-loop designs. The spiral pipe configuration laid flat in a horizontal pit arrangement is used commonly in the northern Midwest of the United States, where sandy soil causes vertical trenches to collapse. A large open pit is excavated by a bulldozer, and then the overlapping pipes laid flat on the bottom of the pit. The bulldozer is also used to cover the pipe, being careful to not damage them with the bulldozer's grousers.

Although most horizontal closed-loop systems (see [Figure 3](#)) are installed with either a chain trencher or a backhoe, horizontal boring machines are also now available for this application. Developed for buried utility applications such as electric or potable water service, these devices simply bore through the ground parallel to the ground surface. A detector at the surface can show the exact point where the boring head is located underground so that the bore does not penetrate other known utilities or cross over into a neighbor's lot.

Most horizontal loop installations place the pipe loops in a parallel rather than a single (series) loop to reduce pumping power ([Figure 28](#)). Splitting the flow into parallel loops increases the fluid-to-soil temperature difference and subsequent heat transfer. Parallel loops may require slightly more pipe, but may use smaller pipe and thus have smaller internal volumes, requiring less antifreeze (if needed). Also, smaller pipe is typically much less expensive for a given length, so total pipe cost should be less for parallel loops. An added benefit is that parallel loops can be flushed out with a smaller purge pump than is required for a larger single-pipe loop. A disadvantage of parallel loops is the potential for unequal flow in the loops and thus nonuniform heat exchange efficiency.

The time required to install a horizontal loop is not much different from that for a vertical system. For the arrangements described, a two-person crew can typically install the ground heat exchanger for an average house in a single day.

Soil characteristics are an important concern for any ground heat exchanger design. With horizontal loops, the soil type can be more easily determined because the excavated soil can be inspected and tested. EPRI (1990) lists criteria and simple test procedures that can be used to classify soil and rock for horizontal ground-loop design.

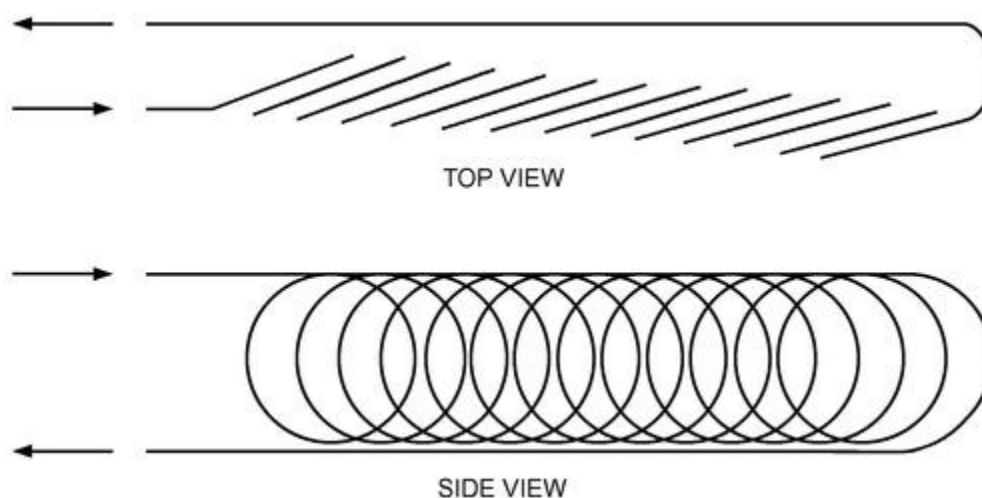


Figure 27. General Layout of Spiral Earth Coil

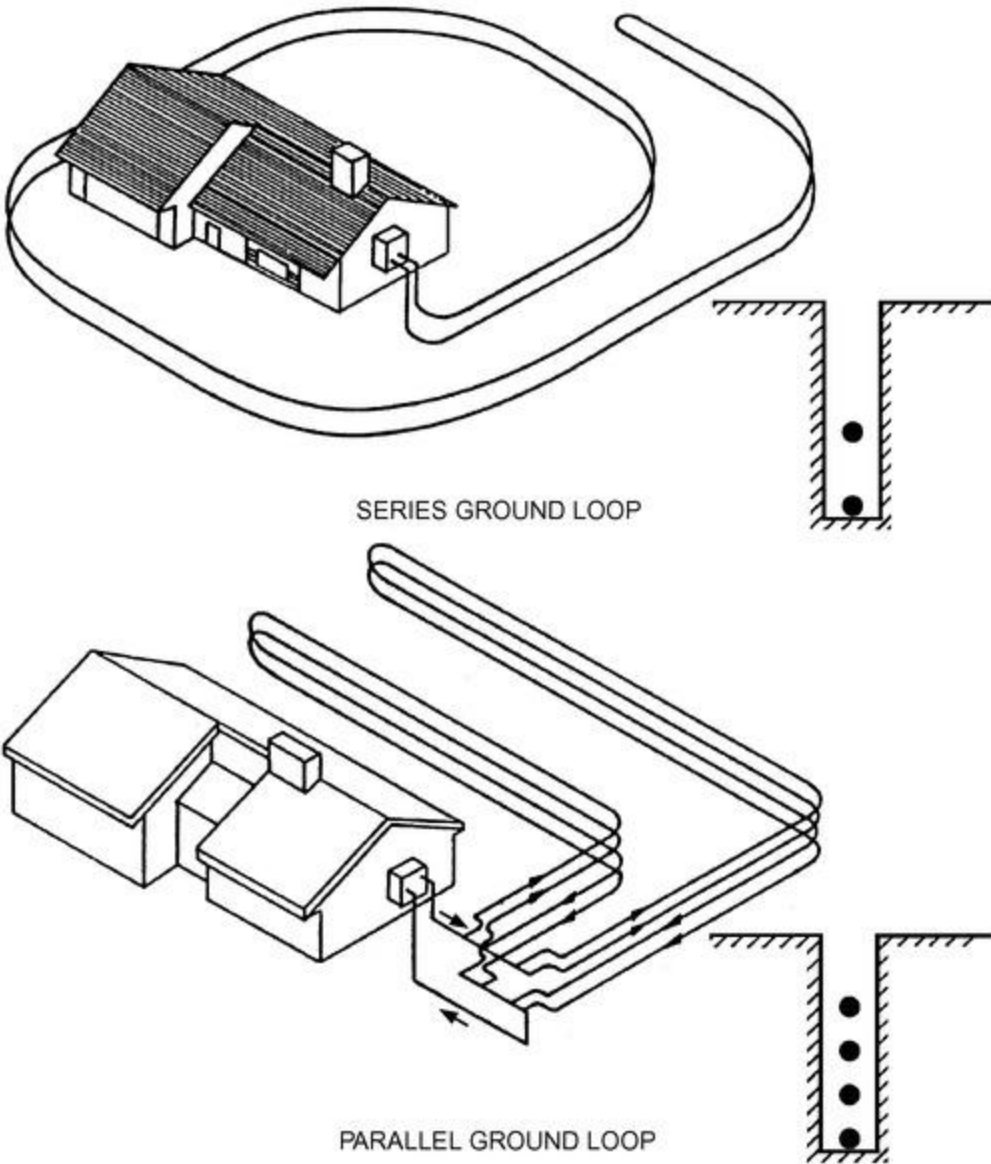


Figure 28. Parallel and Series Ground Heat Exchanger Configurations

Although soil type and moisture content are important considerations in sizing the ground heat exchanger, some design guidelines have been developed based on extensive analysis of monitored systems in mostly southern climates (Kavanaugh and Calvert 1995). [Table 18](#) gives recommended trench lengths for the various types of commonly used excavation methods. Heating-mode run times approaching 100% on a daily basis would be the norm at heating design conditions in heating-dominated climates. In contrast, daily run times of no more than 50% would be encountered at design cooling conditions in cooling-dominated climates. The combination of long run times and ice formation around the pipes makes performance of horizontal systems dependent on both the loop field design and how the system is matched to the building load. Though many thousands of these systems have been installed in heating climates, no comparable analysis has been performed to determine proper design guidelines. The loop length data in [Table 18](#) for soil temperatures below 56°F are based on nominal heat pump capacity and use of supplemental resistance heat at design conditions. If installing such a system for the first time, contact several experienced contractors in the area to determine successful design lengths for the local climate and soil types.

Table 18 Recommended Lengths of Trench or Bore per Ton for Residential GCHPs

		Pitch ^b	Ground Temperature, °F						
Coil Type ^a		Feet of Pipe per Feet Trench/Bore	44 to 47	48 to 51	52 to 55	56 to 59	60 to 63	64 to 67	68 to 70
Horizontal	6-Pipe/6-Pitch Spiral	6	180	160	150	160	180	200	230
	4-Pipe/4-Pitch Spiral	4	220	200	190	200	220	250	300
	2-Pipe	2	300	280	250	280	300	340	400
Vertical U-tube	3/4 in. Pipe	2	180	170	155	170	180	200	230

1 in. Pipe	2	170	160	150	160	170	190	215
1 1/4 in. Pipe	2	160	150	145	150	160	175	200

Multiply [Table 16](#) Values by Bold Values Below to Correct for Other Values of Ground Conductivity

	Ground Thermal Conductivity in Btu/h · ft · °F								
	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
Horizontal loop	1.22	1.0	0.89	0.82	—	—	—	—	—
Vertical loop ^a	—	—	1.23	1.10	1.0	0.93	0.87	0.83	0.79

Source: Kavanaugh and Calvert (1995).

Note: Based on $k = 0.6$ Btu/h · ft · °F for horizontal loops and $k = 1.2$ Btu/h · ft · °F for vertical loops. Figures for soil temperatures < 56°F based on modeling using nominal heat pump capacity and assumption of auxiliary heat at design conditions.

^a Vertical loop values based on an annular fill with $k = 0.85$ Btu/h · ft · °F. Multiply lengths by 1.2 for $k_{annulus} = 0.4$ Btu/h · ft · °F and 0.95 for $k_{annulus} = 1.1$ Btu/h · ft · °F.

^a Lengths based on DR11 high-density polyethylene (HDPE) pipe. See [Figures 24](#) to [26](#) for details.

^b Multiply length of trench by pitch to find required length of pipe.

Trench lengths in [Table 18](#) are based on a minimum trench separation of 10 ft and minimum horizontal loop average burial depth of the greater of 5 ft or 2 ft below the frost line. Bore lengths are based on a vertical bore separation of 20 ft. Design ground heat exchanger temperatures are a maximum of 90°F return and 100°F entering in warm climates and a minimum of 28°F return and 22°F entering in cold climates.

Additional considerations for horizontal loop systems in colder climates arise from the potential for ice formation around the pipe loop. The loop should not pass within 2 ft of any buried water line (potable, sewer, or rainwater). If such proximity cannot be avoided, the GCHP loop can be insulated in that area. Horizontal loops should not be placed closer than 6 ft from a basement or crawl space wall when buried parallel to the wall. Heaving from ice formation could cause structural damage if placed in close proximity to the wall.

Leaks in heat-fused plastic pipe are rare when attention is paid to pipe cleanliness and proper fusion techniques. Should a leak occur, it is usually best to try to isolate the leaking parallel loop and abandon it in place. The effort required to find the source of the leak usually far outweighs the cost of replacing the defective loop. Because the loss of as little as 0.25 gal of water from the system causes the system to lose pressure and shut down, leaks cannot be located by looking for wet soil, as is commonly done with water lines.

Although leaks should be rare with properly thermally fused pipe, some states have adopted restrictions against the use of certain types of antifreeze mixtures in GCHP systems; check local water-quality regulations before selecting a mixture. Methanol has been used extensively because of its low cost and good physical properties when cold. Comprehensive studies by Heinonen and Tapscott (1996) and Heinonen et al. (1997) showed that propylene glycol is a good alternative when issues of flammability or environmental safety are important considerations. A more thorough discussion of antifreeze solutions is given in the Antifreeze Requirements section of this chapter.

Fluid Flow and Loop Circuiting. Residential systems, like commercial applications, sometimes have excessive pumping power. This trend may result from undersized piping, excessive amounts of viscous antifreeze solutions, or conservative pump sizing. Because a 3 ton heat pump with an energy efficiency ratio (EER) of 15 requires a total power (compressor and fan) of 2400 W, the addition of a second 1/6 hp pump (which draws 245 W) reduces system efficiency by 10%. [Table 19](#) provides a guideline to ensure adequate liquid flow rate with the least possible number of pumps. It should be used in conjunction with [Table 18](#) and applies to loops with 0 to 15% propylene glycol solutions (by volume; note that caution and additional treatment may be needed for solutions lower than 20%, because of risks of corrosion and biological growth below this level). This solution has the reputation of being the most difficult of the commonly used solutions to pump when cold. However, it is no more difficult to pump than ethyl alcohol, and pumping penalties can be mitigated by adding only the required amount of antifreeze. Shorter loops may require higher levels of antifreeze solutions. See the section on Antifreeze Requirements for more details. Any exposed piping above the frost line must be insulated with closed-cell insulation with ultraviolet (UV) protection (paint or wrap).

Example 5. Design the vertical ground coupling grid and the pumping loop for a 4 ton (48,000 Btu/h) residential heat pump system. The home is located in Nashville, Tennessee, and the header pipes can be brought into the equipment room where the unit will be located. The driller can bore 4.5 in. holes to a depth of 175 ft in the light limestone and clay at the site. The owner wants the drilling site to be located 75 ft from the house. Thermally enhanced grout with thermal conductivity of 0.85 Btu/h · ft · °F is used to fill the annular region between the U-tubes and borehole walls.

Solution: The soil temperature is estimated to be 59°F in Nashville. [Table 16](#) suggests bore lengths of 170 ft/ton for 3/4 in. U-bends, 160 ft/ton for 1 in., and 150 ft/ton for 1 1/4 in. (bores are deep, greater than 100 ft). However, 1 1/4 in. U-bends are very difficult to install in a 4.5 in. borehole, and are not considered. Therefore, either 680 ft (170 ft/ton × 4 tons) of 3/4 in. U-bend coupling or 640 ft (160 × 4 tons) of 1 in. coupling is required. The latter is used in this example. Also, [Table 18](#) is based on a soil conductivity of 1.2 Btu/h · ft · °F, which is an approximate

average between limestone and clay, and a bore fill (or grout) conductivity of 0.85 Btu/h · ft · °F. If the ground conductivity is higher (i.e., more limestone than clay), the loops should be reduced as noted in [Table 16](#); if lower, the loops should be lengthened. Loop lengths also must be lengthened if the bore fill (or grout) conductivity is lower than 0.85 Btu/h · ft · °F, as noted in [Table 18](#).

Layout is dictated by drilling conditions. The total length of 640 ft requires four bores, because the driller can only drill to 175 ft. This can be accomplished with four 160 to 165 ft holes. [Table 18](#) suggests between three and five parallel circuits for the grid. Three and five circuits do not divide evenly into the four U-bends. Therefore, four circuits (one per U-bend) should be used in an arrangement similar to [Figure 29](#).

Table 19 Recommended Residential GCHP Piping Arrangements and Pumps

Coil Type*	Nominal Heat Pump Capacity, tons				
	2	3	4	5	6
	Required Flow Rate, gpm				
	5 to 6	7 1/2 to 9	10 to 12	12 to 15	15 to 18
	Number of Parallel Loops				
Spiral (10 pt.)	3 to 4	4 to 6	6 to 9	8 to 10	8 to 10
6-Pipe	3 to 4	4 to 6	6 to 9	8 to 10	8 to 10
4-Pipe	2 to 3	4 to 6	5 to 8	6 to 9	6 to 10
2-Pipe	2 to 4	3 to 5	4 to 6	5 to 8	6 to 10
Vertical 3/4 in. pipe	2 to 3	3 to 5	4 to 6	5 to 8	6 to 10
1 in. pipe	2 to 3	2 to 4	3 to 5	4 to 6	4 to 6
1 1/4 in. pipe	1 to 2	1 to 2	2 to 3	2 to 3	2 to 4
Trench Length					
Header Diameter (HDPE Pipe), in.					
Less than 100 ft	1 1/4	1 1/4	1 1/2	1 1/2 to 2	1 1/2 to 2
100 to 200 ft	1 1/4	1 1/2	1 1/2	2	2
Size (No.) of Pumps Required					
	1/12 hp (1)	1/6 hp (1)	1/12 hp (2)	1/6 hp (2)	1/6 hp (2)

Source: Kavanaugh and Calvert (1995).

* Based on DR11 HDPE pipe.

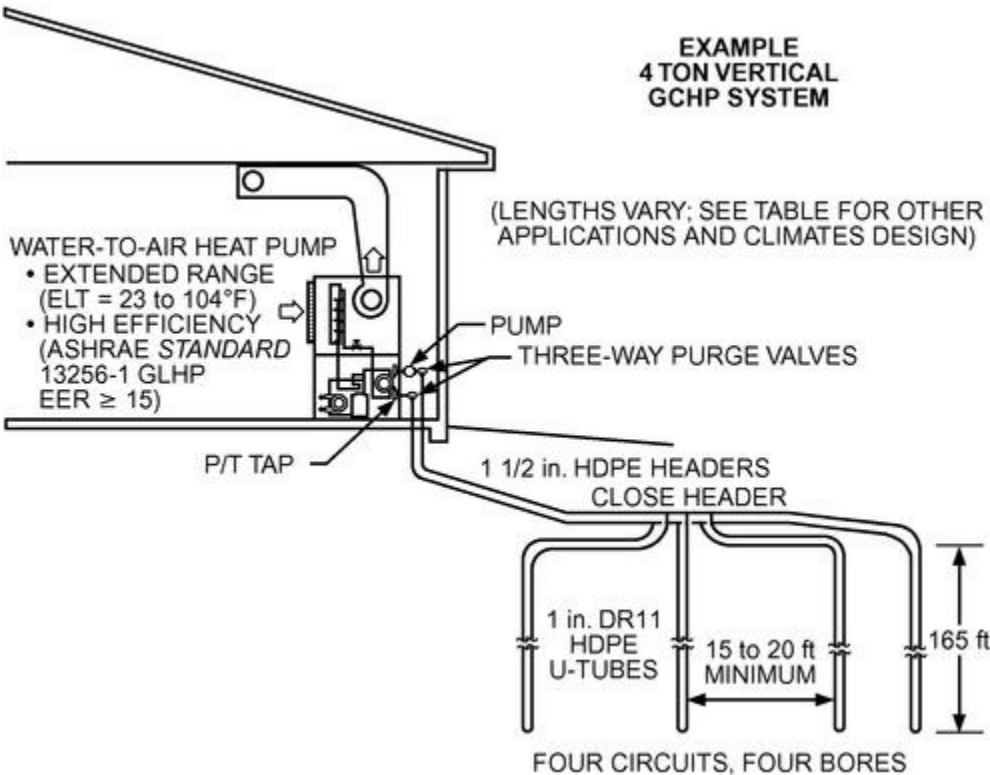


Figure 29. Residential Design Example

Central Plant Systems

Central plant GCHP systems use central water-to-water equipment (e.g., a water-cooled chiller) to move thermal energy between a source loop (the ground coupled heat exchanger), a chilled-water loop, and a hot-water loop. Here, the term *central plant* implies the mechanical equipment is in one centralized location; it does not imply a campus is served, and single buildings are a common application. The central plant source loop is commonly a vertical closed-loop heat exchanger, but any combination of GCHP types may be used. The chilled- and hot-water loops may serve any HVAC distribution equipment (fan coils or radiant systems are efficient options) as long as design hot-water supply temperatures do not exceed heat pump temperature ranges (typically 130°F). Best practice in these applications is largely theoretical or anecdotal; research is needed on the efficacy of the various central plant GCHP applications and designers considering central plant GCHPs must be cautious.

Compared to traditional unitary (water-to-air) GCHP systems, central plant systems offer the potential advantages of (1) incorporating direct heat recovery from hot-water loads to chilled-water loads, (2) sometimes allowing waterside economizing, (3) centralizing equipment maintenance, and (4) expanding retrofit opportunities. However, when such plants are connected to conventional distribution such as variable air volume (VAV), the loss of zone level heating and cooling results in large fan and reheat energy penalties. Pumping power is often higher than equivalent unitary heat pump systems. Central plants also have significantly more complex design, controls, commissioning, operational training, and maintenance. Most existing tools and methods for sizing ground heat exchangers and evaluating energy performance do not accurately represent central plant GCHP operation.

As with unitary systems, central plant systems may take many forms. [Figure 30](#) illustrates the basic building blocks of a central plant GCHP system. Any number of heat recovery chillers and heating or cooling heat pumps may be installed. Either or both of the load-side loops may be connected to the source loop to achieve direct heat transfer (water economizing). The loops may be separated by control valves or heat exchangers (as shown in [Figure 30](#)), but loops should be linked only by control valves where fluid type and pressure are compatible.

Most central plant heat pump equipment is designed for use in one of three basic strategies:

- **Parallel water-to-water heat pumps:** dedicated heat pumps provide hot and/or chilled water in parallel. This type is often used in conjunction with unitary heat pumps or in simple applications where there is little opportunity for heat recovery or water economizing.
- **Packaged modular heat pumps:** multiple units are connected by control valves to any of the loops (source, hot water, or chilled water) such that each individual heat pump may serve for heating, cooling, or heat recovery.
- **Heat pump chiller:** one or more large heat recovery chillers, designed to operate with a large lift, operate to maintain both hot and chilled water at required temperatures. No dedicated cooling heat pump or heating heat pump is provided. The heat exchangers (or direct valve connections between loops) provide the means to achieve heat rejection or heat absorption with the ground heat exchanger.

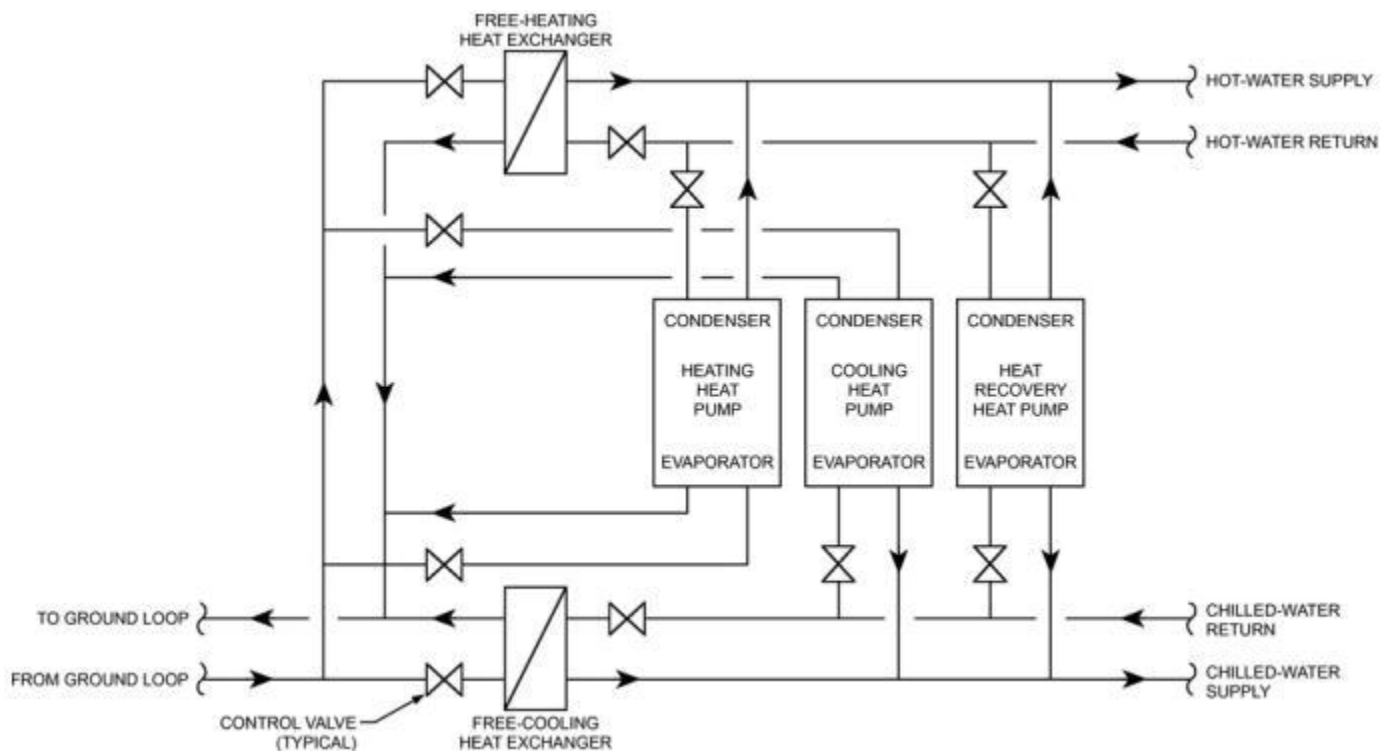


Figure 30. Central Plant GCHP System

To provide stable temperature control and part load operation, central plant GCHP systems can benefit from additional thermal capacitance (buffer tanks) on the load side loops. Any loop flow control methodology may be used; however, variable flow should be used to minimize pumping energy penalties where equipment allows. Consider using water as the working fluid wherever possible, because of the energy and capital cost associated with antifreeze solutions.

Central plant designers have the option to connect hybrid heating and cooling equipment to the load side of the heat pumps instead of the source/ground side, which adds redundancy and reduces pump and compressor energy at the expense of increased controls complexity. Hybrid air-cooled chillers may serve the chilled-water loop directly, whereas evaporative cooling equipment is best placed on the source loop in series or in parallel with the ground heat exchanger. Connecting a hybrid boiler to the hot-water loop reduces thermal stress for boilers designed to operate at higher temperatures and provides direct emergency heat; connecting a hybrid boiler to the source loop reduces control interconnection/complexity and reduces heat pump faults caused by low entering water temperatures.

Antifreeze Requirements

Closed-loop horizontal and surface water heat exchanger systems often require antifreeze in the circulating water in locations with significant heating seasons. Antifreeze may not be needed in a comparable vertical borehole heat exchanger, because the deep ground temperature is essentially constant. At a depth of 6 ft, a typical value for horizontal heat exchangers, ground temperature varies by approximately $\pm 10^{\circ}\text{F}$. Even if the mean ground temperature is 60°F in late winter, ground temperature at a 6 ft depth drops to 50°F . The heat extraction process lowers the temperature even further around the heat exchanger pipes, probably by an additional 10°F or more. Even with good heat transfer to the circulating water, the entering water temperature (leaving the ground heat exchanger) is around 40°F . Lakes that freeze at the surface in the winter approach 39°F at the bottom, yielding nearly the same margin of safety against freezing of the circulating fluid. An additional 10°F temperature difference is usually needed in the heat pump's refrigerant-to-water heat exchanger to transfer heat to the refrigerant. Having a refrigerant-to-water coil surface temperature below the freezing point of water risks growing a layer of ice on the water side of the heat exchanger. In the best case, coil icing restricts and may eventually block the flow of water and cause a shutdown. In the worst case, ice could burst the tubing in the coil and require a major service expense.

Several factors must be considered when selecting an antifreeze for a ground-loop heat exchanger; the most important are (1) effect on system life-cycle cost, (2) corrosivity, (3) leakage, (4) health risks, (5) fire risks, (6) environmental risks from spills or disposal, and (7) risk of future use (acceptability of the antifreeze over the life of the system). A study by Heinonen and Tapscott (1996) evaluating six antifreezes against these seven criteria is summarized in [Table 20](#). No single material satisfies all criteria. Methanol and ethanol have good viscosity characteristics at low temperatures, yielding lower-than-average pumping power requirements. However, in concentrated forms they both pose a significant fire hazard. Methanol is also toxic, eliminating it from consideration in areas that require nontoxic antifreeze to be used. Propylene glycol had no major concerns, with only leakage and pumping-power requirements prompting minor concerns. Potassium acetate, calcium magnesium acetate (CMA), and urea have favorable environmental and safety performance, but are all subject to significant leakage problems, which has limited their use.

Table 20 Suitability of Selected GCHP Antifreeze Solutions

Category	Methanol	Ethanol	Propylene Glycol	Potassium Acetate	CMA	Urea
Life-cycle cost	***	***	**1	**1	**1	***
Corrosion	**2	**3	***	**	**4	*5
Leakage	***	**6	**6	*7	*8	*9
Health risk	*10,11	**10,12	***10	***10	***10	***10
Fire risk	*13	*13	***14	***	***	***
Environment risk	**15	**15	***	**15	**15	***
Future-use risk	*16	**17	***	**18	**19	**19
Category	Notes					
Life-cycle cost	1. Higher-than-average installation and energy costs.					
Corrosion	2. High black iron and cast iron corrosion rates. 3. High black iron, cast iron, copper, and copper alloy corrosion rates. 4. Medium black iron, copper, and copper alloy corrosion rates. 5. Medium black iron, high cast iron, and extremely high copper and copper alloy corrosion rates.					
Leakage	6. Minor leakage observed. 7. Moderate leakage observed. Extensive leakage reported in installed systems. 8. Moderate leakage observed.					

9. Massive leakage observed.

Health risk	10. Protective measures required with use. See Material Safety Data Sheet (MSDS).
	11. Prolonged exposure can cause headaches, nausea, vomiting, dizziness, blindness, liver damage, and death. Use of proper equipment and procedures reduces risk significantly.
	12. Additives make ethanol poisonous for human consumption. See Material Safety Data Sheet (MSDS).
Fire risk	13. Pure fluid only. Little risk when diluted with water in antifreeze.
	14. Very minor potential for pure fluid fire at elevated temperatures.
Environment risk	15. Water pollution.
Future-use risk	16. Toxicity and fire concerns. Prohibited in some locations.
	17. Toxicity, fire, and environmental concerns.
	18. Potential leakage concerns.
	19. Not currently used as GSHP antifreeze solution. May be difficult to obtain approval for use.

Key:
Source: Heinonen and Tapscott (1996).
* Potential problems, caution in use required
** Minor potential for problems
*** Little or no potential for problems

1.4 GROUND-COUPLED HEAT PUMP SYSTEMS USING REFRIGERANT-BASED HEAT TRANSFER FLUIDS (DIRECT EXCHANGE)

Direct-exchange ground-coupled heat pumps (DXGCHPs) circulate refrigerant from the heat pump in sealed copper tubing to directly exchange heat with the ground. In heating mode, the ground loop system functions as an evaporator, absorbing heat from the ground and causing the refrigerant to change phase from liquid to vapor. In cooling mode, the ground loop system functions as a condenser, discharging heat to the ground and changing the refrigerant from vapor to liquid. DXGCHP systems are applied primarily to residential and moderately sized commercial buildings, and are ideal for installation where space is limited because of the smaller ground loop footprint and drilling equipment required. Distribution systems for DXGCHPs use conditioned air delivered through a direct-expansion air handler or cased coil, hydronic heating and cooling, or both. Potable water heating is an option. DXGCHP systems can be applied to various geological formations and typically in locations with ground temperatures of 40 to 80°F. DXGCHPs are generally sized up to 72,000 Btu/h. Multiple systems are specified for higher-capacity applications.

System Design

The DXGCHP system operation is very similar the GCHP shown in [Figure 1](#), except that the refrigerant is piped directly through the ground loop, the buried polyethylene tubing is replaced with copper tubing, and the water circulating pump and refrigerant-to-water heat exchanger are eliminated.

DXGCHP ground heat exchanger systems can be horizontal, diagonal, or vertical ([Figure 31](#)). A typical DXGCHP distribution system illustrating the liquid and vapor manifold arrangement is shown in [Figure 32](#). The ground heat exchangers are comprised of multiple individual ground loops, with the number of ground loops increasing with system capacity. Each loop is typically two copper tubes with a return bend connecting them at one end, and ranges in size from 0.25 to 0.50 in. OD, depending on the specific design. The tube diameter must be small to achieve high velocity, which ensures adequate oil return to the compressor, but large enough so the pressure drop does not significantly lower system performance. In addition, the selected lubricant must maintain a relatively low viscosity down to 5°F for proper oil return during the heating season.

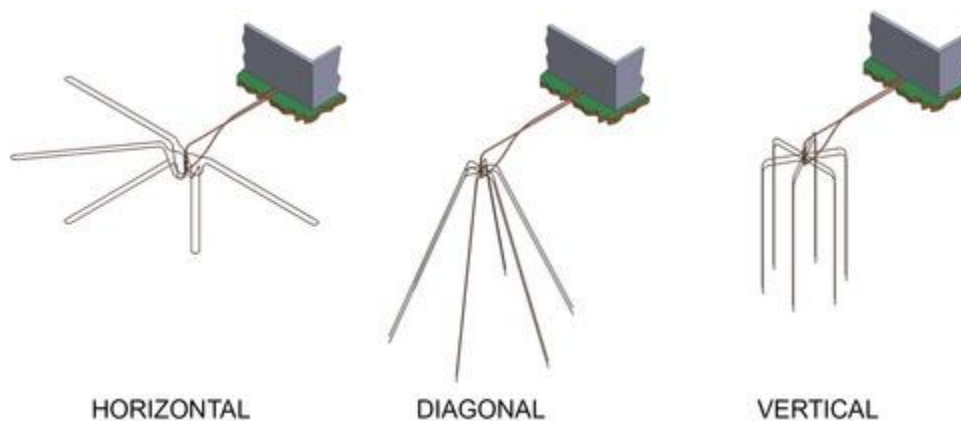


Figure 31. DXGCHP Ground Heat Exchanger Configurations

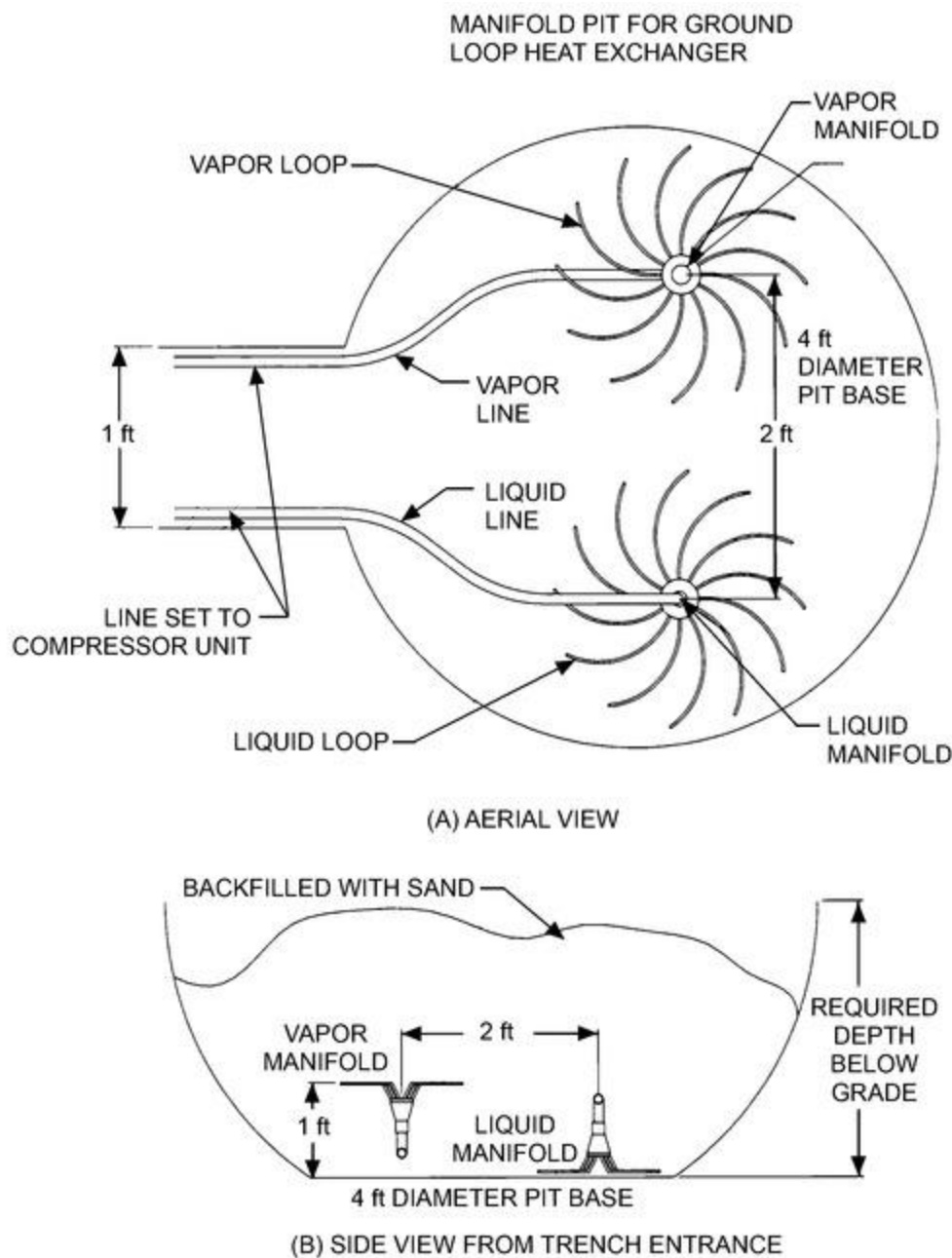


Figure 32. Typical DXGCHP Ground Heat Exchanger Distribution System

The vertical and diagonal ground loop systems in [Figure 31](#) require drilling 3 in. minimum diameter boreholes to accommodate a ground loop and thermal grout. The horizontal earth loop system in [Figure 31](#) can be either a trench, pit or vertical bore. All ground loop systems, including line sets and manifolds, must be at least 5 ft below grade or 18 in. below the local frost line, whichever is deeper, to ensure adequate year-round heat exchange with the surrounding ground.

DXGCHP heat exchanger lengths and configurations are currently selected using DXGCHP manufacturers' performance tables, which are based on empirical laboratory and field test data accrued over the last 45 years. Generalized analytical techniques for the design and application of DXGCHP ground heat exchanger systems are currently being developed. Using a manufacturer's performance tables requires knowledge of the soil temperature (e.g., from [Figure 10](#)), building loads (see ACCA [2008, 2016] and [Chapters 17](#) and [18 of the 2021 ASHRAE Handbook—Fundamentals](#)), and available land area and geology.

Available Land Area and Geology. The available land area, geology at the site, and knowledge and experience of local excavators and drillers influence the selected configuration of the ground heat exchanger. The specifier may make a preliminary selection of the ground loop configuration before sizing the system. In general, if the job site has relatively level land and enough space is available, excavating and installing a horizontal pit or trench ground loop system is economically attractive. There must be adequate space to put the excavated earth. Horizontal ground loops can also be installed by horizontally boring (see [Figure 3](#)).

If vertical ground loops are to be installed, the required drilling will disturb far less surface space than horizontal systems. Vertical systems for capacities up to 72,000 Btu/h can be installed within a surface footprint of only 8 ft diameter, which makes this configuration well suited for installation where ground surface area is limited. In addition, borehole diameters of 3 in. for each ground loop allow use of small, maneuverable drill rigs.

Site geology, including soil composition and associated thermal conductivity, is a major factor in selecting the appropriate ground loop configuration with sufficient ground loop surface area for effective heat transfer. DXGCHP manufacturers provide tables for selecting ground heat exchanger configurations and surface areas that accommodate various soil thermal conductivities. Sufficient soil samples should be taken from the ground loop field and analyzed for pH and potentially corrosive elements. If corrosive elements are present beyond the manufacturer's stated threshold levels, an alternative acceptable location for the ground heat exchanger system is required. New advances in copper coating technologies are also viable options in some soil conditions.

Ground Heat Exchanger Corrosion Protection System

DXGCHP ground heat exchangers are typically constructed of copper because of its high thermal conductivity and compatibility with refrigerant system pressures; annealed copper tubing is malleable, making it easy to install. Because it is a noble metal, it is almost impervious to corrosion from soils found worldwide. However, copper is still vulnerable to corrosion in aggressive soils, and must be protected. The simplest way to prevent corrosion is to apply cathodic protection by connecting the copper to another metal, typically zinc, which is more easily corroded. The sacrificial metal corrodes instead of the protected copper. Although this method of corrosion protection is low cost and easy to apply, a major drawback is that corrosion protection fades over time as the sacrificial metal deteriorates. The sacrificial metal must be replaced to maintain an adequate level of corrosion protection.

A more effective means of protecting copper in aggressive soils is the **impressed current cathodic protection (ICCP)** system ([Figure 33](#)). This system provides an electronically regulated continuous current from a titanium anode covered with a mixed-metal oxide, to the copper ground loops, resulting in superior long-term ground heat exchanger system durability and performance. DXGCHP manufacturers design ICCP systems as integral, matched components of specific DXGCHP systems to ensure superior corrosion protection for the ground heat exchanger, thus eliminating the need for the specifier to attempt designing the ICCP system separately. Part of the specification process includes contacting local utilities to learn whether there are existing underground impressed current protection systems, and the effective range of those systems, the proximity of which could potentially interfere with the DXGCHP ground heat exchanger protection system.

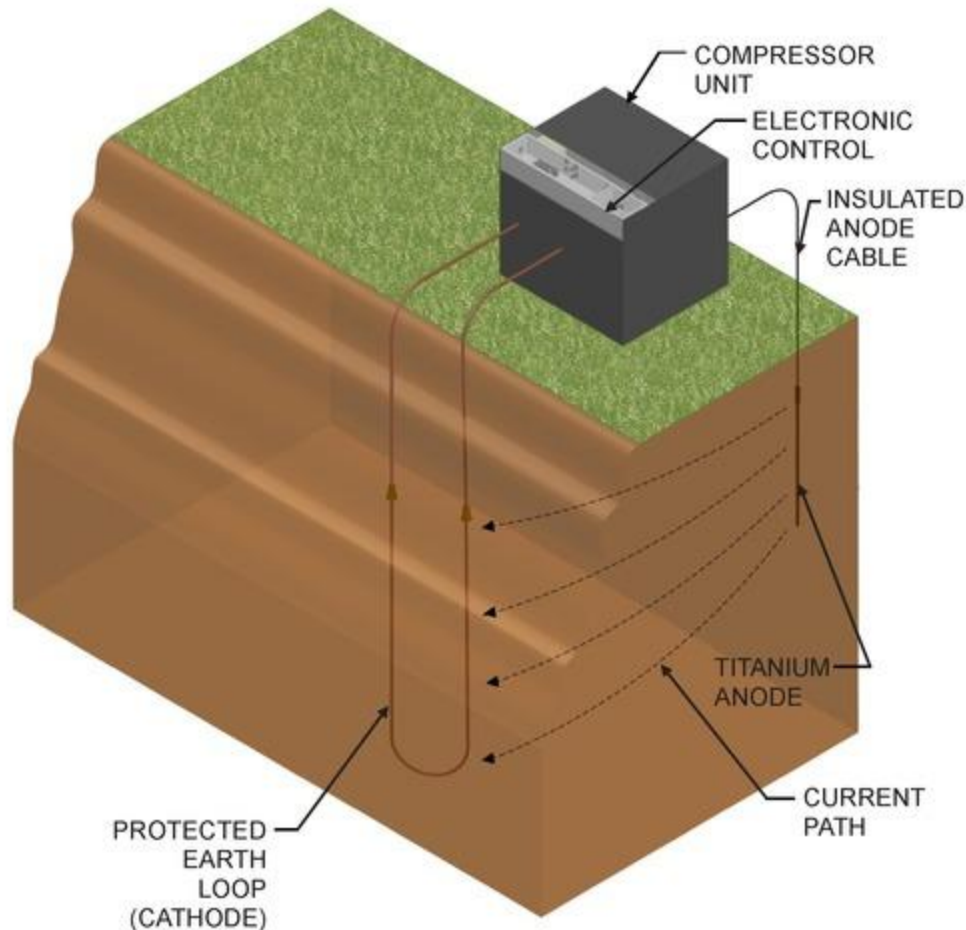


Figure 33. Typical Impressed Current Protection System

1.5 OPEN-LOOP GROUNDWATER HEAT PUMP SYSTEM COMPONENTS

A groundwater heat pump system (GWHP) removes groundwater from a well and delivers it to a heat pump (or an intermediate heat exchanger) where heat is absorbed from or rejected to the water. Both unitary and central plant designs are used. In the unitary type, a large number of small water-to-air heat pumps are distributed throughout the building. The central plant design uses one or a small number of large-capacity chillers supplying hot and chilled water to a two- or four-pipe distribution system. The unitary approach is more common and tends to be more energy efficient.

Direct systems (in which groundwater is pumped directly to the heat pump without an intermediate heat exchanger) are not recommended except on the very smallest installations. Although some systems of this design have been successful, many have had serious difficulty even with groundwater of apparently benign chemistry. Thus, prudent design for commercial/industrial-scale projects isolates groundwater from the building system with a heat exchanger. The increased capital cost of installing the heat exchanger is only a small percentage of the total cost and, in view of these systems' greatly reduced maintenance requirements, is quickly recovered.

Past GWHP systems sometimes used surface disposal (to rivers, lakes, drainage ditches, etc.) of the groundwater. Current standards use reinjection instead, because it eliminates the potential for negative effects on the aquifer water level over time and preserves the positive environmental character associated with GSHP systems.

Regardless of the type of equipment installed in the building, the specific components for handling groundwater are similar. Primary items include (1) wells (supply and injection), (2) well pump and controls, and (3) groundwater heat exchanger. Some specifics of these items are discussed in the Geothermal Energy section of this chapter. In addition to those comments, the following considerations apply specifically to unitary GWHP systems using a groundwater isolation heat exchanger.

Water Wells

This section includes information on water wells that is generally common to both direct-use and groundwater heat pump (GWHP) systems. Water well open-loop systems and standing column well best practices are covered in ANSI/CSA/IGSHPA *Standard* C448-16. Local well regulations must be checked and adhered to. Many jurisdictions have laws that protect their aquifers, especially if they depend on groundwater for municipal use.

An **aquifer** is a geologic unit that can yield groundwater to a well in sufficient quantities to be of practical use (UOP 1975). Aquifers can exist in areas where water is present in conjunction with pore spaces in the subsurface materials sufficient to allow the water to move laterally.

In many projects, construction of the well (or wells) is handled through a separate contract between the owner and the driller or a hydrology/hydrogeology consultant. As a result, the engineer is not responsible for its design. However, because design of the building system depends on the wells' performance, it is critical that the engineer be familiar with water well terminology and test data. The most important consideration with regard to the wells is that they be completed and tested (for flow volume and water quality) before final system design, in much the same way that ground thermal properties testing precedes GCHP system design.

[Figure 34](#) illustrates some important well terms. Several references (Anderson 1984; Campbell and Lehr 1973; EPA 1975; Roscoe Moss Company 1985) cover well drilling and well construction in detail.

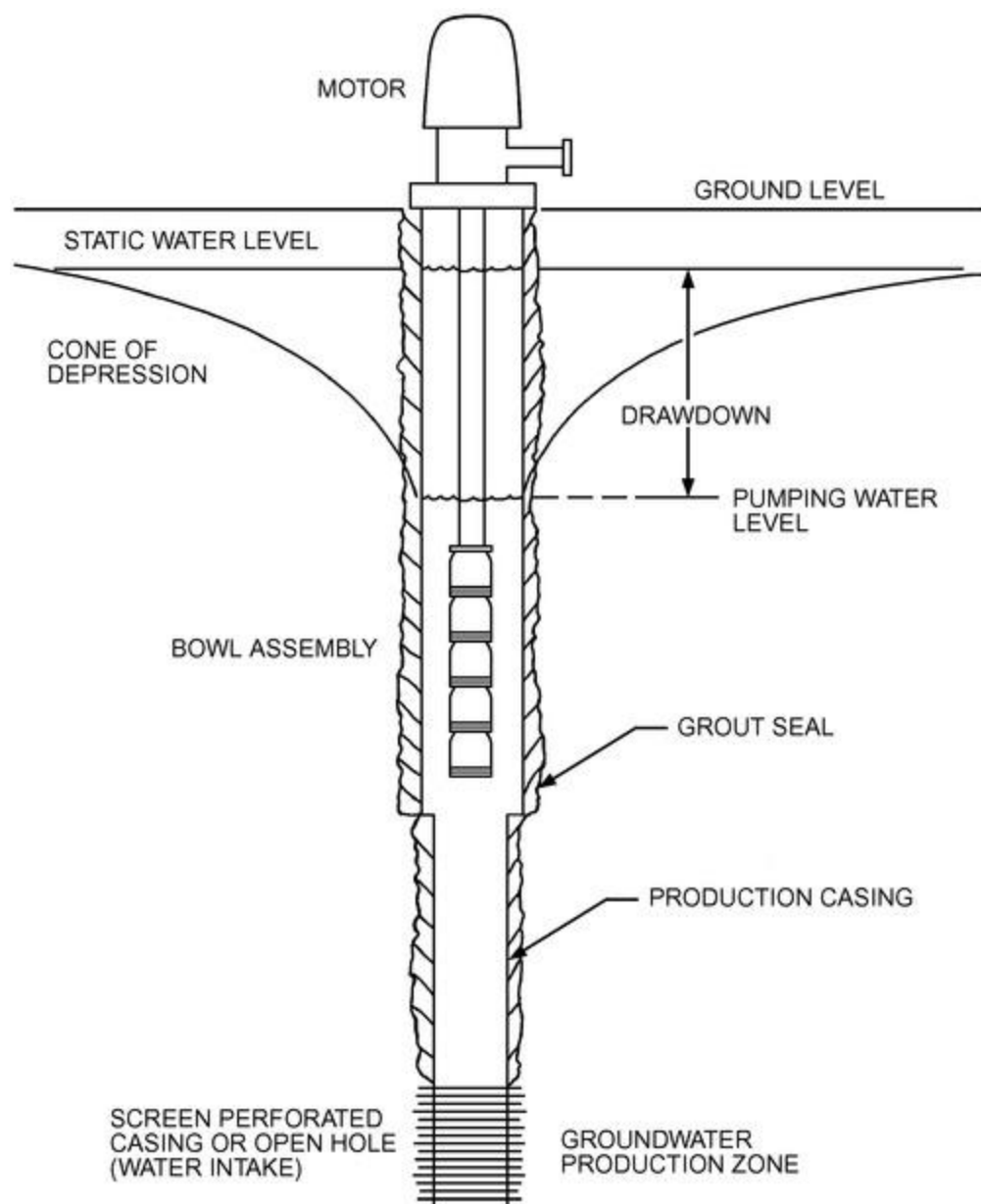


Figure 34. Water Well Terminology

Static water level (SWL) is the level that exists under static (non-pumping) conditions. In some cases, this level is much closer to the surface than that at which the driller encounters water during drilling. **Pumping water level (PWL)** is the level that exists under specific pumping conditions. Generally, this level is different for different pumping rates (higher pumping rates mean lower pumping levels). The difference between the SWL and the PWL is the **drawdown**. The well's **specific capacity** is frequently quoted in gpm per foot of drawdown. For example, for a well with a static level of 50 ft that produces 150 gpm at a pumping level of 95 ft, drawdown = 95 – 50 = 45 ft; specific capacity = 150/45 = 3.33 gpm per foot.

Table 21 Nominal Well Surface Casing Sizes

Pump Bowl Diameter, in.	Suggested Casing Size, in.	Minimum Casing Size, in.	Submersible Flow Range 3450 rpm, gpm	Lineshaft Flow Range, 1750 rpm, gpm
-------------------------	----------------------------	--------------------------	--------------------------------------	-------------------------------------

4	6	5	<80	<50
6	10	8	80 to 350	50 to 175
7	12	10	250 to 600	150 to 275
8	12	10	360 to 800	250 to 500
9	14	12	475 to 850	275 to 550
10	14	12	500 to 1000	
12	16	14	900 to 1300	

Water entrance velocity (through the screen or perforated casing) can be an important design consideration. Velocity should be limited to a maximum of 0.1 fps (0.05 ft/s for injection wells) to avoid incrustation of the entrance openings. The **pump bowl assembly** (impeller housings and impellers) is always placed sufficiently below the expected pumping level to prevent cavitation at the peak production rate. For the previous example, this pump should be placed at least 115 ft below the casing top (pump setting depth = 115 ft) to allow for adequate submergence at peak flow. Along with any expected annual aquifer water level fluctuations, the specific **NPSH pressure** required for a pump varies with each application and should be carefully considered in selecting the setting depth along with any expected annual aquifer water level fluctuations.

For the well pump, **total pump head** is composed of four primary components: lift, column friction, surface requirements, and injection head (pressure). **Lift** is the vertical distance that water must be pumped to reach the surface. In the example, lift is 95 ft. The additional 20 ft of submergence imposes no static pump head (pressure).

Column friction, the friction loss in the pump column between the bowl assembly and the surface, is calculated from pump manufacturer data in a similar manner to other pipe friction calculations (see [Chapter 22 of the 2021 ASHRAE Handbook—Fundamentals](#)). **Surface pressure requirements** account for friction losses through piping, heat exchangers, and controls, and in many applications are between 25 and 35 ft. **Injection pressure requirements** are a function of well design, aquifer conditions, and water quality. In theory, an injection well penetrating the same aquifer as the production well experiences a water level rise (assuming equal flows) that mirrors the drawdown in the production well. Using the earlier example, an injection well with a 50 ft static level would experience a water level rise of 45 ft, resulting in a surface injection pressure of $45 - 50 = -5$ ft (i.e., a water level that remains 5 ft below the ground surface). Thus, no additional pump head is required for injection in the example.

In practice, injection pressure requirements usually exceed the theoretical value. With good (non-scaling) water quality, careful drilling, and little sand production, injection pressure should be near the theoretical value. For poor water quality, high sand production, or poor well construction, injection pressure may be 10 to 40% higher.

The well casing diameter depends on the diameter of the pump (bowl assembly) necessary to produce the required flow rate. [Table 21](#) presents nominal casing sizes for a range of water flow rates.

In addition to the production well, most systems should include an injection well to dispose of the fluid after it has passed through the system. Injection stabilizes the aquifer from which the fluid is withdrawn by reducing or eliminating long-term drawdowns and helps to ensure long-term productivity. Construction of injection wells differs from production wells primarily in the recommended screen velocity (0.05 ft/s, or 1/2 that of production wells) and well sealing design. Injection wells, particularly those likely to be subject to positive injection pressure, should be fully cased and sealed from the top of the injection zone to the surface.

It is commonly thought that wells, particularly injection wells, often fail, but failure is more often attributable to the designer than to the well itself. The most common factors in reduced water well (production and injection) performance are incrustation and biofouling of screens, formation plugging with fines, sand pumping, casing/screen collapse, and pump problems (Driscoll 1986). To a large extent, incrustation and biofouling can be reduced by minimizing drawdown through careful well design and the avoidance of excessive groundwater flows. Material selection appropriate for the water chemistry and avoidance of substandard casing and screen products can reduce or eliminate failures in these components. Sand production should be limited by screen, gravel pack, and development practices, or removed by strainers before injection. With such good practices, maintenance intervals can be reduced to approximately 10 to 15 years in favorable conditions, and 5 to 8 years in unfavorable settings. One key to successful water well operations is effective monitoring: regular testing of well yield, drawdown, specific capacity, and sand production, coupled with periodic review of trends in these parameters.

Flow Testing

When possible, well testing should be completed before mechanical design. Only with actual flow test data and water chemical analysis information can accurate design proceed. It is highly recommended that a hydrogeologist be engaged in the testing process.

Flow testing can be divided into three different types of tests: rig, short-term, and long-term (Stiger et al. 1989). Rig tests are generally very short and are accomplished while the drilling rig is on site. The primary purpose of this test is to purge the well of remaining drilling fluids and cuttings and to get a preliminary indication of yield. The length of the test is generally governed by the time required for the water to run clean. The rate is determined by the available pumping equipment. Frequently, the well is blown (pumped with the drilling rig's air compressor). As a result, limited information about the well's production characteristics is available from a rig test. If the well is air lifted, it may not be

useful to collect water samples for chemical analysis because certain chemical constituents may be oxidized by the compressed air.

Properly conducted, short-term, single-well tests lasting 4 to 24 h yield information about well flow rate, temperature, drawdown, and recovery. These tests are used most frequently for direct-use and GWHP applications. The test is generally run with a temporary electric submersible pump or lineshaft turbine pump driven by an internal combustion engine and are often performed by a well pump contractor.

A step test ([Table 22](#)), the most common type, involves at least three production rates, the largest being equal to the design flow rate for the system served. The three points are the minimum required to determine a productivity curve for the well that relates production to drawdown (Stiger et al. 1989). The key parameters monitored during these tests are well water level and water flow. Water level and pumping rate should be stabilized at each point before flow is increased. In many cases, water level is monitored with a bubbler or an electric sounder, and flow is measured using an orifice meter. More sophisticated instrumentation (e.g., pressure transducers for water level, magnetic flow meters, data loggers) can also be used. Short-term testing is generally used for small projects and provides information on yield, drawdown, and specific capacity.

Table 22 Example Well Flow Test Results SWL 68 ft

Time Since Pump Start, min	Flow, gpm	Water Level, ft	Comments
5	125	78.3	clear
10	127	79.5	clear
15	125	81.1	clear
20	125	82	clear
25	125	83.1	clear
30	126	83.4	clear
40	125	83.3	clear
50	125	83.3	clear
60	125	83.3	clear
65	200	90.6	cloudy
70	200	96.8	clear
75	200	98.9	clear
80	200	99.5	clear
85	201	99.7	clear
90	200	100.1	clear
100	200	100.2	clear
120	200	100.2	clear
125	295	130.6	cloudy
130	300	135.8	cloudy
135	301	140.3	clear
140	300	141.5	clear
160	300	142.3	clear
170	300	145.7	clear
180	300	145.7	clear

Test results should reflect stable flow rates, and individual flow steps are extended until water level readings stabilize. In many cases, brief intervals of turbidity may occur at flow changes, but extensive periods of turbidity indicate instability in the near-well formation.

Long-term tests of up to 30 days provide information on the reservoir. Normally, these tests involve monitoring nearby wells to evaluate interference effects. The data are useful in calculating transmissivity and storage coefficient, reservoir boundaries, and recharge areas (Stiger et al. 1989) but are rarely used for direct-use and GWHP systems.

It is also important to collect background information before the test, and water level recovery data after pumping has ceased. Recovery data in particular can be used to evaluate skin effect, which is a type of well flow resistance caused by residual drilling fluids, insufficient screen or slotted liner area, or improper filter pack.

Testing for Recharge Wells

Most large ground water systems require the construction of a return well system to return water extracted from an aquifer to the same aquifer at a distance from the supply well(s). In addition to flow testing as described above, recharge wells must be tested to determine the return flow capacity and recharge pressure. The system may require more than one return well per supply well, but this is highly dependent on the geology of the site. The recharge test is usually carried out using water pumped from the supply well. It may be conducted as part of the 24-h flow test on the supply well. Data from the testing will be necessary to size the supply pump(s). The recharge pressure will be added to the supply pump(s) head loss calculations.

Groundwater Quality

The importance of groundwater quality depends on the system design. Systems using isolation heat exchangers commonly encounter no water quality issues (other than iron bacteria) that would prevent a GWHP system from operating under reasonable maintenance levels.

Table 23 Water Chemistry Constituents

Quality	Comment
pH	Typical range: 6.5 to 9.0. Lower values typically associated with higher rates of general corrosion in ferrous and copper alloys; higher values associated with scaling.
TDS	Total dissolved solids: gross indicator of quantity of dissolved constituents. Higher levels associated with increased corrosion and/or scaling; used in calculation of scale index.
Fe	Iron: use care to prevent exposure to air; problems possible at >0.5 ppm.
Total M alkalinity	Ability of water to buffer acid; strongly linked to scale and used to calculate scaling index. Usually expressed as ppm CaCO ₃ .
Ca	Calcium ion: linked to scaling of water and used to calculate scaling index. Expressed in ppm Ca × 0.5 = ppm as CaCO ₃ .
CO ₃ /HCO ₃	Carbonate/bicarbonate: varies in concentration with pH.
Hardness	Linked to scaling and used to calculate scale index; at >100 ppm, scaling can occur. Expressed in ppm/17.1 = hardness in gr/gal.
Cl	Chloride: accelerates corrosion of carbon and stainless steels; may be elevated in coastal areas.
Mn	Manganese: causes black scale; possible deposits at >0.2 ppm.
O ₂	Oxygen: dissolved gas; accelerates corrosion; promotes other reactions; test in field.
H ₂ S	Hydrogen sulphide: dissolved gas; rotten egg odor >0.5 ppm; attacks copper alloys; test in field.
CO ₂	Carbon dioxide: dissolved gas, often present at pH < 7.5, test in field. GW pressurization keeps CO ₂ in solution.
Stability index	
(Ryznar index)	Originally developed to predict corrosion but used in GWHP for scaling prediction; calculated from temperature, Ca, TDS, alkalinity, and hardness. Must use temperature reflective of application: 85°F for systems with plate heat exchanger, 150°F for nonisolated systems.
Saturation index	
(Langlier index)	Similar to stability index. Originally developed to predict corrosion but used in GWHP for scaling prediction; calculated from temperature, Ca, TDS, alkalinity, and hardness. Must use temperature reflective of application: 85°F for systems with plate heat exchanger, 150°F for nonisolated systems
BART	Bacteriological activity reaction test: broad indicator of various bacteria. Most common tests are for iron-reducing (IRB), slime-forming (SLYM), and sulfate-reducing (SRB) bacteria.

Source: Rafferty (2008).

For systems that use groundwater directly in heat pump units (e.g., standing-column systems and small residential GWHP systems), several issues are of concern. The primary water quality problem in the United States is scaling, usually of calcium carbonate (lime). Because this type of scaling is partially temperature driven, the temperature of surfaces that groundwater contacts determines the extent to which scaling will occur. In these systems, peak temperatures in the refrigerant-to-water exchanger in cooling mode are likely to be over 160°F. For the same system using an isolation plate heat exchanger, the groundwater is unlikely to encounter temperatures over 90°F. Using the plate heat exchanger reduces the propensity for scaling and limits any scale that does occur to a single heat exchanger. Rafferty (2000b) provides information on water scaling potential on a state-by-state basis.

Hydrogen sulfide can destroy the oxide layer on copper, copper-nickel alloys, and stainless steels, and make these metals vulnerable to acidic corrosion. Titanium heat exchangers are recommended for hydrogen-sulfide-bearing waters.

Table 24 Controller Range Values for Dual Set-Point Well Pump Control*

	Building Loop Thermal Mass in Gallons per Ton of Peak Block Cooling Load						
	2	4	6	8	10	12	14
Cooling range, °F	31	16	11	8	6	5	4
Heating range, °F	18	9	6	4	3	3	3

Source: Rafferty (2000c).

* Table values for pumps > 5 hp. For pumps < 5 hp, three-phase range values may be reduced by 50%.

Excessive iron, particularly ferrous iron, in the water can result in coating of heat transfer surfaces if the water is exposed to air (allowing the iron to oxidize to the ferric state, a form with much lower solubility in water). Periodically removing this iron from the plates of a single heat exchanger is much less labor intensive than removing it from tens or hundreds of individual heat pump heat exchangers. [Table 23](#) summarizes the minimum parameters that should be evaluated for a GWHP application.

Particulate matter (e.g., sand) in the groundwater stream, although usually not a problem in the mechanical system, can effectively plug injection wells. Sand production should be addressed in construction of the production well (screen/gravel pack/development). If it must be dealt with on the surface, a screen or strainer is preferable to a centrifugal separator, which can be ineffective at start-up and shutdown and can experience variable flow (Kavanaugh and Rafferty 2014). Perforation size selection is critical to a strainer's effectiveness, and should be based on 90 to 100% removal of the particulate material. Particle size information can be based on the sieve analysis results of drill cuttings (used to size the well screen) or of a sample taken during well flow testing. In applications with very fine sand, multiple strainers in parallel may be necessary to control pressure drop (Rafferty 2008).

Well Pumps

Submersible pumps have not performed well in higher-temperature, direct-use projects. However, the submersible pump is a cost-effective option with normal groundwater temperatures, as encountered in heat pump applications. The low temperature eliminates the need to specify an industrial design for the motor/protector, thereby greatly reducing the first cost relative to direct use. Caution should still be used for wells that are expected to produce moderate amounts of sand. The high speed (nominal 3600 rpm) of most submersible pumps makes them susceptible to erosion damage. Applications with sand/particles greater than 30 to 40 mesh should specify sand fighter submersible pump configurations.

Small groundwater systems have frequently been identified with excessive well pump energy consumption. The reasons for excessive pump energy consumption (high water flow rate, coupling to the domestic pressure tank, and low efficiency of small submersible pumps) are generally not present in large, commercial groundwater systems. In large systems, the groundwater flow per unit capacity is frequently less than half that of residential systems. Pressure at the wellhead is not the 30 to 50 psi typical of domestic systems, but is rather a function only of pressure losses through the groundwater loop. Finally, large well pumps have efficiencies of up to 83% compared to the 35 to 40% range for small submersible pumps.

In GWHP system design, the control method for the well pump determines the extent to which the optimum relationship between well pump power and heat pump power is preserved at off-peak conditions. There are several ways the pump can be controlled. Multiple pumps can be staged to meet system loads, either with multiple wells or with multiple pumps installed in a single well. A dual set-point control similar to that used in boiler/tower systems energizes the well pump above a given temperature in cooling mode and below a given temperature in heating mode. Between those temperatures, the building loop floats without the addition of groundwater. To control well pump cycling, it is necessary to establish a temperature range (difference between pump-on and pump-off temperatures) over which the pump operates in both the heating and cooling modes. The size of this range is primarily a function of the building loop water volume in terms of gallons per peak per ton of peak block system load (Rafferty 2000c). [Table 24](#) summarizes these data. In the example in [Table 25](#), the optimum system building loop return temperature (at peak system EER) is 80.6°F. If this system had a water volume of 8 gal/ton, from [Table 22](#), a range of 8°F in cooling mode would be required. This range would result in a well pump start temperature of $80.6 + (8/2) = 84.6^\circ\text{F}$ and a well pump stop temperature of $80.6 - (8/2) = 76.6^\circ\text{F}$. A similar calculation can be made for heating mode. From [Table 24](#), for systems with very low thermal mass, the dual set-point method of control becomes impractical because of the very large temperature range required. For these applications, an alternative method of control (variable speed, staging, etc.) is required.

Table 25 Example GWHP System* Design Data

Heat Pump EWT, °F	Heat Pump LWT, °F	Heat Pump EER	Ground-water LWT, °F	Ground-water Flow, gpm	Well Pump Head, ft	Well Pump kW	Loop Pump kW	System EER
61.0	72.4	17.6	68.4	289	256	23.7	4.8	11.8
63.0	74.5	17.3	70.5	233	229	17.5	4.8	12.5
65.0	76.5	16.9	72.5	196	210	13.7	4.8	12.9

67.0	78.6	16.5	74.6	169	197	11.4	4.8	13.0
69.0	80.6	16.1	76.6	149	186	9.7	4.8	13.1
71.0	82.7	15.7	78.7	133	179	8.5	4.8	13.0
73.0	84.7	15.3	80.7	120	172	7.5	4.8	12.9
75.0	86.7	15.1	82.7	110	167	6.7	4.8	12.9
77.0	88.8	14.9	84.8	101	163	6.0	4.8	12.8
79.0	90.8	14.6	86.8	94	159	5.5	4.8	12.6
81.0	92.3	14.2	88.9	88	156	5.1	4.8	12.4
83.0	94.9	13.4	90.9	82	153	4.7	4.8	12.2

* Block cooling load 85 tons, 60°F groundwater, 75 ft well static water level, 2 gpm/ft specific capacity, 37 ft surface head losses, 4°F heat exchanger approach, 213 gpm building loop flow at 65 ft head.

Well pumps may also be controlled using a variable-speed drive, which responds to building loop return temperature by varying groundwater flow to the exchanger to maintain the cooling or heating mode set point. Submersible-motor variable-speed applications are somewhat different than surface motor applications. Most manufacturers limit speed reduction to 50%, and other issues such as minimum water velocity for motor cooling, switching frequency, reactor requirement, and motor protection must be addressed. Additional information on VFD applications for submersible motors is available in Rafferty (2008).

Heat Exchangers

Design of a plate-and-frame heat exchanger is largely a trade-off between pressure drop, which influences pumping (operating cost), and overall heat transfer coefficient, which influences surface area (capital cost). In general, exchangers in GWHP systems can be economically selected for approach temperatures (between loop return and groundwater leaving temperatures) as low as 3°F. Most selections involve an approach of between 3 and 7°F and a pressure drop of less than 10 psi on the building loop side. Excessive fouling factors ($>0.0002 \text{ h}\cdot\text{ft}^2\cdot\text{°F/Btu}$) should not be specified when selecting plate heat exchangers, which can be easily disassembled and cleaned.

Heat exchanger cost may be reduced for groundwater applications by using Type 304 stainless steel plates rather than the Type 316 or titanium plates common in direct-use projects. The low temperature and generally low chloride content of heat pump fluids frequently make the less expensive Type 304 material acceptable. Chloride content of the groundwater, particularly in coastal areas, should always be compared to values in [Figure 45](#) to determine plate material acceptability. Exchanger performance should be checked at minimum system flow rates to ensure adequate heat transfer. In some cases, very low design pressure drop selections can encounter inadequate heat transfer at minimum flows.

1.6 OPEN-LOOP GROUNDWATER HEAT PUMP SYSTEM DESIGN

Extraction Well Commercial Systems

This section applies to systems with an extraction well and means to return the water elsewhere, such as reinjection wells or surface disposal. An open-loop system design must balance well pumping power with heat pump performance. As groundwater flow increases through a system, more favorable average temperatures are produced for the heat pumps. Higher groundwater flow rates, to a point, increase system EER or COP: increased well pump power is outweighed by decreased heat pump power requirements (because of the more favorable temperatures). At some point, additional increases in groundwater flow result in a greater increase in well pump power than the resulting decrease in heat pump power. The key strategy in open-loop system design is identifying the point of maximum system performance with respect to heat pump and well pump power requirements. Once this optimum relationship has been established for the design condition, the method of controlling the well pump determines the extent to which the relationship is preserved at off-peak conditions. This optimization process involves evaluating the performance of the heat pumps and well pump(s) over a range of groundwater flows. Key data necessary to make this calculation include well performance (flow and drawdown at various groundwater flows) and heat pump performance versus entering water temperatures at different flow rates. Well information is generally derived from well pump test results. Heat pump performance data are available from the manufacturer.

GWHP systems employ the same type of extended-range unitary heat pumps as GCHP systems. Building loop pumping guidelines (see [Table 8](#)) in the GCHP portion of this chapter also apply to GWHP systems. In large commercial applications, the head loss associated with the isolation heat exchanger in a GWHP system is typically lower than that of an equivalently sized ground heat exchanger in a GCHP system. A guideline for building loop head loss in a GWHP system can be described as follows:

Building loop head loss (ft of water) = 28 + 0.1*d*

where *d* = pipeline distance ft from plate heat exchanger outlet to most distant heat pump unit inlet.

This calculation assumes a maximum head loss of 4 ft/100 ft, fittings at 25% of total head loss, and a heat pump unit head loss of 12 ft. Because of more extensive fittings, retrofits can sometimes exceed this value.

For moderate-efficiency heat pumps (EER of 14.2), efficient loop pump design (7.5 hp/100 tons), and a heat exchanger approach of 3°F, [Figure 35](#) provides curves for two different groundwater temperatures (GWT = 50 and 65°F) and two well pump situations (static water level [SWL] 75 ft/specific capacity 10 gpm/ft and SWL 300 ft/specific capacity 3 gpm/ft). The curves are plotted for constant well pump head, a situation which does not occur in practice. In reality, well pump head rises with flow but at a rate typically less than that in friction head applications.

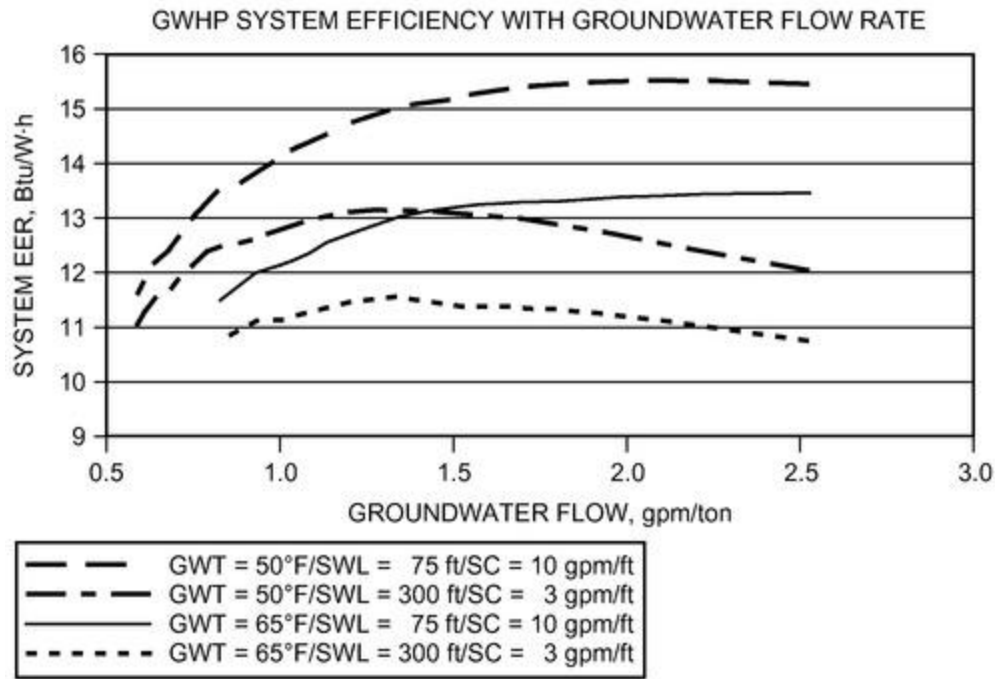


Figure 35. Optimum Groundwater Flow for Maximum System EER SWL is static water level in ft, and SC is specific capacity of well in gpm/ft. (Kavanaugh and Rafferty 2014)

Although the four curves show a clear optimum flow, sometimes operating at a lower groundwater flow reduces well/pump capital cost and the problem of fluid disposal. These considerations are highly project specific, but do afford the designer some latitude in flow selection. Generally, an optimum design results in a groundwater flow rate that is less than the building loop flow rate.

The exception is when groundwater temperatures are less than 47°F or greater than 72°F. In these situations, the groundwater flow requirement is influenced more by avoiding excessive heat pump EWT in the cooling mode (groundwater temperatures above 72°F) and heat pump LWTs that could result in freezing conditions in the heating mode (groundwater temperatures less than 47°F). In the case of low water temperatures, some designers have found it advantageous to use antifreeze in the building loop to slightly broaden the allowable loop temperature range.

[Table 25](#) provides design data for a specific example system.

Central Plant Systems

Central plant systems, in which a conventional or heat recovery central chiller is connected to a four-pipe system, are the oldest type of open-loop design, having first been installed in the late 1940s. Because of the cost and energy requirements of the central plant design, these systems typically do not result in the same level of energy efficiency as unitary GWHP systems.

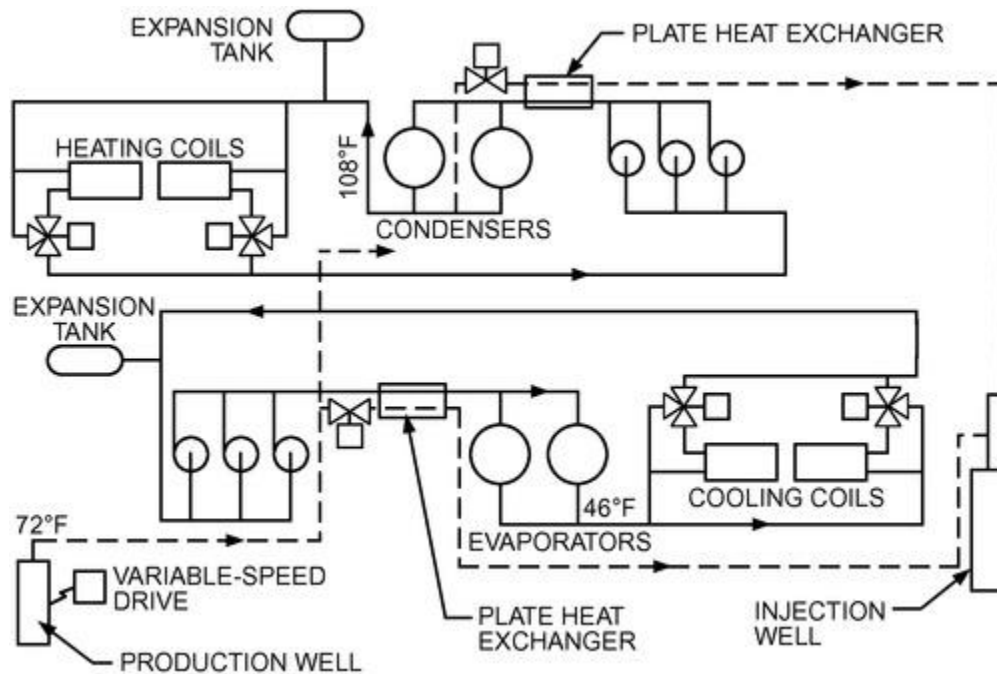


Figure 36. Central Plant Groundwater System

For central plant groundwater systems, two heat exchangers are normally used: one in the chilled-water loop and one in the condenser water loop (Figure 36). The evaporator-loop exchanger provides a heat source for heating-dominated operation and the condenser-loop exchanger provides a heat sink for cooling-dominated operation.

Sizing the **condenser-loop exchanger** is based on providing sufficient capacity to reject the condenser load in the absence of any building heating requirement.

Sizing the **chilled-water-loop exchanger** must consider two loads. The primary criterion is the load required during heating-dominant operation. The exchanger must transfer sufficient heat (when combined with compressor heat) from the groundwater to the chilled-water loop to meet the building's space heating requirement. Depending on the relative groundwater and chilled-water temperatures and on the design temperature rise, exchangers may also provide some free cooling during cooling-dominant operation. If groundwater temperature is lower than that of chilled water returning to the exchanger, some chilled-water load can be met by the exchanger. This mode is most likely available in regions with groundwater temperatures below 60°F.

Central plant chiller controls must also allow for the unique operation with a groundwater source. Controls can be similar to those on a heat recovery chiller with a tower, with one important difference. In a conventional heat recovery chiller, waste heat is available only when there is a building chilled-water (or conditioning) load. In a groundwater system, a heat source (the groundwater) is available year round. To take advantage of this source during the heating season, the chiller must be loaded in response to the heating load instead of the chilled-water load. That is, the control must include a heating-dominant mode and a cooling-dominant mode. Two general designs are available for this:

- Chiller capacity remains controlled by chilled-water (supply or return) temperature, and groundwater flow through the chilled-water exchanger is varied in response to the heating load
- Chiller capacity is controlled by the heating-water (condenser) loop temperature, and groundwater flow through the chilled-water exchanger is controlled by chilled-water temperature

For buildings with a significant heating load, the former may be more attractive, whereas the latter may be appropriate for conventional buildings in moderate to warm climates.

Extraction Well Residential Systems

This section applies to systems with an extraction well and means to return the water elsewhere, such as reinjection wells or surface disposal. Residential groundwater heat pump systems have the same design considerations as commercial groundwater heat pump systems, but differ on three main items: typically they (1) are integrated with a household domestic water system, (2) are single-zone systems, and (3) do not isolate the groundwater from the heat pump unit(s).

Groundwater heat pumps are a prudent choice in residential buildings on well water if the groundwater is of good quality. As such, the heat pump can be integrated into the domestic water system and considered another water-using appliance. Design care must be taken to ensure that the well and pressure tank have adequate capacity to handle the additional flow demand of the heat pump. Well pumps may be of the submersible or jet type, and the design groundwater flow rate should be chosen based on its temperature such that the system COP or EER is maximized. Flow

control valves are recommended in the discharge line to ensure that the well is not over-pumped. Placement of a slow-closing motorized valve also on the discharge line ensures positive pressure on the heat pump water coil, and stops the flow of water when the heat pump is not operating (Figure 37). Flow control valves may be noisy as they meter flow. This noise can be mitigated by placing the motorized shutoff valve, with its associated pressure drop, after (downstream of) the flow control valve.

The pressure tank provides water at pressure on demand without short-cycling the well pump. A prepressurized bladder tank is preferred, and it should be large enough that filling it with the well pump takes at least 1 min.

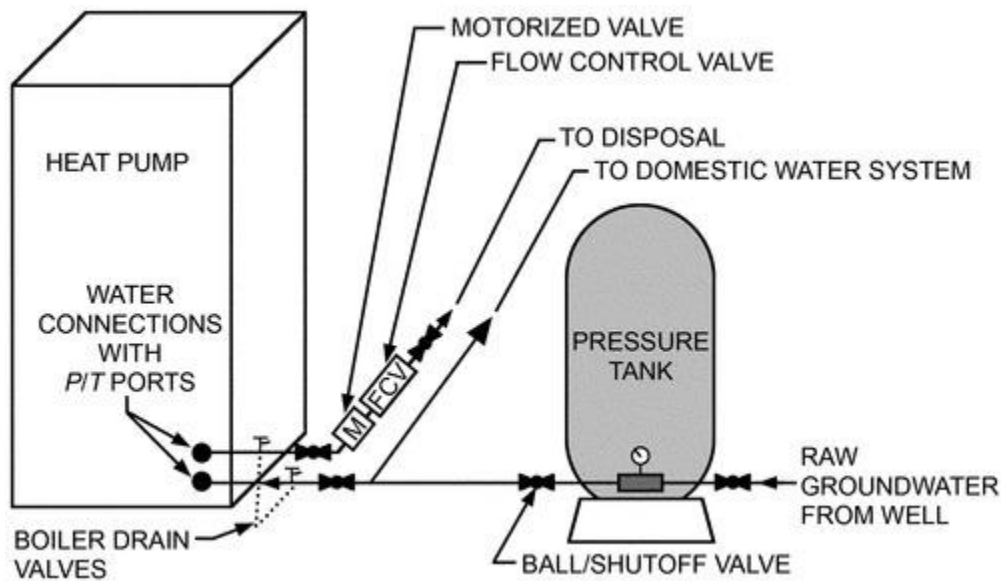


Figure 37. Motorized Valve Placement

Residential groundwater heat pump systems are small enough that the additional cost of an isolation heat exchanger is typically not economically justified. Raw groundwater is generally used as the heat transfer fluid, but provisions must be made (such as hose bibs or boiler drains) to allow for flushing and descaling of the heat pump water/refrigerant coil if necessary.

After exiting the heat pump, groundwater should be returned to a point of discharge in accordance with best practices and/or local codes. Surface discharge to a pond or wetland, or infiltration in to a dry well may be more of an option in these smaller systems than with larger commercial systems due to the correspondingly lower flow rates.

Standing-Column Systems

Standing-column systems use the same well to extract and re-inject the water (Figure 38), and consist of a borehole cased in steel or other material until competent bedrock is reached. The casing must be driven 25 to 50 ft into, and sealed in, the competent bedrock. Bedrock sealing requirements vary by state. The remaining depth of the well is then self supporting through bedrock. Standing-column wells (SCWs) are most practical and cost effective when used in areas with near-surface (<200 ft) consolidated bed rock; the long steel/PVC casing needed to reach deeper bedrock can make the systems very expensive. Though standing-column systems have been applied mostly in the northeastern United States, approximately 60% of the country is underlain by near-surface bedrock suitable for the systems.

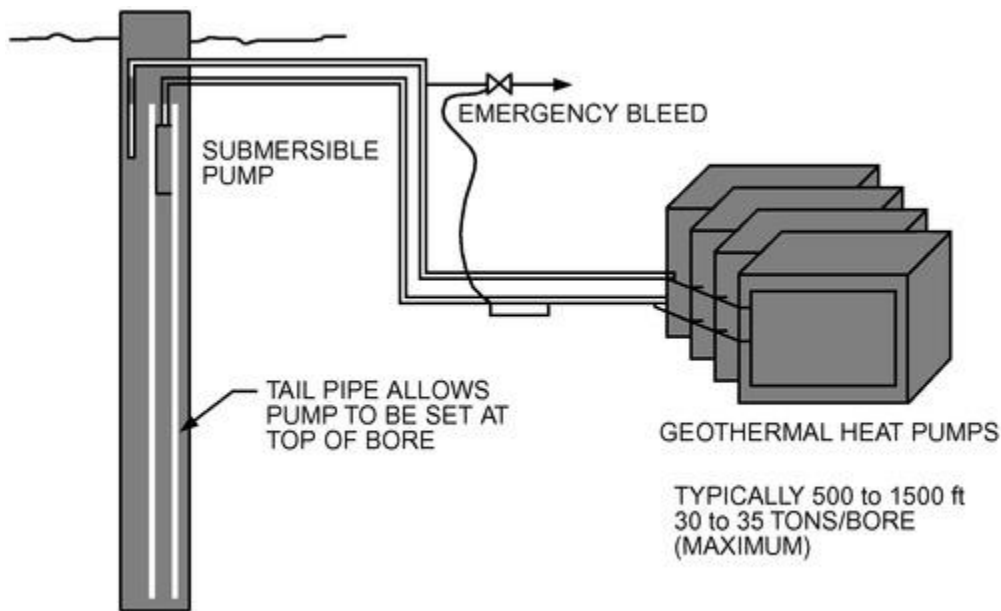


Figure 38. Commercial Standing-Column Well

The SCW combines supply and injection wells into one, and does not depend on the presence or flow of groundwater, beyond that of the typical bleed rate of 10 to 20% of total pumped flow (based on a 3 gpm/ton design flow rate). The bleed circuit effectively extracts water from the SCW by diverting part of the water returning from the heat pump into a reinjection well, storm drain, or roof drain, but generally not into sewer or septic systems. SCWs are always augmented with a bleed circuit to monitor the entering water temperature. Further, the bleed circuit can be used to promote advective flow (bleed circuit reduces water level in SCW and therefore increases flow of groundwater, which is near the undisturbed ground temperature, into the well) to regulate the entering water temperature (Figure 38). The additional advective flow can restabilize (i.e., bring back to far-field temperatures by overflowing smaller amounts of water on command) well water temperatures that are below or above design limits because of variations in rock conductivity, building anomalies, or nonstandard weather patterns. This advective flow is a powerful short-term method of warming and cooling well columns that are beyond design limits. Additional advective flow can be promoted by drawing water from the well for domestic or commercial use. Bleed operation is most critical during winter: entering water temperatures below 40 to 42°F can result in water leaving the heat pump(s) at less than 34°F, for systems designed for flows of 3 gpm/ton. Adequate control to bleed the SCW or shut the heat pump off (and instead use backup heat) must be provided to avoid freezing the water in the heat exchangers.

Water being returned to the bore cannot be allowed to free fall. Free falling water entraps air, which reduces heat exchanger performance and promotes scaling and microbial corrosion. The water should be returned to the SCW using a solid drop pipe, typically 25 to 50 ft below the level of the maximum static water depth. If the drop pipe contains more than 34 ft of water, a perfect vacuum is formed. With a vacuum on the return line, the bleed circuit cannot release water, and air will be drawn in. Therefore, a back-pressure device should be installed in the return line between the bleed-circuit tee and the SCW, to maintain a positive gage pressure of 5 to 10 psi at the bleed-circuit tee.

For residential application, a 250 to 500 ft well provides a heating/cooling capacity of 2 to 8 tons. A relatively simple SCW is used where a submersible well pump is placed at the bottom of the bore. In many jurisdictions, a single well can function with the dual use of ground heat transfer and domestic water. In addition to saving construction costs, this technique enhances advective heat transfer by daily use of domestic water. Dual-use wells typically require (1) the submersible pump to be at a lower elevation than the return drop pipe and (2) installation of a back-flow preventer between the heat pump and the domestic water take-off line.

For commercial application, a heating capacity of 350,000 to 420,000 Btu/h or cooling capacity of 30 to 35 tons can be expected from a single 1500 ft deep standing-column well. These estimated capacities assume a 10% (0.3 gpm/ton) on command and intermittent advective bleed flow. For deep commercial SCWs more than 500 ft, a tail pipe (porter shroud assembly [PSA]) is inserted to form a conduit to draw up water, and an annulus to return water downward (ANSI/CSA/IGSHPA *Standard* C448-16). This tail pipe is perforated at the bottom to form a diffuser. Water is drawn into the diffuser and up the central riser pipe to the submersible pump. The well pump must be located below the water table in line with the central riser pipe. The tail pipe allows a shorter, reduced-power wire size as well as more accessible well pump service. SCW well bore configurations are based upon casing-to-bedrock size, bedrock bore size, and PSA size. The most common SCW is 1500 ft deep and uses an 8 in. casing into bedrock pocket, 6 in. rock bore and a 4 in. PSA. The designer can anticipate stable and slightly higher temperature from the well bottom; below 500 ft ground water temperatures typically increase by 0.25 to 0.5°F per 100 ft. Ideal spacing between SCWs is 50 to 75 ft, to inhibit well-to-well thermal interference (Orio et al. 2005). Typically, spacing between SCWs is greater than vertical closed-loop (GCHP) boreholes. Closer spacing affects well field performance and can be evaluated with design software. Additional information on standing-column systems can be found in Spitler (2002).

In practice, SCWs are a trade-off between extraction well groundwater systems and GCHPs. Flow testing requirements for SCWs are less extensive than for extraction well groundwater systems. The capacity per bore length is less than extraction well systems because SCWs are recharged by advection of only 10 to 20% of total pumped flow, rather than 100% of flow with extraction wells. The SCW capacities are larger than for closed-loop GCHPs because SCWs promote partial advection; they have lower bore thermal resistance since there is no conduction resistance from grout or plastic pipe; and well depths can be deeper without application problems related to large pressure drop in long, narrow pipes.

The U.S. Environmental Protection Agency (EPA) Underground Injection Control program considers standing-column reinjection well water a Class V water use, type 5A7, noncontact cooling water for ground-source heating and cooling. The EPA and equivalent state agencies regard SCW reinjection as a beneficial use. Permitting or notice may be required, depending on average daily water flow rates. SCWs are serviced by qualified well contractors with minimal familiarization training.

1.7 SURFACE WATER HEAT PUMPS

Surface water bodies can be very good heat sources and sinks if properly used. In some cases, lakes can be the very best water supply for cooling. Various water circulation designs are possible; several of the more common are presented here. ANSI/CSA/IGSHPA *Standard* C448-16 contains guidance for both open- and closed-loop surface water heat exchangers.

In a **closed-loop system**, one or more water-to-water or water-to-air heat pumps are linked to one or more submerged coils or flat plate heat exchangers, referred to as **surface water heat exchangers (SWHEs)**. Heat is exchanged to (cooling mode) or from (heating mode) the lake by the fluid (usually a water/antifreeze mixture) circulating inside the SWHE. The heat pump transfers heat to or from the air in the building.

In an **open-loop system**, water is pumped from the lake through a heat exchanger and returned to the lake some distance from the point at which it was removed.

Thermal stratification of water often keeps large quantities of cold water undisturbed near the bottom of deep lakes. This water is cold enough to adequately cool buildings by simply being circulated through heat exchangers. A heat pump is not needed for cooling, and energy use is substantially reduced. Closed-loop coils may also be used in colder lakes. Heating can be provided by a separate source or with heat pumps in heating mode. As noted previously, precooling or supplemental total cooling are also allowed when water returning to the building is near or below 55°F.

Heat Transfer in Lakes

Heat is transferred to lakes by three primary modes: radiant energy from the sun, convective heat transfer from the surrounding air (when the air is warmer than the water), and conduction from the ground. Solar radiation, which can exceed $300 \text{ Btu/h}\cdot\text{ft}^2$ of lake area, is the dominant heating mechanism, but it occurs primarily in the upper portion of the lake unless the lake is very clear. About 40% of solar radiation is absorbed at the surface (Pezent and Kavanaugh 1990). Approximately 93% of the remaining energy is absorbed at depths visible to the human eye.

Convection transfers heat to the lake when the lake surface is cooler than the air. Wind speed increases the rate at which heat is transferred to the lake, but maximum heat gain by convection is usually only 10 to 20% of maximum solar heat gain. Conduction gain from the ground is even less than convection gain (Pezent and Kavanaugh 1990).

Lakes are cooled primarily by evaporative heat transfer at the surface. Convective cooling or heating in warmer months contributes only a small percentage of the total because of the relatively small temperature difference between the air and lake surface. At night when the sky is clear, longwave radiation can account for significant amount of cooling. The relatively warm water surface radiates heat to the cooler sky. For example, on a clear night, a cooling rate of up to $50 \text{ Btu/h}\cdot\text{ft}^2$ is possible from a lake 25°F warmer than the sky. The last mode of heat transfer, conduction to the ground, does not play a major role in lake cooling (Pezent and Kavanaugh 1990), though it does provide significant heating under winter conditions (Gu and Stefan 1990) when the surface of the lake is frozen.

To put these heat transfer rates in perspective, consider a 1 acre ($43,560 \text{ ft}^2$) lake used in connection with a 10 ton ($120,000 \text{ Btu/h}$) heat pump. In cooling mode, the unit rejects approximately $150,000 \text{ Btu/h}$ to the lake. This is $3.4 \text{ Btu/h}\cdot\text{ft}^2$, or approximately 1% of the maximum heat gain from solar radiation in the summer. In winter, a 10 ton heat pump absorbs only about $90,000 \text{ Btu/h}$, or $2.1 \text{ Btu/h}\cdot\text{ft}^2$, from the lake.

Thermal Patterns in Lakes

The maximum density of water occurs at 39.2°F, not at the freezing point of 32°F. This phenomenon, in combination with the normal modes of heat transfer to and from lakes, produces temperature profiles advantageous to efficient heat pump operation. In the winter, the coldest water is at the surface. It tends to remain at the surface and freeze. The bottom of a deep lake stays 5 to 10°F warmer than the surface. This condition is referred to as **winter stagnation**. The warmer water is a better heat source than the colder water at the surface.

As spring approaches, the surface water warms until the temperature approaches the maximum density point of 39.2°F. The winter stratification becomes unstable, and circulation loops begin to develop from top to bottom. This condition of spring overturn (Peirce 1964) causes the lake temperature to become fairly uniform.

Later in the spring, as water temperatures rise above 45°F, the circulation loops are in the upper portion of the lake. This pattern continues throughout the summer. The upper portion of the lake remains relatively warm, with evaporation cooling the lake and solar radiation warming it. The lower portion (hypolimnion) of the lake remains cold because most radiation is absorbed in the upper zone. Circulation loops do not penetrate to the lower zone, and conduction to the ground is quite small. The result is that, in deeper lakes with small or medium inflows, the upper zone is 70 to 90°F, the lower zone is 40 to 55°F, and the intermediate zone (thermocline) has a sharp change in temperature in a small change in depth. This condition is referred to as **summer stagnation**.

As fall begins, the water surface begins to cool by radiation and evaporation. With the approach of winter, the upper portion begins to cool toward the freezing point and the lower levels approach the maximum density temperature of 39.2°F. An ideal temperature-versus-depth chart is shown in [Figure 39](#) for each of the four seasons (Peirce 1964).

Many lakes do exhibit near-ideal temperature profiles. However, (1) high inflow/outflow rates, (2) insufficient depth for stratification, (3) level fluctuation, (4) wind, and (5) lack of enough cold weather to establish sufficient amounts of cold water necessary for summer stratification can disrupt the profile. Therefore, a thermal survey of the lake should be conducted or existing surveys of similar lakes in similar geographic locations should be consulted (Hattermer and Kavanaugh 2005). When interpreting survey data, be aware that there are annual variations in temperature profiles that will not be reflected in measurements for a single year. Shallow ponds and lakes often destratify completely. This does not preclude their use for SWHP systems, but does reduce performance in cooling mode and may require larger heat exchangers compared to lakes that remain stratified.

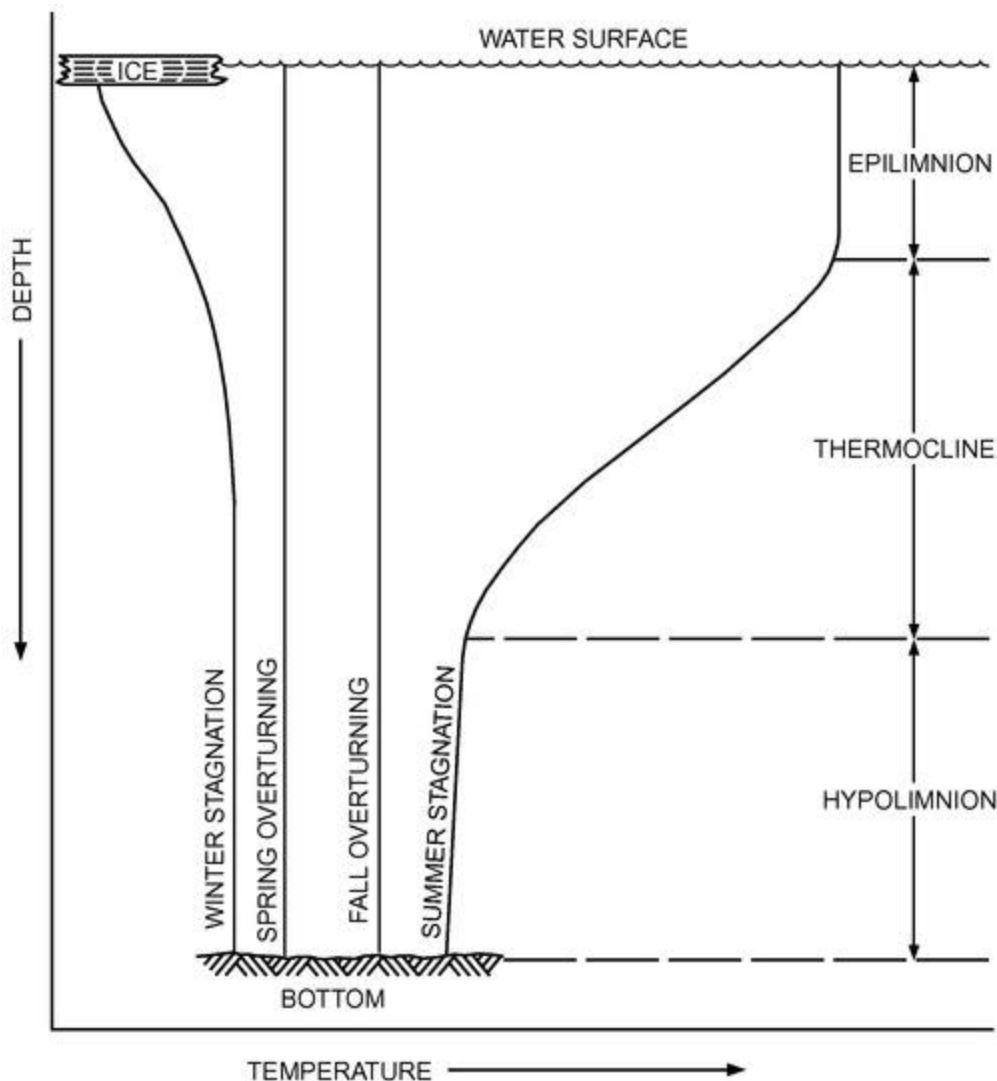


Figure 39. Idealized Diagram of Annual Cycle of Thermal Stratification in Lakes

The thermal and environmental effects of heat rejection and absorption on larger lakes and streams have been studied (Bashyuni et al. 2017; Hattermer et al. 2006; Spitler and Mitchell 2016). However, the effect of SWHPs on thermal stratification profiles is not well characterized, and there is a lack of experimental data. The relative importance of heat transfer modes is not well known, especially during heating mode. There are no publicly available design tools that consider the many heat and mass flow modes of lakes, streams, and oceans. The model of unstratified ponds

developed by Chiasson et al. (2000b) has been implemented in publicly available energy calculation programs, but its lack of accounting for stratification, freezing on the coil, or freezing at the surface limits its usefulness for analysis of deeper lakes and operation in heating mode. The model developed by Spitler et al. (2012) accounts for stratification and for freezing both on the coil and at the surface, but makes several approximations such as neglecting inflows, outflows, and water level variations. Furthermore, validation of all such models is necessarily limited to a small number of lakes for which experimental data are available. Therefore, some caution in using such tools is warranted.

It would be ideal to have a set of statistically characterized temperature profiles (e.g., 1% and 99% design temperature profiles), but the data simply do not exist. With few exceptions, lake temperature profiles are measured infrequently, if at all. Available sources with significant quantities of data include the Consortium of Universities for the Advancement of Hydrologic Science's Hydrologic Information System (CUAHSI HIS 2017), the U.S. EPA's STORET database (EPA 2017), and the U.S. Geological Survey (USGS 2017).

Closed-Loop Lake Water Heat Pump Systems

The closed-loop SWHEs shown in [Figure 40](#) have several advantages over the open loop:

- Fouling is reduced because clean water (or water/antifreeze solution) circulates through the heat pump
- Pumping-power requirement is lower because there is no elevation head from the lake surface to the heat pumps
- It is the only type recommended with unitary heat pumps if a lake temperature below 40°F is possible: fluid outlet temperature is about 6°F below that of the inlet at a flow of 3 gpm per ton, and icing occurs on heat exchanger surfaces when the lake water temperature is in the 34 to 38°F range

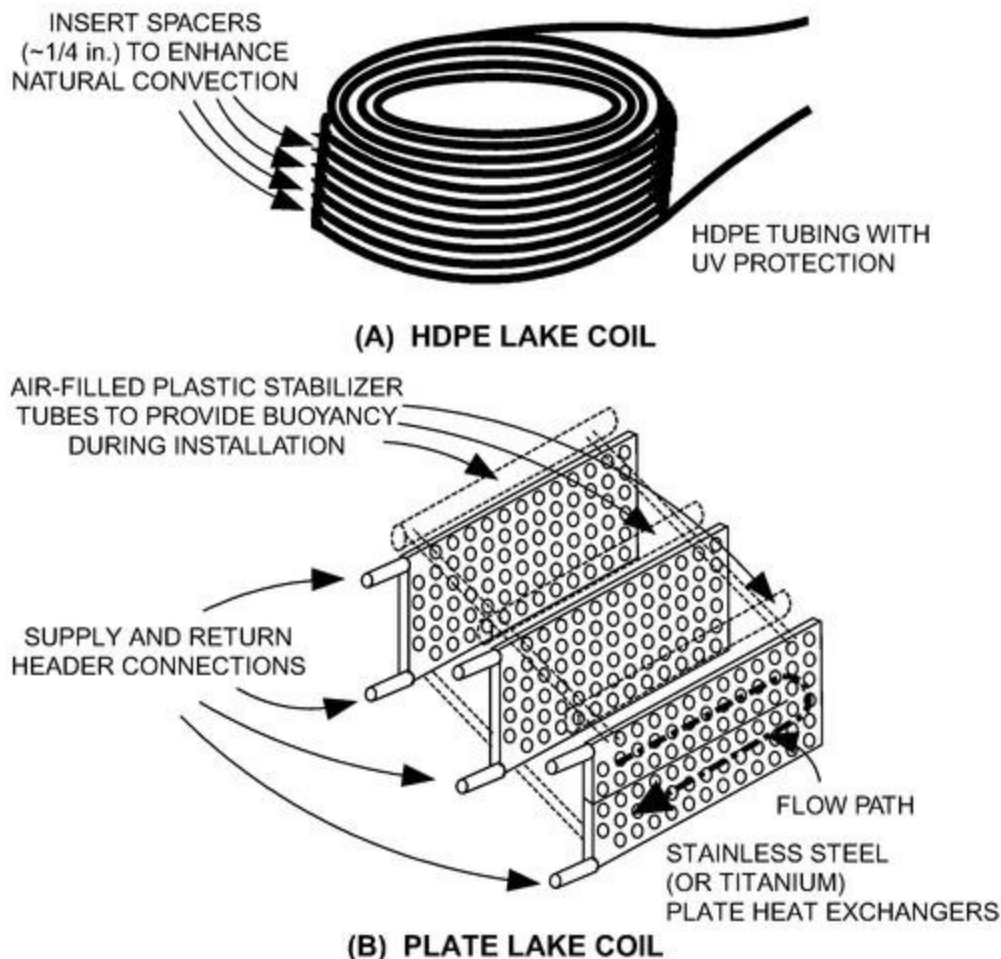


Figure 40. SWHEs: (A) HDPE Coil Type and (B) Plate Type

Disadvantages of closed-loop systems include the following:

- Heat pump performance decreases slightly because circulation fluid temperature drops 4 to 12°F below lake temperature
- Coils may be damaged in public lakes; thermally fused polyethylene loops are much more resistant to damage than copper, glued plastic (PVC), or tubing with band-clamped joints

- Fouling can occur on the outside of the lake coil, particularly in murky lakes or where coils are located on or near the lake bottom.

High-density polyethylene (HDPE 3408) is recommended for in-lake piping. All connections must be either thermally socket-fused or butt-fused. These plastic pipes should also have protection from UV radiation, especially when near the surface. Polyvinyl chloride (PVC) pipe and plastic pipe with band-clamped joints are not recommended.

Plate heat exchangers are also available, and manufacturers typically provide estimated capacities for specified conditions. However, use care in applying metal heat exchangers in cold climates. For an equal temperature difference between the reservoir and fluid inside the coil, the surface temperature of metal heat exchangers is closer to the freezing point of water. The higher thermal resistance of HDPE results in a larger required surface area and a much lower heat transfer per unit of surface area. Thus, the surface temperature of an adequately designed HDPE coil tends to be higher than that of a metal tube or plate and less likely to develop ice on the coil exterior. In certain circumstances, ice build-up may occur on the closed-loop SWHE nonetheless. Use proper methods for anchoring the SWHE to resist the upward buoyant force caused by ice build-up.

The piping networks of closed-loop systems resemble those used in ground-coupled heat pump systems. Both a large-diameter header between the heat pump and lake coil and several parallel loops of piping in the lake are required. Loops are spread out to limit thermal interference, hot spots, and cold pockets. Although this layout is preferred in terms of performance, installation is more time consuming. Many contractors simply unbind plastic pipe coils and submerge them in a loose bundle. Some compensation for thermal interference is obtained by making bundled coils longer than the spread coils. A diagram of this type of installation is shown in [Figure 36A](#).

Copper coils have also been used successfully. Copper tubes have a very high thermal conductivity, so coils only one-fourth to one-third the length of plastic coils are required. However, copper pipe does not have the durability of HDPE 3408, and if fouling is possible, coils must be significantly longer.

Open-Loop Lake Water Heat Pump and Direct Surface Cooling Systems

Open-loop surface water heat pump systems use heat pumps or chillers to provide heating and/or cooling, with surface water circulating through a heat exchanger to provide the heat source and/or sink. The heat pumps may be unitary or custom built.

Direct surface water cooling (DSWC) systems use cold lake or sea water to provide cooling without heat pumps. Because the total (horizontal + vertical) distance between the building(s) being cooled and the actual location of the cold water is often significant, the scale of the system that is economically feasible tends to be quite large.

In cases where the lake or sea water may not be cool enough to meet all demands, hybrid systems that can provide cooling with or without heat pumps or chillers have also been used. Published descriptions of all three types of systems have been reviewed by Mitchell and Spitler (2013).

As noted previously, open-loop systems with unitary heat pumps are not suitable for lake temperatures below 40°F because of the risk of freezing in the evaporator. However, larger installations that use custom-designed heat pumps are successfully operated in Scandinavia under even colder conditions. Their capacities can be as large as 100 million Btu/h. One such system takes water from the Baltic Sea at 37°F and returns it at 33°F using falling film evaporators (i.e., plate evaporators with the sea water sprayed on to the outside of the evaporator) (Mitchell and Spitler 2013).

Open-loop systems are generally designed with either a submersible pump in a wet sump pit connected to the water body or with a conventional centrifugal pump in a dry sump pit. It is also possible in some applications to place the pump just above the water level. Small (e.g., residential) systems are sometimes installed with a submersible pump in the water body.

Design guidelines for open-loop systems may be summarized as follows:

- HDPE is recommended for pipelines because of its flexibility, durability, and high thermal resistance. HDPE is also fusible and floats in water, which makes it possible to connect large sections of pipe together for surface installation.
- Design intakes to avoid entrainment and impingement of sediment and fish. A radial wedge wire intake screen with 0.08 to 0.375 in. openings placed 6.5 to 10 ft above the lake or seabed is recommended. Limit screened face velocity to 0.5 fps.
- For direct surface water cooling applications, intake water temperature should be no higher than 55°F for space air dehumidification. Water with higher temperatures may also be used for sensible only cooling or precooling.
- Pumps may be configured in a wet-sump or dry-sump configuration. For wet-sump pumping designs, a large-diameter intake pipeline or a deep sump pit may be necessary to achieve the required flow rate. For dry-sump pumping configurations, available net positive suction head should be calculated carefully to check against the pump specifications. Pump material should be chosen to resist corrosion and erosion.
- Heat exchangers operating in salt water should be constructed from titanium; those operating in freshwater may use stainless steel. Water quality should be tested and verified.

- Heat exchanger fouling may be a problem in warmer climates. Methods for addressing this concern include chlorine dosing, permanently installed brush systems, or disassembly and cleaning.
- Heat pumps for larger systems are typically custom designed units. If used for heating, multistage compression is common.
- Outfall structures should discharge at a depth that will not promote nutrient enhancement of the outfall area. They should also discharge near the lake or seabed and cover a large enough area so as to prevent a high thermal gradient in the source water body.

2. DIRECT-USE GEOTHERMAL ENERGY

2.1 RESOURCES

Geothermal energy is the thermal energy in the earth's crust: thermal energy in rock and fluid (water, steam, or water containing large amounts of dissolved solids) that fills the pores and fractures in the rock, sand, and gravel. Calculations show that the earth, originating from a completely molten state, would have cooled and become completely solid many thousands of years ago without an energy input beyond that of the sun. It is believed that the ultimate source of geothermal energy is radioactive decay within the earth (Bullard 1973).

Through plate motion and vulcanism, some of this energy is concentrated at high temperature near the surface of the earth. Energy is also transferred from deeper parts of the crust to the earth's surface by conduction and by convection in regions where geological conditions and the presence of water allow.

Because of variation in volcanic activity, radioactive decay, rock conductivities, and fluid circulation, different regions have different heat flows (through the crust to the surface), as well as different temperatures at a particular depth. The normal increase of temperature with depth (i.e., the normal geothermal gradient) is about 13.7°F per 1000 ft of depth, with gradients of 5 to 27°F per 1000 ft being common. Areas that have higher temperature gradients and/or higher-than-average heat flow rates constitute the most interesting and viable economic resources. However, areas with normal gradients may be valuable resources if certain geological features are present. As shown in [Figure 41](#), local gradients also vary with geological condition.

Geothermal resources of the United States are categorized into the following types:

Igneous point resources are associated with magma bodies, which result from volcanic activity. These bodies heat the surrounding and overlying rock by conduction and convection, as allowed by the rock permeability and fluid content in the rock pores.

Hydrothermal convection systems are hot fluids near the earth's surface that result from deep circulation of water in areas of high regional heat flow. A widely used resource, these fluids rise from natural convection between hotter, deeper formations and cooler formations near the surface. The passageway that provides for this deep convection must consist of adequately permeable fractures and faults.

Geopressed resources, present widely in the Gulf Coast of the United States, consist of regional occurrences of confined hot water in deep sedimentary strata, where pressures of greater than 10,000 psi are common. This resource also contains methane, which is dissolved in the geothermal fluid.

Radiogenic heat sources exist in various regions as granitic plutonic rocks that are relatively rich in uranium and thorium. These plutons have a higher heat flow than the surrounding rock; if the plutons are blanketed by sediments of low thermal conductivity, an elevated temperature at the base of the sedimentary section can result. This resource has been identified in the eastern United States.

Deep regional aquifers of commercial value can occur in deep sedimentary basins, even in areas of only normal temperature gradient. For deep aquifers to be of commercial value, (1) basins must be deep enough to provide usable temperature levels at the prevailing gradient, and (2) permeability in the aquifer must be adequate for flow.

Thermal energy in geothermal resources exists primarily in the rocks and only secondarily in the fluids that fill the pores and fractures. Thermal energy is usually extracted by bringing to the surface the hot water or steam that occurs naturally in the open spaces in the rock. Where rock permeability is low, the energy extraction rate is low. In permeable aquifers, fluid produced may be injected back into the aquifer at some distance from the production well to pass through the aquifer again and recover some of the energy in the rock. [Figure 37](#) indicates geothermal resource areas in the United States.

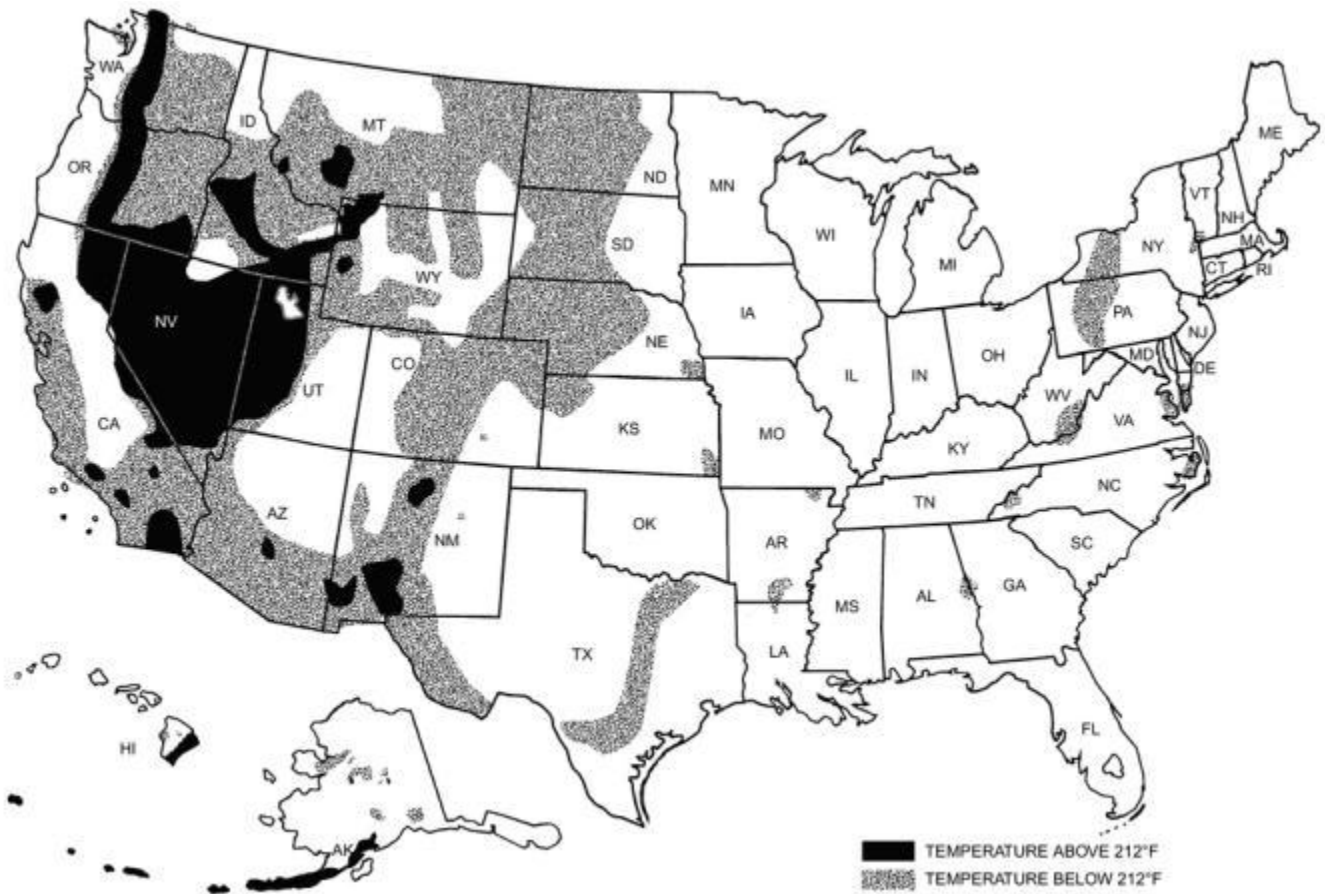


Figure 41. U.S. Hydrothermal Resource Areas (Lienau et al. 1995)

Temperature

The temperature of fluids produced in the earth's crust and used for their thermal energy content varies from above 60°F to 680°F. The lower temperature value represents local undisturbed ground temperature in the absence of geothermal resources (approximately equal to the local average climate temperature) and the higher temperature represents an approximate value for the HGP-A well at Hilo, Hawaii.

The following classification by temperature is used in the geothermal industry:

High temperature	$t > 300^{\circ}\text{F}$
Intermediate temperature	$195^{\circ}\text{F} < t < 300^{\circ}\text{F}$
Low temperature	$60^{\circ}\text{F} \leq t < 195^{\circ}\text{F}$

Electric generation is generally not economical for resources with temperatures below about 300°F, which is the reason for the division between high- and intermediate-temperature. However, binary (organic Rankine cycle) power plants, with the proper set of circumstances, have demonstrated that it is possible to generate electricity economically above 230°F. In 1988, there were 86 binary plants worldwide, generating a total of 126.3 MW (Di Pippo 1988).

Geothermal resources at lower temperatures are more common. The frequency by reservoir temperature of identified convective systems above 194°F is shown in [Figure 42](#).

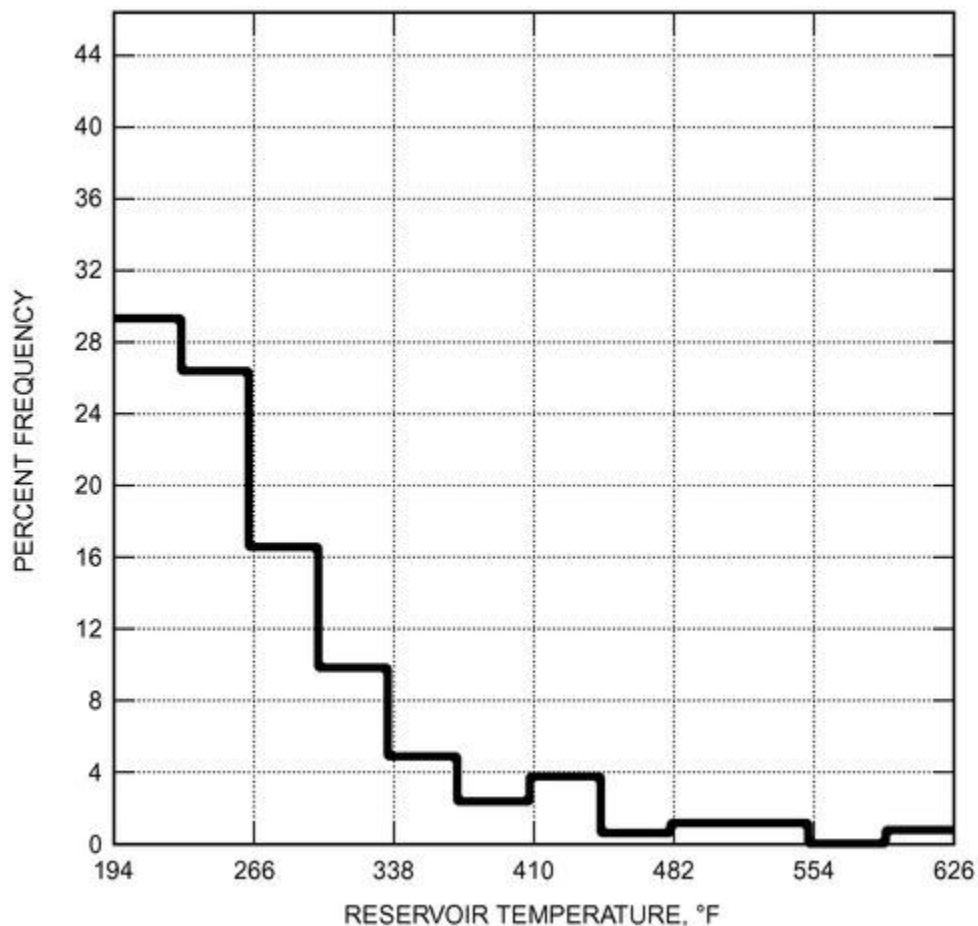


Figure 42. Frequency of Identified Hydrothermal Convection Resources Versus Reservoir Temperature (Muffler 1979)

2.2 FLUIDS

Geothermal energy is extracted from the earth through naturally occurring fluids in rock pores and fractures. Fluids produced are steam, hot water, or a two-phase mixture of both. These may contain various amounts of impurities, notably dissolved gases and dissolved solids.

Geothermal resources that produce essentially dry steam are **vapor dominated**. Although these are valuable resources, they are rare. Hot-water (**fluid-dominated**) resources are much more common and can be produced either as hot water or as a two-phase mixture of steam and hot water, depending on the pressure maintained on the production well. If pressure in the production casing or in the formation around the casing is reduced below the saturation pressure at that temperature, some of the fluid will flash, and a two-phase fluid will result. If pressure is maintained above the saturation pressure, the fluid remains single-phase. In fluid-dominated resources, both dissolved gases and dissolved solids are significant.

Geothermal fluid chemistry varies over a wide range. In the Imperial valley of California, some high-temperature geothermal fluids may contain up to 300,000 ppm of **total dissolved solids (TDS)**. Fluids of this character are extremely difficult to accommodate in systems design and materials selection. In fact, most low-temperature fluids contain less than 3000 ppm and many meet drinking water standards. Despite this, even geothermal fluids of a few hundred ppm TDS can cause substantial problems with standard construction materials.

2.3 PRESENT USE

Discoveries of concentrated radiogenic heat sources and deep regional aquifers in areas of near-normal temperature gradient indicate that 37 states in the United States have economically exploitable direct-use geothermal resources (Interagency Geothermal Coordinating Council 1980). The Geysers, in northern California, is the largest single geothermal development in the world. The U.S. Department of Energy created a database of geothermal system data (including ground resource data) for practitioners to share data about installations (NGDS 2014).

The total electricity generated by geothermal development in the world was 7974 MW in 2000 (Lund et al. 2001). Direct application of geothermal energy for space heating and cooling, water heating, agricultural growth-related heating, and industrial processing represented about 51.6×10^9 Btu/h worldwide in 2000. In the United States in 2000, direct-use installed capacity amounted to 12.9×10^9 Btu/h, providing 19.3×10^{12} Btu/yr.

The major uses of geothermal energy in the United States are for heating greenhouse and aquaculture facilities. The principal industrial use is for food processing.

2.4 DESIGN

A major goal in designing direct-use systems is capturing the most possible heat from each gallon of fluid pumped. System owning and operating costs are composed primarily of well pumping and well capitalization components; maximizing system Δt (i.e., minimizing flow requirements) minimizes well capital cost and pump operating cost. In many cases, system design can benefit from connecting loads in series according to temperature requirements. Direct-use system design is covered in detail in Anderson and Lund (1980) and Rafferty (1989a).

Direct-use systems can be divided into four subsystems: (1) production, including the producing wellbore and associated wellhead equipment; (2) transmission and distribution to transport geothermal energy from the resource site to the user site and then distribute it to the individual user loads; (3) user system; and (4) disposal, which can be either surface disposal or injection back into a formation.

In a typical direct-use system, geothermal fluid is produced from the production borehole by a lineshaft multistage centrifugal pump. When the geothermal fluid reaches the surface, it is delivered to the application site through the transmission and distribution system.

In [Figure 43](#), geothermal fluid is separated from the heating system by a heat exchanger. This secondary loop is especially desirable when the geothermal fluid is particularly corrosive and/or causes scaling. The geothermal fluid is pumped directly back into the ground without loss to the surrounding surface.

2.5 COST FACTORS

The following characteristics influence the cost of energy delivered from geothermal resources:

- Well depth
- Distance between resource location and application site
- Well flow rate
- Resource temperature
- Temperature drop
- Load factor
- Composition of fluid
- Ease of disposal

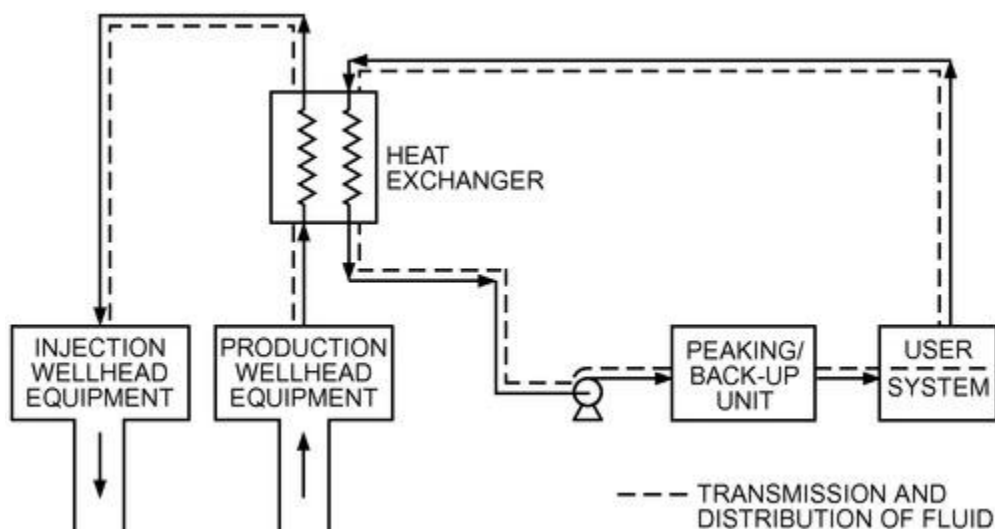


Figure 43. Geothermal Direct-Use System with Wellhead Heat Exchanger and Injection Disposal

Many of these characteristics have a major influence because the cost of geothermal systems is primarily front-end capital cost; annual operating cost is relatively low.

Well Depth

The cost of the wells is usually one of the larger items in the overall cost of a geothermal system, and increases with resource depth. Compared to many geothermal areas worldwide, well depth requirements in the western United States are relatively shallow; most larger geothermal systems there operate with production wells of less than 2000 ft, and many at less than 1000 ft.

Distance Between Resource Location and Application Site

Direct use of geothermal energy must occur near the resource. The reason is primarily economic; although geothermal (or secondary) fluid could be transmitted over moderately long distances (greater than 60 miles) without great temperature loss, such transmission is generally not economically feasible. Most existing geothermal projects have transmission distances of less than 1 mile.

Well Flow Rate

Energy output from a production well varies directly with the fluid flow rate. The energy cost at the wellhead varies inversely with the well flow rate. A typical good resource has a production rate of 400 to 800 gpm per production well; however, geothermal direct-use wells have been designed to produce up to 2000 gpm}.

Resource Temperature

The available temperature is fixed by the prevailing resource. The temperature can restrict applications. It often requires a reevaluation of accepted application temperatures, which were developed for uses served by conventional fuels for which the application temperature could be selected at any value in a relatively broad range. Most existing direct-use projects use fluids in the 130 to 230°F range.

Temperature Drop

Because well flow is limited, power output from a geothermal well is directly proportional to the temperature drop of the geothermal fluid connected to the system. Consequently, a larger temperature drop reduces operating (pumping) and capital (well and production pump) costs.

Cascading geothermal fluid to uses with lower temperature requirements can help achieve a large temperature difference (Δt). Most geothermal systems are designed for a Δt between 30 and 50°F, although one system was designed for a Δt of 100°F with a 190°F resource temperature.

Load Factor

Defined as the ratio of the average load to the design capacity of the system, the load factor effectively reflects the fraction of time that the initial investment in the system is working. Again, because geothermal cost is primarily initial rather than operating cost, this factor significantly affects a geothermal system’s viability. As the load factor increases, so does the economy of using geothermal energy. The two main ways of increasing the load factor are (1) to select applications where it is naturally high, and (2) to use peaking equipment so that the geothermal design load is not the application peak load, but rather a reduced load that occurs over a longer period.

Table 26 Selected Chemical Species Affecting Fluid Disposal

Species	Reason for Control
Hydrogen sulfide (H ₂ S)	Odor
Boron (B ³⁺)	Damage to agricultural crops
Fluoride (F ⁻)	Level limited in drinking water sources
Radioactive species	Levels limited in air, water, and soil

Source: Lunis (1989).

Composition of Fluid

The quality of the produced fluid is site specific and may vary from less than 1000 ppm TDS to heavily brined. Fluid quality influences two aspects of the design: (1) material selection to avoid corrosion and scaling effects, and (2) disposal or ultimate end use of the fluid.

Ease of Disposal

The costs associated with disposal, particularly when injection is involved, can substantially affect development costs. Historically, most geothermal effluent was disposed of on the surface, including discharge to irrigation, rivers, and lakes. This method of disposal is considerably less expensive than constructing injection wells.

Geothermal fluids sometimes contain chemical constituents that make surface disposal problematic. Some of these constituents are listed in [Table 26](#).

Most new, large geothermal systems use injection for disposal to minimize environmental concerns and ensure long-term resource reliability. If injection is chosen, the depth at which the fluid can be injected affects well cost substantially. Many jurisdictions require the fluid be returned to the same or similar aquifers; thus, it may be necessary to bore the injection well to the same depth as the production well. Direct-use injection wells are considered Class V wells under the U.S. Environmental Protection Agency’s Underground Injection Control (UIC) program. Water wells, along with terminology relating to the technology, are discussed in the section on Ground-Source Heat Pumps.

Direct-Use Water Quality Testing

Low-temperature geothermal fluids commonly contain seven key chemical species that can significantly corrode standard materials of construction (Ellis 1989). These include

- Oxygen (generally from aeration)
- Hydrogen ion (pH)
- Chloride ion
- Sulfide species
- Carbon dioxide species
- Ammonia species
- Sulfate ion

The principal effects of these species are summarized in [Table 27](#). Except as noted, the described effects are for carbon steel. Kindle and Woodruff (1981) present recommended procedures for complete chemical analysis of geothermal well water.

Table 27 Principal Effects of Key Corrosive Species

Species	Principal Effects
Oxygen	<ul style="list-style-type: none">• Extremely corrosive to carbon and low-alloy steels; 30 ppb shown to cause fourfold increase in carbon steel corrosion rate.• Concentrations above 50 ppb cause serious pitting.• In conjunction with chloride and high temperature, <100 ppb dissolved oxygen can cause chloride-stress corrosion cracking (chloride-SCC) of some austenitic stainless steels.
Hydrogen ion (pH)	<ul style="list-style-type: none">• Primary cathodic reaction of steel corrosion in air-free brine is hydrogen ion reduction. Corrosion rate decreases sharply above pH 8.• Low pH (5) promotes sulfide stress cracking (SSC) of high-strength low-alloy (HSLA) steels and some other alloys coupled to steel.• Acid attack on cements.
Carbon dioxide species (dissolved carbon dioxide, bicarbonate ion, carbonate ion)	<ul style="list-style-type: none">• Dissolved carbon dioxide lowers pH, increasing carbon and HSLA steel corrosion.• Dissolved carbon dioxide provides alternative proton reduction pathway, further exacerbating carbon and HSLA steel corrosion.• May exacerbate SSC.• Strong link between total alkalinity and corrosion of steel in low-temperature geothermal wells.
Hydrogen sulfide species (hydrogen sulfide, bisulfide ion, sulfide ion)	<ul style="list-style-type: none">• Potent cathodic poison, promoting SSC of HSLA steels and some other alloys coupled to steel.• Highly corrosive to alloys containing both copper and nickel or silver in any proportions.
Ammonia species (ammonia, ammonium ion)	<ul style="list-style-type: none">• Causes stress corrosion cracking (SCC) of some copper-based alloys.

Chloride ion

- Strong promoter of localized corrosion of carbon, HSLA, and stainless steel, as well as of other alloys.
- Chloride-dependent threshold temperature for pitting and SCC. Different for each alloy.
- Little if any effect on SSC.
- Steel passivates at high temperature in 6070 ppm chloride solution (pH = 5) with carbon dioxide. 133,500 ppm chloride destroys passivity above 300°F.

Sulfate ion

- Primary effect is corrosion of cements.

Source: Ellis (1989).

Note: Except as indicated, described effects are for carbon steel.

Two of these species are not reliably detected by standard water chemistry tests and deserve special mention. Dissolved oxygen does not occur naturally in low-temperature (120 to 220°F) geothermal fluids that contain traces of hydrogen sulfide. However, because of slow-reaction kinetics, oxygen from air inleakage may persist for some minutes. Once the geothermal fluid is produced, it is extremely difficult to prevent contamination, especially if pumps used to move the fluid are not downhole-submersible or lineshaft turbine pumps. Even if fluid systems are maintained at positive pressure, air inleakage at pump seals is likely, particularly with the low level of maintenance in many installations.

Hydrogen sulfide is ubiquitous in extremely low concentrations in geothermal fluids above 120°F. This corrosive species also occurs naturally in many cooler groundwaters. For strongly affected alloys, such as cupronickel, hydrogen sulfide concentrations in the low parts per billion (10^9) range may have a serious detrimental effect, especially if oxygen is also present. At these levels, the characteristic rotten egg odor of hydrogen sulfide may be absent, so field testing may be required for detection. Hydrogen sulfide levels down to 50 ppb can be detected using a simple field kit; however, absence of hydrogen sulfide at this low level may not preclude damage by this species.

Two other key species that should be measured in the field are pH and carbon dioxide concentrations. This is necessary because most geothermal fluids release carbon dioxide rapidly, causing a rise in pH.

Production of suspended solids (sand) from a well should be addressed during well construction with gravel pack, screen, or both. Proper selection of the screen/gravel pack is based on sieve analysis of cutting samples from drilling. Surface separation is less desirable because it requires sand to pass first through the pump, reducing its useful life.

Biological fouling is largely a phenomenon of low-temperature (<90°F) wells. The most prominent organisms are various strains (*Galionella*, *Crenothrix*) of what are commonly referred to as **iron bacteria**. These organisms typically inhabit water with a pH range of 6.0 to 8.0, dissolved oxygen content of less than 5 ppm, ferrous iron content of less than 0.2 ppm, and a temperature of 46 to 61°F (Hackett and Lehr 1985). Iron bacteria can be identified microscopically. The most common treatment for iron bacteria infestation is chlorination, surging, and flushing; success depends on maintaining proper pH (less than 8.5), dosage, free residual chlorine content (200 to 500 ppm), contact time (24 h minimum), and agitation or surging. Precleaning (wire brushing) of the screen and redevelopment of the well after treatment are key to effectiveness. Hackett and Lehr (1985) provide additional detail on treatment.

2.6 MATERIALS AND EQUIPMENT

For system parts exposed to the fluid, materials selection is an important part of the design process. Chemical treatment of the geothermal fluid is not an effective strategy in most cases, because of economics and environmental (disposal) considerations. Corrosion and scaling in direct-use systems are generally addressed by isolating the fluid from the majority of the system using a plate heat exchanger.

Performance of Materials

Carbon Steel. The Ryznar index has traditionally been used to estimate the corrosivity and scaling tendencies of potable water supplies. However, one study found no significant correlation (at the 95% confidence level) between carbon steel corrosion and the Ryznar index (Ellis and Smith 1983). Therefore, the Ryznar and other indices based on calcium carbonate saturation should not be used to predict corrosion in geothermal systems, though they remain valid for scaling prediction.

In Class Va geothermal fluids [as described by Ellis (1989); <5000 ppm total key species (TKS), total alkalinity 207 to 1329 ppm as CaCO_3 , pH 6.7 to 7.6], corrosion rates of about 5 to 20 mil/yr can be expected, often with severe pitting.

In Class Vb geothermal fluids [as described by Ellis (1989); <5000 ppm TKS, total alkalinity <210 ppm as CaCO_3 , pH 7.8 to 9.85], carbon steel piping has given good service in a number of systems, as long as system design rigorously excluded oxygen. However, introduction of 30 ppb oxygen under turbulent flow conditions causes a fourfold increase in uniform corrosion. Saturation with air often increases the corrosion rate by at least 15 times. Oxygen contamination at the 50 ppb level often causes severe pitting. Chronic oxygen contamination causes rapid failure.

External surfaces of buried steel pipe must be protected from contact with groundwater. Groundwater is aerated and has caused pipe failures by external corrosion. Required external protection can be obtained by coatings, pipe-wrap, or

preinsulated piping, provided the selected material resists the system operating temperature and thermal stress.

At temperatures above 135°F, galvanizing (zinc coating) does not reliably protect steel from either geothermal fluid or groundwater. Hydrogen blistering can be prevented by using void-free (killed) steels.

Low-alloy steels (steels containing not more than 4% alloying elements) have corrosion resistance similar, in most respects, to carbon steels. As with carbon steels, sulfide promotes entry of atomic hydrogen into the metal lattice. If the steel exceeds a hardness of Rockwell C22, sulfide stress cracking may occur.

Copper and Copper Alloys. Copper-tubed fan-coil units and heat exchangers have consistently poor performance because of traces of sulfide species found in geothermal fluids in the United States. Copper tubing rapidly becomes fouled with cuprous sulfide films more than 0.04 in. thick. Serious crevice corrosion occurs at cracks in the film, and uniform corrosion rates of 2 to 6 mil/yr appear typical, based on failure analyses.

Experience in Iceland also indicates that copper is unsatisfactory for heat exchange service and that most brasses (Cu-Zn) and bronzes (Cu-Sn) are even less suitable. Cupronickel often performs more poorly than copper in low-temperature geothermal service because of trace sulfide.

Much less information is available regarding copper and copper alloys in non-heat-transfer service. Copper pipe shows corrosion behavior similar to copper heat exchange tubes under conditions of moderate turbulence (Reynolds numbers of 40,000 to 70,000). An internal inspection of yellow brass valves showed no significant corrosion. However, silicon bronze CA 875 (12-16Cr, 3-5Si, <0.05Pb, <0.05P), an alloy normally resistant to dealloying, failed in less than three years when used as a pump impeller. Leaded red brass (CA 836 or 838) and leaded red bronze (SAE 67) appear viable as pump internal parts. Based on a few tests at Class Va sites, aluminum bronzes have shown potential for corrosion in heavy-walled components (Ellis 1989).

Solder is yet another problem area for copper equipment. Lead-tin solder (50Pb, 50Sn) was observed to fail by dealloying after a few years' exposure. Silver solder (1Ag, 7P, Cu) was completely removed from joints in under two years. If the designer elects to accept this risk, solders containing at least 70% tin should be used.

Stainless Steel. Unlike copper and cupronickel, stainless steels are not affected by traces of hydrogen sulfide. Their most likely application is heat exchange surfaces. For economic reasons, most heat exchangers are probably of the plate-and-frame type, most of which are fabricated with one of two standard alloys, Type 304 and Type 316 stainless steel. Some pump and valve trim also are fabricated from these or other stainless steels.

These alloys are subject to pitting and crevice corrosion above a threshold chloride level, which depends on the chromium and molybdenum content of the alloy and on the temperature of the geothermal fluid. Above this temperature, the passivation film, which gives the stainless steel its corrosion resistance, is ruptured, and local pitting and crevice corrosion occur. Figure 44 shows the relationship between temperature, chloride level, and occurrence of localized corrosion of Type 304 and Type 316 stainless steel. This figure indicates, for example, that localized corrosion of Type 304 may occur in 80°F geothermal fluid if the chloride level exceeds approximately 210 ppm; Type 316 is resistant at that temperature until the chloride level reaches approximately 510 ppm. Because of its 2 to 3% molybdenum content, Type 316 is always more resistant to chlorides than is Type 304.

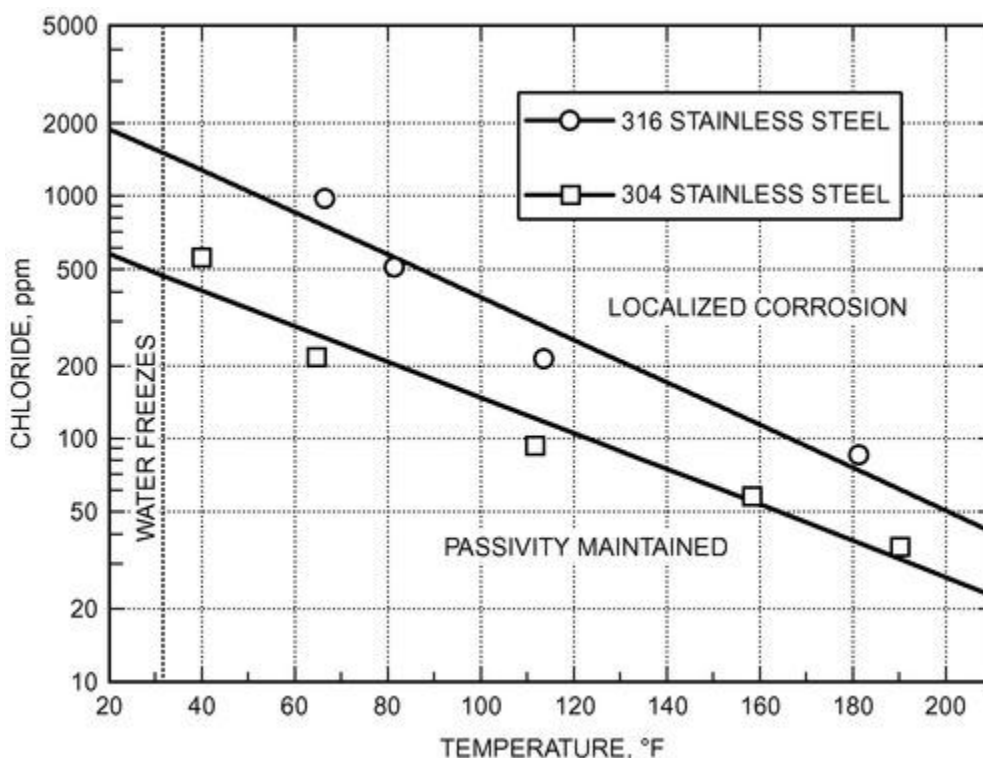


Figure 44. Chloride Concentration Required to Produce Localized Corrosion of Stainless Steel as Function of Temperature (Efrid and Moeller 1978)

Aluminum. Aluminum alloys are not acceptable in most cases because of catastrophic pitting.

Titanium. This material has extremely good corrosion resistance and could be used for heat exchanger plates in any low-temperature geothermal fluid, regardless of dissolved oxygen content. Great care is required if acid cleaning is to be performed. The vendor's instructions must be followed. The titanium should not be scratched with iron or steel tools; this can cause pitting.

Chlorinated Polyvinyl Chloride (CPVC) and Fiber-Reinforced Plastic (FRP). These materials are easily fabricated and are not adversely affected by oxygen intrusion. External protection against groundwater is not required. The mechanical properties of these materials at higher temperatures may vary greatly from those at ambient temperature, and the materials' mechanical limits should not be exceeded. The usual mode of failure is creep rupture: strength decays with time. Follow manufacturer's directions for joining to avoid premature failure of joints.

Elastomeric Seals. Tests on O-ring materials in a low-temperature system in Texas indicated that a fluoroelastomer is the best material for piping of this nature; Buna-N is also acceptable (Ellis 1989). Neoprene, which developed extreme compression set, was a failure. Natural rubber and Buna-S should also be avoided. Ethylene-propylene terpolymer (EPDM) has been used successfully in gasket, O-ring, and valve seats in many systems. EPDM materials have swollen in some systems using oil-lubricated turbine pumps.

Pumps

Production well pumps are among the most critical components in a geothermal system and have been the source of much system downtime. Therefore, proper selection and design of the production well pump is extremely important. Well pumps are available for larger systems in two general configurations: lineshaft and submersible. The lineshaft type is most often used for direct-use systems (Rafferty 1989b).

Lineshaft Pumps. Lineshaft pumps are similar to those typically used in irrigation applications. An aboveground driver, typically an electric motor, rotates a vertical shaft extending down the well to the pump. The shaft rotates pump impellers in the pump bowl assembly, which is positioned at such a depth in the wellbore that adequate net positive suction head pressure (NPSH) is available when the unit is operating. Two designs for the shaft/bearing portion of the pump are available: open and enclosed.

In the **open lineshaft pump**, the shaft bearings are supported in "spiders," which are anchored to the pump column pipe at 5 to 10 ft intervals. The shaft and bearings are lubricated by the fluid flowing up the pump column. In geothermal applications, bearing materials for open lineshaft designs are typically elastomer compounds. The shaft material is typically stainless steel. Experience with this design in geothermal applications has been mixed. It appears that the open lineshaft design is most successful in applications with high (<50 ft) static water levels or flowing artesian conditions. Open lineshaft pumps are generally less expensive than enclosed lineshaft pumps for the same application.

In an **enclosed lineshaft pump**, an enclosing tube protects the shaft and bearings from exposure to the pumped fluid. A lubricating fluid is admitted to the enclosed tube at the wellhead. It flows down the tube, lubricates the bearings, and exits where the column attaches to the bowl assembly. The bowl shaft and bearings are lubricated by the pumped fluid. Oil-lubricated, enclosed lineshaft pumps have the longest service life in low-temperature, direct-use applications.

These pumps typically include carbon or stainless steel shafts and bronze bearings in the lineshaft assembly, and stainless steel shafts and leaded red bronze bearings in the bowl assembly. Keyed-type impeller connections (to the pump shaft) are superior to collet-type connections (Rafferty 1989b).

Because of the lineshaft bearings, lineshaft pump reliability decreases as pump-setting depth increases. Nichols (1978) indicates that, below about 800 ft, reliability is questionable, even under good pumping conditions.

Submersible Pumps. The electrical submersible pump consists of three primary components located downhole: the pump, the drive motor, and the motor protector. The pump is a vertical multistage centrifugal type. The motor is usually a three-phase induction type that is filled with oil for cooling and lubrication; it is cooled by heat transfer to the pumped fluid moving up the well. The motor protector is located between the pump and the motor and isolates the motor from the well fluid while allowing pressure equalization between the pump intake and the motor cavity.

The electrical submersible pump has several advantages over lineshaft pumps, particularly for wells requiring greater pump bowl setting depths. The deeper the well, the greater the economic advantage of the submersible pump. Moreover, it is more versatile, adapting more easily to different depths.

Submersible pumps have not demonstrated acceptable lifetimes in most geothermal applications. Although they are commonly used in high-temperature, downhole applications in the oil and gas industry, the acceptable overhaul interval in that industry is much shorter than in a geothermal application. In addition, most submersibles operate at 3600 rpm, resulting in greater susceptibility to erosion in aquifers that produce moderate amounts of sand. They have, however, been applied in geothermal projects where an existing well of relatively small diameter must be used. At 3600 rpm, they provide greater flow capacity for a given bowl size than an equivalent 1750 rpm lineshaft pump.

Standard cold-water submersible motors can be used at temperatures up to approximately 120°F with adequate precautions. These consist primarily of ensuring adequate water velocity past the motor (minimum 3 ft/s), which may require the use of a sleeve, and a small degree of motor oversizing (Franklin Electric 2001).

Well Pump Control. Well pumps serving variable loads are often controlled using variable-speed drives (VSDs). Submersible pumps can also be controlled using VSDs, but special precautions are required. Drive-rated motors are not commonly available for these applications, so external electronic protection should be used to prevent premature motor

failure. In addition, the motor manufacturer must be aware that the motor will be applied in a variable-speed application. Finally, because of the large static head in many well pump applications, controls should be configured to prevent the pump from operating at no-flow conditions.

Heat Exchangers

Geothermal fluids can be isolated with large central heat exchangers, as in the case of a district heating system, or with exchangers at individual buildings or loads. In both cases, the principle is to isolate the geothermal fluid from complicated systems or those that cannot readily be designed to be compatible with the geothermal fluid. The main types of heat exchangers used in transferring energy from the geothermal fluid are plate and downhole.

Plate Heat Exchangers. For all but the very smallest applications, plate-and-frame heat exchangers are the most commonly used design. Available in corrosion-resistant materials, easily cleanable, and able to accommodate increased loads by adding plates, these exchangers are well suited to geothermal applications. Their high performance is also an asset in many system designs. Because geothermal resource temperatures are often less than those used in conventional hot-water heating system design, minimizing temperature loss at the heat exchanger is frequently a design issue. Approach temperatures of 5°F and less are common.

Materials for plate heat exchangers in direct-use applications normally include Buna-N or EPDM gaskets and 316 or titanium plates. Plate selection is often a function of temperature and chloride content of the water. For applications characterized by chloride contents of >50 ppm at 200°F, titanium would be used. At lower temperatures, much higher chloride exposure can be tolerated (see [Figure 40](#)).

Downhole Heat Exchangers. The downhole heat exchanger (DHE) is an arrangement of pipes or tubes suspended in a wellbore (Culver and Reistad 1978). A secondary fluid circulates from the load through the exchanger and back to the load in a closed loop. The primary advantage of a DHE is that only heat is extracted from the earth, which eliminates the need to dispose of spent fluids. Other advantages are the elimination of (1) well pumps with their initial operating and maintenance costs, (2) the potential for depletion of groundwater, and (3) environmental and institutional restrictions on surface disposal. One disadvantage of a DHE is the limited amount of heat that can be extracted from or rejected to the well. The amount of heat extracted depends on the hydraulic conductivity of the aquifer and well design. Because of the limitations of natural convection, only about 10% of the heat output of the well is available from a DHE in comparison to pumping and using surface heat exchange (Reistad et al. 1979). With wells of approximately 200°F and depths of 500 ft, output under favorable conditions is sufficient to serve the needs of up to five homes.

The DHE in low- to moderate-temperature geothermal wells is installed in a casing, as shown in [Figure 45](#).

Downhole heat exchangers with higher outputs rely on water circulation within the well, whereas lower-output DHEs rely on earth conduction. Circulation in the well can be accomplished by two methods: (1) undersized casing and (2) convection tube. Both methods rely on the difference in density between the water surrounding the DHE and that in the aquifer.

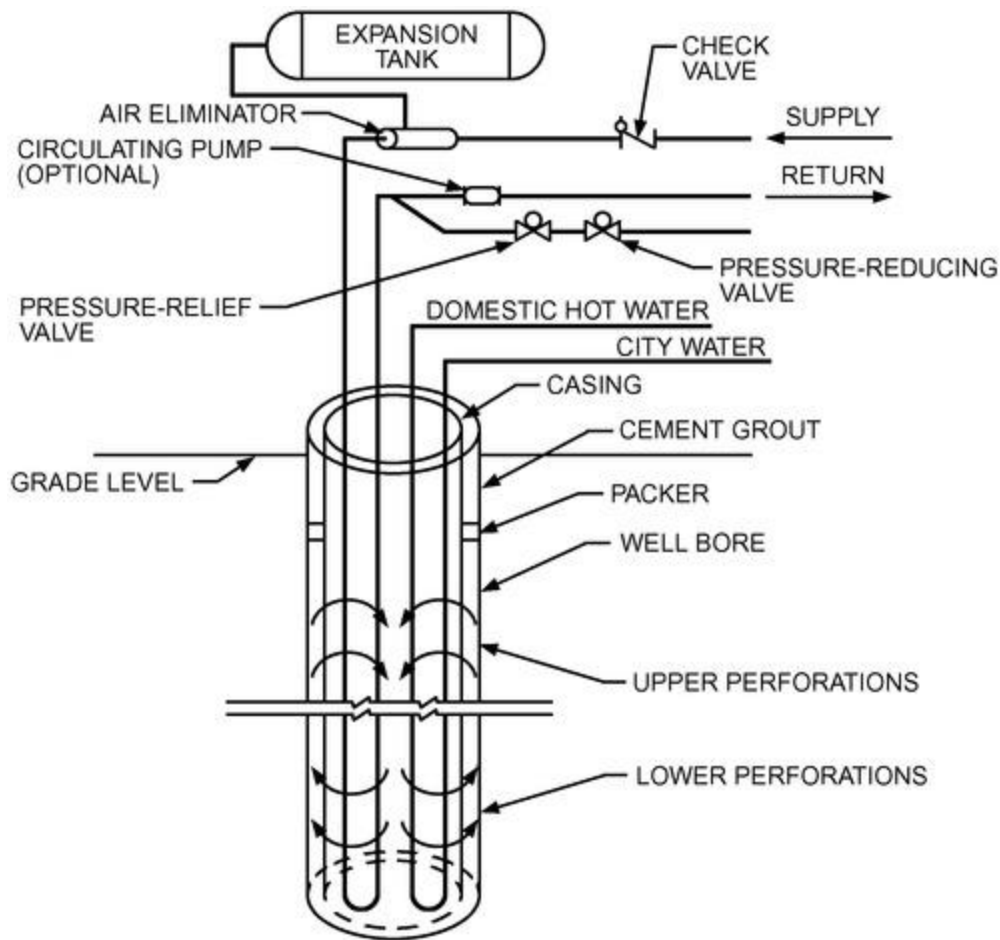


Figure 45. Typical Connection of Downhole Heat Exchanger for Space and Domestic Hot-Water Heating (Reistad et al. 1979)

Circulation provides the following advantages:

- Water circulates around the DHE at velocities that, in optimum conditions, can approach those in the shell of a shell-and-tube exchanger.
- Hot water moving up the annulus heats the upper rocks and the well becomes nearly isothermal.
- Some of the cool water, being denser than the water in the aquifer, sinks into the aquifer and is replaced by hotter water, which flows up the annulus.

Figure 45 shows well construction in competent formation (i.e., where the wellbore will stand open without a casing). An undersized casing with perforations at the lowest producing zone (usually near the bottom) and just below the static water level is installed. A packer near the top of the competent formation allows installation of an annular seal between it and the surface. When the DHE is installed and heat extracted, thermosiphoning causes cooler water inside the casing to move to the bottom, and hotter water moves up the annulus outside the casing.

Because most DHEs are used for space heating (an intermittent operation), heated rocks in the upper portion of the well store heat for the next cycle.

Where the well will not stand open without casing, a convection tube can be used. This is a pipe one-half the diameter of the casing either hung with its lower end above the well bottom and its upper end below the surface or set on the bottom with perforations at the bottom and below the static water level. If a U-bend DHE is used, it can be either inside or outside the convection tube. DHEs operate best in aquifers with a high hydraulic conductivity and that provide water movement for heat and mass transfer.

Valves

In large (>2.5 in.) pipe sizes, resilient-lined butterfly valves are preferred for geothermal applications. The lining material protects the valve body from exposure to the geothermal fluid. The rotary rather than reciprocating motion of the stem makes the valve less susceptible to leakage and build-up of scale deposits. For many direct-use applications, these valves are composed of Buna-N or EPDM seats, stainless steel shafts, and bronze or stainless steel disks. Where oil-lubricated well pumps are used, a seat material of oil-resistant material is recommended. Gate valves have been

used in some larger geothermal systems but have been subject to stem leakage and seizure. After several years of use, they are no longer capable of 100% shutoff.

Piping

Piping in geothermal systems can be divided into two broad groups: pipes used inside buildings and those used outside, typically buried. Indoor piping carrying geothermal water is usually limited to the mechanical room. Carbon steel with grooved end joining is the most common material.

For buried piping, many existing systems use some form of nonmetallic piping, particularly asbestos cement (which is no longer available) and glass fiber. With the cost of glass fiber for larger sizes (>6 in.) sometimes prohibitive, ductile iron is frequently used. Available in sizes >2 in., ductile iron offers several positive characteristics: low cost, familiarity to installation crews, and wide availability. It requires no allowances for thermal expansion if push-on fittings are used.

Most larger-diameter buried piping is preinsulated. The basic ductile iron pipe is surrounded by a layer of insulation (typically polyurethane), which is protected by an outer jacket of PVC or high-density polyethylene (HDPE).

Standard ductile iron used for municipal water systems is sometimes modified for geothermal use. The seal coat used to protect the cement lining of the pipe is not suitable for the temperature of most geothermal applications; in applications where the geothermal water is especially soft or low in pH, the cement lining should be omitted, as well. Special high-temperature gaskets (usually EPDM) are used in geothermal applications. Few problems have been encountered in using ferrous piping with low-temperature geothermal fluids unless high chloride concentration, low (<7.0) pH, or oxygen is present in the fluid. Most cases of corrosion failure have resulted from external attack by soil moisture in buried applications. For more information on piping systems for these applications, see [Chapter 12 in the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#) and ASHRAE (2013a, 2013b).

2.7 RESIDENTIAL AND COMMERCIAL BUILDING APPLICATIONS

The primary applications for direct use of geothermal energy in the residential and commercial area are space and domestic water heating. Space cooling using the absorption process is possible but rarely applied.

Space Heating

[Figure 46](#) illustrates a system that uses geothermal fluid at 170°F (Austin 1978). The geothermal fluid is used in two main equipment components for heating the buildings: (1) a plate heat exchanger that supplies energy to a closed heating loop previously heated by a natural gas boiler (the boiler remains as a standby unit) and (2) a water-to-air coil used for preheating ventilation air. In this system, proper control is crucial for economical operation.

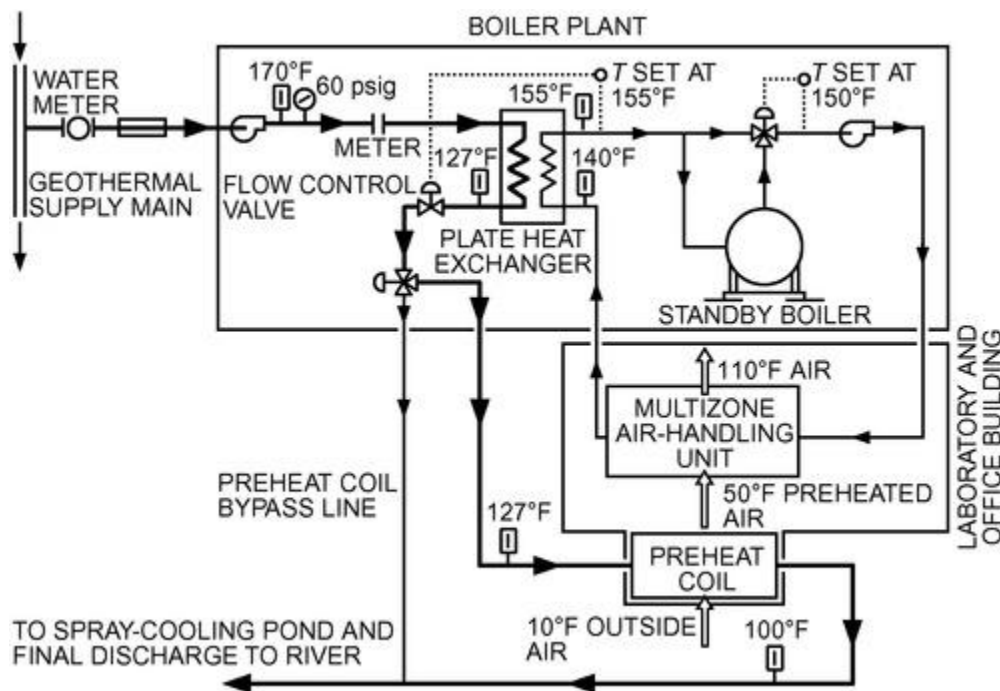


Figure 46. Heating System Schematic

The average temperature of discharged fluid is 120 to 130°F. Geothermal fluid is used directly in the preheat coils in the buildings, which would probably not be the case if the system were designed today (Lienau 1979).

[Figure 47](#) shows a geothermal district heating system that has a unique feature: its design is based on a peak load Δt of 100°F using a 190°F resource. It is of closed-loop design with central heat exchangers. The production well has an artesian shut-in pressure of 25 psi, so the system operates with no production pump for most of the year. During colder weather, a surface centrifugal pump located at the wellhead boosts the pressure.

Geothermal flow from the production well is initially controlled by a throttling valve on the supply line to the main heat exchanger, which responds to a temperature signal from the supply water on the closed-loop side of the heat exchanger. When the throttling valve reaches the full-open position, the production booster pump is enabled. The pump is controlled through a variable-speed drive that responds to the same supply-water signal as the throttling valve. The booster pump is designed for a peak flow rate of 300 gpm of 190°F water.

A few district heating systems have also been installed using an open distribution system. In this design, central heat exchangers (as in [Figure 47](#)) are eliminated and the geothermal water is delivered to individual building heat exchangers. When more than a few buildings are connected to the system, using central heat exchangers is normally more cost effective.

Terminal equipment used in geothermal systems is the same as that used in nongeothermal heating systems. However, certain types of equipment are better suited to geothermal design than others.

In many cases, buildings heated by low-temperature geothermal sources operate at lower supply water temperatures than conventional hydronic designs. Because many geothermal sources are designed to take advantage of a large Δt , proper selection of equipment for low flow and low temperature is important.

Finned-coil, forced-air systems generally function best in this low-temperature/high- Δt situation. One or two additional rows of coil depth compensate for the lower supply water temperature. Although an increased Δt affects coil circuiting, it improves controllability. This type of system should be able to use a supply water temperature as low as 110°F.

Radiant floor panels are well suited to very low water temperatures, particularly in industrial applications with little or no floor covering. In industrial settings, with a bare floor and a relatively low space-temperature requirement, the average water temperature could be as low as 95°F. For a higher space temperature and/or thick floor coverings, a higher water temperature may be required.

Baseboard convectors and similar equipment are the least capable of operating at low supply-water temperature. At 150°F average water temperatures, derating factors for this design load may be affected. This type of equipment can be operated at low temperatures from the geothermal source to provide base-load heating, with peak load supplied by a conventional boiler. Ensure the boiler does not supply a higher fraction of the load than intended by the designer.

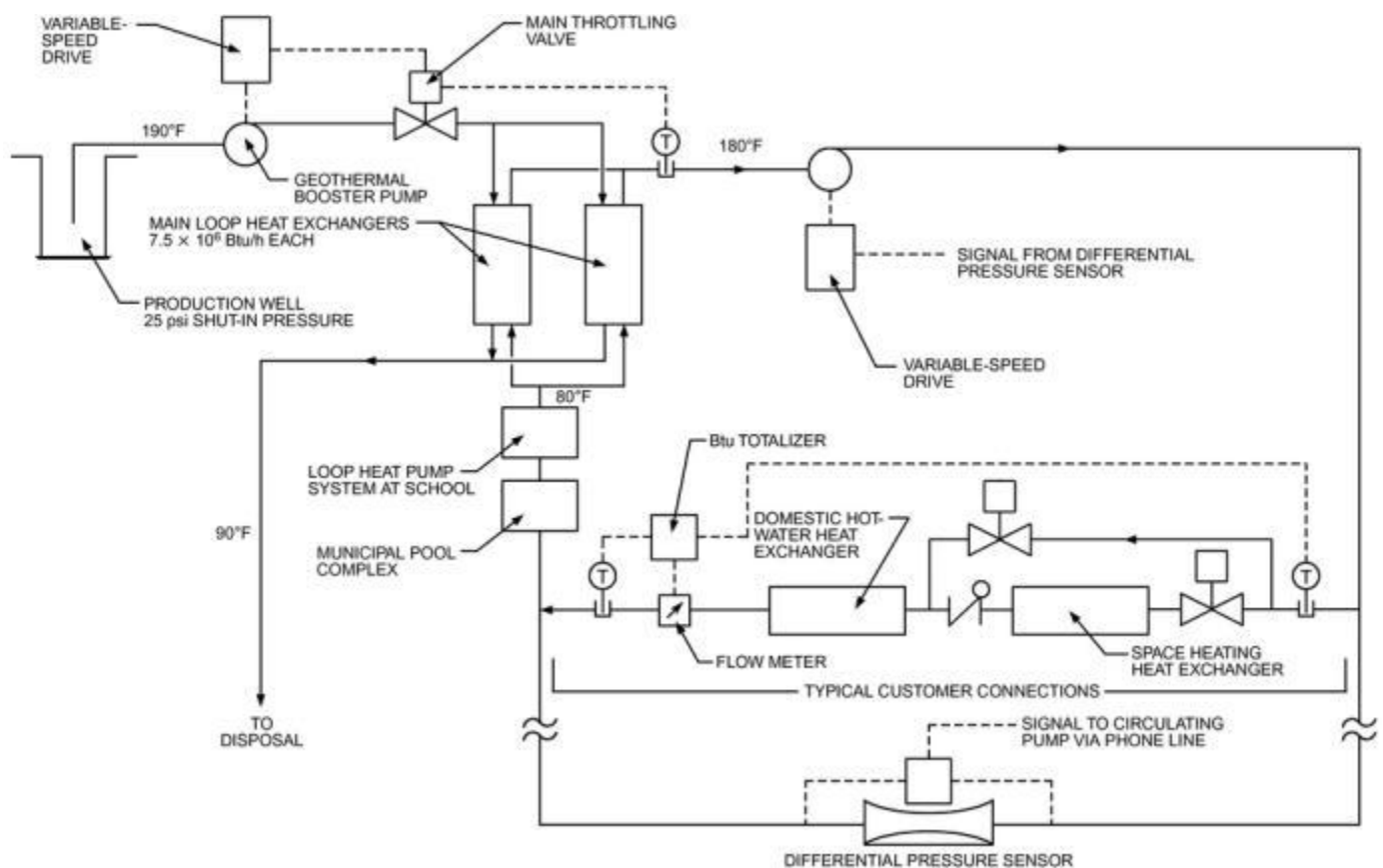


Figure 47. Closed Geothermal District Heating System (Rafferty 1989a)

Domestic Water Heating

Domestic water heating in a district space-heating system is beneficial because it increases the overall size of the energy load, energy demand density, and load factor. For those resources that cannot heat water to the required temperature, preheating is usually possible. Whenever possible, the domestic hot-water load should be placed in series with the space-heating load to reduce system flow rates and increase Δt .

Space Cooling

Geothermal energy has seldom been used for cooling, although emphasis on solar energy and waste heat has created interest in cooling with thermal energy. The absorption cycle is most often used, and lithium bromide/water absorption machines are available in a wide range of capacities. Temperature and flow requirements for absorption chillers run counter to the general design philosophy for geothermal systems: they require high supply water temperatures and a small Δt on the hot-water side. Figure 48 shows the effect of reduced supply water temperature on machine performance. The machine is rated at a 240°F input temperature, so derating factors must be applied if the machine is operated below this temperature. For example, operation at a 200°F supply water temperature results in a 50% decrease in capacity, which seriously affects the economics of absorption cooling at a low resource temperature.

Coefficient of performance (COP) is less seriously affected by reduced supply water temperature. The nominal COP of a single-stage machine at 240°F is 0.65 to 0.70; that is, for each ton of cooling output, a heat input of 12,000 Btu/h divided by 0.65, or 18,460 Btu/h, is required.

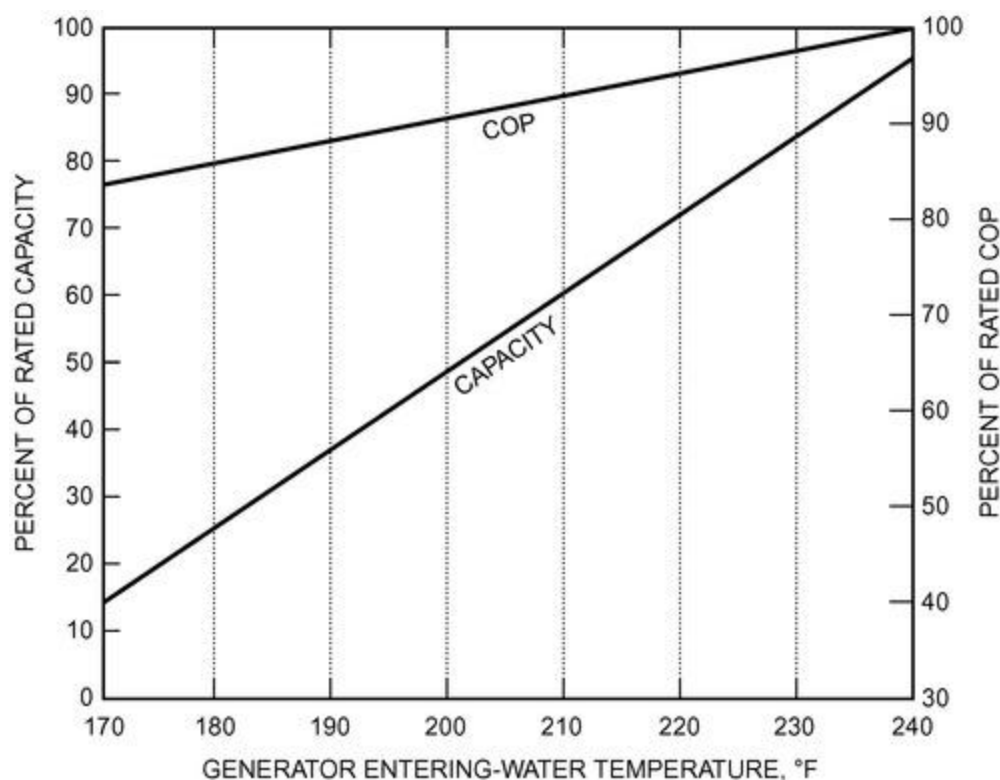


Figure 48. Typical Lithium Bromide Absorption Chiller Performance Versus Temperature (Christen 1977)

Most absorption equipment is designed for steam input (an isothermal process) to the generator section. When this equipment is operated from a hot-water source, a relatively small Δt must be used. This creates a mismatch between building flow requirements for space heating and cooling. For example, assume a 200,000 ft² building is to use a geothermal resource for heating and cooling. At 25 Btu/h · ft² and a design Δt of 40°F, the flow requirement for heating is 250 gpm. At 30 Btu/h · ft², a Δt of 15°F, and a COP of 0.65, the flow requirement for cooling is 1230 gpm.

Some small-capacity (3 to 25 ton) absorption equipment has been optimized for low-temperature operation in conjunction with solar heat. Although this equipment could be applied to geothermal resources, the prospects are questionable. Small absorption equipment generally competes with packaged direct-expansion units in this range; absorption equipment requires a great deal more mechanical auxiliary equipment for a given capacity. The cost of the chilled-water piping, pump, and coil; cooling-water piping, pump, and tower; and hot-water piping raises the capital cost of the absorption equipment substantially. Only in large sizes (>10 tons) and in areas with high electric rates and high cooling requirements (>2000 full-load hours) would this type of equipment offer an attractive investment to the owner (Rafferty 1989a).

Cascading Systems

Cascading geothermal systems have been used for centuries all over the world (Lund 2010). These systems typically use intermediate- ($195 < t < 300^{\circ}\text{F}$) and low-temperature ($t < 195^{\circ}\text{F}$) resources, depending on the first step of the cascade. As shown in [Figure 38](#), low-temperature resources are most frequently identified but are underused. Resources typically include hot/warm fluid well production, springs, ponds, shallow groundwater flows, and geologic features such as hot pots.

The nature of the resource, temperature, volume, and chemical composition determine the heat collection method and use. In ideal cases, the thermal fluid may be used directly to heat or cool, but indirect collection methods are more common. Open water systems may use a heat exchanger immersed in the fluid. Other sources need to be pumped and a plate-and-frame heat exchanger used. The goal is to use as much heat from the fluid as possible. The designer should carefully examine fluid quality to determine the best harvesting method, system material composition, and disposal of the geothermal fluid.

An example of a cascading system is a hot-water well producing 350 gpm of 155°F , low TDS, and approximately neutral pH water used for the following heat cascading temperature loads: domestic hot water at 140°F , a large spa at 104°F , a pool at 85°F , warm-water irrigation storage and a water feature to a pond at about 80 to 72°F , and finally through a lake plate heat exchanger in the pond serving GSHPs to heat and cool selected campus buildings. Another common application is to use a low-temperature cascading system for hydroponic and aquaculture in both remote and urban environments.

2.8 INDUSTRIAL APPLICATIONS

Design philosophy for the use of geothermal energy in industrial applications, including agricultural facilities, is similar to that for space conditioning. However, these applications have the potential for much more economical use of the geothermal resource, primarily because they (1) operate year-round, which gives them greater load factors than possible with space-conditioning applications; (2) do not require extensive (and expensive) distribution to dispersed energy consumers, as is common in district heating; and (3) often require various temperatures and, consequently, may be able to make greater use of a particular resource than space conditioning, which is restricted to a specific temperature. In the United States, the primary non-space-heating applications of direct-use geothermal resources are dehydration (primarily vegetables), gold mining, and aquaculture.

3. RENEWABILITY

Geothermal energy is a renewable resource (see the section on Nonrenewable and Renewable Energy Sources in [Chapter 34 of the 2021 ASHRAE Handbook—Fundamentals](#) for discussion). Quantification of the source may be required for renewable portfolio standards, utility programs, etc.; to do this, measure or calculate the electric or thermal energy that is either generated from, or avoided by, use of the geothermal resource.

Geothermal energy ultimately comes from a variety of sources, mostly the heat transfer from hotter regions below the crust and the heat of radioactive decay in the crust. Direct-use and higher-temperature geothermal resources may be considered renewable, because the heat removed is replaced by natural processes: heat is generated deep in the earth and transferred to more shallow depths. The geothermal resource must be carefully managed, however, and can eventually be depleted if used at too high of a rate.

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