GROUND-SOURCE (GEOTHERMAL) HEAT PUMPS

by John W. LUND

1. INTRODUCTION

When a geothermal resource temperature falls below the 40-50°C range, it is frequently impractical to use the fluid directly for most applications. A heat pump can be used to transfer heat from a low-temperature resource to that of a high-temperature reservoir, thus providing the higher temperature needed for space heating. The process can also be reversed by removing heat from a high-temperature resource and rejecting it to a lower temperature reservoir, thus providing cooling to a space.

Air-source heat pumps have been used for many years for both space heating and cooling; however, their efficiency is influenced by the variation in outside air temperature. When heat is most needed, the outside air is cooler, thus often requiring backup electric resistance heating during the coldest days. Similarly, cooling is most needed during the hottest days, requiring the equipment to work at low efficiencies.

Ground-source heat pumps, often referred to as geothermal heat pumps, overcome the problem of resource variations, as ground temperatures remain fairly constant throughout the year. Depending upon the soil type and moisture conditions, ground (and groundwater) temperatures experience little if any seasonal variations below about 10 m.

The ground-source or geothermal heat pumps (GSHP or GHP), thus have several advantages over air-source heat pumps. These are: (1) they consume less energy to operate, (2) they tap the earth or groundwater, a more stable energy source than air, (3) they do not require supplemental heat during extreme low outside temperature, (4) they use less refrigerant, (5) they have a simpler design and consequently less maintenance, and (6) do not require the unit to be located where it is exposed to weathering.

The main disadvantage is the higher initial capital cost, being about 30 to 50% more expensive than air source units. This is due to the extra expense and effort to bury heat exchangers in the earth or providing a well for the energy source. However, once installed, the annual cost is less over the life of the system, resulting in a net savings. The savings is due to the coefficient of performance (COP), averaging over 3 for GSHP as compared to 2 for air-source heat pumps. A corresponding improvement is obtained in the cooling mode, as measured by the energy efficiency ratio (EER). These terms are defined later in this paper.

NOMENCLATURE (Kavanaugh, undated)

Two basic types of ground-source heat pumps exist: ground-coupled and water-source. There are a variety of names for the ground-coupled heat pumps. These include ground-source heat pumps, earth-coupled heat pumps, earth energy systems, ground source systems, geothermal heat pumps, closed-loop heat pumps, and solar energy heat pumps. Much of the confusion arises from local marketing needs. Sales people may wish to connect GCHPs to renewable energy sources (solar, geothermal), dissociate them from air hear pumps (GS systems), or connect them to environmental awareness (earth energy). A generally (although not universally) recognized nomenclature is shown in Fig. 1.

Ground-coupled heat pumps are a subset of ground-source heat pumps (GSHPs). GSHPs also include groundwater and lake water heat pumps (water source). The distinguishing feature of GCHPs is that they are connected to a closed-loop network of tubing that is buried in the ground. The most common method of ground-coupling is to bury thermally-fused plastic pipe either vertically or horizontally. A water or antifreeze solution is circulated through the inside of the tubing and heat is released to or absorbed from the ground. No water enters the system from the ground. Water-to-air heat pumps are located inside the building and are connected to the water loop with a circulator pump. This type of system is referred to as a secondary fluid GCHP since there is an intermediate liquid between the refrigerant and the ground.

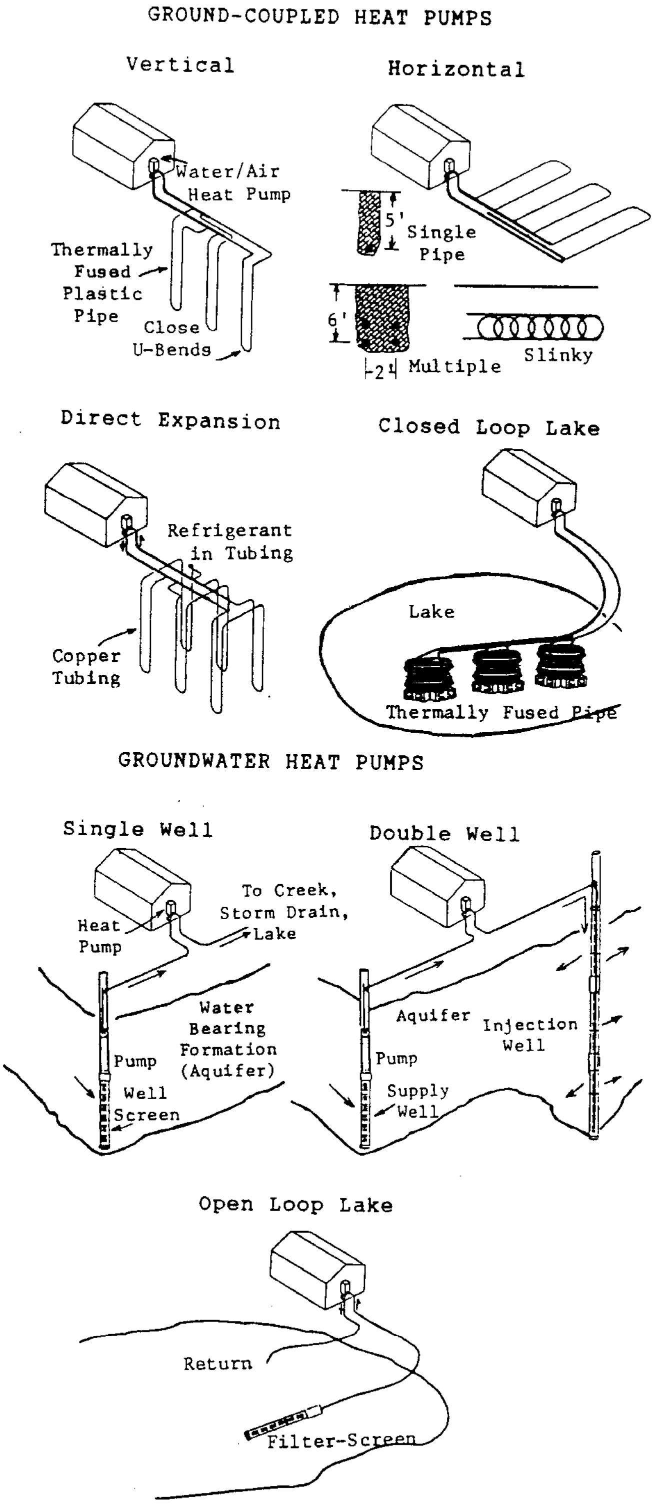

Figure 1: Ground-source heat pump types

A less frequently used system is referred to as a direct expansion (DX) GCHP. Refrigerant lines are buried in the ground in either a vertical or horizontal arrangement. Thus, the intermediate heat exchanger and fluid are eliminated. The possibility of higher efficiency than secondary fluid GCHPs does exist. However, larger charges of refrigerant are required and system reliability is compromised. Therefore, the future of DX GCHP is not clear because of environmental concerns.

A variety of ground-coupled heat exchanger designs have recently been proposed and demonstrated as cost-cutting alternatives. These include the Geo-Bag (a large plastic bag buried in the ground and filled with water), a large diameter borehole with spiral coils, and the horizontal-placed slinky coil (Fig. 2).

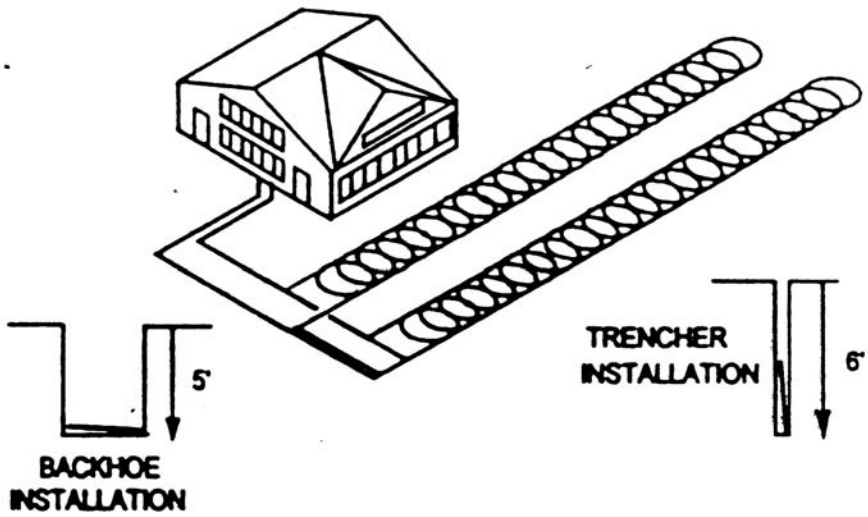

Figure 2: Slinky coil ground-coupled heat pump system

2. DETAILS ON THE COMMON GROUND-SOURCE HEAT PUMPS SYSTEMS (LUND, 1989)

Two major types exist: ground-coupled (closed loop) or water source (open loop). The ground-coupled uses a buried earth coil with circulating fluid in a closed loop of horizontal or vertical pipes to transfer thermal energy to and from the earth. The water-source uses a well or an open pond to provide an energy source or sink. Ground-coupled systems have been used in northern Europe for many years, but were not used on a commercial scale in the U.S. until 1980. Ground coupling is used where insufficient well water is available, where the quality of the well water is a problem, where drilling and casing of wells are expensive, or where disposal of well water is restricted.

In the horizontal mode of the ground-coupled system, pipes are buried in trenches spaced a minimum of 1.5 m apart and from 1.2 to 1.8 m deep. This allows for minimum thermal interference between pipes; however, this system is affected by solar radiation. Solar radiation will cause a cycling of soil temperatures, that lags in time and decreases with depth due to the insulating properties of the soil as shown in Fig. 3; however, the temperature is much more stable than for air- source units. Moist soil will have greater temperature swings than dry soil. The loops can be placed in a double layer as shown in Fig. 4. Vertical installation (Fig. 5) of the coils are used where land space is limited or trenching would disturb the surface landscape, and drilling costs are reasonable. Holes are drilled approximately 45 m deep and 4.5 to 6.0 m apart.

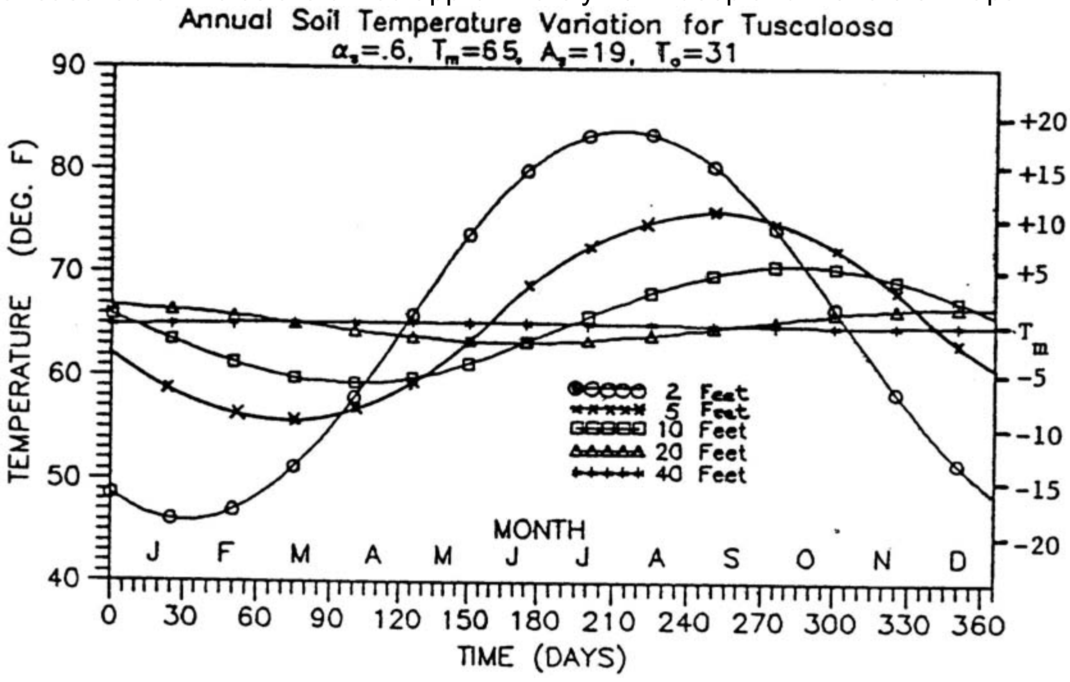

Figure 3: Depth vs. annual ground temperature variation for Tuscaloosa, Alabama

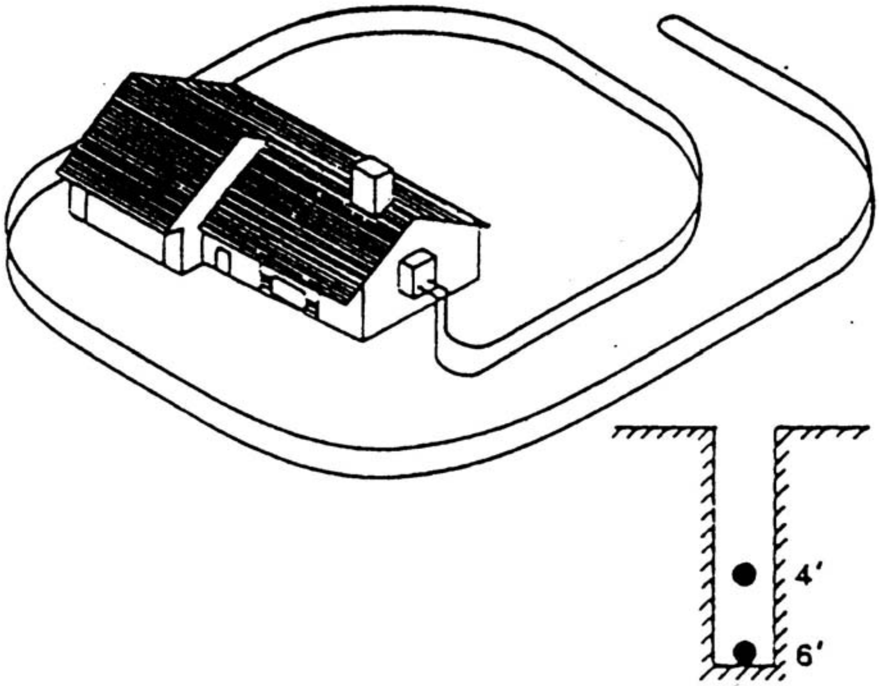

Figure 4: Two-pipe horizontal ground heat exchanger (source: Oklahoma State University):
Earth coil type: Horizontal - two-layer;
Water flow: Series;
Typical pipe size: 3.8-5.1 cm;
Practical length
Double loop: 18.2-26.0 m/kW;
Single loop: 36.4-52.0 m/kW;
Burial depth: 1.2-1.8 m

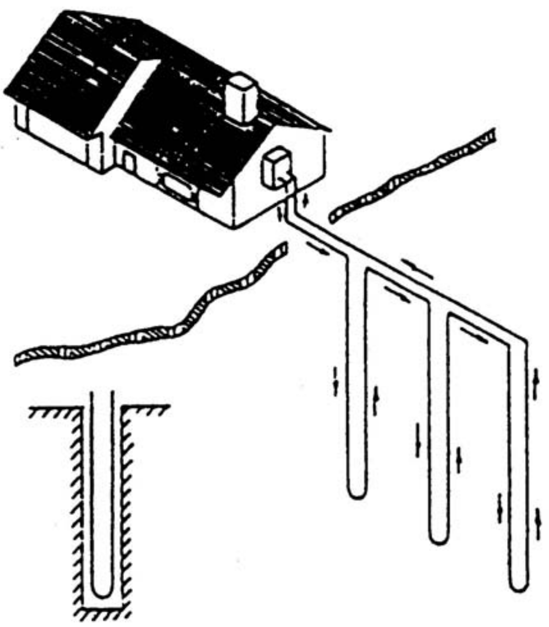

Figure 5: Series vertical ground heat exchanger (source: Oklahoma State University):
Earth coil type: Vertical - Single U-Bend;
Water flow: Series;
Pipe sizes: 2.5, 3.8 & 5.1 cm;
Bore length: 14.3-17.3 m/kW;
Pipe length (single pipe): 19.9-34.7 m/kW

Computer programs have been developed (Dexheimer, 1985) to calculate the length of horizontal earth coils for heating and cooling. Polyethylene pipes are the most popular in use, and along with socket-fusion joining, are usually guaranteed for over 50 years.

Whereas, horizontal loops are affected by solar radiation, rain and wind, the vertical loops are controlled by the mean-annual temperature of the area and the geothermal gradient and thus, have a more stable temperature environment. Water wells are usually used where one is already available, such as for domestic water supply. Normally, a minimum diameter of 15 cm and a production of about 3.23 L/min/kW of heat pump capacity is required. 10.5 kW, a typical residential load, requires about 34.1 L/min. The 15 cm diameter well casing is required to place the pump and return line (Fig. 6). The fluid can either be returned to the well by the return line, placed in an injection well, or disposed on the surface such as for irrigation. Pipes have also been anchored to the bottom of surface ponds (minimum depth of 1.8 m); however, the heating and cooling capacities are affected by solar radiation and other surface weather factors similar to the horizontal loops. Installation is cheaper and heat transfer is more efficient; however, ponds do not maintain a constant temperature as wells do and the pipes are more vulnerable to accidental damage.

3. HEAT PUMP OPERATION (KAVANAUGH, 1991)

The operation of the heat pump unit is the same for air-source and ground-source configuration. The main difference is that the air-source requires an outside unit (accumulator and fan) which may frost up in cold weather, requiring frequent defrosting. They also require a backup heating source (electric or gas) when outside temperatures are too low for efficient operation. The operation and cycle in both cooling and heating mode of the heat pump are shown in Figs. 7 and 8 (Oklahoma State University, 1988).

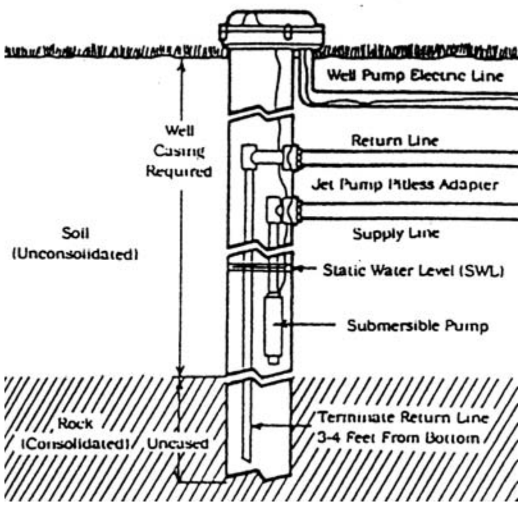

Figure 6: Cross-section view of geothermal well (source: Water-Source Heat Pump Book)

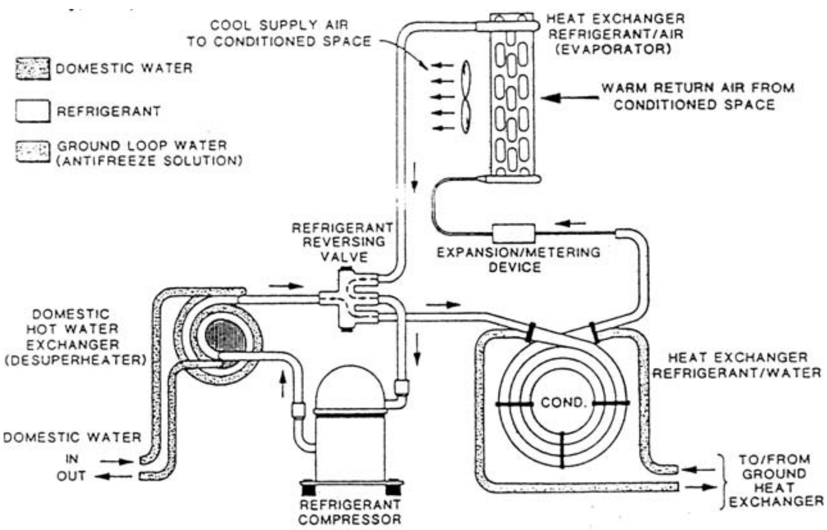

Figure 7; Cooling cycle (source: Oklahoma State University)

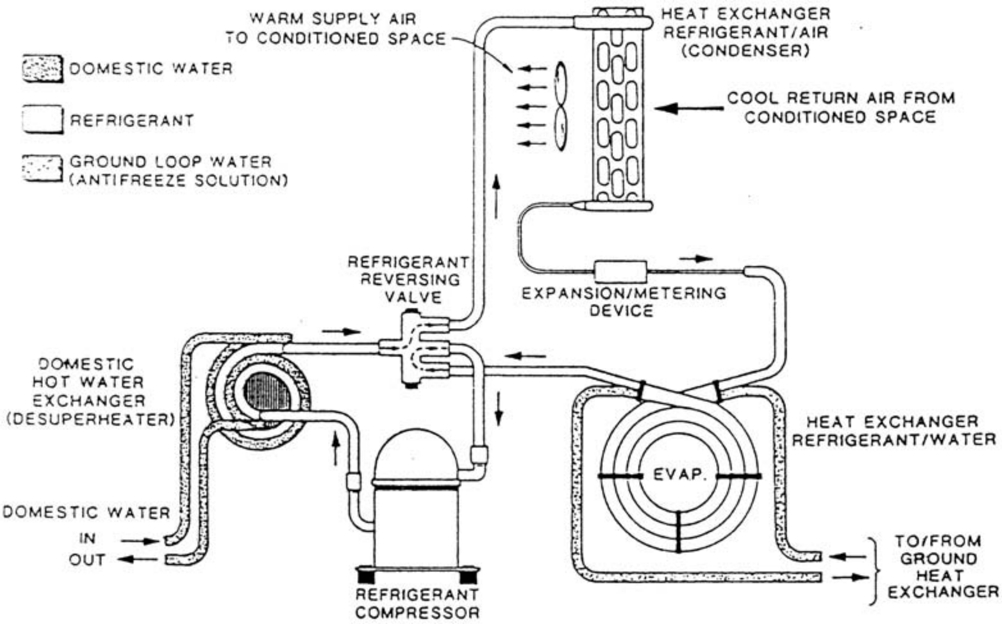

Figure 8: Heating cycle (source: Oklahoma State University)

As seen from these figures, the basic components of a standard heat pump are an electric motor-driven compressor, a reversing valve, an expansion device, and two heat exchangers. A desuperheater can be added to heat domestic hot water as shown on the left side of the figures. One heat exchanger transfers heat between the heat pump and the environment, and the second transfers heat to and from the interior of the building, referred to as the condenser or evaporator (depending on which mode is used).

Refrigerant enters the compressor shell as a low-temperature, low-pressure gas. It passes around the motor and is heated before entering the intake of the compression chambers. The compression process elevates both the pressure and temperature of the refrigerant gas. In the cooling mode (Fig. 7), this gas enters the reversing valve and is routed to the heat exchanger in contact with the environment (ground heat exchanger). Since the gas is at a high temperature, relatively cool air or water from the environment (geothermal source) can be used to remove heat from the refrigerant in the heat exchanger. Removal of heat results in the cooling and <u>condensing</u> of the refrigerant. Pressure loss is usually small in the condenser; therefore, the refrigerant exits the condenser as a liquid with a temperature slightly above the environment's.

The liquid then experiences a drop in pressure across the restriction in the expansion device. This causes a rapid decrease in temperature. The temperature inside the building is much

warmer than the refrigerant entering the indoor heat exchanger. Therefore, the liquid is evaporated, and in the process, heat is removed from the building air in the <u>evaporator</u>. Thus, we have the desired cooling effect. The evaporated gas is then passed through the reversing valve before returning to the compressor.

In the heating mode (Fig. 8), the solenoid in the reversing valve is switched so that the hot compressor discharge gas is routed to the indoor heat exchanger. This exchanger is now used to <u>condense</u> the hot refrigerant with the relatively cool indoor air; therefore, the desired heating effect is carried out on the building. The condensed liquid enters the expansion device in a reverse direction. The pressure loss results in a temperature decline so that the environment (ground) transfers heat to the cold refrigerant. This causes the refrigerant to evaporate. The low-temperature, low-pressure gas returns to the compressor through ports in the reversing valve.

Actual heat pumps may have additional components such as a multiple expansion devices, fans, pumps, additional heat exchangers, auxiliary heat sources, accumulators, and control and safety mechanisms. However, the basic means of “pumping heat" between the building and the environment (geothermal source) is essentially the process shown in Figures 7 and 8.

4. EXAMPLES OF GROUND- AND WATER-SOURCE HEAT PUMP TYPES (KAVANAUGH, 1991)

Figure 9 shows four common ground and water-source heat pumps. The home on the far left utilizes a vertical closed-loop and ground-coupled heat pump. Water is circulated through a water-to-refrigerant heat exchanger (condenser in cooling, evaporator in heating) in the heat pump. Upon leaving out of the house and into underground headers buried 0.6-1.2 m below the surface. In this installation, the flow is split into three vertical U-Bend ground couplings. Bore depth is typically 13.0-21.7 m/kW of heat pump, pipe size of 1.9-3.8 cm, and bore diameter of 10-15cm. Obviously, the driller in this installation stopped when the rock strata was encountered. Deeper or more shallow bore depths are possible as long as the required bore per kW requirement is maintained. Note also that water flow is split into three circuits. This is often necessary to minimize water pump size.

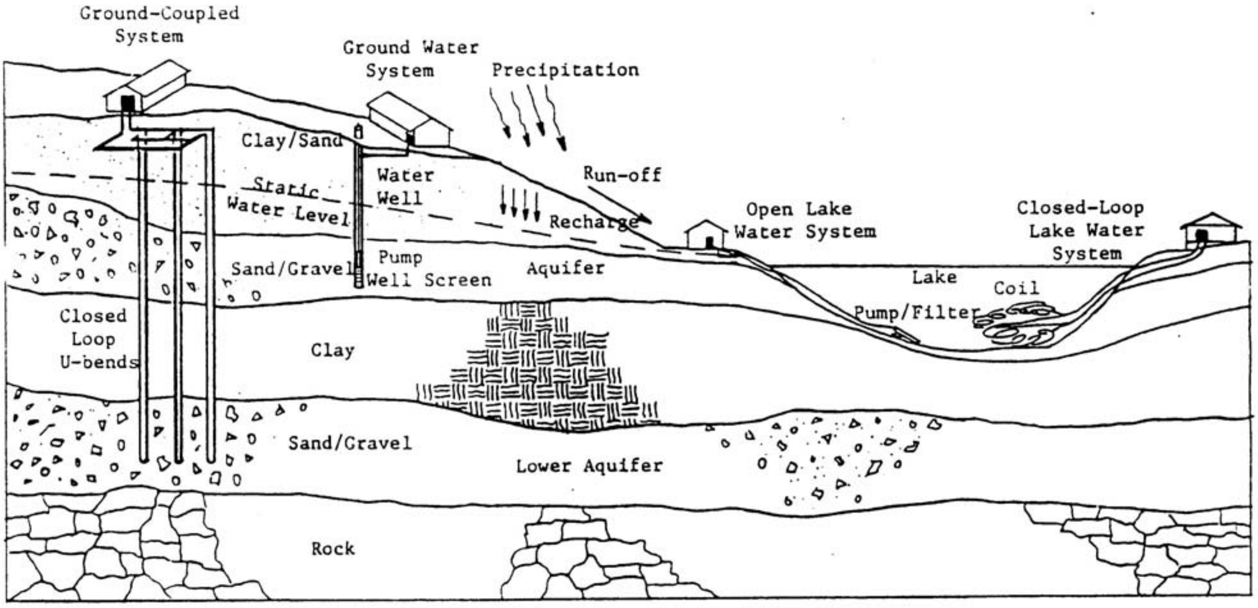

Figure 9: Ground-coupled, groundwater and two lake-water heat pump systems in a typical hydro-geological formation.

The next house is heated and cooled by a groundwater heat pump system. A well (10 cm casing minimum) is drilled into the shallow aquifer. A well screen is placed on the bottom section of the casing through which groundwater is extracted. A submersible pump located slightly above the screen delivers water to the heat pump in the house. In this installation, the water is obviously disposed of at the surface; however, it could also be reinjected if aquifer water levels are critical. Two important requirements in these systems are a clean, well-developed well, and that the pump should be located about a meter below the lowest seasonal static water level. It would not be uncommon to drill to a deeper aquifer; although, the shallow aquifer is preferred in terms of operating and first cost.

Lakes can be used in two manners. The home on the left shore utilizes an open-loop heat pump system. Water is pumped from a submersible pump near (but not on) the lake bottom and through the heat pump, and is returned near the surface of the lake. Above surface pumps can also be used. They require greater power input, freeze precautions and some method of preventing loss of suction during off-periods.

The home on the right lake front is heated and cooled with a closed-loop heat pump. Much like the ground-coupled system, water (or a water/antifreeze mixture) is circulated through a copper or thermally-fused plastic pipe coil and the heat pump. These systems do not require water filtration like the open systems. However, they often do require freeze protection and protection from damage, and performance improvements are moderate in shallow lakes.

4.1 Specific Examples of GCHP Use

Worldwide, GSHP account for 12% of the geothermal energy used for direct applications, amounting to approximately 16,500 TJ (4,580 GWh) annually. Present estimates indicate that there are 150,000 groundwater and 250,000 ground-coupled (55% vertical) heat pump installations in the USA (updated from Ellis, 1989). The annual growth rate is estimated at 10%. Ground-source heat pumps account for 63% of the geothermal direct use in the USA, amounting to 12,000 TJ (3,340 GWh) annually. A typical installation, which would be for a single-family residence, consists of a 10.5 kW (3 tons) using about 30 L/min with a 6°C temperature drop in the circulating fluid. This would shave about 5 kW off winter-peak heating demand and about 2.5 kW from summer demand. Thus, 200,000 homes using GSHPs would avoid constructing a new 1,000 MW power plant. Although the incremental cost of the ground-coupled closed loop adds about US$ 3,000 to the cost of a residential heating system, payback occurs in 3-5 years from money saved on utility bills. Currently, the main GCHP uses in the USA is in the mid-western and southeastern states, where many utilities were offering rebates of US$ 500-2000 to homeowners to install GSHP in order to take advantage of the peak shaving (by increasing load leveling for the utility, referred to as demand side management) (Lund, 1988; Lienau and Lund, 1992).

The largest GSHP installation in the United States is the Galt House East Hotel and Waterfront Office Building in Louisville, Kentucky (Pinckley, 1995). Heating and air conditioning is provided for 600 hotel rooms, 100 apartments, and 89,000 square meters of office space, totaling 161,650 square meters. The system can extract 177 L/s of groundwater from four wells at 14°C and can either remove energy from the well water for heating or add heat to the well water from the air conditioning. The water is then discharged into a storm-water system. The system provides 15.8 MW of cooling and 19.6 MW of heating. The hotel complex energy use is approximately 53% of a similar non-GSHP system in an adjacent unit, for a monthly savings of approximately US $25,000. The emission of CO_2, SO_2, HO_x and particulates are also reduced.

There have also been increased utilization of GSHP in Europe, especially in Germany, Austria and Switzerland. In Switzerland, more than 20,000 borehole heat exchangers (BHE) have been installed and have been operating reliably for decades (Rybach and Sanner, 1999). A similar number are also installed in Germany. A typical single-dwelling house has a capacity demand of about 10 kW; however, the BHE system is 30 to 40% higher in cost in comparison with a conventional oil-fired- system. Environmental awareness, enforced by a governmental subsidy (US$ 200 per heat pump kW), is the main incentive for the BHE installations in Switzerland.

5. EQUIPMENT (KAVANAUGH AND RAFFERTY, 1991)

In the commercially available size range, equipment is in two basic configurations: positive displacement and centrifugal. Centrifugal machines are used for the largest applications with positive displacement equipment for smaller capacities. The following sections briefly discuss each of these types of equipment.

5.1 Positive Displacement

Reciprocating compressor heat pumps, the most common positive displacement type, are available as standard units in sizes generally below 0.88 MW heating output (McQuay, 1986). This equipment employs a 1, 4, 6, or 8-cylinder compressor on smaller equipment and multiple 4 or 6-cylinder compressors on larger units (Carrier Corp., 1981). These units are also used for the smaller residential applications of around 10.5 kW.

Capacity control is accomplished by suction cut-off type cylinder unloading down to –15-20% capacity below which hot gas by-pass must be employed. As a result, it is important, particularly for space heating applications, that equipment selection considers off-peak operation. The number of control steps is dependent upon the number of compressor cylinders, with 4-step control available on the smaller units and up to 8-step control on the large units.

Table 1 illustrates the off-peak performance of small, medium and large (with respect to the 0.88 MW capacity) reciprocating heat pumps.

The increase in efficiency at part load is because of the nature of capacity control employed by the manufacturer from which data were taken. This is a result of a special unloading arrangement and the part load operation of two separate refrigerant circuits on the heat pump. This increases the amount of heat transfer area available in the evaporator and condenser relative to load requirements, thus increasing efficiency.

The refrigerant employed is a function of the temperatures between which the machine is working. Table 2 presents a summary of refrigerant temperature limitations.

Table 1. Off-Peak Performance of Reciprocating Heat Pumps (McQuay, 1986)

Small		Medium		Large	
Capacity (%)	kW (%)	Capacity (%)	kW (%)	Capacity (%)	kW (%)
100	100	100	100	100	100
86	81	92	88	94	90
68	60	84	76	87	80
24	20	72	64	75	62
-	-	60	54	59	46
-	-	31	27	51	37
-	-	-	-	42	29
-	-	-	-	23	14

Evaporators are the shell-and-tube type with water generally on the shell side. However, one major manufacturer produces equipment with water on the tube side. Condensers are also shell-and-tube with water on the tube side. Reciprocating machines do not generally include a separate liquid sub-cooling heat exchanger, though sub-cooling is addressed in condenser circuitry (Carrier Corp., 1987).

Table 2: Reciprocating Heat Pump Refrigerant Temperature Limitations (McQuay, 1986; Carrier Corp., 1981)

Refrigerant	Maximum Condenser Leaving Water Temperature	Minimum/Maximum Evaporation Leaving Water Temperature
	(°C)	(°C)
R-22	54	6/32
R-500	66	4/38
R-12	77	4/38
R-114	104	21/49

Packaged reciprocating heat pumps are supplied from the factory with all safety and operating controls for the machine including, in most cases, compressor starters. The machines need only to be interfaced with system controls and a power source.

Other positive displacement models are rotary (rolling piston, rotary valve, single screw and twin screw), and orbital (scroll and trochoidal). Their performance is similar to the reciprocating compressor. See the ASHRAE Equipment Guide for more details.

5.2 Centrifugal

Centrifugal heat pumps are available in capacities ranging from – 0.29-7.32 MW in a single unit (McQuay, 1983). The equipment features a single or dual compressor, depending upon the size. One large manufacturer of this equipment in the U.S. employs a high-speed wheel, driven by a hermetically-sealed squirrel cage motor through a single helical-gear couple. Motor cooling is provided by controlled liquid refrigerant injection. A second manufacturer employs a 3-stage compressor operating at motor speed. Refrigerant temperature limitations are similar to those shown in Table 2 for reciprocating equipment.

Stable part-load operation is maintained by inlet guide vanes with the assistance of an adjustable diffuser block at the wheel exhaust (McQuay, 1983). Construction of the balance of the machine is similar to that of the reciprocating machine with the exception that the source water in the evaporator flows through the tubes, rather than the shell as in reciprocating equipment. This configuration permits the use of alternate tube construction materials to accommodate (without the use of a heat exchanger loop) aggressive fluids in certain applications.

6. PERFORMANCE (LIENAU, ET AL., 1995)

The energy performance of a GSHP system can be influenced by three primary factors: the heat pump machine, the circulating pump or well pumps, and the ground-coupling or groundwater characteristics.

The heat pump is the largest single energy consumer in the system. Its performance is a function of two things: the rated efficiency of the machine and the water temperature produced by the ground-coupling (either in the heating or cooling mode). The most important strategy in assembling an efficient system is to start with an efficient heat pump. It is difficult and expensive to enlarge a ground-coupling to improve the performance of an inefficient heat pump.

Water-source heat pumps are currently rated under one of three standards by the American Refrigeration Institute (ARI). These standards are ARI-320, ARI-325, and ARI-330. The standard intended for ground-coupled systems is ARI-330 entitled "Ground Source Closed Loop Heat Pump Equipment." Under the standard, ratings for cooling EER and heating COP are published. It's important to consider that these are single-point ratings rather than seasonal values as in the case of the air-source equipment. Cooling EER values are based on an inlet water temperature of 25°C. Heating COP values are based on a heating inlet temperature of 0°C. These values are characterizations of a northern climate.

The current ARI directory contains equipment with EER ratings of less than 10 to a high of 18.6. COP values range from 2.8 to 3.6. It is evident that there is a wide range of equipment performance at the standard rating conditions. Based on these values, it is evident that the performance of the equipment can vary by as much as 100% according to the quality of heat pump purchased.

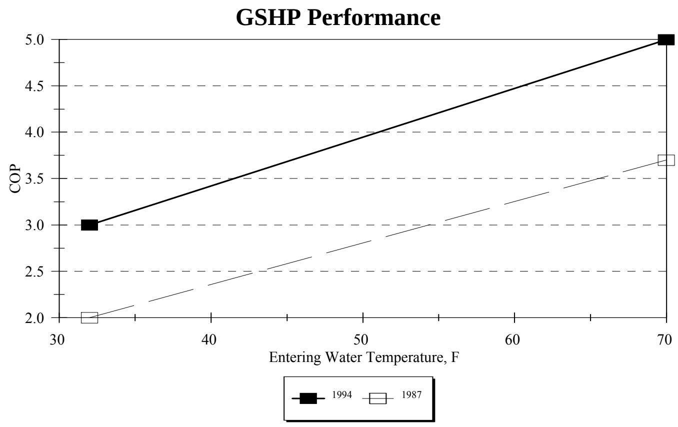

Figure 10: GSHP performance improvements from 1987 to 1994 for heating mode

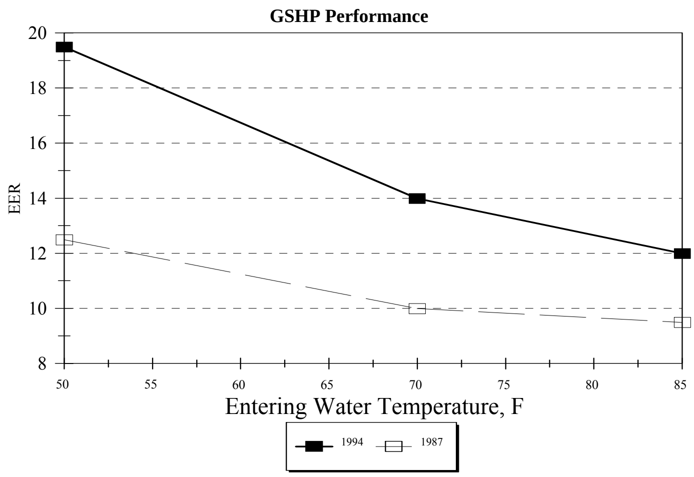

Figure 11: GSHP performance improvements from 1987 to 1994 for cooling mode

In recent years, there has been a substantial increase in the efficiency of GSHP equipment. Based on performance reported in the ARI directory for 1987 and 1994, the increase in EER ranges from 26 to 56 percent, and in COP from 35 to 50 percent depending on the entering water temperature. Figures 10 and 11 show this increase in performance for a typical machine based on average values of EER and COP as a function of entering water temperature. Based on improvements in performance of GSHPs from 1987 to 1994, the date of a GSHP installation should be noted.

The actual performance of the equipment is a function of the water temperature produced by the ground-coupling. The values discussed above are based on standard rating conditions (25°C cooling and 0°C heating). The actual temperatures are a function of the local ground temperatures and the design of the ground-coupling. For example, in a region where the local ground temperature is 16°C and the ground loop is designed for the customary -7 to -4°C aboveground temperature, a heat pump rated at an EER of 16.8 would actually operate at an EER of 14.2 under peak-load conditions. A poorly designed loop, which forces the unit to operate at -1°C aboveground temperature, would reduce the value to less than 13.0. These examples are for cooling operation which is the dominant load in commercial applications. The same relationship holds for heating operations, however.

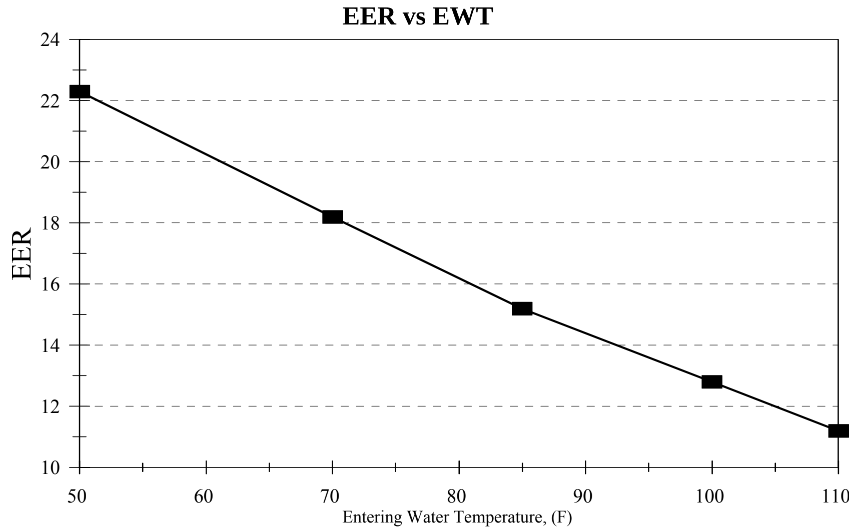

Figure 12: EER for a 10.5 kW GSHP

Figures 12 and 13 show EER and COP as a function of inlet temperatures for a 10.5 kW machine designed for ground-coupled systems suggests the following guidelines for pumping power for commercial ground-coupled systems.

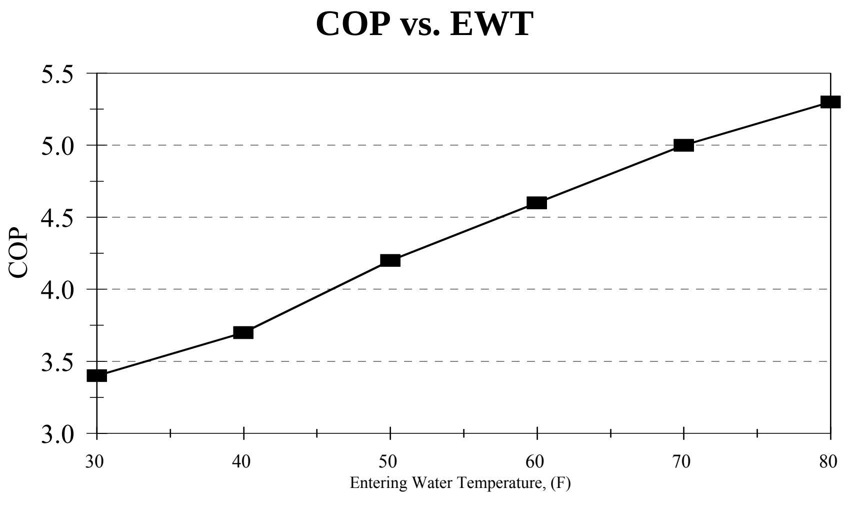

Figure 13: COP for a 10.5 kW GSHP

The system energy performance is also influenced by the pumping energy required to circulate the fluid through the heat pump and the ground loop. One author (Kavanaugh, 1991) in the design of ground-coupled systems suggests the following guidelines for pumping power for commercial ground-coupled systems:

1. Efficient systems: <50 watts/ton (<14 watts/kW)
2. Acceptable systems: 50-100 watts/ton (14-28 watts/kW)
3. Inefficient: >100 watts/ton (>28 watts/kW)

To put these values into perspective, consider an office building with a 175 kW cooling load and heat pump units selected to operate at an EER of 14 under peak conditions.

With an efficient circulating pump design (10 watts/kW}, the energy demand of the circulating pump would amount to 175 kW *x* 10 watts/kW = 1750 watts. Combining the pump demand with the heat pump unit demand results in a system EER of 13.5.

The same building and equipment coupled to a poorly designed pumping system consuming 34 watts (6,000 watts pumping power) per kW would yield a system EER of only 12.2; thus, compromising the premium paid for the higher-efficiency equipment. As indicated above, coupling this system to an inadequate ground-coupling could easily reduce the system EER to between 10 and 11.

In summary, it is necessary when evaluating a ground-coupled system to consider the efficiency of the machine, the adequacy of the ground-coupling and the nature of the pumping design to fully understand the efficiency of the system.

6.1 Basic formulas

Energy Efficiency Ratio (cooling mode):

$$EER = \frac{Q_c}{Q_e} = \frac{coolingc\ apacity\ (kW)}{electric\ power\ input\ (compressor)\ (kW)}$$

Coefficient of Performance (heating mode):

$$COP = \frac{Q_h}{Q_e} = \frac{heating\ capacity\ (kW)}{electric\ power\ input\ (compressor)\ (kW)}$$

Usually the heat extraction rate for heating is less than the heat rejection for cooling; thus, the GCHP should be designed to accommodate the larger value.

7. HEAT PUMP UNITS (KAVANAUGH, 1991)

Currently in the U.S., the water-source heat pump unit most widely used is the packaged water-to-air system. Split systems, air-to-water and water-to-water are offered on a more limited scale. Component variations in water-to-air units occur primarily in the type of expansion device and water-to-refrigerant coils. Before 1975, most units had copper (or copper-nickel) tube-in-tube coils with capillary tube expansion devices that are intended for use with groundwater with temperatures above 13°C. The market today includes units capable of handling inlet solution temperatures between -4 and 38°C. This is accomplished with better heat exchangers, expansion devices, and compressors. Two commonly used water-to-refrigerant exchangers are modified tube-in-shell (water in tube side) and coaxial tube-in-tube (water on inner tube refrigerant side in the annulus). Both have extended surfaces on the refrigerant side to compensate for the lower film coefficients. Manufacturers are selecting heat exchangers with lower water-side pressure losses to minimize pumping requirements.

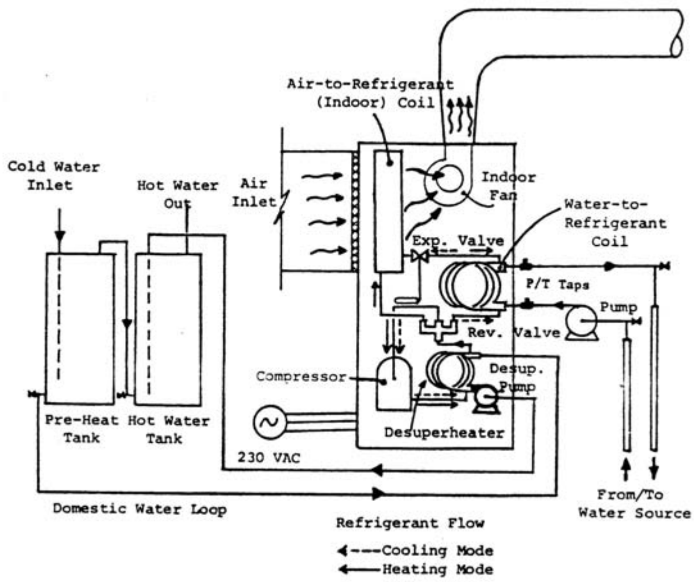

Figure 14: Typical water-to-air heat pump arrangement

Expansion valves permit a much wider acceptable range of refrigerant evaporation and condensation temperatures. This device is especially suited to ground-coupled, lake water, and closed-loop water systems in which temperature fluctuations are experienced.

In hot, humid climates, the addition of a heat exchanger in the high side of the refrigerant loop for heating domestic water is almost always recommended. This device is typically a desuperheater that uses compressor heat to generate hot water either in the cooling mode or with excess heating capacity (available in southern climates) in the heating mode. Units are now available that have larger heat exchangers and control mechanisms that permit the full condensing capacity of the refrigerant circuit to be used for heating water.

Figures 14 and 15 show a typical arrangement of a packaged water-to-air heat pump. The desuperheater water heater and pump are typically available either as part of the package or as a field-installed option. The pump for the primary water-to-refrigerant coil is usually not part of the package because its size and type vary significantly.

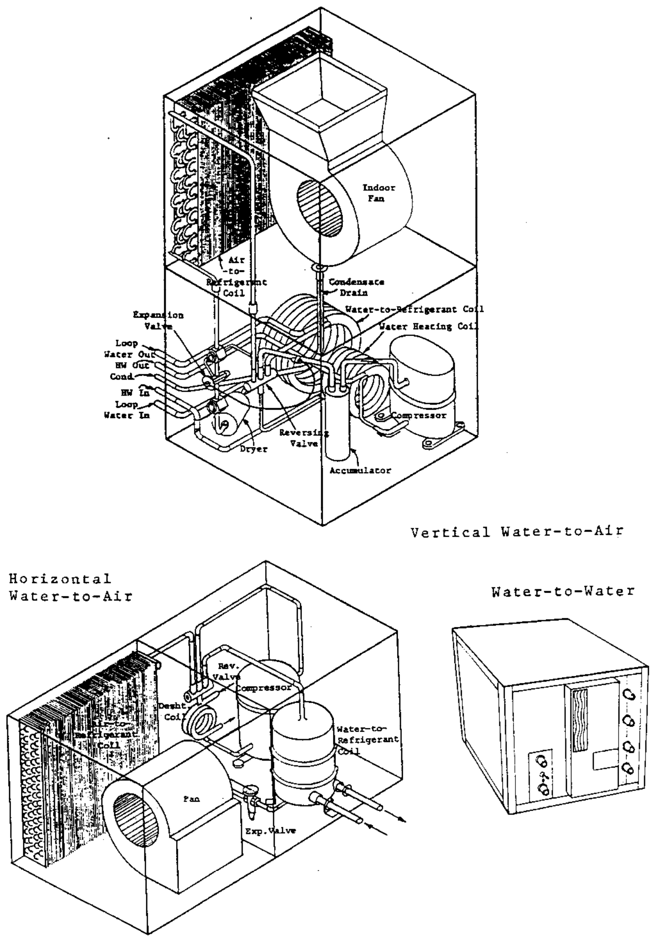

Figure 15: Water-source heat pump packages

7.1 Unit performance

Not all "water-source" heat pumps are alike. Some are optimized for heating in colder climates and some are not intended for low water temperatures. Many have quality, well-designed components; but, some contain "bottom-of-the-line" equipment. The quality of the various units can not be elaborated upon in this paper. However, the rated performance and resulting characteristics can be discussed. Table 3 is included to show the wide variation in the performance of four similarly sized water-to-air heat pumps. Total cooling (TC) and unit energy efficiency ratios (EER) are given at two different entering water temperatures (EWT). Note that the EERs are comparable at 29°C (85°F) for the non-scroll compressor models; but, Brand X has a substantially lower value at 18°C (65°F) EWT. The most recent advances have come with the use of scroll compressors (Brand C). They have outstanding heating and cooling efficiencies. However, work remains to be done in order to enhance heating capacities with lower entering water temperatures.

The differences are even more important in the heating mode. Heating capacity in 10^3 Mbtu/hr (–MJ/hr) and COP are shown for two values of EWT. Note that Brand A has substantially greater capacity and COP with a 7°C (45°F) EWT than either Brands B or X. While Brand X performance may be acceptable with groundwater, its performance will be poor with a ground-coupled or lake water system. Brand C costs more; but, the improvements in cooling efficiency warrant use in cooling-dominated climates. In heating-dominated climates, Brand A may be a better choice.

Table 3: Water-to-Air Heat Pump Capacities and Efficiencies 9 gpm, 1200 cfm, 80°/67°F EAT (34 L/min, 34 m^3/min, 27°/19°C)

	Cooling				Heating				Water coil head loss
	TC (1000 Btu/h)		EER*		TH (1000 Btu/h)		COP*		
EWT (°F)	65	85	65	85	45	65	45	65	(ft H_2O)
Brand A	42	37	14.6	11.2	43	54	4.1	4.5	14
Brand B	38	34	14.6	11.5	32	44	3.3	3.9	15
Brand C (scroll)	42	39	18.7	14.8	34	43	4.1	4.7	7.4
Brand X	37	36	13.0	11.3	21	29	2.3	2.9	20

* If it includes compressor and fan power, deduct 6 to 8% for closed-loop system efficiency and 15 to 25% for open-loop system efficiency

Another important aspect of water-source heat pumps is pressure drop (or head loss) across the water-to-refrigerant coil. When this value is high, additional or larger pumps will be required. The recommended 6 to 8 percent efficiency penalty may substantially increase in closed-loop systems. Brands A and B have marginally high losses and Brand X's is unacceptable. Brand C has very good head loss.

8. COMMERCIAL (LARGE-SCALE) GROUND-SOURCE HEAT PUMP SYSTEMS (RAFFERTY AND KNIPE, 1988; RAFFERTY, 1995; KAVANAUGH & RAFFERTY, 1997)

Unitary ground-source heat pump systems for commercial buildings can be installed in a variety of configurations. The oldest and, until recently, most widely used approach was the groundwater system. In this design (Fig. 16), groundwater from a well or wells is delivered to a heat exchanger installed in the heat pump loop. After passing through the heat exchanger (where it absorbs heat from or delivers heat to the loop), the groundwater is disposed of on the surface or in an injection well. The use of an injection well is desirable in order to conserve the groundwater resource.

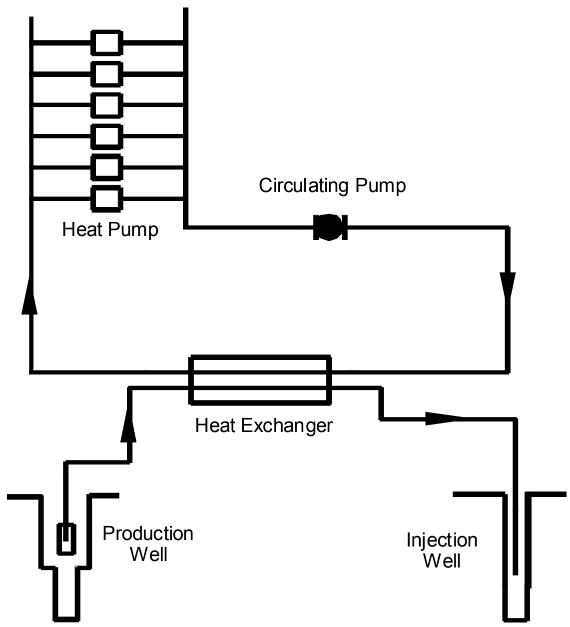

Figure 16: Groundwater heat pump system

A second and increasingly popular design is the ground-coupled heat pump system. In this approach (Fig. 17), a closed loop of buried piping is connected to the building loop. For most larger commercial applications, the buried piping is installed in a grid of vertical boreholes 30-90 m deep. Heat pump loop water is circulated through the buried piping network absorbing heat from or delivering heat to the soil. The quantity of buried piping varies with climate, soil properties and building characteristics, but is generally in the range of 13-22 m/kW of system capacity. Borehole length requirements are almost always dictated by heat rejection (cooling mode) duty for commercial buildings.

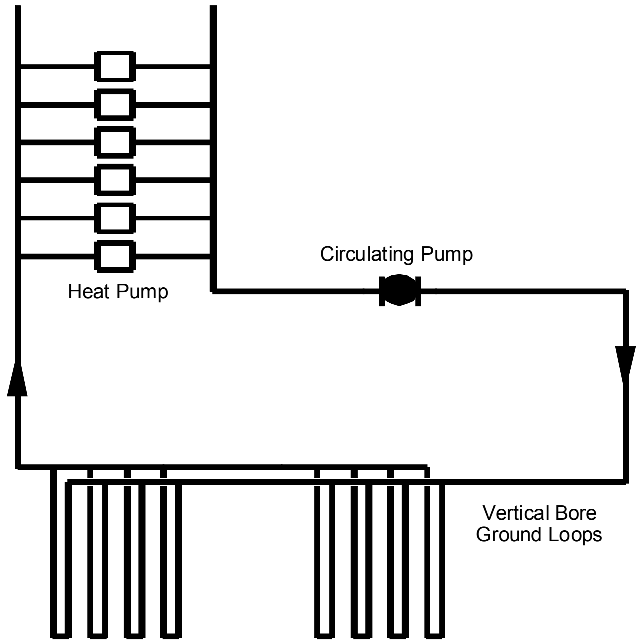

Figure 17: Ground-coupled heat pump system

A third design for ground-source systems in commercial buildings is the "hybrid" system. This approach (Fig. 18) may also be considered a variation of the ground-coupled design. Due to the high cost associated with installing a ground loop to meet the peak cooling load, the hybrid system includes a cooling tower. The use of the tower allows the designer to size the ground loop for the heating load and use it in combination with the tower to meet the peak cooling load. The tower preserves some of the energy efficiency of the system, but reduces the capital cost associated with the ground loop installation.

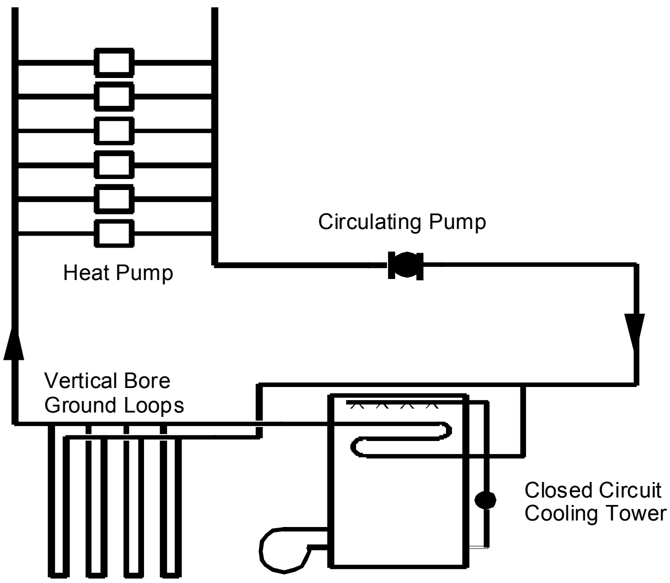

Figure 18: Hybrid ground-coupled heat pump system

In addition to the three designs discussed above, ground-source systems can also be installed using lake water, standing column wells and horizontal ground-coupled approaches. The first three schemes have wide use and broad potential application.

In the USA, it is common in the ground-source heat pump industry to refer to the costs for the ground-source portion of the system on cost per ton (3.5 kW) basis. In keeping with this practice, most cost data presented is expressed in terms of cost per ton. It is important to note, however, that the cost per ton refers to the actual load imposed on the ground-source portion of the system. This is not the same as the installed capacity of the equipment. Due to load diversity, the peak load imposed upon the heat rejection equipment is always less than the total installed capacity. The load used for cost calculations is frequently referred to by engineers as the block load.

8.1 Results

Figure 19 presents the results for the 16°C (60°F) groundwater case assuming the use of a single production/ injection well pair to serve the system. The four curves shown indicate cost (in $/ton) for four different groundwater well depths: 60, 120, 180 and 240 m (200, 400, 600 and 800 ft). In all cases, the values shown include costs for the production wells, well flow testing, production well pump, pump variable-speed drive, buried piping for transport of the groundwater to the building, heat exchanger to isolate the groundwater from the building loop, heat exchanger controls, injection well, injection well flow testing, and a 15% contingency factor. As indicated, the depth requirement for the wells has a substantial impact upon the installed cost. In addition, the unit cost for small systems (50-100 tons [176-352 kW]) is often higher by a factor of 3 compared to costs for larger systems (300-500 tons [1055-1758 kW]).

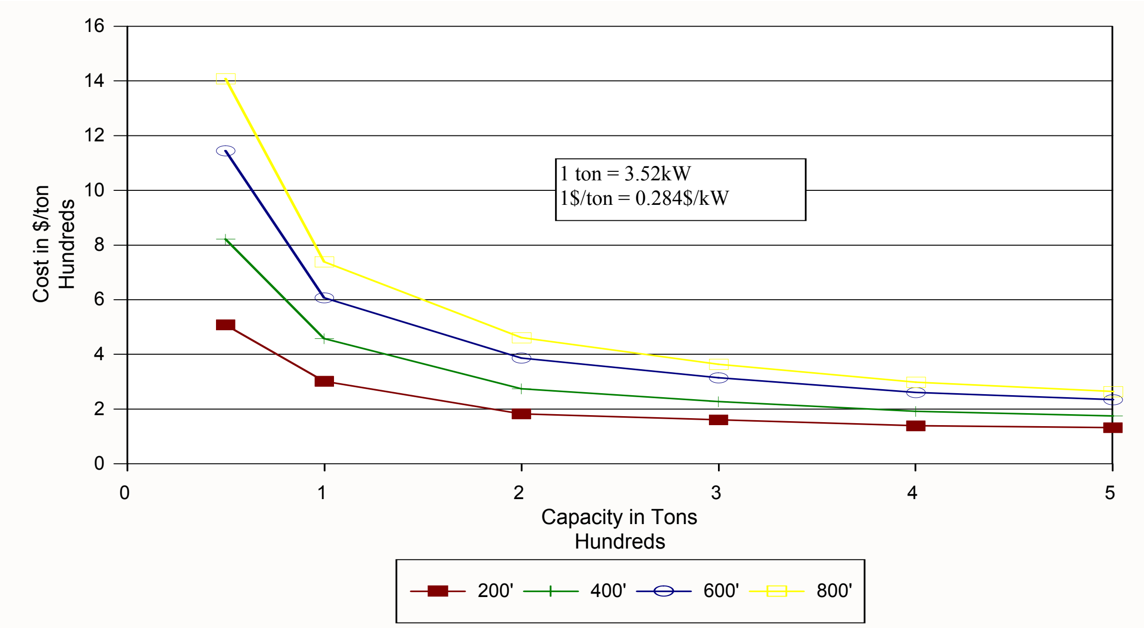

Figure 19: Groundwater system cost (1 well - 60°F [15.5°C] water)

For ground-coupled systems, actual project costs rather than calculations were used. Costs for these systems are a function of two values: the number of feet of borehole necessary per ton of heat rejection, and the cost per foot for completing the vertical bore and installing the piping. To arrive at a cost per ton, a value of US $5 per foot (US $16/m) of bore has been used. Although some recent projects have been the beneficiary of cost as low as US $3.75 per ft (US $12/m), and one as low as US $3 per ft (US $10/m), many areas of the country are still reporting costs of as much as US $15 per ft (US $50/m). Cost estimates for this comparison are based on a line cost of US $5 per ft and 200 ft/ton = US $1000/ton (US$ 285/kW).

Hybrid systems include both a ground loop and a cooling tower. The ground loop is sized to meet the heating load and, it along with the tower, is used to meet the cooling heat rejection load. As a result, hybrid system costs are a combination of ground loop costs and cooling tower costs. Using the US $5 per ft (US $16/m) value for the hybrid ground loop portion and vendor quotes for the cooling tower, Fig. 20 shows the cost per ton for the hybrid system based on 16°C

(60°F) soil temperature. The four curves shown for the hybrid system reflect costs for different ratios of heating loop length versus cooling loop length. As indicated, hybrid systems enjoy more favorable economics as the heating ground loop length decreases as percentage of the cooling ground loop length requirement. This is because the cost per ton of the cooling tower is less than the cost per ton of the ground loop.

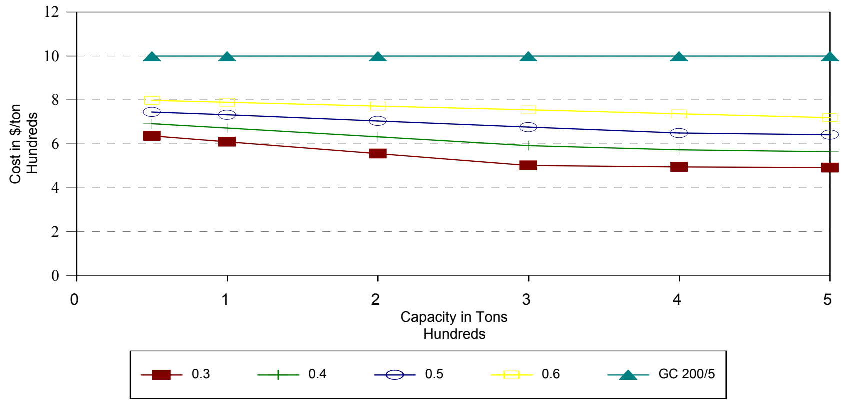

Figure 20: Hybrid system cost - 60°F (16°C) soil

Generally, the hybrid system is attractive in situations where ground loop costs per ton are high, and where the heating loop length requirement is low relative to the cooling loop length requirement.

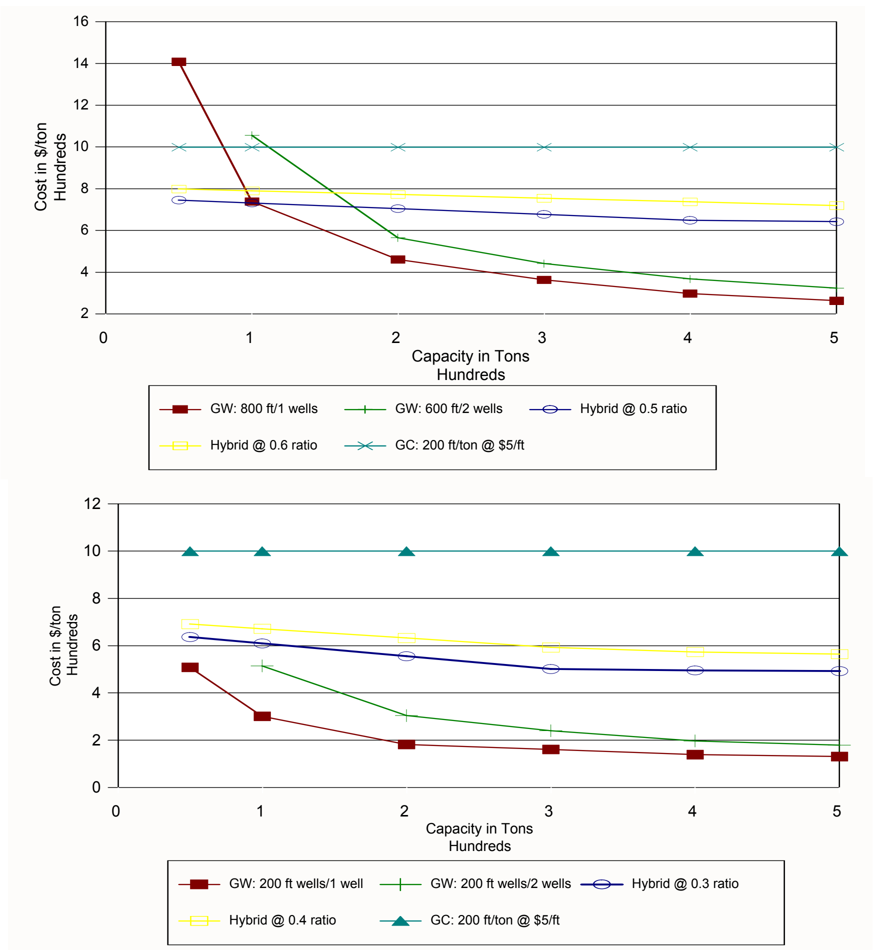

Figure 21: Ground-source system cost, 60°F (16°C) water or soil: a) Low case and b) High case

Figure 21 presents a comparison of the three types of systems for 60°F (16°C) soil. The ground-coupled system cost line is based upon US $5 per ft (US $16/m) and 200 ft per ton (17.3 m/kW) (US $1000 per ton = $285/kW). The two hybrid system curves in Fig. 21a are based upon loop length ratios (heating ÷ cooling) of 0.30 and 0.40; whereas, Fig. 21b used loop ratio lengths 0.5 and 0.6. The former are the most favourable conditions for hybrid systems. The two groundwater curves are based upon 200 ft (60 m) wells and one production/ injection well pairs (upper curve). Again, these are the most favorable conditions calculated for groundwater systems in this paper. It is clear that, based on these conditions, the ground-water system enjoys substantial capital cost advantage over the remaining two systems over the entire range of capacity covered. The loop ratios in Fig. 21b are the least favorable for the hybrid system.

As indicated (Fig. 21b) at system capacities of 100-175 tons (350-610 kW) and above, the groundwater system has the capital cost advantage over hybrid and ground-coupled systems. Below this range, the hybrid system is the most attractive. It is only under conditions of less than 100 tons (350 kW) with well depths of 800 ft (240 m), that the groundwater system capital cost exceeds that of the ground-coupled system.

This discussion addresses only system capital cost. In the process of system selection, other issues should be considered as well. These would include operating costs such as electricity for pumps and fans, water treatment costs (tower), and regulatory issues with respect to groundwater. As a result, system capital cost provides only a portion of the information required for informed decision making.

9. GROUND-COUPLED HEAT EXCHANGER DESIGN (BRAUD ET AL., 1988)

9.1 Heat Exchanger Configurations

Concentric Pipes. The concentric pipe heat exchanger consists of a closed pipe casing with an inner pipe for return flow, Fig.22. The pipes are at the axis of an earth cylinder of radius equal to the radius of thermal influence of the heat effect. Heat transfer between the circulating fluid, as it flows down the annual space, and the surrounding earth is the useful heat transfer. As the fluid returns up the inner pipe, it experiences heat gain due to heat flow across the inner pipe wall, and crossover heat flow is detrimental to the heat exchange process. It can be reduced to practical low values by proper selection of material for the inner pipe.

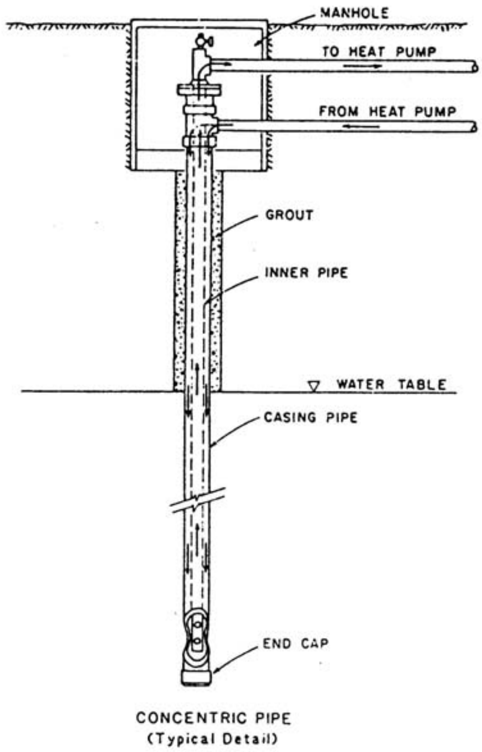

Figure 22: Concentric pipe configuration for vertical heat exchanger

U-Bend. The U-bend configuration consists of two pipes side-by-side and connected to each other with a 180° fitting at the bottom (Fig. 23). Fluid flows down one pipe and returns up the other. As the fluid descends, the temperature difference in the fluid and the earth mass causes heat flow. Because of temperature differences in the two fluid streams, there is also some deterious crossover heat flow. In this configuration, there are two walls of pipe material to impede the crossover heat flow; but, the presence of each pipe interferes with the heat loss to earth of the other.

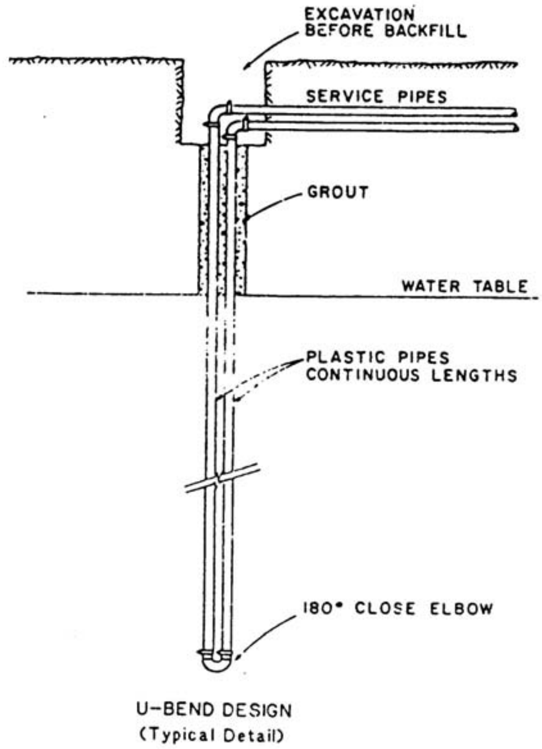

Figure 23: Two pipe side-by-side configuration for vertical heat exchanger

Equation for Heat Flow. The equation for rate of heat transfer from the fluid in the heat exchanger to the earth mass is:

$$\frac{Q}{L} = U \times \Delta T \tag{1}$$

where:

- Q = Rate of heat transfer, Btu/hr (W) for the whole heat exchanger length,
- L = Length of the heat exchanger, m (ft),
- U = Conductance rate for heat transfer from the circulating fluid to the earth, Btu/hr °F (temperature difference per ft of length)(W/°C/m) for the operating conditions,
- ΔT = $(T_2 + T_1)/2 - T_o$, the difference in average fluid temperature in the pipes $(T_2 + T_1)/2$, and the earth temperature T,
- T_2 = Fluid exit temperature, °F (°C),
- T_1 = Fluid entry temperature, °F (°C),
- T_0 = Earth temperature, °F (°C).

As given in many heat transfer texts, the conductance term U for heat flow from fluid in the heat exchanger to the earth can be estimated with the conductance coefficient for composite cylinders. The impedance to heat flow is caused by the thermal resistance of the pipe wall and the soil cylinder around the casing. Fluid surface resistance films are small relative to the other terms and are encompassed in the two resistance terms. We can express U as:

$$U = \frac{2\pi}{Soi\ resistance + Pipe\ resistance} = \frac{2\pi}{R_s + R_p} \tag{2}$$

The thermal influence of the pipe walls which separate the two fluid streams affects the temperature change in the fluid passing through the heat exchanger for given operating conditions, and it is manifest in the magnitude of the U-value as defined first equation. The casing wall resistance R_p can be calculated with sufficient accuracy from handbook values. The earth thermal resistance term R_s values can be solved for in the second equation. In fact, the purpose of field testing is to quantity the R_s value under different operating conditions and heat exchanger designs. Field testing was thus performed for the two heat exchanger configurations in Louisiana.

9.2 Results

Earth Conductance. The rate at which the earth would absorb heat was relatively high at the beginning of a run and declined as time went on. With on-off injection of heat, the instantaneous conductance values were always higher than with constant-on rate as shown in Fig. 24, where conductance values for 25%, 50% and 100% on-time are given. Asymptote values are the best estimates of the conductance (U-value) derived from regression curve fits to the test data, Fig. 25. The constant-on conductance rate (i.e., 100% value) (continuous heat injection for 48 or more hours) was found to be U = 4.86 W/°C/m ; for 50% on time, the effective U-value was 7.51 W/°C/m; and for 25% duty cycle, U was 11.88, see Table 4. By use of Equation 2, the earth resistance term R_s was also calculated, Table 4. With the steel casing, the thermal resistance of the steel was negligible relative to the surrounding earth resistance term (R_p □ 0).

Most of the runs were made with SCH 40 PVC inner return pipe. Runs with thin wall SDR 26 inner pipe exhibited less temperature drop of the circulating water. This effect was manifest in effective conductance values of U = 4.24 W/°C/m in 100% duty cycle. Wall thickness increased the heat transfer rate by only 5% over SCH 40.

Measured comparable heat exchange rates to earth with a steel casing with U-values range from 5.19 W/°C/m to 3.46.

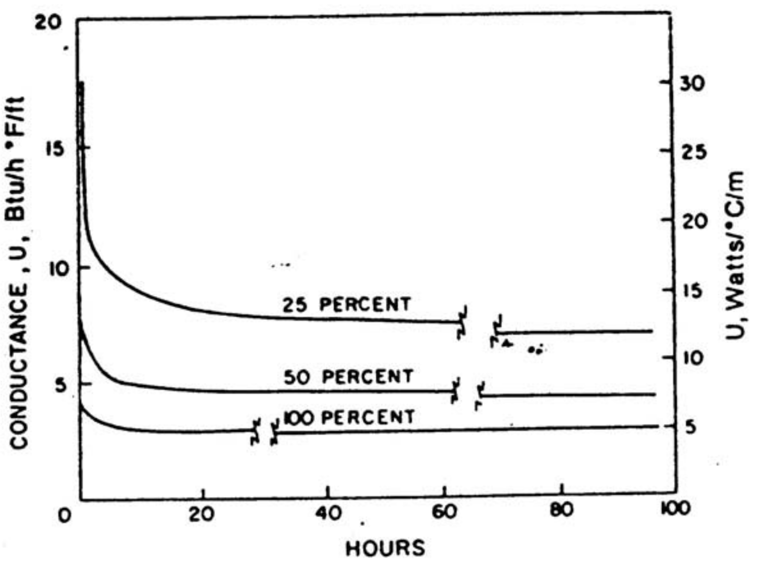

Figure 24: Typical runs of heat injection to earth with three-duty cycle

Table 4: Earth Heat Exchange Rate with Steel Casing and SCH 40 PVC Inner Pipe

Percent Run Time	100	50	25
U: W/^{o}C/mt	4.86	7.51	11.88
Earth Resistance R_s ^{o}C m/W	1.29	0.889	9.526

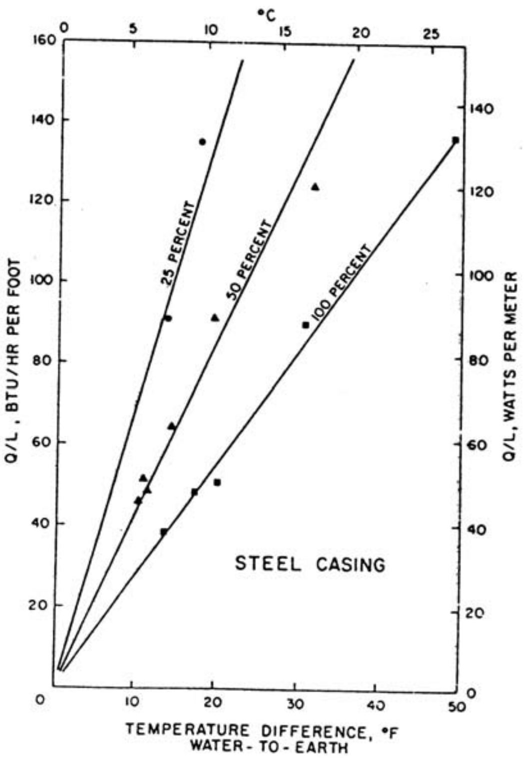

Figure 25: Earth absorption rate, Q/L plotted vs. average water-to-earth temperature difference for 25, 50 and 100% run time (duty cycle)

PVC Pipe Casing. Present day pipe costs dictate that plastic pipe is cheaper pipe than metal. Besides, PVC is widely used for water wells. Earth heat transfer rates with PVC plastic pipe can be calculated with second equation, the earth resistance term, R_s, Table 4, and the known thermal conductivity and wall thickness for PVC pipe. The U-value is given in Table 5.

Table 5: Heat Exchanger Parameters for PVC Casing and Inner Pipe

Percent run time	100	50	25
U: W/°C/m	3.58	4.85	6.37
PVC pipe resistance R_p ° C m/W	0.461	0.461	0.461

Thin wall PVC casing and SCH 40 PVC inner pipe are the most cost-effective combination

A residential heat pump using a PVC concentric pipe heat exchanger was monitored with good agreement to the experimental values. U-values ranged from 2.9 to 5.1 depending on run-time for the particular test day in June and July, 1981. Heating mode operation during severe cold weather in January, 1982, caused circulation water to drop to a minimum of 15°C. This value is a safe temperature and indicates that circulation water does not need antifreeze protection in South Louisiana.

Conductance values in Table 4 agree with other field tests of vertical earth-coupled heat exchangers. Bose, et al. (1980) determined the long-term conductance to earth in a 12.7 cm PVC casing with 3.2 cm SCH 40 PVC inner pipe. The U-value was 2.95.

U-Bend. Twelve runs of heat injection were made with the 81 m (265 ft) polyethylene pipe U-bend heat exchanger. Values of earth conductance are given in Table 6.

Table 6: Conductance to Earth with Polyethylene U-bend Heat Exchanger

Percent run time	100	50	25
U: W/°C/m	3.46	4.71	11.60

These values are close to those for PVC pipe except for 25% duty cycle. Because the earth resistance to heat flow is so much greater than the pipe wall resistance, the two designs (concentric pipe vs. U-bend) provide about the same performance.

9.3 Heat Exchanger Size (Length)

Besides the energy parameters for the heat pump, one must know the highest supply water temperature acceptable for cooling mode and the lowest temperature acceptable for heating mode. These values along with local earth temperature establish the design temperatures for sizing an earth-coupled heat exchanger.

Example: Find the heat exchanger length needed for a heat pump with 7,032 W cooling capacity at 35°C in an area with earth temperature T = 21°C. The heat pump duty cycle is estimated to be 50% run time during warmest summer days. Manufacturer specifications give a high temperature limit of 35°C for entering water. The heat pump discharge water will be 5.6°C warmer than entry. Total heat rejection of the heat pump = 4,747 W per ton or 9,493 W. In the heating mode, the heat pump has a heat of absorption value of 7,325 W at 7.2°C. Discharge water will be 3.2°C cooler than entry. Low temperature limit for entering water is 7.2°C. Design with PVC pipe, concentric pipe configurations.

Solution:

Cooling mode:

1. Find the design water-to-earth temperature difference, ΔT:

$$\Delta T = \frac{35 + (35 + 5.6)}{2} - 21 = 16.8$$

2. In Table 5.5 read the effective conductance rate for PVC pipe casing with 50% duty cycle,

$$U = 4.85\ W/°C/m)$$

3. Solve for L in the Equation 1:

$$L = \frac{9{,}483\,W}{4.85\,W/°C/m \times 16.8°C} = 117\ m$$

Heating mode:

1. Find design water-to-earth temperature difference, ΔT:

$$\Delta T = 21 - \frac{(7.2 - 3.2) * 7.2}{2} = 15.4°C$$

2. Assume that the heat pump will run 12 h in 24 h in coldest weather. The 50% duty cycle value for U then applies.

$$U = 4.85\ W/°C/m$$

3. Solve for L in the Equation 1:

$$L = \frac{7{,}325\,W}{4.85\,W/°C/m \times 15.6°C} = 97\ m$$

As this example shows, the heat exchanger length needed is the larger value which is about 117 m of PVC casing for the heat pump in cooling mode.

10. ADDITIONAL INFORMATION ON GROUND-COUPLED INSTALLATIONS

Ground-loop installed cost currently represents a large portion of system cost. The initial key to simulating market interest in GSHPs is bringing down those costs, according to an Electric Power Research Institute (EPRI) report, "Ground-Source and Hydronic Heat Pump Market Study." The report provides detailed information on the residential GSHP market, emphasizing the requirements for performance improvement and cost reduction.

The size and installed cost of the ground loop depends on the thermal conductivity of surrounding soil or rock. See Fig. 26, in which thermal resistivity (the inverse of conductivity) is plotted versus moisture content with soil type as the parameter. Thermal conductivity is a sensitive function of the type of soil or rock and its moisture content. The ground-loop designer must be able to identify the soil or rock, choose a design value for the minimum moisture content, and arrive at a design value for thermal conductivity. Until recently, there was no rational approach to this task. EPRI research shows that four soil texture classes (sand, silt, clay and loam) may be used to characterize soils and provides design values of thermal conductivity for each. The range of values for identified rock types has also been developed. This information and its rationale are presented in a report, "Soil and Rock Classification According to Thermal Conductivity."

A companion volume, "Soil and Rock Classification for the Design of Ground-Coupled Heat Pump System: Field Manual," published in cooperation with the National Rural Electric Cooperative Association (NRECA), Oklahoma State University (OSU), and the International Ground-Source Heat Pump Association (IGSHPA), describes simple field procedures for identifying soil and rock types, and provides appropriate thermal property design values. The soil and rock field manual is scheduled for incorporation in future editions of the authoritative, "Closed-Loop/Ground-Source Heat Pump Systems Installation Guide," published by NRECA, OSU and IGSHPA.

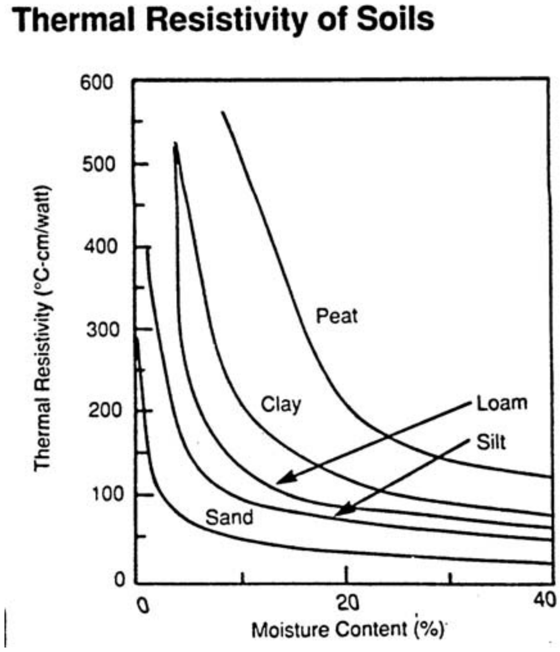

Figure 26: Ground loops function more efficiently in wetter soils where thermal resistivity is lower.

11. GROUT FOR GROUND-COUPLED SYSTEMS

Recent work by Brookhaven National Laboratory (Allan, 1998) studied the thermal conductivity of various cement-sand grouts used in GCHP. Their research investigated the suitability of this material for grouting vertical boreholes. The role of mix variables such as water/cement ratio, sand/cement ratio and super-plasticizers dosage were measured. The work summarized the thermal conductivity, permeability and bond strength results for selected grout mixes. Addition of sand and use of super-plasticizers to reduce water/cement ratio improved grout performance compared with neat cements. Field trials have been undertaken.

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