Chapter 2.4

DESIGN OF CLOSED-LOOP GEOTHERMAL HEAT EXCHANGERS IN THE U.S.

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INTRODUCTION

Geothermal heat pumps (aka ground-source heat pumps) (GHP or GSHP) are used in two basic modes: ground coupled (vertical or horizontal) – closed loop, or groundwater types – open loop (Figure 1 and 2). The installation and use of geothermal heat pumps worldwide have had a large increase over the past ten years at almost 10% annually. Most of this growth has occurred in the United States and Europe, though interest is developing in other countries such as Japan and Turkey (see Geo-Heat Center Quarterly Bulletin, Vol. 22, No. 1, March, 2001). At the beginning of 2000 the worldwide installed capacity in 27 countries was 6,875 MWt and the annual energy use was 23,287 TJ/yr (Lund, 2001a). The actual number of installed units is around 500,000, but the data are incomplete. The equivalent number of 12 kW units installed is slightly over 570,000. The 12 kW (3.4 tons) equivalent is used as typical of homes in the United States and some western European countries. The size of individual units, however, range from 5.5 kW (Poland and Sweden) for residential use to large units over 150 kW (Germany and the United States) for commercial and institutional installations (Sanner, 2001).

In the United States, most units are sized for the peak cooling load and are oversized for heating (except in the northern states) and, thus, are estimated to average only 1,000 full-load heating hours per year (capacity factor of 0.11). In Europe most units are sized for the heating load and are often designed to provide just the base load with peaking by fossil fuel. As a result, these units may operate from 2,000 to 6,000 full-load hours per year (capacity factor of 0.23 to 0.68). An average value for European countries is approximately 2,200 full-load hours per year (Rybach and Sanner, 1999; Sanner 2001).

In the United States, geothermal heat pump installations have steadily increased over the past 10 years with an annual growth rate of about 12%, mostly in the midwestern and eastern states from North Dakota to Florida. Today there are an estimated 500,000 units (12 kW equivalent) installed with 50,000 installed annually. Of these 46% are vertical closed loop, 38% horizontal closed loop and 15% open loop systems (Lund, 2001). Projections for the future are that the growth rate will increase about 12% annually, so that by 2010 an estimated 140,000 new units would be installed in that year, thus, adding almost one million units for a total of about 1.5 million units. Over 600 schools have installled these units for heating and cooling, especially in Texas.

Using a Coefficient of Performance (COP) of 4.0 and 1,000 full-load hours per year in the heating mode, the 500,000 equivalent units remove approximately 16,200 TJ/yr from the ground. The cooling mode energy is not considered geothermal, since this rejects heat to the ground; however, the cooling mode does replace other forms of energy and is, thus considered in fossil fuel and greenhouse gases emission savings. It should be noted at this point, that in the United States, heat pumps are rated on tonnage (i.e., one ton of cooling power – produced by a ton of ice) is equivalent to 12,000 Btu/hr or 3.52 kW.

Figure 1. Ground-coupled (closed-loop) types.

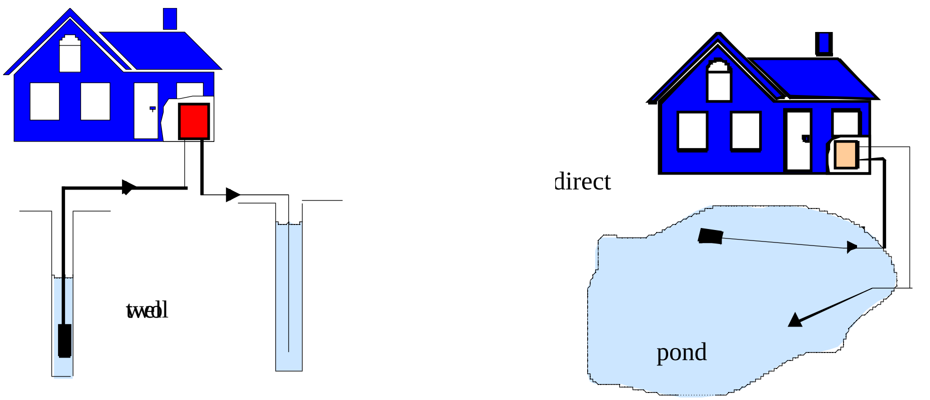

Figure 2. Groundwater (open-loop) types.

Recent converts to this form of energy sa-vings is U. S. President George W. Bush, who recently installed a 49 kW closed loop geothermal heat pump on his Texas ranch.

A general introduction to geothermal heat pumps can be found in Lund (2001b).

BACKGROUND: CLOSED LOOP SYSTEMS

The following presentation is a summary of material (with the appropriate unit conversions) presented in *"Ground-Source Heat Pumps -- Design of Geothermal Systems for Commercial and Institutional Buildings"* by Stephen P. Kavanaugh and Kevin Rafferty, and published by the American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., (ASHRAE) Atlanta, GA (1997), and in *"Earth-Source Heat Exchanger for Heat Pumps"* by Harry J. Braud, James Oliver and Henry Klimkowski, and published in the Geo-Heat Center Quarterly Bulletin, Vol. 11, No. 1, Summer, 1988.

As stated in Chapter 3 of Kavanaugh and Rafferty (1997) on *"Fundamentals of Vertical Ground Heat Exchanger Design"*:

"The design of vertical ground heat exchangers is complicated by the variety of geological formations and properties that affect thermal performance. Proper identification of materials, moisture content, and water movement is an involved process and cannot be economically justified for every project. Therefore, the necessary information for complex analysis is usually unavailable. A more prudent design approach is to apply empirical data to a simple solution of heated (or cooled) pipes place in the ground. The thermal properties can be estimated by using values for soils in a particular group and moisture content that is characteristic of local conditions."

"This method has proved successful for residential and light commercial systems.... Some permanent change in the local ground temperature may be expected for systems with large annual differences between the amount of heat extracted (heating mode) and the amount rejected (cooling mode). This problem is compounded in commercial systems where earth heat exchangers are more likely to be installed in close proximity because the available ground area is more limited."

Two schools of thought are that the maximum and minimum ground temperatures may take several years to occur depending on the amount of heat extracted or rejected to the ground (Claesson and Eskilson, 1987). This is especially true for multiple boreholes close to each other, as the worst case condition may occur several years after installation. Thus, in this case, the design should consider the long term performance. On the other hand, Kavanaugh (1984) suggests that detailed simulations for a great many years is unnecessary. He feels that an estimate of the annual net amount is sufficient for accurate determination of performance as variations in heat transfer are damped by the large thermal mass of the ground surrounding the coil.

RULES OF THUMB

Rules of thumb that is often used for the initial planning and cost estimate in the U.S. (Kavanaugh and Gilbreath, 1995; Rafferty, 2001).

1. 150 to 200 feet/ton (13 to 17 m/kW) for vertical loops
2. Approximately 30% to 50% longer for horizontal loops under the same conditions.
3. US$ 750/ton ($213/kW) for horizontal ground loop installation.
4. US$1,050/ton ($300/kW) for vertical ground loop installation.
5. US$8,200/ton ($2,330/kW) total installed cost for horizontal system (loop, heat pump, ductwork, etc.)
6. US$9,000/ton ($2,560/kW) total installed cost for vertical system (loop, heat pump, ductwork, etc.)
7. Typical U.S. residence of 2,000 - 2,400 ft^2 (186 to 223 m^2) requires 3 to 4 tons (10.6 to 11.4 kW) of heating supply, depending on climate.

It should be noted at this point, that design of horizontal loops, buried in trenches 4 to 6 feet (1.2 to 1.8 m) deep, are heavily influence by solar radiation and the number of loops in each trench. Since the solar radiation is difficult to quantify, no design methods are available for horizontal loops.

THE SIMPLIFIED METHOD

Ingersoll et al. (1954) uses a simple steady-state heat transfer equation to solve for the shorter term variation:

$$q = L\,(t_g - t_w) / R \qquad [1]$$

where,

q = rate of heat transfer for the heat exchanger length (Btu/hr or W),
L = length of heat exchanger (bore length) (ft or m),
t_g = temperature of the ground (^oF or ^oC),
t_w = average temperature of the fluid in the pipes (^oF or ^oC), and
R = thermal resistance of the ground ($h \cdot ft \cdot {}^oF/Btu$ or $m \cdot {}^oC/W$)

This equation can be rewritten as (Braud, et al., 1988):

$$q = L\,(U \cdot \partial T) \qquad [2]$$

where,

U = the conductance rate for heat transfer from the circulating fluid
To the earth ($Btu/h \cdot ft \cdot {}^oF$ or $W/m \cdot {}^oC$), and
$\partial T = (T_2 - T_1)/2 - T_o$, the difference in the average fluid temperature in the pipes ($(T_2 - T_1)/2)$), and the earth temperature (T_o).

THE LONG METHOD

Kavanaugh and Rafferty (1997 - chapters 3 and 4) describe a more precise method of calculating coil length which transforms the steady-state equation to give the variable heat rate of a ground heat exchanger by using a series of constant heat rate pulses. The thermal resistance of the ground per unit length is calculated as a function of time, which corresponds to the time for a particular heat pulse. A term is also included to account for the thermal resistance of the pipe wall and interfaces between the pipe and fluid and the pipe and the ground. The equations for cooling bore length (L_c) and heating bore length (L_h) are as follows:

For cooling:

$$L_c = q_a R_{ga} + (q_{lc} - 3.41\,W_c)(R_b + PLF_m R_{gm} + R_{gd} F_{sc})/(t_g - (t_{wi} + t_{wo})/2 - t_p) \qquad [3]$$

For heating

$$L_h = q_a R_{ga} + (q_{lh} - 3.41\,W_h)(R_b + PLF_m R_{gm} + R_{gd} F_{sc})/(t_g - (t_{wi} + t_{wo})/2 - t_p) \qquad [4]$$

Where,

F_{sc} = short-circuit heat loss factor
PLF_m = part-load factor during design month
q_a = net annual average heat transfer to the ground (Btu/h)
q_{lc} = building design cooling block load (Btu/h)
q_{lh} = building design heating block load (Btu/h)
R_{ga} = effective thermal resistance of the ground, annual pulse ($h \cdot ft \cdot {}^oF/Btu$)
R_{gd} = effective thermal resistance of the ground, daily pulse ($h \cdot ft \cdot {}^oF/Btu$)
R_{gm} = effective thermal resistance of the ground, monthly pulse ($h \cdot ft \cdot {}^oF/Btu$)
R_b = thermal resistance of the bore ($h \cdot ft \cdot {}^oF/Btu$)
t_g = undisturbed ground temperature (^oF)
t_p = temperature penalty for interference of adjacent bores (^oF)
t_{wi} = liquid temperature at heat pump inlet (^oF)
t_{wo} = liquid temperature of heat pump outlet (^oF)
W_c = power input at design cooling load (W)
W_h = power input at design heating load (W)

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

According to Kavanaugh and Rafferty (1997) the above equations [3, 4] consider three different "pulses" of heat to account for (1) long-term heat imbalances (q_a), (2) average monthly heat rates during the design month, and (3) the maximum heat rates for a short-term period during the design day - which could be as short as one hour, but a four-hour block is recommended.

The required bore length is the larger of the two determined by the above equations [3, 4]. In many cases, especially in commercial and institutional buildings, the cooling load is larger resulting in a longer length (L_c) of pipe for cooling. If this length is use, then excess heat will be available during the heating season. An alterative, is to design for the shorter heating bore length (L_h) and then use a cooling tower to compensate for the additional cooling load as described in Rafferty (1995) and discussed later in this paper.

Again, according to Kavanaugh and Rafferty (1997), equations [3, 4] can be simplified as follows:

$$L_c = q_a R_{ga} + (C_{fc} \cdot q_{lc})(R_b + PLF_m R_{gm} + R_{gd} F_{sc})/(t_g - (t_{wi} + t_{wo})/2 - t_p) \qquad [5]$$

$$L_h = q_a R_{ga} + (C_{fh} \cdot q_{lh})(R_b + PLF_m R_{gm} + R_{gd} F_{sc})/(t_g - (t_{wi} + t_{wo})/2 - t_p) \qquad [6]$$

The correction factors, C_{fc} and C_{fh}, account for the amount of heat rejected or absorbed by the heat pumps, which is dependent on the respective EER and COP of the units.

Equations [5, 6] require the average EER and COP of all the units in a particular building.

In addition, the average annual heat rate to the ground (q_a) can be found by using the annual equivalent full-load cooling and heating hours.

$$q_a = (C_{fc} \cdot q_{lc} \cdot \text{EFL hours}_c + C_{fh} \cdot q_{lh} \cdot \text{EFL hours}_h)/8760 \text{ hours} \qquad [7]$$

The heat pumps correction factors used in equations [5, 6 and 7] are given in Table 1 (Kavanaugh and Rafferty, 1997).

Table 1. Heat Pumps Correction Factors.

Cooling EER	C_{fc}	Heating COP	C_{fh}
11 (3.22)	1.31	3.0	0.75
13 (3.81)	1.26	3.5	0.77
15 (4.40)	1.23	4.0	0.80
17 (4.98)	1.20	4.5	0.82

Note: The heating performance, COP (coefficient of performance) is the heating affect produced by the unit (Btu/h or W) divided by the energy equivalent of the electrical input (Btu/h or W) resulting in a dimensionless numbers (the same in the U.S. and in Europe). However, the cooling performance, EER (energy efficiency ratio) is different between the U.S. and Europe. In the U.S. it is defined by the cooling affect produced by the unit (Btu/h) divided by the electrical input (W) resulting in units of Btu/watt · h as shown in Table 1. European units would

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be W/W and is dimensionless - multiple the EER number in Table 1 by 0.293 (shown in parentheses).

All heat pumps are rated by the Air Conditioning and Refrigerant Institute (ARI) and are published every six months in the *Directory of Certified Applied Air Conditioning Products* for Geothermal Heat Pumps. Closed loop systems were rated under ARI 330 and open loops under ARI 325. As of January 2000 these ratings are combined under ISO 13256-1 rating system.

EXAMPLE PROPERTIES

Tables with examples of various properties associated with closed-loop systems are presented in Kavanaugh and Rafferty (1997). Abridged samples of these tables are reproduced below:

Table 2. **Thermal Resistances (R_b) for Polyethylene U-Tubes***

[Pipe (bore) thermal resistance: h·ft·°F/Btu (m·°C/W)]

U-tube Dia.	Type	For water Flows above 2.0 gpm (0.13 L/s)	20% Prop. Glycol flow 3.0 gpm (0.19 L/s)	20% Prop. Glycol Flow 5.0 gpm (0.32 L/s)	20% Prop. Glycol Flow 10.0 gpm (0.63 L/s)
0.75 in. (19 mm)	Sch 40	0.10 (0.058)	0.14 (0.081)	NR	NR
1.0 in. (25 mm)	Sch 40	0.10 (0.058	0.15 (0.087)	0.11(0.064)	NR
1.24 in. (32 mm)	Sch 40	0.09 (0.052)	0.15 (0.087)	0.12 (0.069)	0.09 (0.052)

- Based on using borehole cuttings for backfilling around U-tube. Use Table 3 corrections for other conditions

NR = Not Recommended

Table 3. **Thermal Resistance Adjustments for Other Borehole Backfills or Grouts.**
(Add values to base resistance in Table 2).

Natural Soil Cond.	0.9 Btu/h·ft·°F (1.56 W/m·°C)		1.3 Btu/h·ft·°F (2.25 W/m·°C)			1.7 Btu/h·ft·°F (2.94 W/m·°C)	
Backfill or Grout Conductivity	0.5 Btu/ h·ft·°F*	2.0 Btu/ h·ft·°F	0.5 Btu/ h·ft·°F	1.0 Btu/ h·ft·°F	2.0 Btu/ h·ft·°F	0.5 Btu/ h·ft·°F	1.0Btu/ h·ft·°F
4 in. bore (102 mm)							
0.75 in. U-tube (19 mm)	0.11 (NR)	- 0.05	0.14 (NR)	0.03	-0.02	0.17 (NR)	0.05
1.0 in. U-tube (25 mm)	0.07	-0.03	0.09	0.02	-0.02	0.13 (NR)	0.04
5 in. bore (127 mm)							
0.75 in. U-tube (19 mm)	0.14 (NR)	-0.06	0.18 (NR)	0.04	-0.04	0.21 (NR)	0.06
1.0 in. U-tube (25 mm)	0.11 (NR)	-0.04	0.14 (NR)	0.03	-0.02	0.16 (NR)	0.05
1.25 in. U-tube (32 mm)	0.06	-0.03	0.09	0.02	-0.02	0.12 (NR)	0.04

NR = Not Recommended - For low thermal conductivity grouts, use small bore diameter.
Air Gaps add 0.2 to 0.4 h·ft·°F/Btu to bore resistance (0.12 to 0.23 m·°C/W).
* to convert to W/m·°C - multiply Btu/h·ft·°F x 1.73.
Note: some adjustments are negative, which indicates a thermal enhancement and a lower net thermal resistance compared to natural backfills.

Table 4. **Thermal Conductivity of Sand and Clay Soils in Btu/h·ft·°F (W/m·°C).**

Soil Type	Dry Density	5% moisture	10% moisture	15% moisture	20% moisture
Coarse*	100 lb/ft^3	0.8 - 1.4	1.2 - 1.5	1.3 - 1.6	1.4 - 1.7
100% sand	1.60 g/cm^3	1.38 - 2.42	2.08 - 2.60	2.25 - 2.77	2.42 - 2.94
	120 lb/ft^3	1.2 - 1.9	1.4 - 2.0	1.6 - 2.2	–
	1.92 g/cm^3	2.08 - 3.29	2.42 - 3.46	2.77 - 3.81	--
Fine grained	100 lb/ft^3	0.5 - 0.6	0.5 - 0.6	0.6 - 0.7	0.6 - 0.8
100% clay	1.60 g/cm^3	0.87 - 1.04	0.87 - 1.04	1.04 - 1.21	1.04 - 1.38

* Coarse grained = 0.075 to 5 mm; fine grained = <0.075 mm diameter particles.

Table 5. **Thermal Properties of Rocks at 77^{0}F (25°C)**

Rock Type	Thermal Conductivity* Btu/h·ft·°F (W/m·°C)	Density lb/ft^3 (g/cm^3)
Granite (25% quartz)	1.5 - 2.1 (2.60 - 3.63)	165 (2.64)
Andesite	0.9 - 1.4 (1.56 - 2.42)	160 (2.56)
Basalt	1.2 - 1.4 (2.08 - 2.42)	180 (2.88)
Limestone	1.4 - 2.2 (2.42 - 3.81)	150 - 175 (2.40 - 2.80)
Sandstone	1.2 - 2.0 (2.08 - 3.46)	160 - 170 (2.56 - 2.72)
Wet Shale (no quartz)	0.6 - 0.9 (1.04 - 1.56)	130 - 165 (2.08 - 2.64)
Dry Shale (no quartz)	0.5 - 0.8 (0.86 - 1.38)	130 - 165 (2.08 - 2.64)
Gneiss	1.3 - 2.0 (2.25 - 3.46)	160 - 175 (2.56 - 2.80)
Schist	1.4 - 2.2 (2.42 - 3.81)	170 - 200 (2.72 - 3.20)

* this represents the mid-range for samples of rock

Table 6. **Thermal Conductivities of Typical Grouts and Backfills.**

	Btu/h·ft·°F (W/m·°C)
Grouts without additives	
20% bentonite	0.42 (0.73)
Cement mortar	0.40 - 0.45 (0.69 - 0.78)
Concrete (150 lbs/ft^3 - 2.40 g/cm^3)	0.80 (1.38)

Thermally enchanced grouts	
20% bentonite - 40% quartzite	0.85 (1.47)
30% bentonite - 30% iron ore	0.45 (0.78)
60% quartzite - flowable fill (Cement + fly ash + sand)	1.07 (1.85)

Table 7. **Approximate Liquid Temperature Change Through GCHP Units.**

System Flow gpm/ton (L/s/kW)	Temperature Rise in Cooling - °F (°C)	Temperature Drop In Heating - °F (°C)
3.0 (0.054)	10 (5.6)	6 (3.3)
2.5 (0.045)	13 (7.2)	7 - 8 (3.9 - 4.4)
2.0 (0.036)	15 (8.3)	9 (5.0)

The details of how to determined the vertical bore hole lengths for heating and cooling using the above tables and the more detailed formulas ("The Long Way") can be found in Chapters 3 and 4 of Kavanaugh and Rafferty (1997). Since this computational method is very time consuming, and it is difficult to evaluate the impact of design alternatives as calculations must be repeated - a computer program currently available on the market is then discussed.

COMPUTER PROGRAM

A computer program is available from Stephen Kavanaugh, Energy Information Services, PO Box 861462, Tuscaloosa, AL 35486 (email: geokiss@home.com) - called *Geothermal Heat Pump Design Software for Commercial Buildings* – "GchpCalc". The developer states:

> *"GchpCalc is a program for sizing commercial ground coupled heat pump (GCHP) systems. This program integrates concepts familiar to HVAC engineers with information necessary to design ground heat exchangers."*
>
> *"The program user begins by entering the results of a cooling load and heat loss calculation. After a heat pump model line is selected (data from 12 manufacturers has been preloaded), the designer specifies desired water flows, temperatures, and vertical ground coil arrangement. Ground properties are entered with help from information tables. Data is entered in easy to use screens, which can be easily modified. Output appears in four formats. The program selects the heat pump model for each zone and specifies the required coil length. Hybrid GHP systems can also be designed. A variety of system performance data are listed. These include maximum block loads, required water flow rates, peak demands and average heat pump and system efficiencies."*
>
> *"The program allows the user to easily change any input and quickly determine resulting effects on ground coil size and system performance."*

The Geothermal Heat Pump Design Software for Commercial Version 3.1 features:

* Easy Hybrid System Design
* Hybrid Cooler Sized to Meet Peak Demand or Balance Annual Ground Heat
* Rejection Load
* Automatic or Manual Cooler Fan Motor Sizing
* New and Updated Heat Pump Manufacturer List
* Automatic Calculation of
* Bore Resistance (no more tables)
* Automatic Pump Size Selection
* Expanded Tables of Rock and Soil Thermal Properties
* New Table of Grout and Backfill Thermal Properties
* Expanded and Simplified Zone Load Handling Capabilities

An evaluation copy of the computer program can be obtained for a 10-day period by ac-cessing it on the Internet at: [http://bama.ua.edu/~geocool/Software.htm.](http://bama.ua.edu/~geocool/Software.htm)

A number of other computer design systems are available in the United States, including the International Ground Source Heat Pump Association (GLHEPRO and CLGS); RIGHT-LOOP by Wright Associ-ates, Lexington, MA; ECA by Elite Software Inc., Bryan, TX; WFEA by WaterFurnace Int'l., Inc., Fort Wayne, IN, GL-Source by Kansas Electric Utility, Topeka, KS; and GEOCALC by HVACR Programs, Ferris State Univ., Big Rapids, MI (Sanner, et al, 1999).

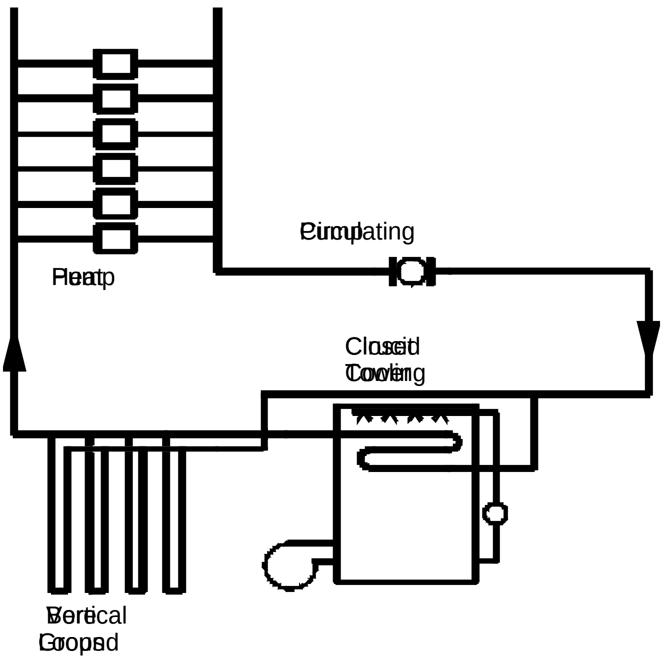

Figure 3. Hybrid ground-coupled heat pump system.

HYBRID SYSTEM

Due to the high cost associated with installing a ground loop to meet the peak cooling load in commercial buildings , a hybrid system using a cooling tower should be considered (Figure 3). The use of the cooling 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 (Raffety, 1995).

Generally, the hybrid system is attractive in situations where ground loop costs per ton (kW) are high, and where the heating loop length requirement is low relative to the cooling loop length requirement. According to Rafferty (1995), the most favorable conditions for a hybrid system is where the loop length ratios (heating/ cooling) are in the range of 0.3 to 0.4. The conclusions are based on 60^{o}F (16^{o}C) soil, ground coupled line cost of $5.00/ft ($16.40/m) and 200 ft/ton (17 m/kW).

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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. The 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 net cements. The conclusion were:

"Super-plasticized cement-silica sand grounts have thermal conductivities in the range of 1.249 to 1.463 Btu/h$\cdot$ft^{o}F (2.161 to 2.531 W/m$\cdot^o$C) for sand/cement ratios by mass of 2. To 2.5. Cement-sand grouts have significantly higher thermal conductivity than neat cement or bentonite grouts and retain conductive properties under drying conditions. Bond strength and interfacial impermeability are improved by addition of sand and control of water/cement ratio."

Field trials were also undertaken.

ECONOMICS

According to Kavanaugh and Gilbreath (1995), and Rafferty (2001), residential geothermal heat pump systems would have the following costs.

1. Installed cost - ground loop

Ground coupled - horizontal:	$ 741/ton ($210/kW)
Ground coupled - slinky	$ 904/ton ($258/kW)
Ground coupled - vertical	$1,028/ton ($292/kW)
Ground water - 3 ton (10.5 kW)	$ 675/ton ($192/kW)

2. Cost of water-to-air heat pump units

2.5 tons (8.8 kW)	$2,143
3.0 tons (10.6 kW)	$2,453
4.0 tons (14.1 kW)	$3,038

3. Total installed cost - 3 ton (10.6 kW) system - includes ground loop, unit, ductwork, etc.

Ground coupled - horizontal	$8,136
Ground coupled - slinky	$8,625

Ground coupled - vertical	\$8,997
Ground water	\$6,873
Air-source heat pump	\$4,430
Gas with air conditioning	\$4,331

4. Estimation of payback

Assuming a typical residence in the U.S. using 40,000 Btu/ft^2/ yr (126 kWh/m^2/ yr) for heating and at 2,000 ft^2 (186 m^2) and with natural gas at \$9.13/million Btu (\$0.0312/ kWh) - the annual heating cost would be: \$730. The difference in cost between a vertical ground coupled system and the gas system is \$4,666. Assuming that the electricity input into the heat pump (COP = 4.0) is 25% of the energy output (5,860 kWh/yr), the simple pay back of the geothermal system for heating would be 14.5 years. Using an electric furnace (assuming cost is the same as the gas furnace) and with electricity at \$0.07/kWh - the annual cost would be \$1,640. The simple payback for heating would then be 3.8 years. These numbers are for the state of Oregon, and will be higher for other parts of the U.S (the payback period would be less).

If cooling is also considered (at 40,000 Btu/ft^2/yr - 126 kWh/m^2/yr and EER = 13.6/4.0) then the payback period compared to natural gas would be 7.3 years, and compared to electricity would be 1.9 years.

EXAMPLE CALCULATIONS (SHORT METHOD)

A sample calculation is provided by Braud et al. (1988) using the "short method" with formula [2].

Given the following data:

Table 8. **Earth Heat Exchange Rate with Steel Casing and SCH 40 PVC Inner Pipe.**

Percent run time	100	50	25
U (W/^{o}C/m)	4.86	7.51	11.88
Earth Resistance R_s (^{O}C·m/W)	1.29	0.839	0.526

Table 9**. Heat Exchanger Parameters for PVC Casing and Inner Pipe.**

Percent run time	100	50	25
U (W/^{o}C/m)	3.58	4.85	6.37
PVC Pipe Resistance R_p 0.461 (^{O}C·m/W)	0.461	0.461	

Table 10. **Conductance to Earth with Polyethylene U-Bend Heat Exchanger.**

Percent run time	100	50	25
U (W/^{o}C/m)	3.46	4.71	11.60

The problem is to determine the vertical heat exchanger length for the following conditions:

* Heat pump with cooling capacity of 7032 W at 35^{o}C
* Earth temperature, T = 21^{o}C
* Heat pump duty cycle is estimated at 50% run time during warmest summer days
* Manufacturer specifications: high

temperature limit of 35°C entering water

* Heat pump discharge water will be 5.6°C warmer than entry
* Total heat rejection of the heat pump = 9493 W
* In the heating mode the heat pump has a heat absorption value of 7325 W at 7.2°C
* Discharge water will be 3.23°C cooler than entry
* Low temperature limit for entering water is 7.2 °C

Design with PVC pipe, concentric pipe configuration.

1. Cooling mode:

Find the design water-to-earth temperature difference, ∂T.

$$\partial T = [(35 + (35 + 5.6))/2] - 21 = 16.8^{o}C$$

From Table 9, read the effective conductance rate for PVC pipe casing with 50% duty cycle, U = 4.85 W/°C/m.

Solve for L in equation [2].

$$L = 9483\ W/(4.85\ W/^{o}C/m \times 16.8^{o}C) = \mathbf{117\ m}$$

2. Heating mode:

Find design water-to-earth temperature difference, _T.

$$\partial T = 21.1 - ((7.2 - 3.3) + 7.2)/2 = 15.6^{o}C$$

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.

Solve for L in equation [2].

$$L = 7325\ W/(4.85\ W/^{o}C/m \times 15.6^{o}C) = \mathbf{97\ m}$$

As this example shows, the heat exchanger length needed is the larger value of 117 m of bore hole (loop of PVC heat exchange pipe) for the heat pump in the cooling mode.

CONCLUSIONS.

There are several methods available for designing geothermal heat pump ground-coupled heat exchanger used in the vertical configuration. Calculation can be done by a simplified ("short method") equation that will give approximate results, but not take into consideration the long term of affects of removing and/or rejecting heat to the ground. A more involved ("long way") method takes these long term effects and the effects of adjacent heat exchangers into consideration. Computer programs are available to look at various alternatives using the more involved calculation procedure.

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