Comparison of Practical Vertical Ground Heat Exchanger Sizing Methods to a Fort Polk Data/Model Benchmark

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ABSTRACT

The results of five practical vertical ground heat exchanger sizing programs are compared against a detailed simulation model that has been calibrated to monitored data taken from one military family housing unit at Fort Polk, Louisiana. The calibration of the detailed model to data is described in a companion paper. The assertion that the data/detailed model is a useful benchmark for practical sizing methods is based on this calibration. The results from the comparisons demonstrate the current level of agreement between vertical ground heat exchanger sizing methods in common use. It is recommended that the calibration and comparison exercise be repeated with data sets from additional sites in order to build confidence in the practical sizing methods.

INTRODUCTION

With the implementation of a large-scale energy savings performance contract (ESPC) at Fort Polk, Louisiana, the heating and cooling systems in each of the base's 4,003 military family housing units were retrofit with geothermal (ground-source) heat pumps (GHPs). Each of the GHPs at Fort Polk features multiple vertical U-tube ground heat exchangers plumbed in parallel to reject/absorb energy to/from the earth. Independently of the ESPC, an evaluation was conducted to verify the energy and demand savings and explore means of improving the economics of future projects. The evaluation included field data collection at the electric feeder, apartment, end-use, and "energy balance (technology assessment)" levels. The calibration of a detailed model against 15-minute "energy balance" data from one of the housing units is described in detail in a companion paper (Thornton et al. 1997). This calibration resulted in a useful data/model benchmark for practical vertical sizing methods. This paper documents the comparison of five practical methods to the data/model benchmark and to each other in order to establish the current levels of agreement.

GROUND-SOURCE HEAT PUMP CONFIGURATION

Of the five "energy balance" monitoring sites at Fort Polk, one housing unit was chosen for the detailed model calibration documented in the companion paper (Thornton et al. 1997). The same apartment is the basis of the comparisons reported here. The unit selected is one of the lower floor apartments in a two-story five-plex building. This unit has a conditioned floor area of 1,052 ft^2 (98 m^2) and sits on a slab floor. There is an apartment above and next to the selected unit. This unit was equipped with a GHP with the following characteristics: nominal 1.5-ton (17,300-Btu/h) (5.1-kW) total cooling capacity and 15.4 EER at ARI 330 rating conditions, 11,800-Btu/h (3.5-kW) heating capacity and 3.5 coefficient of performance (COP) at ARI 330 rating conditions.

This GHP used water as the ground-loop working fluid and came equipped with a desuperheater attachment for supplying domestic hot water. However, the desuperheater was inactive during the monitoring period so as not to confound the calibration of the detailed model for heating and cooling operation. The comparisons documented here are also for the no-desuperheater case.

Two vertical U-tube ground heat exchangers connected in a parallel arrangement were used to reject/absorb heat to/from the earth. Each of the vertical U-tube ground heat exchangers was placed in a vertical borehole of 4.125 in. (0.1048 m) diameter and 258 ft (78.6 m) depth. These boreholes were spaced 16 ft (4.88 m) apart, 25 ft (7.62 m) from the exterior wall, and were backfilled with a bentonite-based grout after the installation of the U-tubes. The U-tubes themselves are composed of 1 in. (0.0254 m) nominal polyethylene pipe (1.08 in. ID, 1.31 in. OD [0.0027 m ID, 0.0033 m OD]) with a nominal center-to-center spacing of 2.565 in. (0.06515 m). The center-to-center U-tube spacing exists at the bottom of the U-tube heat exchanger (the

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bottom of the bore). No extraordinary measures were taken to maintain this spacing along the length of the bore. The horizontal runouts to the boreholes, and the horizontal piping between the bores, are buried at a depth of 3 ft (0.9 m) with outbound and return legs in separate trenches. Refer to Figures 1 and 2 for a graphical representation of the ground heat exchanger configuration.

One-time site measurements revealed that the water flow rate was 4.64 gal/min (0.293 L/s) and that the airflow rate through the heat pump unit was approximately 600 cfm (0.283 m^3/s). The water flow rate for the 1½-ton unit is then 3.22 gpm/ton (0.0578 L/s/kW).

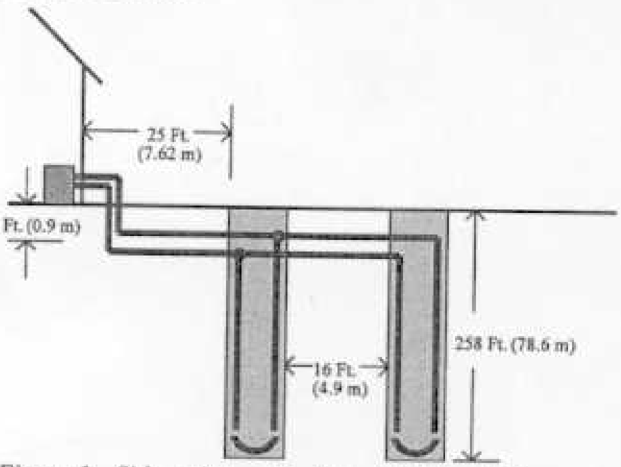

Figure 1 *Side view of the ground heat exchanger configuration.*

CALIBRATION OF THE SIMULATION MODEL

The TRNSYS simulation software package (TRNSYS 1996) was chosen as the tool for the creation of the detailed simulations of the geothermal heat pump system and housing unit. TRNSYS is a modular system simulation package where the user describes the components that compose the system and the manner in which these components are interconnected. Components may be typical pieces of equipment, such as a pump or thermostat, or utility modules, such as occupancy forcing functions, weather data readers, integrators, and printers. Because the program is modular, new component models for the heat pump and vertical ground heat exchangers were easily added to the existing component libraries to expand the capabilities of the program. The program was also chosen for its relative ease in driving the simulation models with measured data for the performance comparisons.

In order to characterize the performance of the system, the performance of each of the components that makes up the system must be characterized. In this case, the system components were chosen to be the building and its associated forcing functions, the heat pump, the ground heat exchanger, the thermostat, the ground-loop pump, and the heat pump's blower. The basis for each component model is described briefly in the companion paper (Thornton et al. 1997).

Due to its modular nature, calibrating a TRNSYS system model implies calibration of each of the individual component models. The calibrations are documented in detail in the companion paper (Thornton et al. 1997) and include heat pump controls power draw, blower power draw, thermostat setpoints, loop pump power draw, ground heat exchanger (best-fit deep-earth temperature and soil thermal properties), compressor power draw (start-up and steady-state), and building heating and cooling loads.

With the detailed model calibrated to measured data, users can have greater confidence in the detailed model's predictions, which makes the data/detailed model a useful comparative benchmark for practical vertical ground heat exchanger sizing methods for monitored installations. Broad-based comparisons are also possible since the calibrated detailed model is capable of addressing most elements of vertical ground heat exchanger design (ground properties, multibore interactions, long-term consequences of annual heat imbalance, bore spacing, bore diameter, pipe spacing, pipe diameter, grout properties, soil layers with different properties, etc.).

PRACTICAL SIZING PROGRAMS

For the purposes of this paper, the five practical design sizing methods will be referred to by a letter designation (A through E) instead of by their software titles. Each of the five practical methods requires a different set of user inputs. The general factors that influence the design size of the vertical ground heat exchangers are the building design loads, the building loads (monthly and annual), the weather, the soil thermal properties, the ground-loop properties (both geometric and thermal), the working fluid, and the installed heat pump. The values used for these topics, and the method of deriving these inputs from the detailed simulation model, are discussed below.

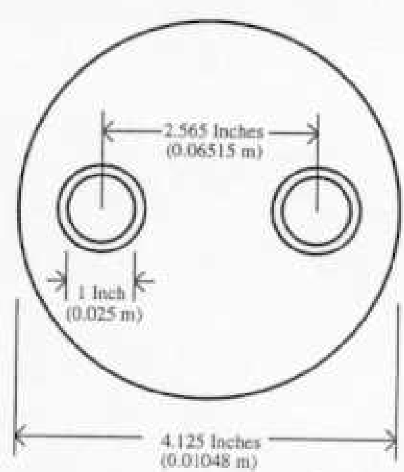

Figure 2 *Top view of one U-tube ground heat exchanger.*

Weather

Unlike the detailed simulation model, none of the five practical design sizing programs required detailed hourly weather

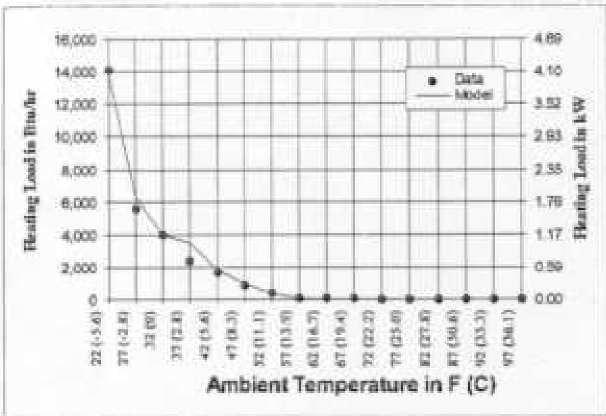

Figure 3 *Average heating load line comparison.*

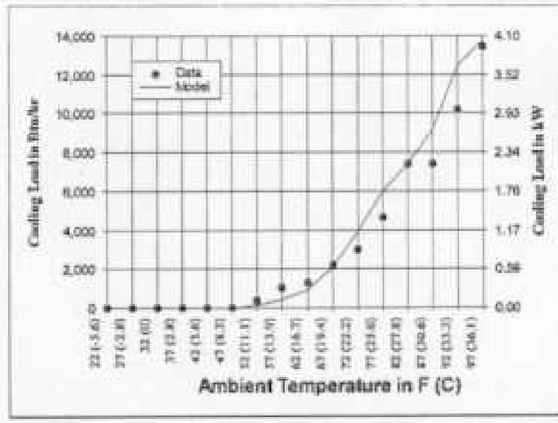

Figure 4 *Average total cooling (sensible and latent) load line comparison.*

information. In fact, only two of the practical methods require any type of weather information at all. These two programs each require the winter and summer design temperatures and the daily temperature range. Alexandria, Louisiana, is the closest city to Fort Polk with reported weather information. The accepted 99% design values for Alexandria are:

- Winter Design Temperature= 23°F (−5.0°C)
- Summer Design Temperature= 95°F (35.0°C)
- Daily Temperature Range= 20°F (11.1°C)

The inputs to the five practical sizing programs and the detailed simulation program are summarized below:

- Program A—Does not require weather information.
- Program B—Does not require weather information.
- Program C—Does not require weather information.
- Program D—Fort Polk weather was added to the program. The design temperatures were added as specified above and the hourly temperature bins were calculated from the Lufkin, Texas, TMY weather (NCC 1981) that was used to drive the simulation.
- Program E—Alexandria, Louisiana, was chosen from a list of available sites with properties as specified above.
- TRNSYS—Used TMY weather for Lufkin, Texas, the closest inland TMY weather station.

Building Design Loads

The building model in the detailed simulation was calibrated to the measured data by "tuning" the parameters of the infiltration model until the resulting average load lines from the simulation matched the average load lines observed at the site (Thornton et al. 1997). The average heating and cooling load lines from the simulation model and the measured data are plotted against the binned ambient temperature in Figures 3 and 4.

As reported earlier, the design temperatures for the site are 23°F (−5.0°C) in heating and 95°F (35.0°C) in cooling. The design heating and cooling loads cannot be read from Figures 3 and 4 at 23°F (−5.0°C) and 95°F (35.0°C), respectively, as this value represents the average load line, not the maximum load line. In fact, the design loads in the building cannot be calculated from the measured data, as the building load calculation is constrained by the capacity of the equipment. However, the building model can predict the design heating and cooling loads. The model predicts a design heating load of 15,760 Btu/h (4.62 kW) and a design total cooling (sensible and latent) load of 19,348 Btu/h (5.67 kW). The data support these relatively large design loads for this building, as the machine was seen to run for periods of several hours (up to eight) in both heating and cooling modes with a nominal heat pump heating capacity of 11,800 Btu/h at 2 gpm (3.46 kW at 0.13 L/s) and a nominal cooling capacity of 17,300 Btu/h at 2 gpm (5.07 kW at 0.13 L/s). The building heating and cooling design loads are required by programs D and E.

Program D also requires the winter and summer balance-point temperatures for the building. Looking at Figure 3, the winter balance-point temperature (the ambient temperature at which the building heating load is exactly met by the internal gains) is seen to be around 60°F (16°C). This value is within the range recommended by the program of 50°F to 65°F (10°C to 18°C). The 60°F (16°C) balance-point temperature taken from Figure 3 is with a relatively constant heating setpoint of near 70°F (21°C) (as seen from the measured data taken at the site). This 10°F (5°C) difference (70°F − 60°F, 21°C − 16°C) can be attributed directly to the solar and internal gains of the space. Unfortunately, due to the great range of cooling setpoint temperatures measured in the actual building, the summer balance-point temperature was more difficult to calculate. A cooling balance-point temperature of 64°F (18°C) was chosen for the calculations, as this represents the average cooling setpoint (74°F) (23°C) minus the effect of the solar and internal gains (10°F) (5°C). This cooling balance-point temperature of 64°F (18°C) falls within the typical range of 50°F to 70°F (10°C to 21°C) recommended by the program. For reference, the design ground heat exchanger lengths calculated by this program are extremely sensitive to these balance-point temperatures.

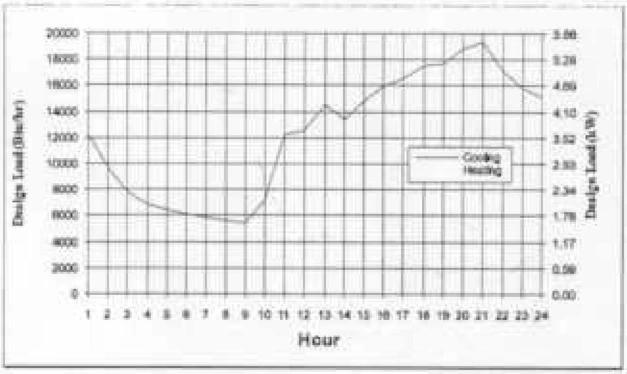

Figure 5 Hourly heating and cooling loads during the heating and cooling design days.

Program C requires the average building design loads in four bins that define the design day: 8 a.m. to noon, noon to 4 p.m., 4 p.m. to 8 p.m., and 8 p.m. to 8 a.m. The values that were used were taken from the detailed simulation program on the day of maximum heating and cooling loads. The design values are shown in Figure 5 and summarized in Table 1. Program C did not allow the calculated building loads to be used with only one heat pump for maximum entering water temperatures (EWTs) above 95°F (35°C), as the heating and cooling loads are higher than the heat pump capacities at the measured flow rate.

Building Monthly Loads

With the calibrated building model in place, determination of the monthly heating and cooling loads was straightforward. The simulation was run for the year and the energy delivered to/removed from the zone was integrated in heating and cooling modes for each month of the year. The monthly building loads are shown in Table 2 and are required by program B. Program E requires the annual heating and cooling loads; the sums of the monthly values are listed in Table 2.

Heat Pump Run-Time

Program C requires the equivalent full-load heat pump run-time in both heating and cooling modes. Using the detailed simulation, the heat pump run-time was integrated into monthly totals in both heating and cooling modes. In the months where the monitored data set was complete, the integrated heat pump run-time values from the simulation were compared against the measured values to double-check the accuracy of the simulation model. The results from this comparison showed a less than 15% difference in the run-times. Although a difference of 15% seems high, it should be pointed out that the simulation is being driven with typical meteorological year (TMY) weather conditions for Lufkin, Texas, and not the conditions measured at the site. The number of full-load heating hours from the simulation was found to be 538 hours, with 1,852 full-load cooling hours. These values were inputs to program C.

TABLE 1
Design Heating and Cooling Loads for Program C

Bin	Heating Design Values	Cooling Design Values
8 a.m. to Noon	11.2 kBtu/h (3.28 kW)	7.6 kBtu/h (2.2 kW)
Noon to 4 p.m.	3.8 kBtu/h (1.1 kW)	13.8 kBtu/h (4.04 kW)
4 p.m. to 8 p.m.	2.2 kBtu/h (0.64 kW)	16.9 kBtu/h (4.95 kW)
8 p.m. to 8 a.m.	8.3 kBtu/h (2.4 kW)	11.8 kBtu/h (3.46 kW)

TABLE 2
Monthly Heating and Cooling Loads

Month	Heating Load kBtu (kWh)	Cooling Load kBtu (kWh)
January	2,962 (868)	0
February	1,878 (550)	0
March	1,174 (344)	0
April	0	286 (84)
May	0	1,818 (533)
June	0	2,856 (837)
July	0	6,214 (1821)
August	0	6,229 (1826)
September	0	4,114 (1206)
October	296 (87)	332 (97)
November	121 (35)	0
December	1,030 (302)	0
Annual Total	7,461 (219)	21,849 (6403)

Ground-Loop Monthly Loads

Program A requires the monthly totals of heat rejection to the water from the heat pump (cooling mode) and heat absorption from the water to the heat pump (heating mode). The heat pump component model in the detailed simulation calculates the rejection/absorption at each time step based on the entering water temperature and flow rate by performing a table look-up on the manufacturer's published performance data. Correction factors are also applied based on the airflow rate and indoor air dry- and wet-bulb temperatures (from the published catalog data). As described in the companion paper (Thornton et al. 1997), several trends were noticed in the data that were not originally accounted for in the heat pump component model. First, the published power was seen to be slightly higher than that observed in the data. Second, the capacity of the heat pump was seen to asymptotically rise to a steady-state value with heat pump run-time. Both of these trends were then accounted for in the heat pump component model to more accurately simulate the installed heat pump. Energy balances and psychrometric checks on the heat pump's reported values are then done to ensure that the results are reasonable at each time step. Integrating the model's heat rejection to the water and heat absorption from the water on a monthly basis provides the results shown in Table 3.

TABLE 3
Monthly Heat Absorbed by and Rejected to Water

Month	Heat Absorbed kBtu (kWh)	Heat Rejected kBtu (kWh)
January	2,201 (645)	0
February	1,393 (408)	0
March	867 (254)	0
April	0	331 (97)
May	0	2,133 (625)
June	0	3,369 (987)
July	0	7,724 (2,263)
August	0	7,751 (2,271)
September	0	5,089 (1,491)
October	201 (59)	410 (120)
November	89 (26)	0
December	761 (223)	0

Soil Properties

The results of the ground heat exchanger calibration to test data are based on "best-fit" soil thermal properties. The "best fit" soil thermal properties corresponded almost exactly to the ASHRAE heavy saturated soil: a density of 200 lbm/ft^3 (3,200 kg/m^3), a specific heat of 0.20 Btu/lbm·R (0.84 kJ/kg·K), and a thermal conductivity of 1.40 Btu/h·ft·R (8.722 kJ/h·m·K) (ASHRAE 1991). An earlier independent analysis of the soil (Ewbanks 1995) at three locations around the base are shown below (the housing unit used for this study is located in the South Fort area).

South Fort
  Soil Type = Sand
  Thermal Conductivity = 1.156 Btu/h·ft·°F (2.00 W/m·K)
  Deep-Earth Temperature = 67.8°F (19.9°C)

Mid Fort
  Soil Type = Clay
  Deep-Earth Temperature = 67.8°F (19.9°C)
  Thermal Conductivity = 0.802 Btu/h·ft·°F (1.39 W/m·K)

North Fort
  Soil Type = Clay/Sand
  Thermal Conductivity = 0.964 Btu/h·ft·°F (1.67 W/m·K)

The thermal conductivity that was found for the "best fit" soil of 1.40 Btu/h·ft·R (2.42 W/m·K) was significantly higher than those found by Ewbanks. However, the thermal conductivity values for sand and clay (which the Ewbanks report characterized the soil as) from three of the five design sizing programs are given below. From these reported conductivity values, a value of 1.40 Btu/h·ft·R (2.42 W/m·K) is certainly reasonable.

- Program B—Thermal conductivity of sand 1.30 to 1.50 Btu/h·ft·°F (2.25 to 2.60 W/m·K)
  Thermal conductivity of clay 1.30 to 1.50 Btu/h·ft·°F (2.25 to 2.60 W/m·K)
- Program C—Thermal conductivity of sand 1.20 to 1.50 Btu/h·ft·°F (2.08 to 2.60 W/m·K)
  Thermal conductivity of clay 1.00 to 1.40 Btu/h·ft·°F (1.07 to 2.42 W/m·K)
- Program D—Thermal conductivity of sand 1.75 to 2.00 Btu/h·ft·°F (3.03 to 3.46 W/m·K)
  Thermal conductivity of clay 0.82 to 0.93 Btu/h·ft·°F (1.42 to 1.61 W/m·K)

The other parameter that was varied in the ground heat exchanger calibration was the deep-earth temperature. The ground heat exchanger calibration was performed in the month of May, the end of the heating season and the beginning of the cooling season. The resulting "best fit" soil temperature of 62°F (16.7°C) is thought to be on the low side, as three months of heating would have removed substantial energy from the ground around the boreholes. However, the value of 62°F (16.7°C) does provide an excellent lower bound for the comparison of the practical methods and was therefore used in this study. One of the five practical methods (program E) reports that the deep-earth temperature (average surface temperature) for Alexandria, Louisiana, is 69°F (20.6°C). Program E also reports that the amplitude of the surface temperature is 17°F (9.44°C), with the minimum surface temperature occurring at the thirty-second day of the year. Since no data were collected that substantiated or refuted the amplitude and day of minimum temperature values, these values were used in all the programs (including the detailed component simulations).

It was desired to run the programs at the "best fit" soil type and deep-earth temperature. However, since there are some discrepancies between the thermal properties of the "best fit" soil and the Ewbanks reported soil and because of the reported deep-earth temperature differences, the practical design programs were run with the following conditions for the comparison to the calibrated simulation:

- 62°F (16.7°C) deep-earth temperature, ASHRAE heavy saturated soil
- 69°F (20.6°C) deep-earth temperature, ASHRAE heavy saturated soil

For reference, the ASHRAE heavy saturated soil type is characterized as follows (ASHRAE 1991):

- thermal conductivity = 1.40 Btu/h·ft·°F (2.42 W/m·K)
- density = 200 lbm/ft^3 (3,204 kg/m^3)
- specific heat = 0.20 Btu/lbm·°F (0.84 kJ/kg·K)
- thermal diffusivity = 0.035 ft^2/h (0.0033 m^2/h)

The inputs to the five practical methods are summarized as follows:

- Program A—Added new soil type to the available list since the thermal diffusivity reported in the program was inconsistent with other sources.

- Program B—Selection of heavy saturated soil type from the list of available soils.
- Program C—Requires the input of thermal conductivity and thermal diffusivity. The values for heavy saturated soil were used (as specified above).
- Program D—Saturated sand was substituted for heavy saturated soil as this was not an available option.
- Program E—Saturated soil was substituted for heavy saturated soil as this was not an available option. Only the deep-earth temperature corresponding to Alexandria, Louisiana (69°F [16.7°C]), could be run, as this value is hardwired into the program.
- TRNSYS—Thermal conductivity, density, and specific heat required as parameters.

The soil temperature inputs to the five methods are summarized as follows:

- Program A—Requires the average surface temperature, surface temperature amplitude, and day of minimum surface temperature.
- Program B—Requires only the average surface temperature.
- Program C—Requires only the average surface temperature.
- Program D—Requires only the average surface temperature.
- Program E—Requires the average surface temperature, surface temperature amplitude, and day of minimum surface temperature. These values are automatically chosen based on the selected weather city.
- TRNSYS—Requires the average surface temperature, surface temperature amplitude, and day of minimum surface temperature as parameters.

Working Fluid

The working fluid for the ground-source heat pumps installed at the Fort Polk site is water. The inputs to the five practical methods are summarized as follows.

- Program A—The program does not allow the user to enter pure water. Therefore, a new antifreeze that had the properties of pure water was added to the available list.
- Program B—Pure water chosen from list of available fluids.
- Program C—Assumes a working fluid.
- Program D—Water chosen from list of available fluids.
- Program E—Assumes a working fluid (methanol 20%).
- TRNSYS—Takes the fluid thermal properties (density, specific heat, and thermal conductivity) as parameters.

Ground-Loop Configuration

The five practical sizing programs require different inputs to define the ground-loop configuration. Four of the programs require geometric and thermal property information that is readily available. Two of the programs require the user to input the borehole thermal resistance (the resistance to heat transfer from the working fluid to the borehole wall). Luckily, the detailed simulation model calculates and outputs the borehole thermal resistance. The calculated value of thermal resistance for the chosen pipe, borehole geometry, and thermal properties is 0.2281 h·ft·°F/Btu (0.1318 m·K/W). Two of the programs require the B/H ratio—the ratio between borehole depth and interborehole spacing. For the installed system, the B/H ratio is 0.062. A value of 0.05 was used for the program that only allowed discrete values of the B/H ratio. The ground-loop configuration inputs are summarized as follows.

- Program A—The type of U-tube pipe and the U-tube configuration were chosen from lists of available types. The distance between U-tube centers, the distance below the ground surface of the top of the U-tube, and the number of boreholes are inputs to the program.
- Program B—The borehole configuration (two in a line) and B/H ratio (0.05) are chosen from lists of available types. The borehole radius and borehole thermal resistance are inputs to the program.
- Program C—The borehole thermal resistance, equivalent diameter, borehole configuration, separation distance, and number of boreholes per parallel loop (one) are inputs to the program. The equivalent diameter was chosen from a table of equivalent diameters based on the U-tube pipe type and size.
- Program D—The pipe size, pipe type, and borehole configuration are chosen from lists of available types. The borehole multiplier is required as an input to the program.
- Program E—The borehole configuration and pipe size are chosen from lists of available types. The number of boreholes is internally selected to be one per ton.
- TRNSYS—The user enters the thermal and geometric parameters of the pipe, the thermal properties of the borehole backfill material, and the geometry of the borehole.

The thermal conductivity of the polyethylene pipe chosen from program A was different from that used in the detailed simulation and different from other reported sources. A comparison was performed between the assumed value from the simulation and the default value from program A. Bore sizing differences on the order of magnitude of 1 ft (0.3048 m) were observed in program A. Since the error introduced is small, the results from program A were based on the default thermal conductivity value from program A and not on the simulation value.

TABLE 4
Design Borehole Depths in ft/bore (m/bore) for Various Maximum EWTs at a Deep-Earth Temperature of 69°F (20.6°C)

	Program A	Program B	Program C	Program D	Program E	TRNSYS
85° F (29.4°C)	221.1 (67.4)	164.3 (50.1)	307.0 (93.6)	388.8 (118.5)	242.5 (73.9)	304.0 (92.7)
90° F (37.2°C)	163.1 (49.7)	138.6 (42.2)	249.0 (75.9)	297.0 (90.5)	197.5 (60.2)	241.3 (73.5)
95° F (35.0°C)	129.3 (39.4)	120.4 (36.7)	211.0 (64.3)	242.8 (74.0)	165.0 (50.3)	198.0 (60.4)
100° F (37.8°C)	107.3 (32.7)	106.8 (32.6)	N/A	206.0 (62.8)	142.5 (43.4)	172.8 (52.7)
105° F (40.6°C)	91.8 (28.0)	96.3 (29.4)	N/A	179.3 (54.7)	125.0 (38.1)	148.6 (45.3)
Observed Value of 85.1°F (29.5°C)	219.2 (66.8)	163.7 (49.9)	305.0 (93.0)	388.8 (118.5)	242.5 (73.9)	302.3 (92.1)
Simulation Maximum of 81.1°F (27.3°C)	307.1 (93.6)	192.7 (58.7)	376.0 (115.0)	514.8 (156.9)	275.0 (83.8)	390.5 (119.0)

TABLE 5
Design Borehole Depths in ft/bore (m/bore) for Various Maximum EWT's at a Deep-Earth Temperature of 62°F (16.7°C)

	Program A	Program B	Program C	Program D	Program E	TRNSYS
85° F (29.4°C)	147.6 (45.0)	128.6 (39.2)	229.0 (69.8)	268.3 (81.8)	N/A	218.0 (66.4)
90° F (37.2°C)	119.4 (36.4)	112.8 (34.4)	196.0 (59.7)	223.0 (68.0)	N/A	182.0 (55.5)
95° F (35.0°C)	100.4 (30.6)	100.7 (30.7)	172.0 (52.4)	191.3 (58.3)	N/A	158.0 (48.2)
100° F (37.8°C)	86.9 (26.5)	91.3 (27.8)	N/A	168.0 (51.2)	N/A	142.0 (43.3)
105° F (40.6°C)	Program Failure	83.6 (25.5)	N/A	150.0 (45.7)	N/A	130.0 (39.6)
Observed Value of 85.1°F (295°C)	146.7 (44.7)	128.2 (39.1)	228.0 (69.5)	268.3 (81.8)	N/A	217.0 (66.1)
Simulation Maximum of 81.1°F (27.3°C)	180.8 (55.1)	144.5 (44.0)	265.0 (80.8)	325.0 (99.1)	N/A	258.0 (78.6)

Heat Pump

Each of the five practical methods required a different method of inputting the heat pump characteristics. The input information to each of the programs is summarized as follows.

- Program A—Requires the total installed heat pump nominal capacity (1.5 tons [5.3 kW]) and the total heat pump water flow rate.
- Program B—Requires quadratic curve-fit coefficients for power and capacity in heating and cooling modes. The detailed model was exercised at the measured site flow rate and then curve-fit to determine the curve-fit coefficients for the program.
- Program C—An external utility program in this package was run to generate a new heat pump data file for the installed system. The inputs to this utility program were the capacity and power at two flow rates and two inlet water temperatures.
- Program D—Because the heat pump installed at the site is not part of the equipment database that came with this program, the user must enter the heating and cooling capacities of the heat pump and the COP/EER for the heat pump at each design inlet water temperature for which the ground heat exchanger length will be calculated. These values were found by exercising the detailed heat pump component model at the measured flow rate.
- Program E—Selected the unit from the internal equipment database.
- TRNSYS—A detailed component model of the heat pump was created for this study. Refer to a companion paper (Thornton et al. 1997) for a detailed discussion of this model.

The curve-fit coefficients required by program B were determined from an external utility program, as the internal curve-fitting routine in program B gave unrealistic results.

COMPARISON RESULTS

Each of the five practical sizing methods reports the required borehole depths as a function of the maximum water temperature entering the heat pump unit. For the purposes of this comparison, the design lengths were calculated using a single-year period of analysis at 85°F, 90°F, 95°F, 100°F, 105°F,

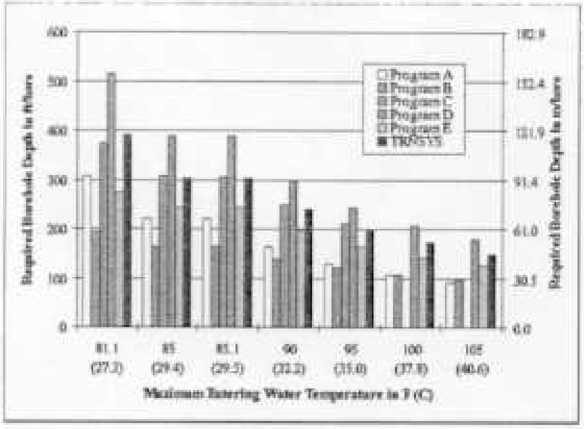

Figure 6 *Design borehole depths for various maximum EWTs at a deep-earth temperature of 69°F (20.6°C).*

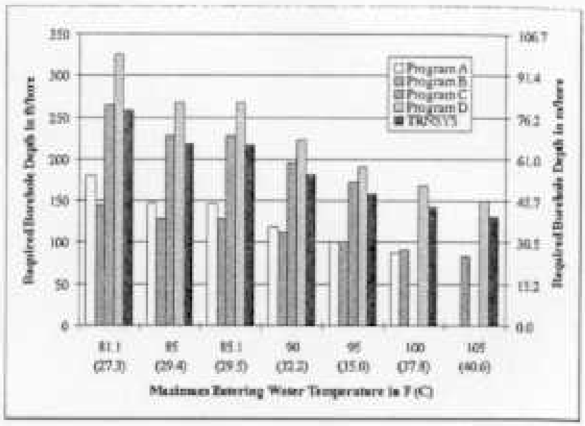

Figure 7 *Design borehole depths for various maximum EWTs at a deep-earth temperature of 62°F (16.7°C).*

81.1°F (the predicted maximum EWT from the simulation), and 85.1°F (the measured maximum EWT at the site) (29.4°C, 37.2°C, 35.0°C, 37.8°C, 40.6°C, 27.3°C, and 29.5°C) at deep-earth temperatures of 62°F and 69°F (16.7°C and 20.6°C). The results from this comparison show a range of borehole depths ranging from 83.6 ft (25.5 m) per bore (at a maximum EWT of 105°F [40.6°C] with a deep-earth temperature of 62°F [16.7°C]) to 514.8 ft (156.9 m) per bore (at a maximum EWT of 81.1°F [27.3°C] with a deep-earth temperature of 69°F [20.6°C]). The results from the comparisons are listed in Tables 4 and 5 and shown graphically in Figures 6 and 7.

The most important spread in the comparison results can be seen by moving horizontally across rows of Tables 4 and 5. Differences on the order of 88% at a deep-earth temperature of 69°F (20.6°C) and of 83% at a deep-earth temperature of 62°F (16.7°C) are seen at the measured maximum heat pump entering water temperature of 85.1°F (29.5°C).

With this amount of spread between the five practical sizing methods when driven with a detailed benchmark to provide consistent (where possible) inputs to the models, it is recommended that the calibration and comparison exercise be repeated with data sets from additional sites in order to build confidence in the practical sizing methods. With the cost of drilling vertical bores to house the ground heat exchangers representing a major expense, accurate and reasonable design results from the practical methods become paramount.

CONCLUSIONS

Five practical vertical ground heat exchanger design programs were exercised with inputs taken from a detailed simulation model calibrated to measured data. Even with consistent (to the extent possible) inputs, these five practical methods calculate very different required borehole depths necessary to keep the heat pump entering water temperature below a user-specified maximum. Further calibration/comparison exercises should be initiated to resolve the widespread differences between these programs by giving the developers an opportunity to calibrate and improve their methods. This activity will build confidence in the practical sizing methods.

ACKNOWLEDGMENTS

The opportunity to evaluate the energy savings performance contract at Fort Polk was created by the efforts of numerous organizations. Personnel at Fort Polk championed the contract and continue to administer it. The Huntsville Division of the Army Corps of Engineers was instrumental in determining the feasibility of the contract, developing the request for proposal, and awarding the contract. The selected energy services company, Co-Energy Group (CEG), was responsible for designing, financing, and building the energy conservation retrofits in return for a share of the energy savings and is responsible for maintaining the installed equipment for the duration of the 20-year contract. Applied Energy Management Techniques, under subcontract to CEG, was responsible for surveying the family housing, developing the energy consumption baseline from historical data, and developing the retrofit designs and prior cost and savings estimates. Oak Ridge National Laboratory (ORNL) carried out an independent evaluation of the contract with sponsorship from the U.S. Department of Defense (DOD), the U.S. Department of Energy (DOE), and Climate Master, Inc. Under subcontract to ORNL, field data collection was provided by Science Applications International Corporation, and TRNSYS modeling was provided by Thermal Energy Systems Specialists.

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DISCUSSION

Jeffrey D. Spitler, Associate Professor, Oklahoma State University, Stillwater, Okla.: I agree that further calibration/comparison exercises would be very useful. When they are performed, the same recommendation made on the companion paper should apply: that independent measurements of the undisturbed ground temperature and individual thermal properties should be made.

Also, I have to express some of the same reservations that I made with regards to the companion paper. Although the "best-fit" soil properties may work very well with the DST model, that does not imply that they are therefore correct. In fact, different models will have different sensitivities to different parameters.

Jeff W. Thornton: The authors agree that the different sizing methods will have different sensitivities to the soil thermal properties than the "best-fit" properties found for the DST model at this site. However, the goal of this paper was to compare the different sizing methods to each other and to the data taken from the site. The "best-fit" soil properties simply provide another point at which the models may be compared.