A Comparison of Vertical Ground Heat Exchanger Design Software for Commercial Applications

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ABSTRACT

In previous work, the authors compared a number of commercially available programs for the design of vertical borehole heat exchangers (BHEx) in residential applications. The objective of this paper is to compare four BHEx design programs and a benchmark simulation for a commercial application. An energy use model of an elementary school served by geothermal heat pumps was calibrated with site-collected data to form the benchmark; the school's operation was then simulated for a typical meteorological year at the site, and the outputs from the simulation were used as inputs to the four design programs. Since loads at the school are dominated by heating, the programs were exercised to design borefields with minimum inlet water temperatures of 30°F (−1.1°C), 35°F (1.7°C), and 40°F (4.4°C). On average, the depths predicted by the design programs agreed with the depths predicted by the benchmark program to within about ±14%. Three of the programs were found to provide relatively consistent results: their design lengths varied from the benchmark lengths by -7% to +12%, while designs from the other program varied by about 16% from the benchmark lengths. This is consistent with the results obtained for the residential comparison.

INTRODUCTION

In two previous papers (Thornton et al. 1997a; Shonder et al. 1999), the authors compared several commercially available software packages for the design of vertical borehole heat exchangers (BHEx) for residential applications. In this paper, the comparison is extended to a commercial application. Using the TRNSYS transient simulation software, an energy use model was developed for a 69,000 ft^2 (6410 m^2) elementary school in Lincoln, Nebraska. Calibration with one year of site-collected data ensured that the model accurately represented the behavior of the building during the monitored year. The model was then driven with data for a typical meteorological year to generate a consistent set of inputs for four borefield design programs. Since loads at the site are dominated by heating, borefield designs from the four programs were compared for minimum entering water temperatures of temperatures 30°F (−1.1°C), 35°F (1.7°C), and 40°F (4.4°C).

Calibrated models are necessary for these comparisons because all of the design programs use different algorithms to size the BHEx. The programs call for different inputs (e.g., monthly heat absorption/rejection to the ground, peak heating and cooling loads per month, equivalent full-load heating and cooling hours, and others), and there is no consistent way to derive all of these inputs from a given monitored data set. Even if it were possible to develop inputs for all of the methods from site-collected data, those data represent the behavior of the system only for the period during which the data were collected. BHEx design algorithms require load information for a typical year at the site. Without a calibrated model, there is no generally accepted method of predicting typical year performance using data from an actual year.

DESCRIPTION OF SITE AND EQUIPMENT

Maxey Elementary School is one of four identical facilities constructed in 1995 in Lincoln, Nebraska, and was equipped with geothermal heat pumps to provide heating and cooling. The school has a floor area of approximately 69,000 ft^2 (6410 m^2) divided between classrooms, offices, meeting rooms, storage areas, a gymnasium, and a cafeteria. Figure 1 is a photograph of the school as seen from the front entrance; a floor plan is presented in Figure 2. The school

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Figure 1 Maxey elementary, one of four identical schools in Lincoln, Nebraska, served by geothermal heat pumps.

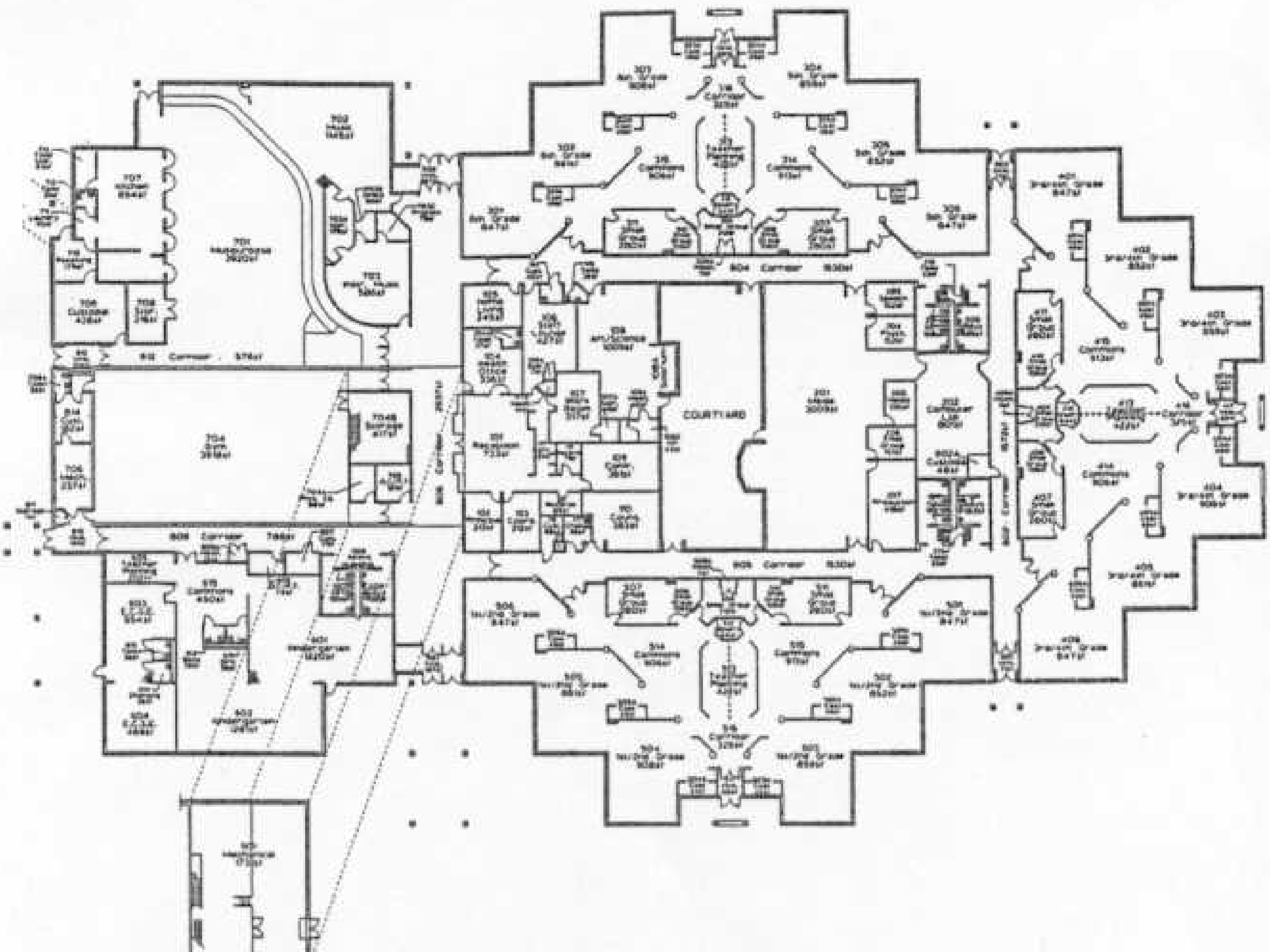

Figure 2 Floor plan of elementary school.

serves about 500 students with a staff of approximately 50, including teachers, office workers, cafeteria staff, and others.

The classrooms are mostly situated on the perimeter of the building with the offices and meeting rooms situated near the core. The school was designed with an open floor plan, including low-rise walls and sliding wall partitions to allow for greater visibility. The school was also designed to provide significant natural lighting, with large windows in each classroom, skylights in the main corridors, and a courtyard in the center of the building. The building is mainly of single-story design but does include a small second floor near the gymnasium, where the main mechanical room for the building is located.

The school was designed to meet *ANSI/ASHRAE Standard 62-1989*, which calls for at least 15 cfm (7 L/s) of fresh air per person. Preconditioned outdoor air is provided to the zone heat pumps by way of two nominal 15-ton (53 kW) heat pumps located in the main mechanical room. Each of these 15-ton (53 kW) units operates on 100% outdoor air and supplies ventilation air to zone heat pumps through two central ducts running along the school's main corridors. Additional outdoor air is provided to assembly areas, such as the multi-purpose cafeteria and gymnasium, by a nominal 10-ton (35 kW) unit operating on 40% outdoor air and a nominal 4.5-ton (16 kW) unit with 45% outdoor air. In all outdoor air units, preheat is provided by a hot water coil when ambient temperatures fall below 40°F (4.4°C). Hot water—produced by four natural gas-fired boilers, each with a capacity of 330,000 Btu/h (97 kW)—is also supplied to unit heaters located in vestibules and other perimeter areas. Domestic hot water is provided by two 100-gallon (379 L) gas-fired water heaters with capacities of 250,000 Btu/h (73 kW) and 197,000 Btu/h (58 kW).

The remaining heat pumps, ranging in size from 1.4 to 4.5 nominal tons (4.9 to 15.8 kW), serve individual zones: classrooms, offices, and common group study areas. For the most part these units are located above the central corridors outside the zones they serve and are easily accessible to maintenance personnel. Table 1 lists the sizes of each heat pump installed at the school, the flow rate of water through the heat pump, and the nominal flow rate of air through the unit. Altogether, the school contains 54 heat pumps with a total nominal cooling capacity of 204 tons (718 kW).

The 54 heat pumps absorb and reject heat through a common loop to a borefield consisting of 120 vertical loops arranged in a 12-by-10 pattern with 20 ft (6 m) center-to-center spacing. Figure 3 shows a layout of the system; note that the borefield is located under the school's soccer field. The bores are 240 ft (73 m) deep—or about 140 feet of bore per ton of cooling capacity (12 meters per kW)—with diameters of 4 ¼ in. (10.8 cm) on the lower 220 ft (67 m) and 6 in. (15.2 cm) on the top 20 ft (6.1 m). Fine gravel was used to backfill the boreholes to within 10 ft (3 m) of the surface, at which point a bentonite plug was used to seal the borehole (in compliance with Nebraska state regulations). Since bores at the site do not penetrate multiple aquifers, a surface plug is sufficient to protect groundwater. Fine gravel was judged to provide adequate pipe thermal contact because the static water level is 40 ft (12 m).

TABLE 1
Heat Pumps at Maxey Elementary School

Zone	Nominal size, tons (kW)	Water flow, gpm (L/s)	Total air flow, cfm (L/s)	Outdoor air, cfm (L/s)
Classroom 401	3.5 (12.3)	8.8 (.56)	1361 (642)	264 (125)
Classroom 402	3.5 (12.3)	8.7 (.55)	1324 (625)	263 (124)
Classroom 403	3.5 (12.3)	8.8 (.56)	1368 (646)	261 (123)
Corridor 802	1.5 (5.3)	4.7 (.3)	718 (339)	0
Common Area 415,410,411	3.5 (12.3)	8.8 (.56)	1263 (596)	243 (115)
Computer Room 202	3.5 (12.3)	9.4 (.59)	1451 (685)	309 (146)
Teacher Planning 409,413,412	2.5 (8.8)	8. (.5)	849 (401)	203 (96)
Classroom 404	3.5 (12.3)	9.1 (.57)	1319 (623)	259 (122)
Common Area 414,407,408	3.5 (12.3)	9.3 (.59)	1359 (641)	240 (113)
Classroom 405	3.5 (12.3)	9.5 (.6)	1358 (641)	256 (121)
Classroom 406	3.5 (12.3)	9.1 (.57)	1411 (666)	254 (120)
Classroom 301	3.5 (12.3)	9.2 (.58)	1433 (676)	254 (120)
Art/Science 108	3.5 (12.3)	9.4 (.59)	1476 (697)	216 (102)
Classroom 302	3.5 (12.3)	8.8 (.56)	1455 (687)	262 (124)
Common Area 315,310,311	3.5 (12.3)	8.7 (.55)	1380 (651)	241 (114)
Classroom 303	3.5 (12.3)	9.2 (.58)	1271 (600)	265 (125)

TABLE 1 (Continued)
Heat Pumps at Maxey Elementary School

Zone	Nominal size, tons (kW)	Water flow, gpm (L/s)	Total air flow, cfm (L/s)	Outdoor air, cfm (L/s)
Teacher Planning 309,312,313	3.5 (12.3)	8.8 (.56)	1221 (576)	215 (101)
Classroom 304	3.5 (12.3)	8.6 (.54)	1412 (666)	267 (126)
Common Area 307,308,314	3.5 (12.3)	8.7 (.55)	1373 (648)	240 (113)
Classroom 305	3.5 (12.3)	9.6 (.61)	1480 (699)	265 (125)
Corridor 804	1.5 (5.3)	4.9 (.31)	524 (247)	0
Classroom 306	3.5 (12.3)	9.1 (.57)	1459 (689)	273 (129)
Restrooms 210,204,205	3.5 (12.3)	9.4 (.59)	1349 (637)	269 (127)
Media Room 201	3.5 (12.3)	8.6 (.54)	1355 (640)	270 (127)
Classroom 506	3.5 (12.3)	9.3 (.59)	1358 (641)	268 (126)
Classroom 505	3.5 (12.3)	9.5 (.6)	1452 (685)	271 (128)
Common Area 507,508,514	3.5 (12.3)	9.2 (.58)	1303 (615)	247 (117)
Classroom 504	3.5 (12.3)	9.0 (0.57)	1470 (694)	269 (127)
Teacher Planning 509,512,513	3.5 (12.3)	9.6 (0.61)	1275 (602)	204 (96)
Classroom 503	3.5 (12.3)	9.5 (0.6)	1432 (676)	276 (130)
Common Area 510,511,515	3.5 (12.3)	9.4 (0.59)	1282 (605)	252 (119)
Classroom 502	3.5 (12.3)	9.1 (0.57)	1486 (701)	265 (125)
Corridor 805	1.5 (5.3)	4.8 (0.3)	509 (240)	0
Media Area 206,207,201	3.5 (12.3)	9.5 (0.6)	1532 (723)	271 (128)
Classroom 501	3.5 (12.3)	9.1 (0.57)	1460 (689)	267 (126)
Offices 101,102,103,117	1.5 (5.3)	4.9 (0.31)	641 (303)	120 (57)
Counseling 110	3.5 (12.3)	6.3 (0.4)	792 (374)	144 (68)
Records/Storage 109,111,112	2.0(7.0)	5.8 (0.37)	810 (382)	147 (69)
Art/Science 107,108	3.5 (12.3)	9.1 (0.57)	1259 (594)	229 (108)
Lounge 104,105,106	2.0 (7.0)	6.0 (0.38)	874 (413)	216 (102)
Corridor 806	3.5 (12.3)	9.3 (0.59)	1424 (672)	0
Music 703	1.5 (5.3)	4.9 (0.31)	690 (326)	123 (58)
Music 702	3.5 (12.3)	9.1 (0.57)	1451 (685)	256 (121)
Multi-Purpose 701	10.0 (35.2)	31.1 (1.96)	3423 (1616)	1500 (708)
Kitchen 707,708,709,710,711	5.0 (17.6)	16.0 (1.01)	1974 (932)	0
Gymnasium 704	5.0 (17.6)	15.1 (0.95)	1603 (757)	750 (354)
P.E. Offices 704A,705,809	3.5 (12.3)	4.4 (0.28)	581 (274)	0
Kindergarten 601	3.5 (12.3)	8.9 (0.56)	1546 (730)	282 (133)
Kindergarten 602	3.5 (12.3)	8.6 (0.54)	1475 (696)	299 (141)
Common Area 605,606,615,611,612	3.5 (12.3)	6.0 (0.38)	787 (371)	159 (75)
Early Childhood 603	2.0 (7.0)	5.5 (0.35)	907 (428)	166 (78)
Early Childhood 604	3.5 (12.3)	5.8 (0.37)	964 (455)	162 (76)
Ventilation Unit HP7V EAST	14.0 (49.2)	45.0 (2.84)	5076 (2396)	5076 (2396)
Ventilation Unit HP7V WEST	14.0 (49.2)	45.0 (2.84)	5172 (2441)	5172 (2441)

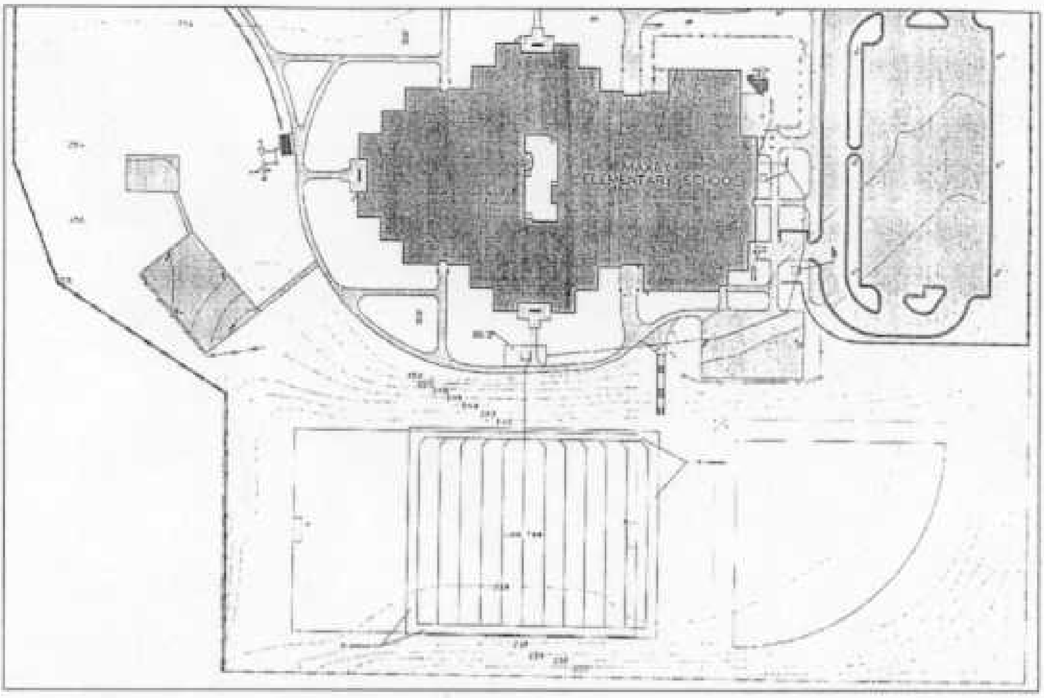

Figure 3 Site plan with bore field layout for elementary school.

The vertical loops consist of thermally fused, 1 in. (2.54 cm) diameter high-density polyethylene piping. The common loop contains approximately 10,000 gal (37,850 L) of water containing 22% (by volume) propylene glycol circulated continuously through the system by a 30 hp (22.4 kW) pump controlled by a variable-frequency drive.

A unique feature of the design at these four schools is the existence of extensive data on the operation of the mechanical systems collected at ten-minute intervals by the building energy management system (EMS). Table 2 lists the information collected by the EMS since the school began operation. Due to sensor drift, temporary equipment failures, and other problems, not all of the data were found to be usable. Because data from 1996 were the most complete, that year was used as the basis for calibration of the simulation model.

SIMULATION MODEL

The model for the school and associated HVAC systems was created in the TRNSYS simulation software package (Klein et al. 1996). TRNSYS is a modular system simulation package well suited to study commercial HVAC systems including GHPs. Previous calibration exercises (Thornton et al. 1997b) have proved its usefulness in calibrating detailed building HVAC systems to measured data.

There were two main goals in calibrating the model of the school:

1. Create a model of the school and its HVAC equipment that, when simulated, matches the observed operation of the school over the duration of the monitoring period.
2. Remove some of the idiosyncrasies that were discovered in the operation of the school during the monitoring period, and drive the school model with typical weather to produce typical year operation.

The school was divided into 65 thermal zones for the calibration: 54 zones were served by GHPs and the remaining 11 had either hot water unit heaters or were unconditioned. This level of detail in the building model was required in order to investigate design aspects not covered in this paper, such as the thermal "fighting" between heat pumps serving adjacent zones in open floor plan areas. The characteristics of the walls, windows, doors, floors, and ceilings (size, material, orientation, etc.) were obtained from as-built architectural drawings and included in the TRNSYS building model. Occupancy data for each zone were obtained from the school for the nine-month school year in 1996. The lighting data were taken from the architectural drawings, while equipment gains were estimated based on the function of the zone. Setback and setup temperatures were determined from the measured data and from conversations with school employees. The building model was then run to determine the heating, cooling, and ventilation loads that must be met by the HVAC equipment at the school.

Component models of the pumps, unit heaters, GHPs, boilers, domestic hot water tanks and distribution systems, ground heat exchanger, and controllers were interconnected in TRNSYS to form the model of the complete HVAC system at the school. Manufacturers' catalog performance data and as-built drawing schedules were then used to estimate the parameters for each of the component models (unit heater fan size,

TABLE 2
Data Collected at Ten-Minute Intervals by EMS

Heat Pumps with Outdoor Air
Return air temperature
Outdoor air temperature
Preheat coil discharge air temperature
Mixed air temperature
Supply air temperature
Supply air humidity
Compressor status (on/off)
Reversing valve status (heating/cooling)
Fan status (on/off)

Zone Heat Pumps
Space temperature
Compressor status
Reversing valve status
Fan status

Loop Field
Supply temperature
Return temperature
Water flow rate

Building Energy Use
Total electric use
HVAC electric use

for example). Where possible, each of the component models was then calibrated to the measured data (if available for that piece of equipment).

Two subroutines were available to model the borehole heat exchangers: the superposition borehole model (SBM) (Eskilson 1986) and the duct storage model (DST) (Hellstrom et al. 1996). SBM is intended to model ground heat flow in situations where a number of thermally interacting bores are present, whereas in DST multibore interaction is treated heuristically. The two subroutines were found to give essentially the same results, which indicates either that thermal interactions between the bores at this site are minimal or that the method DST uses to treat multibore interactions is adequate for this application. Ultimately the DST subroutine was included in the simulation because of its shorter runtime.

The ground heat exchanger model required a separate calibration. Using site-collected data on the flow rate of working fluid through the borefield and inlet and outlet temperatures, the thermal properties of the soil were adjusted until the predicted leaving water temperature from the ground heat exchanger model matched the measured leaving water temperature in a least-squares sense for the 1996 data set. The final "best fit" properties are presented in Table 3. It should be noted that these are effective properties that lump together vertical variations in soil properties as well as the impact of the horizontal runouts and the horizontal buried pipe between the ground heat exchangers. They are the properties that cause the DST model to best fit the data, and it is possible that use of a different ground heat transfer model would result in different best fit properties. Nevertheless, the best fit thermal conductivity of 1.3 Btu/(h·ft·°F) (2.25 W/m·K) is within 4% of an independent thermal conductivity test performed at the same site (Shonder and Beck 2000) in which a value of 1.36 Btu/(h·ft·°F) (2.35 W/m·K) was obtained.

TABLE 3
Properties of Soil and Grout at the Site

Deep earth temperature	54°F (12.41°C)
Density-specific heat product	43 Btu/(ft^3·°F) (2877 kJ/(m^3·K))
Soil thermal conductivity	1.3 Btu/(h·ft·°F) (2.25 W/m·K)
Fill material thermal conductivity	1.0 Btu/(h·ft·°F) (1.73 W/m·K)

With the building and ventilation loads and ambient conditions as driving forces, the TRNSYS system model was run in order to estimate the power consumption of the entire HVAC system. The infiltration was then used as the tuning device on the building loads until the simulated energy consumption matched the measured energy consumption.

The final results for the simulated and measured energy consumption for the school show excellent agreement. According to the monitored data, during calendar year 1996 the school's HVAC system used a total of 323,232 kWh of electrical energy. A correlation of daily energy use vs. daily average temperature indicates that in a typical meteorological year (TMY) the HVAC system would have used 311,372 kWh. When run with TMY weather data, the calibrated model predicts HVAC electrical use of 309,539 kWh. The two values agree to within about 0.6%.

The only data available for natural gas use were for monthly consumption. Figure 4 presents a comparison of actual 1996 monthly natural gas use vs. heating degree-days in the billing period and predicted natural gas use for a TMY vs. heating degree-days. The model predicts a gas use of 12,424 therms (364.2 MWh) in a TMY. For comparison, a correlation of the 1996 data vs. heating degree-days per month predicts that the school would use 12,787 therms (374.7 MWh) in a TMY. The two values agree to within 3%.

Another measure of the accuracy of the calibrated simulation is how well it is able to predict annual minimum and maximum entering water temperatures. Unfortunately, however, not enough 1996 weather data were available to drive the calibrated simulation, so it is necessary to compare the model's predictions for a typical meteorological year with

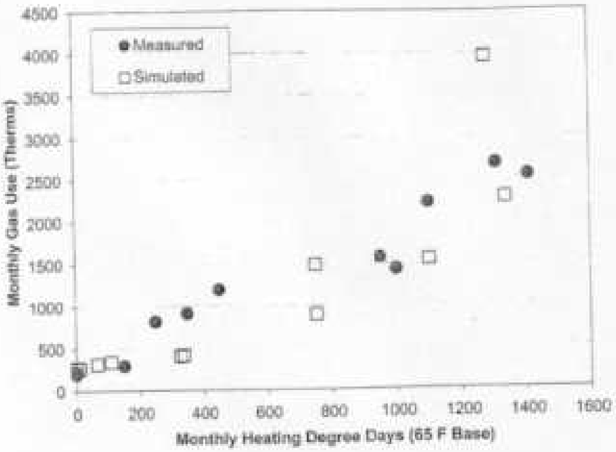

Figure 4 Measured (1996) monthly natural gas use and simulated (TMY) natural gas use vs. monthly base-65°F heating degree-days.

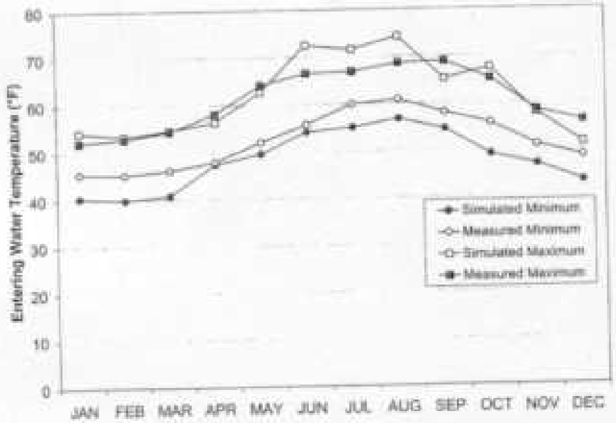

Figure 5 Measured minimum and maximum EWTs for 1996 and simulated minimum and maximum EWT for a typical meteorological year.

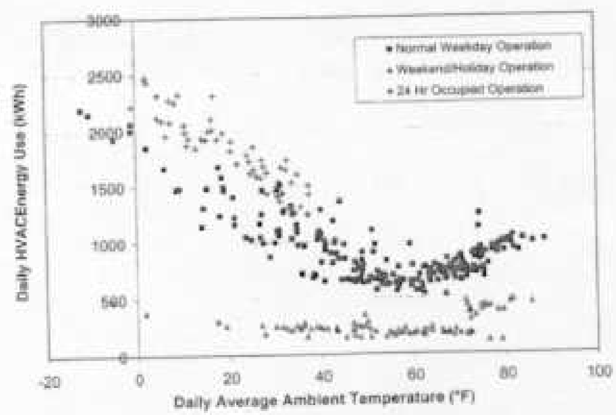

Figure 6 Daily HVAC electrical energy use vs. daily average temperature for 1996.

actual data from 1996. The comparison is presented in Figure 5. Because the cooling season for 1996 was less severe than the TMY (1069 base 65°F [18.3°C] cooling degree-days in 1996 vs.1215 for the TMY) and the 1996 heating season was more severe than the TMY (6924 base 65°F [18.3°C] heating degree-days in 1996 vs. 6241 for the TMY), the measured and simulated EWTs are not directly comparable. Nevertheless, the two appear to agree within about 5°F (2.8°C).

While the completeness of the 1996 data made it useful for calibration purposes, the set did have some problems. In Figure 6, the daily electrical energy use by the HVAC system is plotted vs. daily average temperature for every day in 1996. Three modes of energy use are evident: normal weekday operation when school is in session (filled squares), weekend/holiday operation (open triangles), and an eight-week winter period during which the HVAC system used much more energy than normal. As shown in Figure 7, we were able to model this behavior by eliminating night setback during the eight-week period and heating the school at its daytime setpoint 24 hours per day. Apparently the control system was inadvertently set this way for the period in question. In order to generate inputs for the design methods, the controller was assumed to operate with the normal night setback.

Although schools in the Lincoln district presently operate on a 9-month schedule, the geothermal heat pump schools were designed assuming a 12-month schedule. Consequently, after calibration, the model was exercised to generate inputs for the design methods assuming a 12-month operating schedule.

In addition to providing inputs for the four design programs, the calibrated simulation was exercised to determine benchmark borefield designs at various minimum EWTs for comparison with the programs' designs.

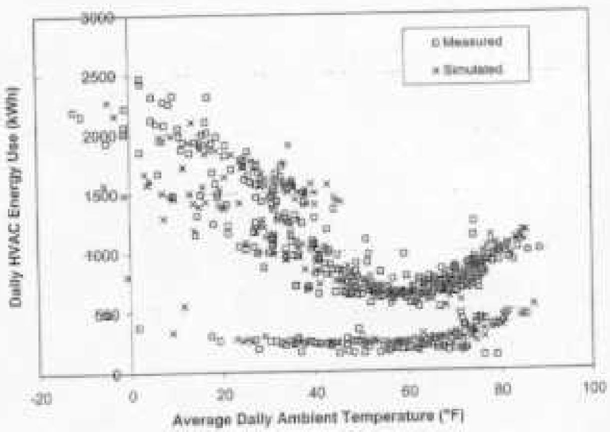

Figure 7 Daily 1996 HVAC electrical energy use and TMY simulated HVAC energy use vs. daily average temperature.

GROUND HEAT EXCHANGER SIZING PROGRAMS

As in the previous residential comparisons, the sizing programs are referred to by letter designation. In this case, only those programs suitable for commercial borefield sizing (A, B, C, and F) were evaluated.

Each of the four programs requires a different set of user inputs. The general factors that influence the design size of the ground heat exchanger (GHX) are the building design loads, monthly and annual heating and cooling loads, soil thermal and temperature properties, heat exchanger geometry and pipe thermal properties, and the capacities and efficiencies of the heat pumps connected to the ground loop. The inputs used and the method of deriving these inputs from the detailed simulation model are presented below.

Program A

Program A allows the user to select a horizontal or vertical heat exchanger configuration from a set of standard arrangements or a rectangular borefield of any dimension. A 120 bore GHX of 10 × 12 dimension and 20 ft (6.096 m) bore-to-bore spacing was input matching the geometry of the actual school borefield. In addition, the program required the distance between the u-tubes, the u-tube pipe material and nominal diameter, distance from the surface of the top of the u-tube, and a fill material (grout) specification. Values corresponding to those of the actual site were used for all of these inputs except for the grout material. The thermal conductivity of the fill material used in the detailed simulation was 1.0 Btu/(h·ft·°F) (1.73 W/(m·K)). From the menu of choices for grout material available in Program A, a 64% solids thermal grout was chosen as the closest match to the actual material. Smith and Perry (1999) report a value for thermal conductivity for a 63.5% solids grout of 0.85 Btu/(h·ft·°F) (1.47 W/(m·K)). Soil type and thermal properties and ground temperature data were selected from menus. A new soil with the "best fit" properties of the school site was added to the Program A menu and used. A user-defined ground temperature data set with school site data was added to the built-in menu and used.

Program A also gives the user the option of selecting a heat transfer fluid from a built-in menu or of including user-defined fluids. A new fluid was added to the menu with the thermal properties of the actual fluid used at the site. Maximum overall design flow rate must be input along with total rated heating and cooling capacity (at standard rating conditions) and heating COP and cooling EER (at design EWTs) of the building heat pumps. Values from the detailed simulation were available for the system flow, 460 gpm (0.029 m^3/s), and for the average heat pump efficiency and capacity at the simulation minimum and maximum EWTs (40°F [4.44°C] and 74.1°F [23.39°C], respectively). These are presented in Table 4. For other EWTs used in the sizing cases, heat pump efficiency was adjusted from the detailed simulation values based on data for the most prevalent heat pump unit used at the school. Rated performance data for this heat pump are presented in Table 5.

TABLE 4
Average Seasonal COPs, Rated Equipment Capacities, Entering Water Temperatures (EWT), and Maximum System Flow from Detailed Simulation

Heating seasonal COP	4.00
Minimum EWT	40.0°F (4.44°C)
Average rated heating capacity at min. EWT	2039 kBtu/h (598 kW)
Average rated heating capacity at 32°F (0°C)	1788 kBtu/h (524 kW)
Cooling seasonal COP	4.62
Maximum EWT	74.1°F (23.39°C)
Average rated cooling capacity at max. EWT	2635 kBtu/h (772 kW)
Average rated cooling capacity at 77°F (25°C)	2610 kBtu/h (765 kW)
Maximum system flow rate	460 gpm (0.029 m^3/s)

TABLE 5
Capacity and Efficiency Data for Most Prevalent Heat Pump Model Used at School Site

EWT °F (°C)	COP (heating) or EER (cooling)	Capacity kBtu/h (kW)
	Heating	
25.0 (−3.89)	3.46	28.2 (8.26)
30.0 (−1.11)	3.67	30.8 (9.03)
35.0 (1.67)	3.86	33.4 (9.80)
40.0 (4.44)	4.09	36.1 (10.58)
	Cooling	
74.1 (23.39)	16.07	45.1 (13.20)
90.0 (32.22)	13.60	42.9 (12.57)
95.0 (35.00)	12.43	40.5 (11.88)
100.0 (37.78)	11.36	38.2 (11.18)

NOTE: Values for fluid flow rate of 11 gpm (0.69 L/s) and air flow rate of 1450 cfm (684 L/s).

The only other input required for Program A was either monthly ground heat absorption and heat rejection or monthly building loads. Values from the detailed simulation for both options are listed in Table 6. The ground load values were the ones used in the sizing simulations discussed later.

It should be noted that Program A provides two methods for computing design lengths: an "average monthly load" basis and a "peak load" basis. The peak option requires two additional inputs—winter and summer peak runtime ratios. The user's manual for Program A defines this simply as the runtime ratio during peak conditions. For the detailed simula-

TABLE 6
Monthly Total and Peak Heating and Cooling Loads and Monthly Ground Heat Absorption and Rejection for School, Simulated for a TMY

Month	Total heating, kBtu (MJ)	Total cooling, kBtu (MJ)	Peak heating, kBtu/h (kW)	Peak heating hours	Peak cooling, kBtu/h (kW)	Peak cooling hours	Heat absorbed, kBtu (MJ)	Heat rejected, kBtu (MJ)
Jan.	343,444 (362,333)	3,895 (4,109)	1,786 (523)	11	152 (45)	2	253,894 (267,858)	10 (11)
Feb.	240,506 (253,734)	1,495 (1,577)	1,696 (497)	5	59 (17)	1	178,038 (187,830)	41 (43)
Mar.	145,415 (153,413)	11,024 (11,630)	1,693 (496)	2	301 (88)	1	100,429 (105,953)	4,480 (4,726)
Apr.	70,820 (74,715)	25,348 (26,742)	905 (265)	3	396 (116)	2	41,360 (43,635)	17,517 (18,480)
May	26,609 (28,072)	92,481 (97,567)	490 (144)	1	941 (276)	2	6,111 (6,447)	95,376 (100,622)
Jun.	6,386 (6,737)	170,592 (179,975)	227 (67)	1	1,508 (442)	3	96 (101)	199,240 (210,198)
Jul.	5,323 (5,616)	227,177 (239,672)	210 (62)	1	1,432 (420)	5	3 (3)	266,829 (281,505)
Aug.	4,807 (5,071)	235,520 (248,474)	154 (45)	1	1,491 (437)	7	34 (36)	276,723 (291,943)
Sept.	20,600 (21,733)	88,148 (92,996)	287 (84)	6	863 (253)	2	3,555 (3,751)	92,602 (97,695)
Oct.	58,755 (61,987)	43,803 (46,212)	1,081 (317)	1	1,101 (323)	3	30,595 (32,278)	37,860 (39,942)
Nov.	203,565 (214,761)	7,045 (7,432)	1,098 (322)	2	217 (64)	2	147,112 (155,203)	645 (680)
Dec.	379,297 (400,158)	3,962 (4,180)	1,274 (373)	4	104 (30)	2	282,797 (298,351)	0

tion maximum and minimum EWTs, the runtime ratios for the peak hourly loads were determined to be 1.00 for heating and 0.58 for cooling based on the standard rated capacities. Discussions with the program developers indicated that their intent is that runtime ratios should be the average values for the peak two-day periods (again based on the standard rated capacity of the heat pumps). Peak two-day runtime ratios were determined to be 0.405 and 0.25 for heating and cooling, respectively, from the detailed simulation.

Sizing runs were made using both methods and with both peak-hour and average two-day runtime ratios for the peak method. Using the peak-hour runtime ratios, the peak load option produced designs are more than twice the length as those for the average monthly load method. Using the average two-day peak runtime ratios, peak method designs were about 30% longer than those of the monthly method. The peak load method, using two-day average runtime ratio values, is the most consistent with the other three programs. Therefore, this paper presents designs from that method only. It is evident that careful determination of the runtime ratio input values is critical to achieving accurate loop designs with Program A's peak option as its results are extremely sensitive to this parameter.

Program B

Program B requires the same basic design parameters as Program A. In addition, it requires the user to input a value for borehole resistance (fluid-to-ground heat transfer resistance). In the absence of a value from the detailed simulation, a utility program included with Program B was used to compute a value of 0.211 h·ft·°F/Btu (0.122 m·K/W). Operating data for the most prevalent heat pump used at the school is included in the Program B database. This was used for the heat pump input required.

One drawback to Program B is that it is limited to a maximum borefield size of 10 × 10. This meant that the actual school field could not be input directly. The 10 × 10 grid was used, but the loads and total flow rate had to be adjusted as discussed below. A bore spacing-to-depth ratio is also required as input. Although the actual value for the school is 0.083 (20/240), Program B only allows the user to choose between a built-in library of discreet values. The closest value available to the actual was 0.10 and this was used.

For each month in a design year, the program requires total heating and cooling load and peak heating and cooling loads. In order to account for the borefield size limitation noted above, values obtained from the detailed simulation were adjusted by a factor of 0.833 (100/120), as recommended by the developers of Program B. In addition, the program requires the number of hours of occurrence of peak load in any one month. A default of six hours is recommended by the developers. From the detailed simulation, maximum hours of occurrence for heating and cooling peaks were determined to be 11 and 7, respectively (see discussion under Program F). These values were input to Program B.

A number of heat transfer fluid choices are built into Program B. The 23.5% propylene glycol option was chosen as the closest match to the actual fluid used (22% propylene glycol by volume). The maximum fluid flow from the detailed simulation, 460 gpm (0.029 m^3/s), was adjusted by the 0.833 factor to account for the smaller effective borefield.

Program C

Program C requires basic information about the ground heat exchanger array: the nominal diameter of the u-tube pipe, thermal conductivity of the fill material, heat transfer fluid flow rates (in gpm/ton), bore separation distance, and borefield dimensions. The program also requires the user to specify whether the fluid flow at design conditions is turbulent, transitional, or laminar. The flow was determined to be turbulent for this case. Given this information, the program calculates a borehole resistance of 0.196 h·ft·°F/Btu (0.113 m·K/W).

Operating data for the school heat pumps were available in the program's database. Data for the most prevalent size unit was used as input. Best fit soil properties from the detailed simulation were used.

As opposed to the monthly loads required by other programs, Program C requires the average loads in each of four time periods (blocks) for a heating and a cooling design day. Design days were determined from the detailed simulation as those days where the daily heating and cooling loads were a maximum. These block loads are given in Table 7. The program also requires annual equivalent full-load heating and cooling hours. These were determined by taking the annual loads from the detailed simulation and dividing them by the average rated capacity of the system heat pumps at the simulation maximum and minimum EWTs (see Table 4 for values). This calculation yielded 738.3 full-load heating hours and 345.5 full-load cooling hours (for comparison, in the simulation model the average runtime for all the heat pumps was 537 hours in heating mode and 397 hours in cooling mode). It should be noted that full-load hours will vary somewhat depending on the EWT used to determine heat pump capacity. Fortunately, borelength results from Program C are not very sensitive to this parameter. Increasing the cooling load hours by 18% resulted in a change in calculated borelength of less than one percent.

TABLE 7
Average Block Loads on Peak Heating and Cooling Days

Block	Average heating load, kBtu/h (kW)	Average cooling load, kBtu/h (kW)
8 a.m. - Noon	1671 (490)	1322 (388)
Noon - 4 p.m.	1305 (382)	1441 (422)
4 p.m. - 8 p.m.	113 (33)	283 (83)
8 p.m. - 8 a.m.	547 (160)	286 (84)

Program F

With the exceptions of the block design loads (Table 7), full-load hours, and monthly ground loads, Program F requires all of the information needed by the other three programs. The heat pump information required by Program F consists of an estimate of the heating and cooling seasonal performance factors. For our purposes, these were determined from the detailed simulation and are presented in Table 4. For EWTs other than the simulation minimum and maximum, these values were adjusted in the same manner as discussed under Program A. If it is desired to consider peak load periods in the design analysis, then the hours of occurrence of peak heating and cooling loads must be provided for each month of the design year. For purposes of the present study, this information was determined from the detailed simulation by taking the number of hours where the heating or cooling load was within 95% of the absolute hourly peak. Peak hours of occurrence are included in Table 6.

The program gives the user the option of including or not including monthly peak loads in the sizing analysis. We ran the program using both approaches. Design borelengths obtained when ignoring peak effects were about 30% to 40% of the lengths obtained when the peak loads were included. Since the "peak load" design lengths were most consistent with the three other design methods, only those design lengths are reported here.

TABLE 8
One-Year Design Lengths in Bore ft/ton (Bore m/kW) from the Four Programs

	Design Program				
Min. EWT	A	B	C	F	TRNSYS
30°F (−1.1°C)	73.5 (6.4)	75.3 (6.5)	91.8 (7.9)	102.9 (8.9)	78.8 (6.8)
35°F (1.7°C)	91.2 (7.9)	92.9 (8.0)	115.3 (10.0)	129.4 (11.2)	103.5 (9.0)
40°F (4.4°C)	118.8 (10.3)	120.6 (10.4)	152.4 (13.2)	170. (14.7)	142.9 (12.4)

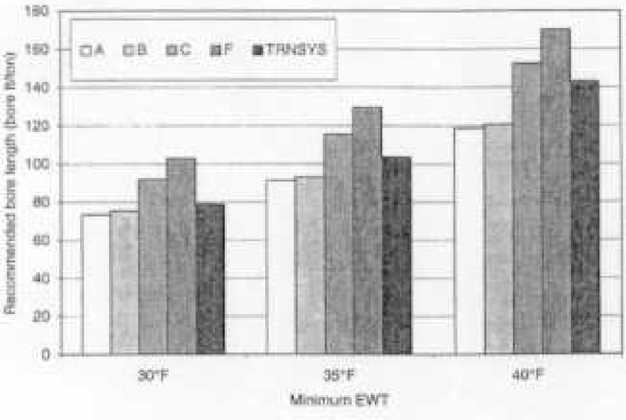

Figure 8 *One-year heat exchanger lengths for various minimum EWTs from the four design programs and the TRNSYS benchmark.*

TABLE 9
Ten-Year Design Lengths in Bore ft/ton (Bore m/kW) from the Four Programs

	Design Program				
Min. EWT	A	B	C	F	TRNSYS
30°F (−1.1°C)	85.3 (7.4)	77.6 (6.7)	97.6 (8.5)	100.6 (8.7)	84.1 (7.3)
35°F (1.7°C)	105.9 (9.2)	97.1 (8.4)	121.8 (10.5)	126.5 (11.0)	109.4 (9.5)
40°F (4.4°C)	137.1 (11.9)	127.1 (11.0)	160.0 (13.9)	168.8 (14.6)	148.2 (12.8)

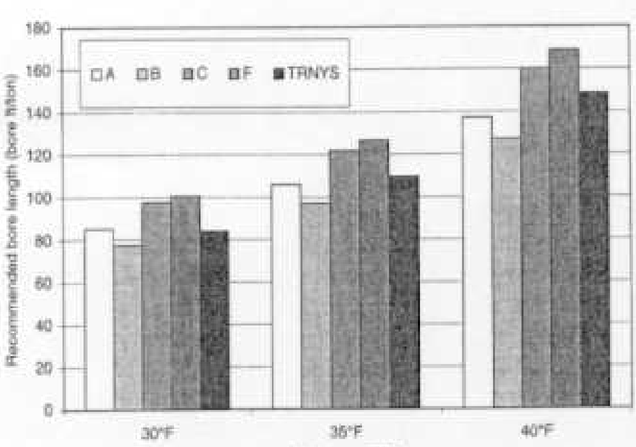

Figure 9 *Ten-year heat exchanger lengths for various minimum EWTs from the four design programs and the TRNSYS benchmark.*

COMPARISON OF RESULTS FROM THE FOUR DESIGN PROGRAMS

Table 8 compares the results of one-year heat exchanger designs from the four programs and the calibrated benchmark. In all cases, the system was determined to be dominated by the heating load; thus, designs were produced to limit the minimum entering water to temperatures of 30°F (−1.1°C), 35°F (1.7°C), and 40°F (4.4°C). These are the heat exchanger lengths required such that the minimum EWT does not fall below the given temperature in the first year of operation. One-year design lengths are most appropriate for applications where heat rejection and extraction roughly balance over the year but are sometimes used for commercial sizing when the borefield has modest multi-year effects. The lengths are plotted in Figure 8. On average, there is a difference of ±16% between the designs from the four programs and the TRNSYS benchmark design. Note, however, that programs A, B, and C agree more closely with the TRNSYS benchmark than does program F. On average, there is a difference of ±12% between the designs of programs A, B, and C and the benchmark, while the designs of program F differ by ±25%, on average, from the benchmark.

The ten-year heat exchanger design lengths are presented in Table 9 and plotted in Figure 9. On average, these lengths are about 7% higher than the one-year lengths, indicating only modest multi-year effects. Overall, the heat exchanger lengths differ by an average of ±12% from the TRNSYS benchmark, somewhat less than the ±16% difference for the one-year lengths. As with the one-year lengths, the designs of programs A, B, and C agree more closely with the benchmark than do the designs of program F. On average, there is a difference of ±11% between the designs of programs A, B, and C and the benchmark, while the designs of program F differ by ±17%, on average, from the benchmark.

CONCLUSIONS

An energy use model was developed for a 69,000 ft^2 (6410 m^2) elementary school in Lincoln, Nebraska. The model was calibrated with one year of site-collected data to ensure the accuracy of its predictions. The model was then driven with climate data from a typical meteorological year to generate a consistent set of inputs for four borefield design programs. Since loads at the site are dominated by heating, borefield designs from the four programs were generated for minimum entering water temperatures of 30°F (−1.1°C), 35°F (1.7°C), and 40°F (4.4°C). For comparison, benchmark BHEx designs were also obtained from the calibrated simulation at these three entering water temperatures.

Three of the programs tested, programs A, B, and C, agreed with the benchmark lengths to within about ±12%, which is comparable to the accuracy seen in the most recent comparison of designs for residential systems. However, one of the programs, F, differed by ±25% on average from the benchmark designs. These results indicate that the publishers of these programs may need to reexamine the methods used to calculate the design lengths.

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DISCUSSION

Byron Bakenhus, Senior Engineer, Lincoln Electric System, Lincoln, Nebr.: Gas was used for fresh air preconditioning at the schools, not because of loopfield and heat pumps limited capacity, but the inability of the fresh air heat pumps to operate with low entering air temperatures (for example 0°F).

Gas boiler and coils were decided to be a more cost effective option than water-to-water heat pumps and coils due to the limited energy required for this load.

John A. Shonder: Thank you for clarifying this. The authors note that it was due mostly to the efforts of Lincoln Electric System, Mr. Bakenhus in particular, that geothermal heat pumps were selected for use in the Lincoln schools.

J.B. Singh, President, J&P Engineers, Kendall Park, NJ: Having a 22% solution of antifreeze in the circulating fluid and a gas boiler seem to be redundant. A better application would have been to use an air-to-water heat pump for outside **air heating.**

Shonder: As Mr. Bakenhus stated in his comment, fresh air heat pumps will not operate at low entering air temperatures, so some type of pretreatment of outdoor air is necessary in this climate. Since gas was available at the site, the boiler and coils were decided to be a more cost effective option for the application.