4413

Optimal Sizing of Hybrid Ground-Source Heat Pump Systems That Use a Cooling Pond as a Supplemental Heat Rejecter— A System Simulation Approach

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ABSTRACT

Cooling-dominated commercial and institutional buildings served by ground-source heat pump (GSHP) systems generally reject more heat to a closed ground-loop heat exchanger (GLHE) than they extract over the annual cycle. This imbalance may result in a significantly larger GLHE than would be required for a system with annually balanced heat rejection and extraction. So-called "hybrid GSHP systems" use supplemental heat rejecters (such as cooling towers, fluid coolers, cooling ponds, or pavement heating systems) to reject excess heat on a seasonal or diurnal basis, thereby reducing the required size of the GLHE and, hence, the first cost of the system. The design challenge lies in finding the optimum size of both the GLHE and the supplemental heat rejecter, which directly depends upon the control strategy used to reject the excess heat. This study uses a system simulation approach to investigate various design alternatives with the aim of optimally sizing a GLHE with a cooling pond supplemental heat rejecter. A control strategy is selected based on the results of a previous study, and a life-cycle cost comparison is conducted for various design cases in two different climatic regions. This study demonstrates the usefulness of system simulation as a tool for determining the optimal design of hybrid ground-source heat pump systems.

INTRODUCTION

Ground-source heat pump (GSHP) systems offer an attractive alternative for both residential and commercial heating and cooling applications because of their higher energy efficiency compared with conventional systems, but their higher first cost has been a significant drawback to wider acceptance of the technology. This is especially true in commercial and institutional applications where the vertical closed-loop configuration is commonly preferred. These types of buildings are generally cooling-dominated and, therefore, reject more heat to the ground than they extract on an annual basis. As a result, the required ground-loop heat exchanger (GLHE) length is significantly greater than the required length if the annual loads were balanced. One option to reduce the size of the GLHE, and therefore the first cost of the system, is to effectively balance the ground thermal loads by incorporating a supplemental heat rejecter into the system. GSHP systems that incorporate a supplemental heat rejecter have been referred to as "hybrid GSHP systems."

Supplemental heat rejection can be accomplished with a cooling tower, fluid cooler, cooling pond, or pavement heating system. Currently suggested design methods for hybrid GSHP systems attempt to size the GLHE based on the annual heating load and then size the supplemental heat rejecter to balance the annual ground loads. However, the design of the system components also depends on the strategy used to control the supplemental heat rejecter. A smaller supplemental heat rejecter operated for more time may reject the same amount of heat as a larger supplemental heat rejecter operated for less time. Hence, a balance between the size of ground loop, size of the supplemental heat rejecter, and the control strategy is required to achieve the best economic alternative.

The work presented in this paper is a follow-up study to that presented by Yavuzturk and Spitler (2000), where various operating strategies of a cooling tower in a hybrid GSHP system were compared by simulating the system with TRNSYS (SEL 1997). The purpose of the Yavuzturk and Spitler (2000) work was not to find the optimum size of the system but to compare different control strategies for the

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supplemental heat rejecter, assuming the GLHE had been economically sized. The objective of this paper is different in that it uses the best supplemental heat rejecter control strategy found by Yavuzturk and Spitler (2000) and then uses system simulation to determine an optimum size for the ground-loop heat exchanger and shallow heat rejecting pond of a hybrid GSHP system. System performance is evaluated by a life-cycle cost comparison, determined through system simulations with TRNSYS (SEL 1997). An example small office building is chosen, and the hybrid GSHP system is optimized for two climatic regions.

BACKGROUND

There are only a modest number of documented studies and reports in the literature dealing with hybrid GSHP systems (Yavuzturk and Spitler 2000). These works are summarized below.

ASHRAE (1995) discusses the benefits and design of hybrid GSHPs for commercial/institutional buildings. A design procedure is suggested for cooling-dominated buildings that estimates the capacity of the supplemental heat rejecter based on the difference between the monthly average cooling and heating loads for the building. The ground-loop heat exchanger is sized to satisfy the building's heating load and the cooling load requirement for the ground loop in excess of that of the heating load is met through supplemental heat rejection. A series of general guidelines for installing and operating the supplemental heat rejecter is presented.

Kavanaugh and Rafferty (1997) review a few alternatives for ground-loop heat exchanger design. The high cost of excessively long ground loops is one of the primary factors that may lead to the consideration of a hybrid system. Other factors include limited land area, the cost of the land, or the high cost of high-efficiency heat pumps. The size of the supplemental heat rejecter is based on peak block load at the design condition. Similar to that of ASHRAE (1995), the nominal capacity is calculated based on the difference between the ground-loop heat exchanger lengths required for cooling and heating. Some recommendations are presented for the integration of the heat rejecter into the system.

Kavanaugh (1998) revises and extends the design procedures recommended by ASHRAE (1995) and Kavanaugh and Rafferty (1997). The revisions to the practice of hybrid ground-source heat pump system design involve balancing the heat flow to the ground on an annual basis in order to limit heat buildup in the borehole field. The annual operating hours of the supplemental heat rejecter needed to balance the heat rejection and extraction in the ground are calculated based on a set point control of the ground loop temperature (a typical range of 80°F [27°C] to 90°F [32°C]). The procedure is demonstrated on a multi-story office building placed in three different climatic regions. The author's results indicate that warm climates are most appropriate for the hybrid application since the savings in required bore length are much more significant than for moderate and cold climates.

Phetteplace and Sullivan (1998) describe a study that has been undertaken to collect performance data from an operating hybrid GSHP system at a 24,000 ft^2 (2230 m^2) military base administration building in Fort Polk, La. Results are reported for a 22-month period. The system consists of 70 vertical closed-loop boreholes, each 200 ft (61 m) deep with a 10 ft (3.3 m) spacing. The supplemental heat rejecter is a 275 kW cooling tower and is controlled with a differential controller that activates the cooling tower fans when the heat pump exiting fluid temperature reaches 97°F (36°C) and deactivates it when this temperature falls below 95°F (35°C). The authors report that for the monitoring period, approximately 43 times as much heat was rejected to the ground as was extracted. The relative energy consumption of the major system components over the study period is reported as 77% for the heat pumps, 19% for the circulation pumps, 3% for the cooling tower fan, and 1% for the cooling tower pump.

Singh and Foster (1998) report on first cost savings that resulted from using a hybrid ground-source heat pump design on the Paragon Center building located in Allentown, Penn., and an elementary school building in West Atlantic City, N.J. In the case of the Paragon Center building, a hybrid GSHP was installed because drilling difficulties precluded the feasibility of installing the required loop length. The hybrid system consists of 88 boreholes, each approximately 125 ft (38 m) deep, with a closed-circuit, 422 kW capacity fluid cooler. In the case of the elementary school in West Atlantic City, a hybrid GSHP was installed because of insufficient land area to accommodate the number of boreholes required to meet the building's cooling loads. A closed-circuit fluid cooler of 411 kW capacity is used, decreasing the required number of boreholes by more than 25%. In both of the reported examples, a significant system first-cost saving is achieved though slightly higher operating and maintenance costs are realized.

Gilbreath (1996) presents design suggestions for hybrid GSHP systems using the Paragon Center as an example and attempts to establish methods for monitoring system performance through the measurement of energy consumption, demand, and loop temperatures. The impact of various control options based on the percentage assistance of the heat rejecter is investigated. Effects of heat recovery and fluid flow control are also discussed. An installation and operating cost analysis is provided comparing the hybrid application to the conventional GSHP to assess and quantify potential cost savings.

Yavuzturk and Spitler (2000) use a system simulation approach to compare the advantages and disadvantages of various control strategies for the operation of a hybrid GSHP in a small office building. The supplemental heat rejection is accomplished with a cooling tower. The benefit of each control strategy is evaluated by comparing the 20-year life-cycle cost of the system for two different climatic regions. The control strategies investigated may be broadly categorized into three groups: (1) a set point control to operate the cooling tower when the heat pump entering or exiting fluid temperature exceeds a set value, (2) a differential control to operate the cooling tower when the difference between the heat pump entering or exiting temperature and the ambient wet-bulb temperature exceed a set value, and (3) a scheduled control to decrease heat buildup in the ground by operating the cooling tower for a given period of time during the night. In general, the system simulation results showed that all of the operating strategies resulted in significant total cost savings in the hybrid GSHP system over the conventional GSHP system. The most beneficial control strategies were found to be those that operate the supplemental heat rejecter primarily when heat rejection conditions are most favorable (i.e., the second category described above).

METHODOLOGY FOR SYSTEM SIMULATION AND ANALYSIS

Building Description and Loads Calculation

A small office building was chosen for simulating the performance of hybrid GSHP systems. The total area of the building is approximately 14,205 ft^2 (1320 m^2). The annual building loads are determined using *Building Loads Analysis and System Thermodynamics* (BLAST 1986) simulation software. The following assumptions have been used to determine the annual building loads.

1. The building is divided into eight different thermal zones.
2. For each zone, a single zone draw-through fan system is specified. The total coil loads obtained from system simulation are equal to the loads to be met with a ground-source heat pump system.
3. The office occupancy is taken as 1 person per 100 ft^2 (9.3 m^2) with a 70% radiant heat gain of 450 Btu/h.
4. A 1.1 W/ft^2 (11.8 W/m^2) of office equipment plug load, as suggested by Komor (1997), is used.
5. The lighting loads are assumed to be 1 W/ ft^2 (10.8 W/m^2).
6. Thermostat set points of 68.0ºF (20.0ºC) during the day (8 a.m. - 6 p.m.) and 58.0ºF (14.4ºC) during the night are used for all zones in the building. Only heating is provided during the night, depending on the requirement.
7. Schedules for office occupancy, lighting, equipment, and thermostat controls are specified.

The example building was simulated for two different climatic regions using typical meteorological year (TMY) weather data. The regions selected were Houston, Tex., for its hot and humid climate and Tulsa, Okla., for its moderate climate. The annual building loads determined on an hourly basis are shown in Figure 1.

Hybrid System Configuration and Component Model Description

A schematic of the hybrid GSHP is shown in Figure 2. The system uses two fluid circulation pumps so that there can be savings in pumping energy when the cooling pond is not

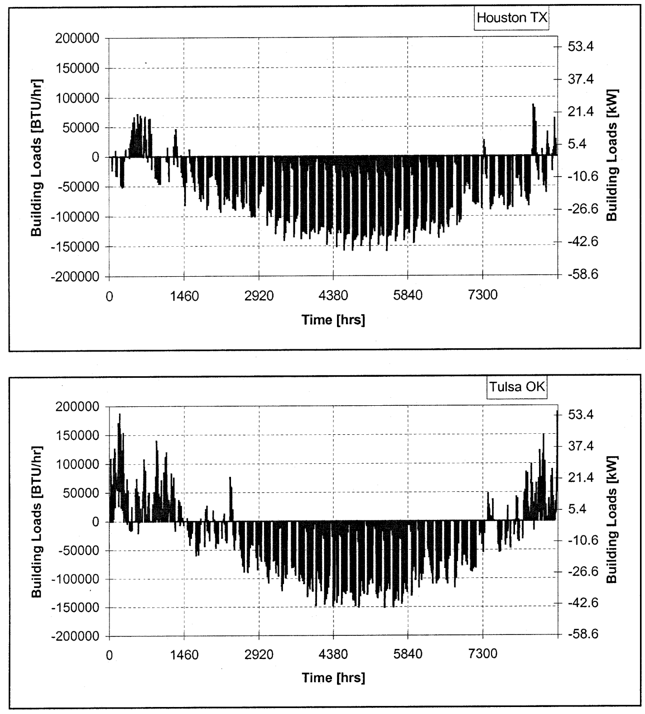


Figure 1 *Annual building loads for climatic conditions typical of Houston, Tex., and Tulsa, Okla. (cooling loads are negative and heating loads are positive).*

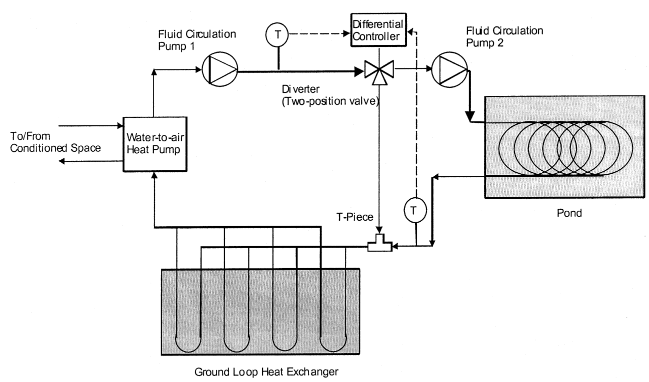


Figure 2 *Schematic diagram of the hybrid system component configuration.*

being used for heat rejection. A constant pumping rate is assumed. The system has been constructed in the TRNSYS modeling environment using standard and nonstandard component models. The standard TRNSYS component models for components such as pumps, t-pieces, flow diverters, and the differential controller are described by SEL (1997). The nonstandard component models are described below.

The building is not modeled explicitly in this application. The hourly building thermal loads described previously are read from a file and passed to the heat pump subroutine, which is a simple water-to-air heat pump model that has been developed for this and other GSHP system simulations. Inputs to the heat pump model include sensible and latent building loads, entering fluid temperature, and fluid mass flow rate. The model uses quadratic curve-fit equations to manufacturers' catalog data to compute the heat of rejection in cooling mode, heat of absorption in heating mode, and the heat pump energy consumption. Outputs provided by the model include exiting fluid temperature, energy consumption, and fluid mass flow rate.

The shallow pond model used in this study is that described by Chiasson et al. (2000). The model accounts for several natural heat transfer mechanisms within a shallow water body plus convective heat transfer from a closed-loop heat exchanger coil. Environmental heat transfer mechanisms that are simulated by the model include solar radiant heat gain, heat and mass transfer due to evaporation, convective heat transfer to the atmosphere, thermal or long-wave radiant heat transfer, conductive heat transfer to the surrounding soil or fill material, and groundwater discharge contributions. A lumped-capacitance approach is taken and the resulting first-order differential equation describing the overall energy balance on the pond is solved numerically. Outputs provided by the model include average pond temperature, exiting fluid temperature, and heat rejected to the pond.

The GLHE model used in this study is that described by Yavuzturk and Spitler (1999), which is an extension of the long time step temperature response factor model of Eskilson (1987). It is based on dimensionless, time-dependent temperature response factors known as g-functions, which are unique for various borehole field geometries. In order to compute the average temperature of the borehole field for each time step, the time-dependent building loads profile is decomposed into unit pulses and superimposed in time using the corresponding temperature response factors. The model includes a flexible load aggregation algorithm that significantly reduces computing time. The main output provided by the model includes the exiting fluid temperature.

Ground and Pond Loop Sizing

The peak entering fluid temperature (EFT) to the heat pump is one of the critical factors in the sizing of any GSHP system. The ground loop is sized to maintain the heat pump entering fluid temperature between approximately 25ºF and 105ºF (–3.4ºC and 40.6ºC). The design peak EFT usually varies from 85.0ºF to 95.0ºF (29.4ºC to 35.0ºC), depending upon the manufacturer and make of the heat pump. The peak EFT can be as high as 110.0ºF (43.3ºC) for high-efficiency rated heat pumps. Similarly, the heat pump entering fluid temperature is also constrained by a lower limit, depending on the heat pump and the heat exchanger fluid used in the ground loop. For colder climates, antifreeze is required to prevent the working fluid from freezing. Hence, the sizing of the ground loop and shallow heat rejection pond are bound by the upper and lower limits of the heat pump entering fluid temperatures.

The study of Yavuzturk and Spitler (2000) essentially showed that the most efficient control of the supplemental heat rejecter (in that case, a cooling tower) was to reject heat under favorable conditions. More specifically, the best results were achieved when the cooling tower was operated when the difference between the ambient wet-bulb temperature and the heat pump exiting fluid temperature exceeded a set value. In this present study, we adopt a similar control strategy, which is to reject heat to the pond when the difference between the average pond temperature and the heat pump exiting fluid temperature exceeds a set value. For Tulsa, an additional set point control based on the heat pump exiting fluid temperature is used to prevent operation of the pond heat exchanger when there is a danger of freezing the heat exchange fluid. The ground and pond loop sizing method is described below for each simulation case. In all cases, the simulation time was 20 years.

Case 1 (Base Case). For this case, the ground-loop heat exchanger was sized for use without any supplemental heat rejecter. The system simulation for this case included the heat pump, GLHE, and the circulation pump for the main loop. The optimal ground loop size for each climatic condition was found by adjusting the borehole depth such that the peak EFT was kept below 96.6ºF (35.8ºC). The borehole field for Houston for the base case consisted of 36 boreholes in a 6 × 6 configuration with a borehole depth of 250 ft (76.2 m) and a bore spacing of 12.5 ft (3.8 m). The borehole field for Tulsa for the base case consisted of 16 boreholes in a 4 × 4 configuration with a borehole depth of 240 ft (73.2 m) and a bore spacing of 12 ft (3.7 m).

A larger loop was required for Houston because of the greater imbalance in cooling load with respect to the heating load. The heat transfer fluid for both cases was water with a flow rate of 3.0 gpm (0.1893 m^3/s) per borehole. Undisturbed ground temperatures of 73ºF (22.8ºC) for Houston and 63ºF (17.2ºC) for Tulsa were chosen for the system simulation. Other configurations of the borehole geometry included a constant thermal conductivity of 1.2 Btu/h·ft·ºF (2.8 W/m·K) for the ground, borehole radius of 3.5 in. (88.9 mm), U-tube pipe nominal diameter of 1.25 in. (31.75 mm), and conductivity of the thermally enhanced grout at 0.85 Btu/h·ft·ºF (1.47 W/m·K) for both climatic conditions.

Cases 2, 3, 4, and 5. For these cases, the borehole field was reduced from 36 (6 × 6 configuration) boreholes to 12 boreholes (3 × 4 configuration) for Houston and from 16 (4 × 4 configuration) to 9 boreholes (3 × 3 configuration) for Tulsa. The shallow heat rejection pond-loop circuit was tied to the reduced size GLHE loop circuit, as shown in Figure 2. Heat was rejected to the pond by operating the circulation pump (pump 2 as shown in Figure 2) using the differential control strategy as described above for Houston. For Tulsa, in addition to the differential control strategy, a set point control was used.

The set point control shuts off the pond when the heat pump exit fluid temperature falls below 50ºF (10ºC). This ensured that the loop circulating fluid temperature did not fall below freezing and thereby avoided the use of antifreeze solutions. The temperature differential selected was 14.4ºF (8ºC) with a dead band range of 9ºF (5ºC). The pond model was set up to simulate a 2 ft (0.61 m) deep pond with a series of horizontally positioned, 500 ft (152.40 m) long, 3/4 in. (19.4 mm) nominal diameter, high-density polyethylene "slinky" heat exchanger coils. Each slinky was configured such that the resultant coil was 40 ft (12.19 m) long with a diameter of 3 ft (0.91 m), thus occupying an area of 120 ft^2 (11.1 m^2).

For each increasing case number (2 through 5), the pond area and the number of slinky coils were progressively increased, keeping the control strategy constant. The number of slinky heat exchanger coils that occupy the pond dictated the pond area. As the pond area was increased, the borehole depth was decreased so that the peak entering fluid temperature to the heat pump determined after a 20-year simulation was within acceptable limits. For the Houston case, the critical design temperature was the *maximum* heat pump EFT because of the predominant cooling load. However, for the Tulsa case, once a pond was added, the critical design temperature was the *minimum* heat pump exit fluid temperature (ExFT) because of the relatively higher peak heating loads. Therefore, borehole depths were adjusted for the Houston case to keep the maximum heat pump EFT below 96.6ºF (35.9ºC) and for Tulsa to keep the heat pump minimum ExFT above 35.6ºF (2ºC).

Table 1 summarizes the pond surface area, the number and depth of boreholes, and the differential control strategy for each case for Houston and Tulsa, respectively.

RESULTS AND DISCUSSION

Analysis of System Performance

Table 2 summarizes the borehole depths, operating hours, operating temperatures, and energy consumption for each case. The details are discussed below.

Case 1 (Base Case). The heat pump EFTs for the 20-year simulations are shown in Figure 3. The gradual increase in the

TABLE 1
Summary of Design Parameters for Each Simulation Case

Houston, Tex.

Case	No. of Pond Slinky Coils	Pond Area, ft^2 (m^2)	No. of Boreholes (arrangement)	Borehole Depth, ft (m)	Differential Control (HP_ExFT-T_Pond) Dead Band Temperature, ºF (ºC) High	Differential Control (HP_ExFT-T_Pond) Dead Band Temperature, ºF (ºC) Low	Set Point Temperature, ºF (ºC)
Case 1	None	N/A*	36 (6 × 6)	250 (76.2)	N/A	N/A	N/A
Case 2	2	240 (22.3)	12 (3 × 4)	258.7 (78.9)	14.4 (8)	5.4 (3)	N/A
Case 3	4	480 (44.6)	12 (3 × 4)	170.0 (51.8)	14.4 (8)	5.4 (3)	N/A
Case 4	6	720 (66.9)	12 (3 × 4)	101.96 (31.1)	14.4 (8)	5.4 (3)	N/A
Case 5	8	960 (89.2)	12 (3 × 4)	85.6 (26.1)	14.4 (8)	5.4 (3)	N/A

Tulsa, Okla.

Case	No. of Pond Slinky Coils	Pond Area, ft^2 (m^2)	No. of Boreholes (arrangement)	Borehole Depth, ft (m)	Differential Control (HP_ExFT-T_Pond) Dead Band Temperature, ºF (ºC) High	Differential Control (HP_ExFT-T_Pond) Dead Band Temperature, ºF (ºC) Low	Set Point Temperature, ºF (ºC)
Case 1	None	N/A	16 (4 × 4)	240 (73.2)	N/A	N/A	50 (10)
Case 2	1	120 (11.2)	9 (3 × 3)	300.8 (91.7)	14.4 (8)	5.4 (3)	50 (10)
Case 3	2	240 (22.3)	9 (3 × 3)	304.7 (92.9)	14.4 (8)	5.4 (3)	50 (10)
Case 4	4	480 (44.6)	9 (3 × 3)	322.4 (98.3)	14.4 (8)	5.4 (3)	50 (10)
Case 5	6	720 (66.9)	9 (3 × 3)	340.2 (103.7)	14.4 (8)	5.4 (3)	50 (10)

* Note: N/A = not applicable.

TABLE 2
Summary of the System Performance for All Cases

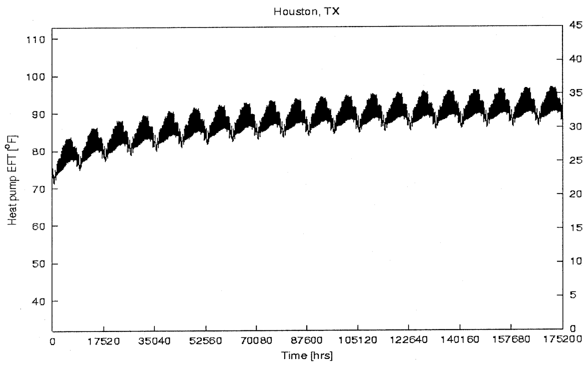
Houston, Tex.

		Case 1	Case 2	Case 3	Case 4	Case 5
Number of boreholes (arrangement)		36 (6 × 6)	12 (3 × 4)	12 (3 × 4)	12 (3 × 4)	12 (3 × 4)
Depth of boreholes, ft (m)		250 (76.2)	258.7 (78.9)	170.0 (51.8)	102.0 (31.1)	85.6 (26.1)
Pond area, ft^2 (m^2)		N/A*	240 (22.3)	480 (44.6)	720 (66.9)	960 (89.2)
Max. flow rate (gpm)	GLHE	108	36	36	36	36
	Pond	N/A	8	16	24	32
Operation of the pond (hrs)	Year 1	N/A	3800	3308	2497	2273
	Year 20	N/A	4549	3472	2451	2209
	Average	N/A	4422	3447	2452	2212
Average annual energy consumption (kWh)						
1. Main circulation pump		11996	4092	3146	2420	2246
2. Pond circulation pump		N/A	299	466	497	598
3. Heat pump		24245	23101	22198	21583	20452
Heat Pump EWT, ºF (ºC) during 20-year operation	Max.	96.6 (35.9)	96.6 (35.9)	96.6 (35.9)	96.6 (35.9)	96.8 (36.0)
	Min.	71.3 (21.8)	59.4 (15.2)	48.95 (9.42)	38.62 (3.68)	35.58 (1.99)
Heat Pump ExWT, ºF (ºC) during 20-year operation	Min.	N/A	56.9 (13.8)	46.6 (8.1)	36.7 (2.6)	35.9 (2.0)

Tulsa, Okla.

		Case 1	Case 2	Case 3	Case 4	Case 5
Number of boreholes (arrangement)		16 (4 × 4)	9 (3 × 3)	9 (3 × 3)	9 (3 × 3)	9 (3 × 3)
Depth of boreholes, ft (m)		240 (73.2)	300.8 (91.7)	304.7 (92.9)	322.4 (98.3)	340.2 (103.7)
Pond area, ft^2 (m^2)		N/A	120 (11.2)	240 (22.3)	480 (44.6)	720 (66.9)
Max. flow rate (gpm)	GLHE	48	27	27	27	27
	Pond	N/A	4	8	16	24
Operation of the pond (hrs)	Year 1	N/A	3002	2861	2475	2173
	Year 20	N/A	4177	3434	2589	2089
	Average	N/A	3940	3315	2569	2102
Average annual energy consumption (kWh)						
1. Main circulation pump		5190	3405	3436	3578	3720
2. Pond circulation pump		N/A	133	224	347	426
3. Heat pump		19927	19160	18080	17041	16664
Heat Pump EWT, ºF (ºC) during 20-year operation	Max.	96.4 (35.8)	93.7 (34.3)	87.7 (30.9)	81.9 (27.7)	80.3 (26.7)
	Min.	50.2 (10.1)	44.8 (7.1)	44.8 (7.1)	43.8 (6.7)	41.2 (5.1)
Heat Pump ExWT, ºF (ºC) during 20-year operation	Min.	N/A	35.6 (2.00)	35.6 (2.00)	35.6 (2.00)	35.6 (2.00)

* Note: N/A = not applicable.



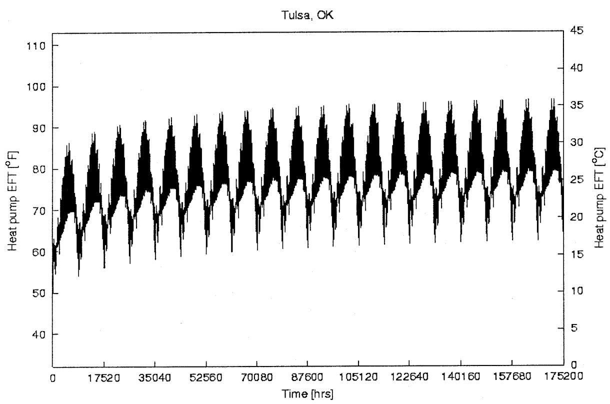


Figure 3 *Hourly entering fluid temperatures to the heat pump for Houston, Tex., and Tulsa, Okla., climatic conditions—case 1 (base case).*

maximum peak EFT from year to year is typical of cooling-dominated buildings. For the Houston case, the maximum peak EFT to the heat pump was 96.8ºF (36.0ºC), occurring at the end of the 20th year, and the minimum EFT is 71.2ºF (21.8ºC), which occurred during first year. For the Tulsa case, the peak EFT was 96.9ºF (36.1ºC) and minimum EFT to the heat pump was 50.2ºF (10.1ºC).

The total power consumed by the heat pump was 67% of the total energy consumption for the Houston case and 79% for the Tulsa case. The total energy consumption of the main circulation pump for Tulsa was significantly smaller than for Houston due to the shorter length of the Tulsa GLHE.

Case 2. For Houston, a pond of surface area 240 ft^2 (22.3 m^2) with two slinky heat exchanger coils was added as the supplemental heat rejecter, reducing the total GLHE length by 65.5%. The pond was observed to operate approximately 50% of the year. The total energy consumption of the system was reduced relative to the base case by 24.2%, mainly due to a reduction in pumping energy caused by the reduction in the GLHE size. A 4.7% decrease in heat pump energy consumption results from lower entering fluid temperatures.

For Tulsa, a pond of surface area 120 ft^2 (11.2 m^2) with one slinky heat exchanger coil was added as the supplemental heat rejecter, reducing the total GLHE length by 29.5%. The pond was observed to operate approximately 45% of the year. The total energy consumption was reduced relative to the base case by 9.6%. Note that the peak EFT for Tulsa case 2 was less than the original target limit of 96.6ºF (35.9ºC) (Table 2). This implies that the GLHE length could be further reduced. However, when the GLHE was further reduced, the heat pump minimum exiting fluid temperature during the heating season was observed to be unacceptable, falling below the freezing point of water. For this reason, the set point control was necessary to limit heat rejection from the pond. Alternatively, it is possible that a more sophisticated control strategy would limit heat rejection from the pond in such a way as to further improve the performance by shifting some of the seasonal heat rejection until after the bulk of the heating season has occurred.

Case 3. For Houston, the number of slinky heat exchanger coils was increased to four, reducing the total GLHE length by 77.3% from the base case. The pond was observed to operate approximately 39% of the year. The total energy consumption was reduced relative to the base case by 28.8%, about 4.6% lower than in case 2.

For Tulsa, the number of slinky heat exchanger coils was increased to two, reducing the total GLHE length by 28.6% from the base case. The pond was observed to operate approximately 38% of the year. The total energy consumption was reduced relative to the base case by 13.4%, about 3.8% lower than case 2. However, note that the GLHE size for case 3 has increased from case 2 by about 0.9%. This increase in the necessary size of the GLHE was due to the increased amount of heat rejected to the larger pond, which decreased the amount of heat rejected to the ground. With less heat rejected to the ground, a larger GLHE was needed to meet the peak heating load in the winter. At this stage, the point of diminishing returns for increasing the size of the pond has been reached for Tulsa. Figure 4 shows the hourly heat pump fluid temperatures for Tulsa for comparison purposes to the base case. The impact of the cooling pond on the heat pump EFT is evident: year-to-year increases in the maximum EFT, as observed in the base case, are eliminated.

Case 4. For Houston, the number of slinky heat exchanger coils was increased to six, reducing the total GLHE length by 86.4% from the base case. The pond was observed to operate approximately 28% of the year. The total energy consumption was reduced relative to the base case by 32.4%, about 3.6% lower than in case 3.

For Tulsa, the number of slinky heat exchanger coils was increased to four, reducing the total GLHE length by 24.4% from the base case. For reasons described above, the GLHE size for case 4 needed to be about 4.2% larger than for case 3 to meet the heating loads. The pond was observed to operate approximately 29% of the year. The total energy consumption

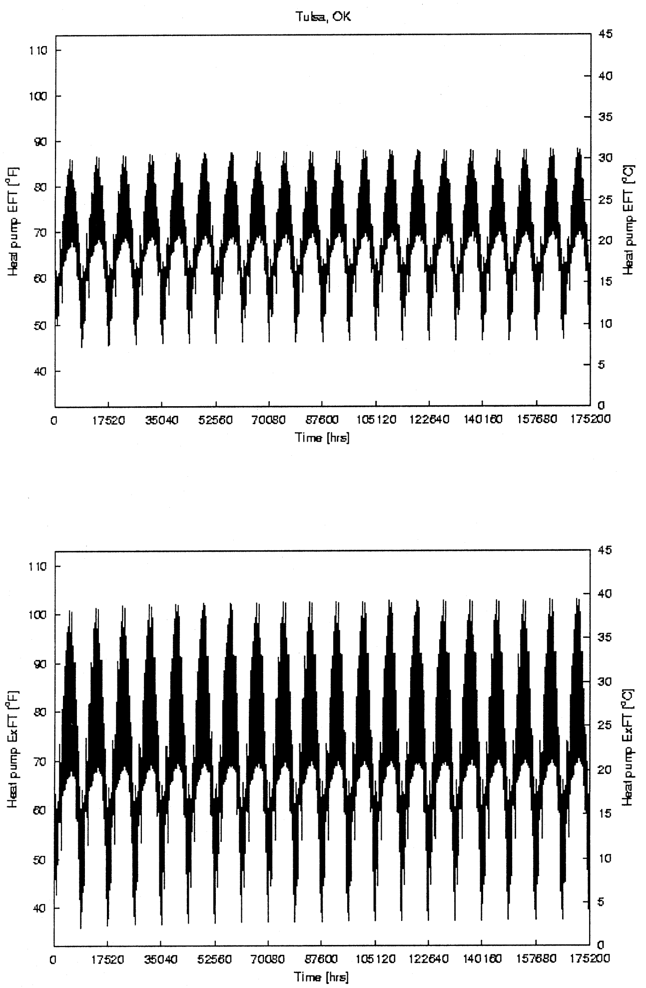


Figure 4 *Hourly heat pump entering fluid (EFT) and exiting fluid (ExFT) temperatures for Tulsa, Okla., climatic conditions—case 3.*

was reduced relative to the base case by 16.5%, about 3.1% lower than case 3.

Case 5. For Houston, the number of slinky heat exchanger coils was increased to eight, reducing the total GLHE length by 88.6% from the base case. The pond was observed to operate approximately 25% of the year. The total energy consumption was reduced relative to the base case by 35.7%, about 3.3% lower than in case 4. Since the pond loop is connected in series with the ground loop, this case represents the maximum possible size of the pond loop, without either reducing the flow in the individual pond loops or increasing the flow in the GLHE. The number of slinky coils results in the total flow rate through the pond loop to be equal to the total flow rate through the ground loop (i.e., all flow is diverted to the pond when the pond circulation pump is operational for this case). Although a larger pond would result in more surface area available for evaporative cooling to occur, increasing the number of slinky coils in the pond would result in a lower flow rate through each coil and lower convection coefficients inside the pipe. Figure 5 shows the hourly heat

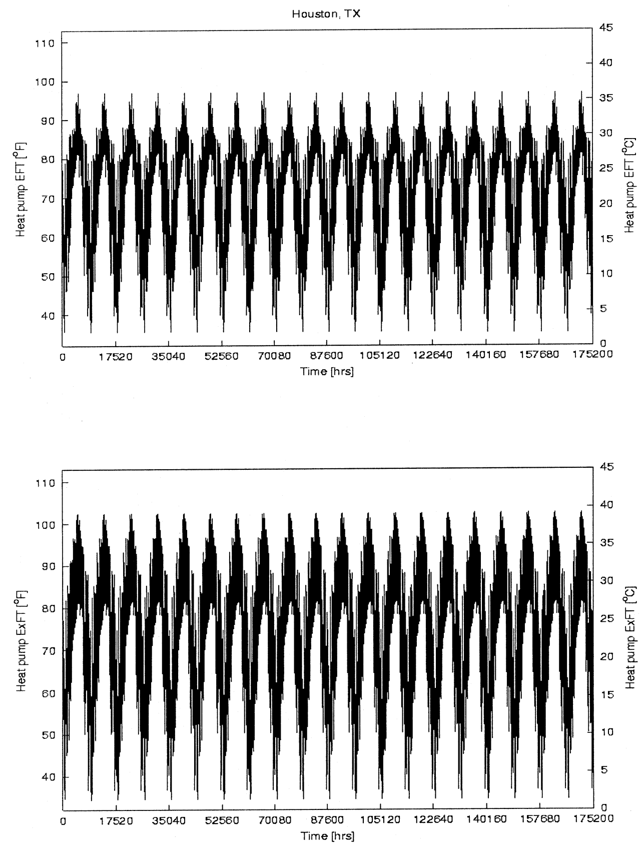


Figure 5 *Hourly heat pump entering fluid (EFT) and exiting fluid (ExFT) temperatures to the heat pump for typical Houston, Tex., climatic conditions; 20-year simulation—case 5.*

pump fluid temperatures for Houston for comparison purposes to the base case. The impact of the cooling pond on the heat pump EFT is evident: annual temperature increases observed in the base case are eliminated.

For Tulsa, the number of slinky heat exchanger coils was increased to six, reducing the total GLHE length by 20.3% from the base case. For reasons described above, the GLHE size for case 5 needed to be about 4.1% larger than for case 4 to meet the heating loads. The pond was observed to operate approximately 24% of the year. The total energy consumption was reduced relative to the base case by 17.1%, about 1.4% lower than case 4.

Summary of System Performance. The trend of decreased total system energy consumption is observed for all cases as the pond size increases. This decrease is observed because of decreased heat pump energy consumption, which is due to reduced heat pump entering fluid temperatures. For Houston, the pumping energy consumption also decreases with increasing pond size because of the associated decrease in GLHE size. While this was not true for Tulsa, the reduction in heat pump energy consumption still offsets the increases in pumping power beyond case 2.

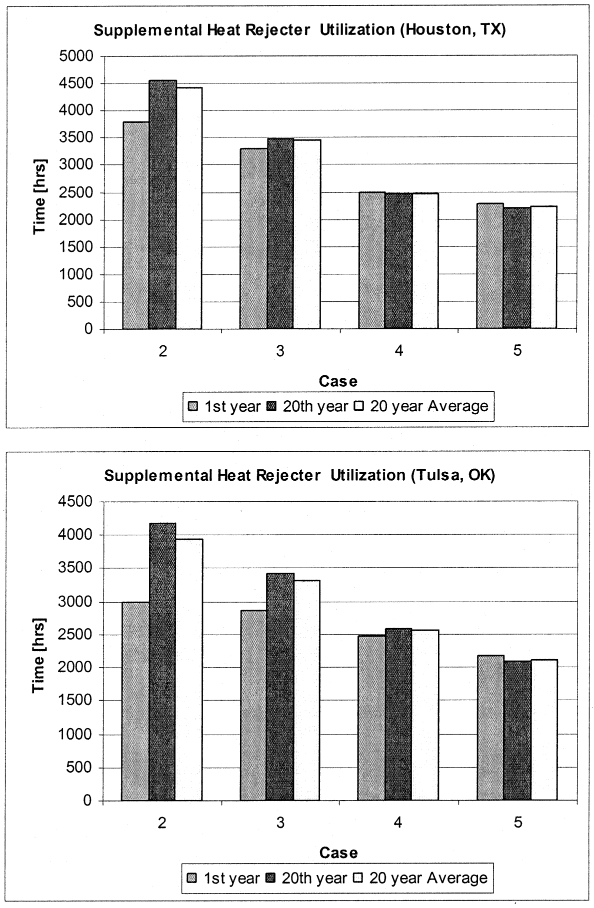


Figure 6 *Annual operating hours for the pond supplemental heat rejector for Houston, Tex., and Tulsa, Okla.*

Figure 6 shows the frequency of the pond usage in the first year, the 20th year, and the 20-year average for all cases. For cases 4 and 5 in both the Houston and Tulsa examples, the annual utilization of the pond remains nearly constant over the years. This constant pond usage means that it was sized in such a way that the annual heat rejection to the ground loop is approximately balanced with the annual heat extraction, ensuring approximately a steady periodic annual fluctuation in GLHE fluid temperatures. An imbalance in the ground loads, due to the under- or oversizing of the pond or due to the control strategy used, is reflected by the change in frequency of pond utilization as demonstrated by cases 2 and 3. An undersized pond results in less heat rejected by the pond and more heat rejected by the ground, thereby increasing the loop temperatures over time and increasing the operation of the pond. Conversely, an oversized pond results in less heat rejected by the ground loop, thereby decreasing the pond utilization over time.

Given the above, it would appear that the larger pond sizes are the best design options in both climatic regions. However, as further economic analysis shows, the economics of the system are dominated by other factors.

Life-Cycle Cost Analysis

A life-cycle cost analysis was performed to evaluate the economics of the various cases that were simulated. A present value approach was selected to compare the alternatives—the present value represents the life-cycle cost in present dollars. The results of the economic analysis are summarized in Tables 3 and 4. One major assumption in the analysis was that land is

TABLE 3
Life-Cycle Cost Analysis Summary for Each Case for Houston, Texas

	Case 1	Case 2	Case 3	Case 4	Case 5
Number of boreholes	6 × 6	3 × 4	3 × 4	3 × 4	3 × 4
Depth of the boreholes (ft)	250	258.7	170	101.96	85.63
Total bore length (ft)	9000	3104	2040	1224	1028
Ground loop installation cost	$54,000	$18,626	$12,240	$7341	$6165
Savings due to reduced size of the borehole field		$35,374	$41,760	$46,659	$47,835
Pond area (ft^2)		240	480	720	960
Pond excavation and other costs		$975	$1544	$2186	$2751
Number of spools in the pond	0	2	4	6	8
Cost of the slinky coils		$200	$400	$600	$800
Total first cost of the pond		$1175	$1944	$2786	$3551
Annual Operating cost:					
1. Main circulation pump	$840	$286	$220	$169	$157
2. Pond circulation pump		$21	$33	$35	$42
3. Heat pump	$1697	$1617	$1554	$1511	$1432
Total annual operating cost	$2537	$1924	$1807	$1715	$1631
Present value of the operating cost of the system	$29,098	$22,073	$20,723	$19,672	$18,704
Net present value of the system	$83,098	$41,874	$34,907	$29,798	**$28,421**

TABLE 4
Life-Cycle Cost Analysis Summary for Each Case for Tulsa, Oklahoma

	Case 1	Case 2	Case 3	Case 4	Case 5
Number of boreholes	4 × 4	3 × 3	3 × 3	3 × 3	3 × 3
Depth of the boreholes (ft)	240	300.8	304.7	322.4	340.2
Total bore length (ft)	3840	2707	2742	2902	3061
Ground loop installation cost	$23,040	$16,241	$16,452	$17,410	$18,369
Savings due to reduced size of the borehole field		$37,760	$37,548	$36,590	$35,631
Pond area (ft^2)		120	240	480	720
Pond excavation and other costs		$692	$975	$1544	$2186
Number of spools in the pond	0	1	2	4	6
Cost of the slinky coils		$100	$200	$400	$600
Total first cost of the pond		$792	$1175	$1944	$2786
Annual Operating cost:					
1. Main circulation pump	$363	$238	$241	$250	$260
2. Pond circulation pump		$9	$16	$24	$30
3. Heat pump	$1395	$1341	$1266	$1193	$1166
Total annual operating cost	$1758	$1589	$1522	$1468	$1457
Present value of the operating cost of the system	$20,166	$18,224	$17,455	$16,834	$16,709
Net present value of the system	$43,206	$35,257	**$35,082**	$36,188	$37,863

available for the pond construction at no cost and that the resale value of the property is not diminished by the pond construction. Additional assumptions are as follows:

a. Ground loop costs: $6.00/ft ($19.69/m) of bore, which includes the material cost of the pipe, horizontal runs and connections, and labor.

b. Pond loop costs: $21/yd^3 ($27/m^3) of pond volume, which includes costs of excavation, labor, fabrication, and installation of slinky coils and assumes an excavation rate of 12.5 yd^3/h (9.56 m^3/h). Additional costs are equipment rental inclusive of pickup and delivery ($350 half day, $430 full day), HDPE pipes ($0.20/ft, $0.66/m), pond liner ($0.75/ft^2, $8.07/m^2), and pump and controls.

c. Electricity rate cost: $0.07 per kWh.

d. Interest rate: 6% annual percentage rate (compounded annually over the 20-year design period).

e. A head loss of 2.67 ft/100 ft (2.67 m/100 m) of pipe length is taken for the ground loop and pond loop.

f. The cost of make-up water, if it must be purchased, has been neglected. Of all the cases considered here, case 5 for Houston will require the most make-up water—31,000 gallons (118 m^3). This is equivalent to adding 52 in. (1.3 m) of water. The annual rainfall for Houston is about 48 in. (1.2 m), so much of the make-up water would be provided naturally, particularly if some of the surrounding terrain or building rooftop drained into the pond. Nevertheless, if all the make-up water had to be provided from the water utility, it would still be very inexpensive: at a rate of $0.075/1000 gallons ($0.20/m^3), the annual cost would be about $23.

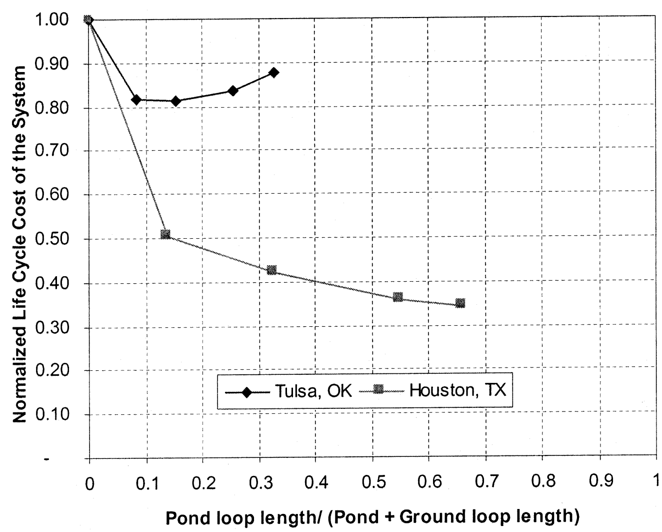


Figure 7 *Normalized life-cycle cost of the GSHP system vs. the ratio of pond loop length to total loop length for Houston, Tex., and Tulsa, Okla., climatic conditions.*

An analysis of the data presented in Tables 3 and 4 shows that case 5 is the lowest cost alternative for Houston and case 3 is the lowest cost alternative for Tulsa. It is evident from this economic analysis that the system life-cycle cost is mainly dominated by the ground loop first costs, given, of course, that an acceptable control strategy is employed.

The system life-cycle cost for each case was normalized to the base case and plotted versus the ratio of pond loop length to total loop length in Figure 7. A review of this figure demonstrates that the higher the annual demand for cooling in a particular building, the greater the economic benefit can be realized by incorporating a pond supplemental heat rejecter. For the Tulsa example, the lowest life-cycle cost alternative (case 3) has a ratio of 8.5% pond loop length to total loop length, and there is a cost savings of 18% relative to the base case. At this same ratio for the Houston example, a much greater cost savings of about 50% is realized. However, the optimum cost savings for the Houston example is 66% at the point when the ratio of pond loop length to total loop length is at a maximum of 66%.

Sensitivity Analysis of the Control Strategy

The impact of varying the differential control strategy on the system performance for case 5 (Houston) and case 2 (Tulsa) was examined. The borehole depth and pond size remained unaltered and the system performance was again simulated for 20 years and the economic analysis repeated. Eight simulations were conducted.

The upper dead band was fixed at 14.4ºF (8.0ºC), while the lower dead band was increased from 1.8ºF (1ºC) to 9ºF (5ºC) in steps of 1.8ºF (1ºC). Similarly, the dead band lower limit was fixed at 5.4ºF (3ºC), while the upper limit is increased from 10.8ºF (6ºC) to 18ºF (10ºC).

Variations in the dead band temperatures resulted in only marginal differences in the system life-cycle cost. These marginal differences were due to the fact that the system cost was governed by the heat pump energy consumption. Changes to the dead band temperatures mainly impacted the cyclic operation of the pond but had little impact on heat pump performance. The life-cycle cost for different control strategies varied within 1% when compared to case 5 for Houston and within 0.2% when compared to case 2 for Tulsa.

CONCLUSION AND RECOMMENDATIONS

A system simulation approach to determining the optimum size of a hybrid GSHP system that uses a cooling pond as a supplemental heat rejecter has been presented. Since the design is strongly influenced by the strategy used to control the supplemental heat rejection, the most efficient control strategy from the work of Yavuzturk and Spitler (2000) was adopted for this study. A control scheme was used to operate the pond when the difference between the heat pump exiting fluid temperature and the average pond temperature exceeded a set value.

This study has shown through system simulation that the optimum size of a GSHP system with a supplemental pond heat rejecter can be approached by adjusting borehole depths and pond loop lengths until a minimum life-cycle cost has been found. This has been done for four configurations of a hybrid pond GSHP system for typical climates of Houston, Tex., and Tulsa, Okla. A sensitivity analysis of the differential control strategy was also conducted.

Some specific conclusions of this study are as follows.

1. The Houston example shows, for highly cooling-dominated buildings, that regardless of the size of the pond supplemental heat rejecter, significant economic benefits on the 20-year life-cycle cost can be realized. The savings in the GSHP system cost by including the pond supplemental heat rejecter in this example is approximately 50-65%.
2. The Tulsa example shows, in buildings with a dominant demand for cooling but also with a significant heating load, that the most economical design of the hybrid system is dependent on the heating load. There is a point of trade-off in the pond size where too much heat is rejected from the system, and, hence, the heat pump EFT is too low during part of the winter season. To prevent this from occurring, a more sophisticated control strategy is needed. We adopted a set point control for limiting the minimum heat pump exiting fluid temperature. Alternatively, it may have been possible to use the pond itself for supplemental heat extraction.
3. The choice of the dead band range used in the differential control strategy appears to have no significant impact on the economics of the system.

This work opens a number of areas for further study. The pond hybrid GSHP systems that were simulated in this work were not completely optimized, and there are still some system options that remain to be examined. Some of these include the following:

a. Implementation of an optimization routine into the system simulation to find “true” optimum values of desired parameters. In particular, it would be useful to find the optimal balance between the GLHE size and the pond size. This would also allow much more flexibility in the choice of the parameters to be optimized, as well as streamline the design process.

b. Examination of variable-speed pumping rates on the system performance. One shortcoming of this study was that a constant pumping rate was assumed. Variable-speed pumping has the potential to significantly reduce operating costs.

c. A quantitative comparison between the life-cycle cost of optimally designed hybrid GSHP systems with a shallow pond versus hybrid GSHP systems with other supplemental heat rejecters, such as cooling towers and shallow horizontal ground-coupled coils.

d. Further refinements to the control strategy. For example, consideration of time-of-day electricity rates in the heat rejecter operation strategy may be beneficial.

e. Life-cycle cost comparison of a cooling pond hybrid GSHP system to a pond-only system. This would require additional considerations not currently implemented in the pond model used in this work, such as pond freezing and seasonal stratification of the pond water.

ACKNOWLEDGMENTS

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DISCUSSION

Van Baxter, R&D staff member, Oak Ridge National Lab, Oak Ridge, Tenn.: Follow up to question about why pumping cost decreased: Regarding pumping energy and ground loop flow, the author indicated that ground loop flow was reduced as ground loop size was reduced. How did lower loop flow affect building heat pump performance? Did you maintain a minimum gpm/ton level or did the flow just float wherever? WSHPs are subject to minimum gpm/ton limits per the manufacturer's specification.

Jeffrey Spitler: Mr. Baxter is correct that water source heat pumps are subject to minimum flow rates per the manufacturer's specification. Typically, the flow rate is also set to maintain turbulent flow in the ground loop, at least at peak flow rate conditions. In our case, we used 3 gpm per borehole, which for water is in excess of what would be needed to maintain turbulent flow in 3/4 in. pipe. For the Houston base case, this assumption, coupled with the large required borehole field, resulted in large pumping costs.

For Houston, the base case had approximately 8.1 gpm/ton, and cases 2-5 had flow rates of 2.7 gpm/ton. For Tulsa, the base case had a flow rate of 3.8 gpm/ton and cases 2-5 had flow rates of 2.2 gpm/ton. These are all feasible, though 2.2 gpm/ton is lower than typical.

For this study, we did not use a detailed heat pump model. This would be a useful extension to the work in the paper, and would allow consideration of trade-offs between flow rates and equipment capacity.

Stephen Tweedie, Director of Engineering Services, Enerplan Consultants Ltd., Moncton, New Brunswick, Canada: Information was well presented, clear, and easy to follow. How can I get more information from the author with regard to current projects I am undertaking?

Spitler: Thank you. Dr. Spitler's e-mail address is spitler@okstate.edu.

Dennis Knight, CEO, Engineering Technology, Inc., Mt. Pleasant, S.C.: (1) Regarding your projected pump savings, are the savings due to lower loop flow per connected ton of HPs or are they due to lower pressure drop from less loop piping? (2) Was there actually less loop piping required or just less bore hole length required?

Spitler: To answer question (1), the savings are due to lower loop flow per connected ton; the pressure drop remained approximately the same because the flow rate in each borehole was held constant, and the pipe diameter was kept the same between cases. With respect to question (2), all boreholes had a single U-tube; therefore, both total borehole length and loop piping were reduced in cases 2-5 when compared to the base case.