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Comparative Study of Operating and Control Strategies for Hybrid Ground-Source Heat Pump Systems Using a Short Time Step Simulation Model

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ABSTRACT

Ground-source heat pumps for cooling-dominated commercial buildings may utilize supplemental heat rejecters such as cooling towers, fluid coolers, or surface heat rejecters to reduce system first cost and to improve system performance. The use of supplemental heat rejecters for cooling-dominated buildings allows the use of smaller borehole fields. Degradation of the heat pump performance is avoided by offsetting the annual load imbalance in the borefield. A comparative study is presented that investigates the advantages and disadvantages of various system operating and control strategies in a hybrid ground-source heat pump application using an hourly system simulation model under different climatic conditions. An actual small office building is used as the example building. The use of a short time step simulation model enables the detailed assessment of the ground heat exchanger's behavior and the determination of system energy consumption on an hour-by-hour basis. A life-cycle cost analysis is conducted to compare each operating and control strategy to determine the lowest cost alternative for a given climate.

INTRODUCTION

The advantages of ground-source heat pumps over their conventional alternatives make these systems a very attractive choice for space conditioning, not only for residential buildings but increasingly also for institutional and commercial buildings. A significant number of commercial buildings are cooling-dominated, especially in southern climates. When used in cooling-dominated buildings, ground-source heat pumps that utilize vertical, closed-loop ground heat exchangers can experience performance degradation as the entering fluid temperature to the heat pump increases over time. This temperature increase is due to the imbalance between the amount of heat extracted from the ground and the amount of heat rejected into the ground. For systems with severely undersized ground heat exchangers, the entering fluid temperature to the heat pump may be so high that the heat pump fails.

Nevertheless, it is possible to avoid this problem by either increasing the total length of the installed ground-loop heat exchanger and/or increasing the spacing between the ground-loop heat exchanger boreholes. However, first costs may be significantly higher so that a ground-source heat pump system may not be competitive with conventional alternatives. For many commercial buildings, there may not be enough land area for a properly sized ground-loop heat exchanger.

In order to decrease the system first cost and to improve the system performance, one of the available options is a hybrid ground-source heat pump application. Hybrid systems utilize supplemental heat rejecters, such as open cooling towers, closed-circuit fluid coolers, or surface heat rejecters interconnected on the building return side between the heat pump and the ground-loop heat exchangers. The supplemental heat rejecter is typically sized so that the annual heat rejection to the ground approximately balances the annual heat extraction from it. Excess heat is then rejected through one or more supplemental heat rejecters. With the supplemental heat rejecter(s), the ground-loop heat exchanger may be significantly smaller.

It should be noted, however, that supplemental heat rejecters, especially open cooling towers and fluid coolers, require periodic maintenance. Additional operating costs also result from cooling tower and pump electricity consumption. If the fluid circulation system is not carefully designed, the cost of fan and pump energy consumption may become signif-

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icant, negating the potential savings attained through a hybrid system. The first cost of supplemental heat rejecters and increased operating costs due to additional fan and pump circulation energy consumption are expected to be small compared to the savings in drilling costs and heat pump operating costs for cooling-dominated buildings.

The actual amount of heat transferred to and from the ground-loop heat exchanger varies continuously due to changing building energy requirements. These changes result in short time-step fluctuations in the supply and return temperatures of the ground heat exchanger that can typically vary up to 10ºF-18°F (5.6ºC-10.0°C) over a given day. The coefficient of performance (COP) of the heat pump is affected by these short-time temperature variations. In cases where time-of-day electricity rates are applicable, the impact of fluctuating performance on the system operating cost may be even more significant. For a detailed building energy analysis, a ground-loop heat exchanger simulation model is called for that can reliably and efficiently predict the short-term fluctuations in the heat pump entering fluid temperatures. This enables the determination of energy consumption and demand on an hour-by-hour basis.

Although the size and the number of total annual operating hours of the supplemental heat rejecters may be estimated based on the annual building loads and the maximum available size of the borefield for a given area, the decision of under what conditions to activate the heat rejection and its short-time impact on the ground-loop heat exchangers is somewhat complex. Recently published works (Kavanaugh 1998; Kavanaugh and Rafferty 1997; Phetteplace and Sullivan 1998) only use a set point control, usually an upper temperature limit, for entering fluid temperature to the heat pump returning from the ground heat exchanger and do not consider more sophisticated system control strategies. In order to quantify the impact of various operating strategies on ground-loop heat exchanger size and operating cost, a simulation model that can account for changes in the hourly load profile and interaction between the ground-loop heat exchanger and heat rejecter is highly desirable.

Therefore, in this study, a short time step simulation model that allows for an hour-by-hour building energy analysis is used. Using hourly weather data from a typical meteorological year for a specific location, the simulation model is capable of predicting the entering and exiting heat transfer fluid temperatures on the borefield on hourly or subhourly intervals. An hour-by-hour system analysis allows for more sophisticated and flexible control strategies. An example strategy may be the "recharge" of the borefield at certain time intervals during the day to lower the heat pump entering fluid temperatures.

The objective of this paper is to present a comparative study that investigates the advantages and disadvantages of several system operating and control strategies using an hour-by-hour system simulation model for two different climates. The short time step simulation approach taken allows for a more detailed assessment of the ground-loop heat exchanger behavior as well as for analysis of the impact of various control strategies on system operating costs. A small office building (located in Stillwater, Oklahoma) is used as the example building. The building loads analysis for each climatic region was performed using BLAST (1986). The simulations for the short time step building energy analysis and ground-loop heat exchanger temperatures were performed using TRNSYS (Klein et al. 1996).

BACKGROUND

A review of recent literature on hybrid ground-source heat pump systems yielded only a modest number of references to research articles and a few references to reports dealing with actual applications.

ASHRAE (1995) discusses the advantages of hybrid ground-source heat pump applications considering capital costs and available surface area limitations for a 100% ground-coupled system. A design procedure is suggested for cooling-dominated buildings that sizes the capacity of the supplemental heat rejecters based on the difference between the monthly average cooling and heating loads of a given building rather than the peak loads. The ground loop is sized to meet the building heating loads, while the cooling load in excess of the heating load is met through supplemental heat rejection. For closely spaced vertical boreholes, it is suggested that it may be advantageous to operate the supplemental heat rejection unit during night hours for cold storage in the ground. A series of general guidelines is given, which discusses the integration of the supplemental heat rejecters into internal piping, the need for an isolation plate heat exchanger when an open cooling tower is used, the set point control of heat rejection based on an upper limit of heat pump entering fluid temperatures, cold storage in the ground through night operation, and the possible year-round operation of the rejecters in southern climates.

Kavanaugh and Rafferty (1997) discuss hybrid ground-source heat pump systems within the framework of ground-loop heat exchanger design alternatives. Primary factors that may mandate the consideration of a hybrid system are the high cost of long loops when the design relies on the ground to meet 100% of the building's heating and cooling requirements, the unavailability or cost of space and the high cost of high-efficiency heat pumps. The sizing of the supplemental heat rejecters is based on peak block load at the design condition. The nominal capacity is calculated based on the difference between the ground-loop heat exchanger lengths required for cooling and heating. Recommendations are made for the integration of the supplemental heat rejecters into the ground-source heat pump piping system.

Kavanaugh (1998) revises and extends the existing design procedures as recommended in ASHRAE (1995) and in Kavanaugh and Rafferty (1997). This revised design procedure addresses issues such as ground heat exchange and heat buildup, system control methods, piping arrangements, freeze protection, auxiliary energy consumption, and maintainability. The revised method, in addition to sizing the ground-loop heat exchanger of the hybrid system and the supplemental heat rejecter, proposes a method for balancing the heat transfer in the ground formation on an annual basis in order to limit heat pump performance degradation due to heat buildup in the borefield. The annual operating hours of the supplemental heat rejecter that are 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] is given). The revised procedure is then applied to a multi-story office building considering three different climates to investigate the appropriateness of the hybrid application. Installation cost savings and operating cost issues are discussed. The author concludes that the economic value of hybrid systems is most apparent in warm and hot climates where cooling loads are the highest. Although hybrid systems with heat recovery options are deemed somewhat attractive for regions of moderate climate, no economic value could be justified for cold climates even with heat recovery.

Phetteplace and Sullivan (1998) describe a 24,000 ft^2 (2230 m^2) military base administration building in Fort Polk, Louisiana, that uses a hybrid ground-source heat pump system. The system uses 70 vertical closed-loop boreholes, each 200 ft (61 m) deep with 10 ft (3.3 m) spacing. The paper presents performance data for a period of 22 months, including performance data from portions of two heating and cooling seasons. The observed data show that, over the period of monitoring, the amount of heat rejected to the ground is about 43 times higher than the amount of heat extracted from it. This is indicative of a very heavily cooling-dominated building. The supplemental heat rejecter is a 275 kW (938 kBtu/h) 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 some heat buildup in the ground due to an imbalance of heat extraction and rejection in the ground. This is attributed to differential controller set point temperatures that are too high. Lowering of these control points is expected to dissipate the heat buildup at the cost of increasing the operating hours of the cooling tower. The relative energy consumption of the major system components over the study period is provided where the heat pumps account for 77% of the total energy consumption, the circulating pumps for 19%, the cooling tower fan for 3%, and the cooling tower pump for 1%.

Singh and Foster (1998) explore 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, New Jersey. The Paragon Center illustrates the need for a hybrid application as a direct result of geological conditions at the site where boreholes drilled deeper than 110 ft (33.5 m) collapsed due to high groundwater flow in limestone strata. The building area is 80,000 ft^2 (7436 m^2). The hybrid system consists of 88 boreholes, each approximately 125 ft (38 m) deep, and a closed-circuit fluid cooler of 422 kW maximum capacity. The elementary school expansion building in West Atlantic City is an example of a hybrid system where the available space for the borehole field was not sufficient to accommodate the number of boreholes required to fully meet the building's cooling loads. The building area is approximately 63,000 ft^2 (5856 m^2). A closed-circuit fluid cooler of 411 kW (1402 kBtu/h) capacity is used, decreasing the required number of boreholes by more than 25% to 66 bores, each about 400 ft (122 m) deep. In both of the reported examples, a significant system first-cost savings is achieved, though with slightly higher operating and maintenance costs.

A more detailed study of the hybrid ground-source heat pump system in the Paragon Center office building is provided by Gilbreath (1996). The study gives design suggestions for hybrid 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 cooling tower in rejecting excess heat is investigated. Effects of heat recovery and fluid flow control are discussed. An installation and operating cost analysis is provided comparing the hybrid application to the ground-source heat pump system without supplemental heat rejection to assess and quantify potential cost savings.

SUPPLEMENTAL HEAT REJECTION

Open-circuit cooling towers and closed-circuit fluid coolers are commonly used for supplemental heat rejection in hybrid ground-source heat pump systems. Open-circuit cooling towers are typically used in conjunction with isolation plate heat exchangers, in order to avoid a mixing of the loop heat transfer fluid and the cooling water. Air-cooled closed-circuit fluid coolers are modular units that accomplish the cooling effect by directly rejecting heat to the atmosphere. However, the first cost and the fan energy consumption of these devices are generally high. Kavanaugh (1998) estimates lower first costs for open-circuit cooling towers with isolation plate heat exchangers than for fluid coolers.

Recent research on hybrid ground-source heat pumps focuses on surface heat rejecters, such as shallow heat rejecters under pavements or in ponds (Chiasson 1999). Surface heat rejecters consist of a series of pipes inserted in the concrete layers of pavements for heating of parking lots during winter months or laid out close to the bottom surface of ponds. In this study, the supplemental heat rejecter is a mechanical draft, open-circuit cooling tower used in combination with an isolation plate heat exchanger.

SYSTEM OPERATION ANALYSIS USING THE SHORT TIME STEP SIMULATION MODEL

Example Hybrid Application System Description

The example small office building was completed in 1997 and is located in Stillwater, Oklahoma. The total area of the building is approximately 14,205 ft^2 (1320 m^2). The building was a candidate for a ground-source heat pump system application although a conventional system was installed.

In order to determine the annual building loads for the example building using BLAST (1986), the following approach was taken:

1. Eight different thermal zones were identified in the building. For each zone, a single-zone draw-through fan system is specified as a surrogate for a ground-source heat pump. The coil loads on this system are equivalent to those of a ground-source heat pump system.
2. The office occupancy is set to one person per 100 ft^2 (9.3 m^2) with a heat gain of 450 Btu/h (131.9 W), 70% of which is radiant.
3. The office equipment heat gains are set to 1.1 W/ft^2 (12.2 W/m^2) as suggested by Komor (1997).
4. The lighting heat gains are set to 1 W/ft^2 (11.1W/m^2).
5. Daytime (8 a.m. - 6 p.m. Monday - Friday), nighttime, and weekend thermostat settings are specified for each zone. During the day, the temperature set point is 68.0°F (20.0°C). For the night, only heating is provided, if necessary, and the set point is 58.0°F (14.4°C).

Climatic Considerations—Building Loads

The example building is analyzed considering two different climatic regions, each represented by the Typical Meteorological Year (TMY) weather data: a typical hot and humid climate is simulated using Houston, Texas; a more moderate climate is simulated using Tulsa, Oklahoma. The results of the BLAST building loads analysis are shown in Figure 1 for both regions considered. The building loads are determined on an hour-by-hour basis for 8760 hours. The cooling loads are shown as negative loads on the building.

As expected, the cooling loads are greatest for Houston typical weather conditions, where the example building is heavily cooling-dominated. As the example building is considered in a relatively cooler climate (Tulsa), the building becomes somewhat less cooling-dominated, and an increase in heating loads is observed.

Hybrid System Component Configuration

A schematic of the hybrid ground-source heat pump system application is shown in Figure 2. The hybrid system uses an open cooling tower with an isolation plate heat exchanger. Two independent fluid circulation loops are designed that are serviced with fluid circulation pumps #1 and #2.

The design contains a bypass (Diverter-1, T-piece-1) so that pumping energy may be conserved when the cooling tower is not being used.

The operation and performance of the hybrid system was simulated using TRNSYS (Klein et al. 1996). Standard TRNSYS library component models were used for components, such as the diverters, T-pieces, fluid circulation pumps, plate heat exchanger, and the cooling tower. A simple heat pump component model as described by Yavuzturk and Spitler (1999) was used. The ground-loop heat exchanger model is described in the next section.

Short Time Step Ground-Loop Heat Exchanger Model

The short time step ground-loop heat exchanger model was developed by Yavuzturk and Spitler (1999) as an extension of the long time step temperature response factor model

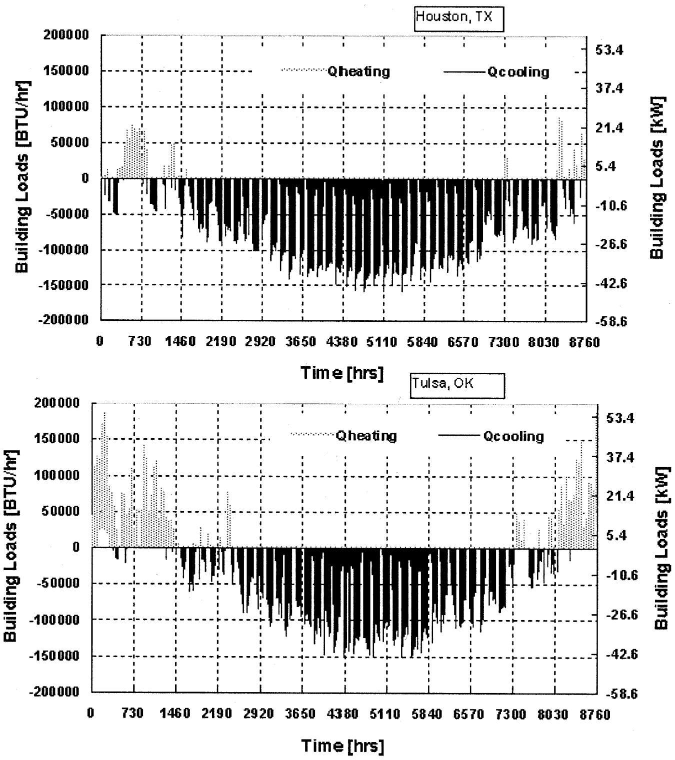

Figure 1 *Annual hourly building loads considering typical climatic conditions in Houston, Texas, and Tulsa, Oklahoma.*

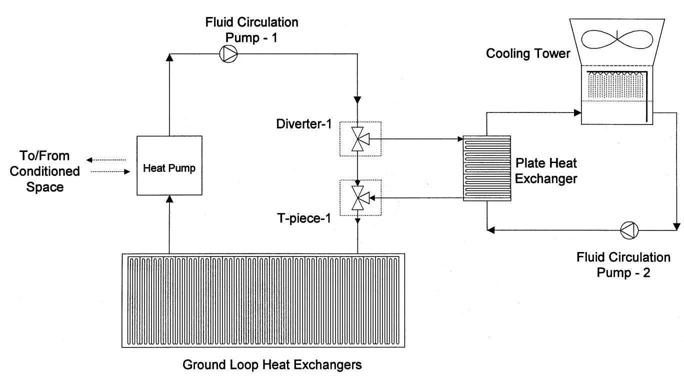

Figure 2 *Hybrid ground-source heat pump system component configuration diagram.*

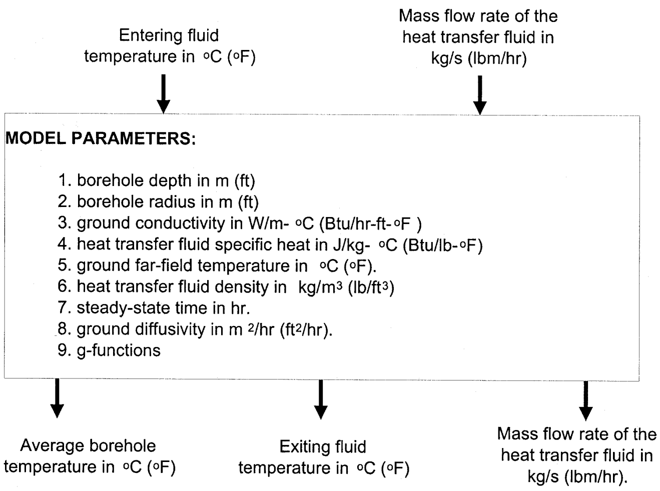

Figure 3 *Short time step ground-loop heat exchanger, model component configuration.*

of Eskilson (1987). It is based on dimensionless, time-dependent temperature response factors, g-functions, which are unique for various borehole field geometries. The temperature response factor model was cast as a TRNSYS component model and includes a flexible load aggregation algorithm that significantly reduces computing time. The component configuration of the model used is provided in Figure 3.

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 following equation is used:

$$T_{borehole} = T_{ground} + \sum_{i=1}^{n} \frac{(Q_i - Q_{i-1})}{2\pi k} g\left(\frac{t_n - t_{i-1}}{t_s}, \frac{r_b}{H}\right) \quad (1)$$

where

t = time (s)
t_s = time scale = $H^2/9\alpha$
H = borehole depth, ft (m)
k = ground thermal conductivity, Btu/h·ft·°F (W/m·°C)
$T_{borehole}$ = average borehole temperature, °F (°C)
T_{ground} = undisturbed ground temperature, °F (°C)
Q = step heat rejection pulse, Btu/h·ft (W/m)
r_b = borehole radius, ft (m)
i = index to denote the end of a time step
α = ground diffusivity
g = dimensionless temperature response factor (g-function)

The entering and exiting fluid temperatures of the ground-loop heat exchanger are computed based on the average temperature of the borefield. A detailed discussion on the development of the short time step g-functions and the load aggregation algorithm in the component model used is given by Yavuzturk and Spitler (1999).

Ground-Loop Heat Exchanger and Cooling Tower Sizing

One of the determining factors in sizing the length of the ground-loop heat exchangers and in determining the capacity of the cooling tower in a hybrid ground-source heat pump system design is the peak entering fluid temperature (EFT) to the heat pump from the borehole field. A significant number of "off-the-shelf" heat pumps are designed by their manufacturers for peak EFTs ranging between 85.0°F (29.4°C) to 95.0°F (35.0°C). The EFTs can be as high as 110.0°F (43.3°C) with high-efficiency rated heat pumps. Heat pump peak entering fluid temperatures above the rated operating temperatures degrade the performance of the heat pump. Similarly, there are lower limits for the heat pump entering fluid temperature that depend on the heat pump and the type of heat transfer fluid used in the loops. For high heating demands during winter months, this temperature may be near the freezing point of the working fluid. Any ground-loop and supplemental heat rejecter design must, therefore, be constrained by limits on the peak EFT to the heat pump. Currently, available methods for determining the required total length of ground loops use approaches that iterate between the total loop length and the maximum and the minimum heat pump EFTs for a specified duration (e.g., 20-25 years) of system operation.

Currently, available methods for the sizing of supplemental heat rejecters attempt to balance the annual ground energy rejection with the annual ground energy extraction. Theoretically, the average borehole field temperature will then not increase from year to year because no long-term temperature rise in the ground is thus allowed to occur. However, some control strategy must be implemented to achieve an annual balance. A very common approach is to activate the supplemental heat rejecters when the loop temperature is greater than a certain upper limit. It is, therefore, possible to decrease the size of the supplemental heat rejecter by increasing the required operating hours (settle for a smaller unit but operate it longer) or to increase the size of it by decreasing the required operating hours (settle for a bigger unit but operate it less). Accordingly, the supplemental heat rejecter sizing procedure may be somewhat flexible. The component model used allows for an hour-by-hour computation of the total amount of heat rejected through the cooling tower. For a control strategy, it is thus possible to determine the actual size of the supplemental heat rejecter based on the integrated amount of hourly heat rejection. It should be noted that the objective of this study is not to develop or recommend procedures for sizing supplemental heat rejecters but rather to investigate the effects of various control strategies on the system operation. Nevertheless, an optimal design procedure would be an excellent topic for future research.

Initially, a large cooling tower was selected. Then, the final required cooling tower capacity was determined by simulating with the large cooling tower and determining the required cooling tower capacity when the peak entering fluid temperature to the heat pump occurs. The cooling towers are probably slightly oversized, as the peak capacity was specified at the design wet-bulb temperature, even though that may not be coincident with the peak EFT. The simulated capacity of the cooling tower changes from one climatic region to another (Houston, Texas vs. Tulsa, Oklahoma) due to the local wet-bulb temperatures and the required fluid flow rates. An air flow rate of about 5300 cfm (9000 m^3/hr) is drawn through the cooling tower, operating on a simple on/off switch.

Operating and Control Strategies

The control strategies are selected to provide comparisons between system operations with and without the use of supplemental heat rejection. Admittedly, the selection of the system operating and control strategies for systems with supplemental heat rejection can be somewhat arbitrary, although an attempt has been made to include commonly employed control schemes. The objective here is to investigate the impact of each control strategy on the system operation rather than suggest a specific operating procedure. Including the case of optimum ground-loop heat exchanger design for a climatic region, ten system operating and control strategies were investigated.

Base case: The ground-loop heat exchanger length is designed without the use of any supplemental heat rejecters. System fully relies on the ground-loop heat exchanger to meet the building loads.

Case 2: The ground-loop heat exchanger length is designed considering the use of supplemental heat rejecters, yielding a smaller ground-loop heat exchanger size. However, no supplemental heat rejection is included in the simulations. This “undersized” ground loop case is of interest to illustrate the heat buildup and its effects on the loop temperatures at the heat pump.

Case 3: In this control strategy, the cooling tower is activated when the heat pump entering or exiting fluid temperatures are greater than a set value. The following two strategies are considered:

3a) ExFT > 96.5°F (35.8°C).

3b) EFT > 96.5°F (35.8°C)

Case 4: This case uses a differential temperature control approach for the operation of the cooling tower and the circulation pump on the secondary system loop. The difference between either the heat pump entering or the exiting fluid temperatures and the ambient wet-bulb temperature is used as the control criterion. It is subdivided into three strategies. The operation of the cooling tower may be based on the following:

4a) The cooling tower fan and the secondary fluid circulation loop pump are activated whenever the difference between the heat pump entering fluid temperature and the ambient air wet-bulb temperature is greater than 3.6°F (2.0°C). The cooling tower fan and the secondary fluid circulation loop pump are turned off when this difference is less than 2.7°F (1.5°C).

4b) The cooling tower fan and the secondary fluid circulation loop pump are activated whenever the difference between the heat pump entering fluid temperature and the ambient air wet-bulb temperature is greater than 14.4°F (8.0°C). The cooling tower fan and the secondary fluid circulation loop pump are turned off when this difference is less than 2.7°F (1.5°C).

4c) The cooling tower fan and the secondary fluid circulation loop pump are activated whenever the difference between the heat pump exiting fluid temperature and the ambient air wet-bulb temperature is greater than 3.6°F (2.0°C). The cooling tower fan and the secondary fluid circulation loop pump are turned off when this difference is less than 2.7°F (1.5°C).

Case 5: The operating and control strategy is based on cool storage in the ground to avoid a long-term temperature rise. The cool storage effect is achieved by operating the supplemental heat rejecters for six hours during the night. As a precaution to avoid potentially high loop temperatures, a set point control is also built in. Any heating load during the recharge period is neglected. Three different substrategies are considered to assess the impact of ground recharge in different seasons:

5a) The cooling tower fan and the secondary loop circulation pump are activated between 12:00 a.m. and 6:00 a.m. year-round. In addition, the supplemental heat rejecter is operated when the entering fluid temperature to the heat pump exceeds 96.5°F (35.8°C).

5b) This strategy is very similar to 5a. The only difference is that the cooling tower fan and the secondary loop circulation pump are activated between 12:00 a.m. and 6:00 a.m. only during the months of January through March (ground recharge during cold season).

5c) Similar to 5a but the cooling tower fan and the secondary loop circulation pump are activated between 12:00 a.m. and 6:00 a.m. only during the months of June through August (ground recharge during hot season).

SIMULATION RESULTS WITHOUT SUPPLEMENTAL HEAT REJECTION

Base Case—Optimum Design of the Borehole Field without Supplemental Heat Rejection

This is the reference case to which all other cases will need to be compared. For the base case the ground-loop heat exchanger is sized for use without any supplemental heat rejection. In the analyses of this paper, the optimal ground loop size for each climatic region is based on a peak EFT of approximately 96.5°F (35.8°C). The size was determined by adjusting the borehole depth so that the maximum temperature determined with a 20-year simulation just reached the specified peak EFT. The borehole depth was then rounded to the nearest 10 ft (3.1 m).

The system simulation for this case included only the heat pump, the ground heat exchangers, and the circulation pump of the main loop. Using the building loads for the two climatic regions, the model is run on an hour-by-hour basis for the design simulation period of 20 years. Heat pump EFTs for the first two years are plotted in Figure 4. These results are based on a fluid flow rate of 3.0 gpm (0.68 m^3/h) of water per borehole and on undisturbed ground temperatures of $T_{FarField\ Houston}$ = 73.0°F (22.8°C), $T_{FarField\ Tulsa}$ = 63.0°F (17.2°C). For both climates, a constant thermal conductivity of 1.2 Btu/h·ft·°F (2.08 W/m·K) is assumed for the ground formation. Identical single borehole geometries with constant borehole resistance (borehole radius of 3.5 in. [88.9 mm], U-tube pipe size of 1.25 in. [31.75 mm], and thermally enhanced grout with k_{grout} = 0.85 Btu/h·ft·F [1.47 W/m·K] are assumed) are configured for the comparison.

The ground-loop heat exchanger for Houston comprised 36 boreholes in a 6×6 configuration, each borehole drilled to 250.0 ft (76.2 m) deep, 12.5 ft (3.8 m) apart. The maximum predicted EFT to the heat pump after two years is about 86.0°F (30.0°C), rising to a maximum of 96.6°F (35.9°C) after 20 years of simulation. The minimum EFT of the 20-year simulation is 71.3°F (21.8°C), occurring in the first year. The design for Tulsa has 16 boreholes in a 4×4 configuration, each borehole drilled to 240 ft (73.2 m), 12 ft (3.7 m) apart. The maximum EFT after two years of simulation is about 89.4°F (31.9°C), rising to 96.4°F (35.8°C) after 20 years. For Tulsa, the minimum EFT to the heat pump is 50.2°F (10.1°C).

The daily fluctuations in the heat pump EFTs increase significantly as the example building is considered in relatively colder climates. This is because in colder climates a smaller ground heat exchanger is required to meet the peak EFT for a given peak cooling load. Naturally, the smaller ground heat exchanger results in a larger daily fluctuation in EFT over a day.

The energy consumption of the heat pump and the circulating pump for the base case are provided in Table 1 for both Tulsa and Houston TMY conditions. The percent energy consumption distribution between the fluid circulation pump and the heat pump is 40%/60% for Houston and 23.5%/76.5% for Tulsa, respectively. The energy consumption of the fluid circulation pump is significantly smaller in Tulsa than in Houston due to the shorter loop length for Tulsa.

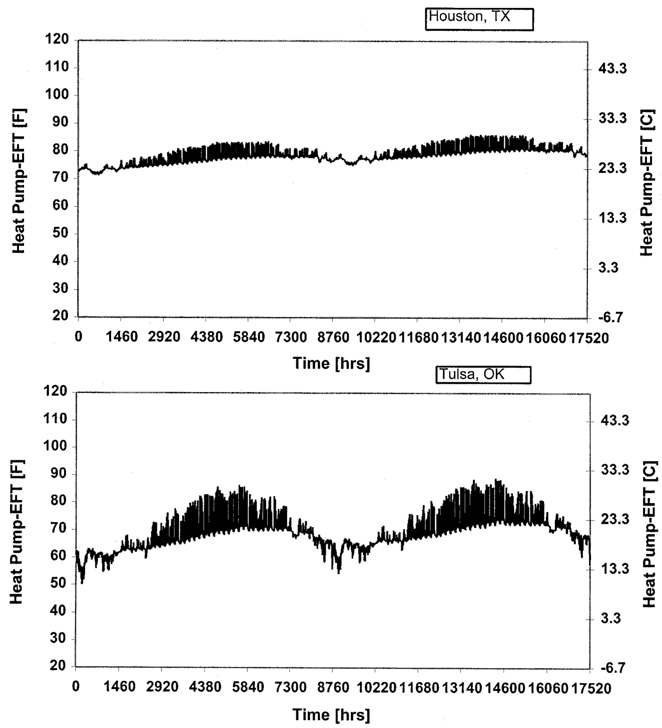

Figure 4 *Hourly entering fluid temperatures to the heat pump considering Houston, Texas, and Tulsa, Oklahoma, typical climatic conditions—base case.*

TABLE 1
System Simulation Summary for the Base Case

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Energy Consumption, Heat Pump (kWh)	20,399	25,904	24,245	17,931	20,660	19,927
Energy Consumption, Fluid Circulation Pump (kWh)	16,177	16,177	16,177	7190	7190	7190
Total Energy Consumption (kWh)	36,577	42,082	40,423	25,122	27,850	27,117

Case 2—Undersized Design of the Borehole Field without Supplemental Heat Rejection

The operation of the undersized borehole field without any supplemental heat rejection is interesting because it illustrates the effects of the long-term temperature rise in the ground due to reduced ground-loop heat exchanger length. The borefield for Houston TMY conditions is designed with 12 boreholes in a 3 × 4 configuration, each 250 ft (76.2 m) deep. This represents a two-thirds reduction from the base case. For Tulsa TMY conditions, the borefield is reduced from 16 boreholes to nine boreholes arranged in a 3 × 3 configuration with each borehole drilled to 240 ft (73.2 m).

The hourly heat pump entering fluid temperatures for Houston and Tulsa are shown in Figure 5. Even for the first two years of simulation, the EFTs to the heat pump are already over 110.0°F (43.3°C). A 20-year simulation predicts heat pump EFTs in excess of 120.0°F (48.9°C). The temperature fluctuations are observed to occur in a significantly wider band than in the base case. This is because an unchanged amount of heat is required to be rejected through a shorter loop length. Accordingly, the heat transfer fluid entering the heat pump from the ground is at a higher temperature. If this excess heat were not to be dissipated through supplemental rejection, the COP of the heat pump would deteriorate significantly over time.

Table 2 shows the energy consumption of the heat pump and the fluid circulation pump for Tulsa and Houston. In this case without any supplemental heat rejection, the energy consumption on the fluid circulation pumps is lowered significantly due to shorter loop lengths. However, since a long-term temperature rise is allowed to occur in the borefield, the heat pump operates with lower efficiency. This results in a sizeable increase in the energy consumption of the heat pump for both Tulsa and Houston TMY conditions. Although the total system energy consumption remains almost unchanged as compared to the base case, the heat pump energy consumption for Houston and Tulsa is significantly higher. It should be noted here that the energy consumptions are based on curve fits (Yavuzturk and Spitler 1999) of catalog data. Necessarily, they are extrapolated to higher temperatures than are supported by the catalog data. Therefore, the accuracy of the energy consumption data may be reduced. Perhaps it should suffice to say that the heat pumps are running with EFTs outside the recommended operating range, most probably with additional detrimental effects, such as insufficient capacity to meet the demand.

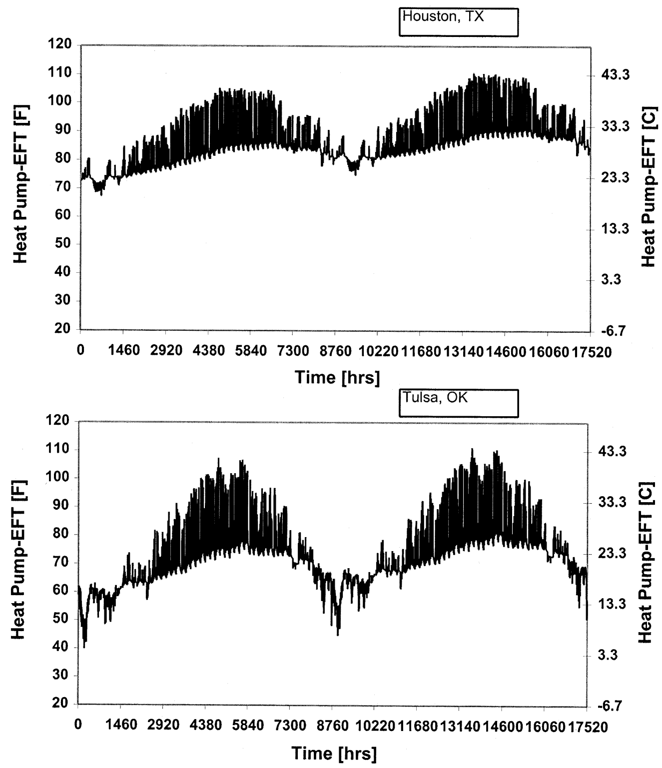

Figure 5 *Hourly entering fluid temperatures to the heat pump for typical Houston, Texas, and Tulsa, Oklahoma, climatic conditions—case 2.*

TABLE 2
System Simulation Summary for Case 2

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Energy Consumption, Heat Pump (kWh)	26,583	37,458	34,424	21,680	25,985	24,855
Energy Consumption, Fluid Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	31,976	42,851	39,817	25,724	30,030	28,900

SIMULATION RESULTS WITH SUPPLEMENTAL HEAT REJECTION

Case 3—Set Point Control for the Heat Pump Entering and Exiting Fluid Temperatures

A set point control for the operation of the supplemental heat rejecters is straightforward. With this control strategy, the cooling tower is activated whenever the heat pump exiting (Case 3a) or entering (Case 3b) fluid temperature reaches 96.5°F (35.8°C). The upper limit of 96.5°F (35.8°C) is selected considering the design maximum entering fluid temperature in the base case design. The operating hours of the cooling tower, the energy consumption resulting from supplemental heat rejection, including the cooling tower fan and the pumping energy consumption for the secondary fluid circulation loop (circulation pump #2), the energy consumption due to the heat pump operation, and the main fluid circulation loop (circulation pump #1), are given in Tables 3 and 4 for both climatic regions.

In cooling-dominated buildings, the temperature of the fluid exiting from the heat pump to the borefield will typically be higher than the temperature entering the heat pump. A set point control scheme that is based on the heat pump exiting temperature will, therefore, activate the cooling tower more often. Similarly, the duration of the cooling tower operation in general will depend on building cooling loads. The higher the building cooling loads, the more heat will need to be rejected, the longer and/or the more often the supplemental heat rejecters will be activated.

A comparison between Tables 1 and 3 shows that the annual average energy consumption of the heat pump is slightly decreased for both climatic regions when compared to the base case. The heat pump operates more efficiently due to slightly lower entering fluid temperatures to the heat pump. Overall, however, the savings in electricity consumption are somewhat larger due to reduced pumping costs associated with a smaller borefield.

TABLE 3
Hybrid System Simulation for Control Strategy 3a

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	523	750	704	317	440	415
Energy Consumption, Cooling Tower Pump (kWh)	31	45	42	19	26	25
Energy Consumption, Cooling Tower (kWh)	193	277	260	117	162	153
Energy Consumption, Heat Pump (kWh)	22,734	24,086	23,877	19,227	19,953	19,813
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	28,351	29,802	29,573	23,408	24,187	24,036

TABLE 4
Hybrid System Simulation for Control Strategy 3b

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	236	604	541	84	272	233
Energy Consumption, Cooling Tower Pump (kWh)	14	36	32	5	16	14
Energy Consumption, Cooling Tower (kWh)	87	223	200	31	100	86
Energy Consumption, Heat Pump (kWh)	24,459	25,653	25,413	20,742	21,384	21,264
Energy Consumption, Main Circulation Pump (kWh)	5,392	5,392	5,392	4,044	4,044	4,044
Total Energy Consumption (kWh)	29,953	31,306	31,039	24,823	25,546	25,409

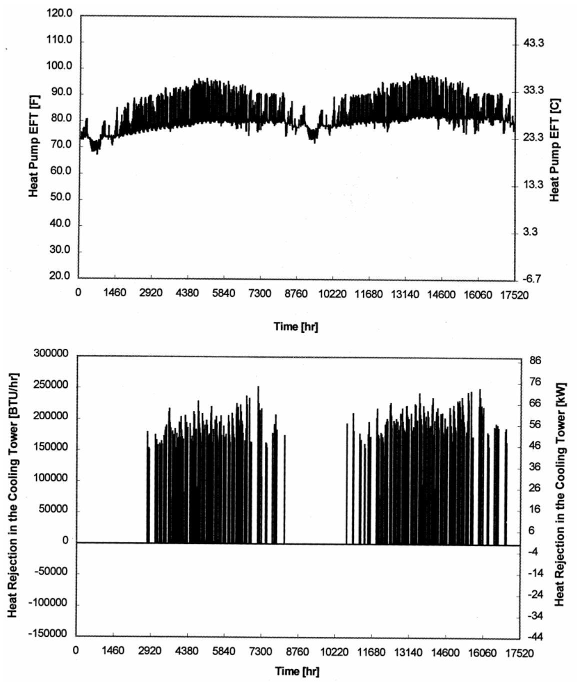

Figure 6 *Hourly entering fluid temperatures to the heat pump and heat rejection in the cooling tower for typical Houston, Texas, climatic conditions—two-year simulation, case 3a.*

The increase in operating hours for both control substrategies in later years of simulation is due to small temperature rises in the ground. Accordingly, the cooling tower must run somewhat longer. The set point temperature may be lowered to reduce the long-term temperature rise.

Hourly heat pump entering fluid temperatures and heat rejection in the cooling tower for case 3a are provided in Figure 6 for Houston TMY conditions. The maximum entering fluid temperature to the heat pump is 96.8°F (36.0°C), occurring in the 20th year of the simulation. The results for Tulsa are qualitatively similar.

Case 4—Differential Control for the Heat Pump Entering and Exiting Fluid Temperatures

An operating control strategy based on the difference between the heat pump entering or exiting fluid temperature and the ambient air wet-bulb temperature is designed to reject heat whenever the weather conditions are advantageous. The ambient air wet-bulb temperature is preferred to the dry-bulb temperature since the effectiveness of the cooling tower is based on the difference between the cooling tower inlet water temperature and the ambient air wet-bulb temperature.

In this strategy, the cooling tower and the secondary loop water circulation pump are activated when the difference between EFT or ExFT and $T_{WetBulb}$ are greater than the specified dead band high point (upper temperature difference). The cooling continues until this temperature difference falls below the dead band low point (lower temperature difference). For this analysis, a dead band low point of 2.7°F (1.5°C) is selected while two different dead band high points are used, 3.6°F (2.0°C) and 14.4°F (8.0°C), to investigate the effects of the size of the dead band. When the control strategy is based on the heat pump exiting fluid temperature, a dead band with a low point of 2.7°F (1.5°C) and high point of 3.6°F (2.0°C) is defined.

Due to the higher cooling demand of the example building in Houston, a higher frequency for the cooling tower operation can be expected. The increased frequency for cooling tower operation strongly depends on the size of the borehole field as well as on the cooling demand of the building. The higher the cooling demand of the building and the smaller the borehole field, the more often will the cooling tower be operated.

Summaries of the simulation results for control strategies 4a, 4b, and 4c are shown in Tables 5, 6, and 7. For both climatic regions, a general decrease in the annual operating hours of the

TABLE 5
Hybrid System Simulation Summary for Control Strategy 4a

	Houston, Texas			Tulsa, Oklahoma		
	1st year	**20th year**	**20-year average**	**1st year**	**20th year**	**20-year average**
Operation of the Cooling Tower (h)	5140	4470	4569	5159	4647	4723
Energy Consumption, Cooling Tower Pump (kWh)	311	270	276	312	281	286
Energy Consumption, Cooling Tower (kWh)	1901	1653	1690	1908	1719	1747
Energy Consumption, Heat Pump (kWh)	19,199	19,016	19,045	17,664	17,542	17,568
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	26,804	26,333	26,405	23,929	23,587	23,646

TABLE 6
Hybrid System Simulation Summary for Control Strategy 4b

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	3961	3483	3550	3818	3430	3481
Energy Consumption, Cooling Tower Pump (kWh)	239	210	215	231	207	210
Energy Consumption, Cooling Tower (kWh)	1465	1288	1313	1412	1269	1288
Energy Consumption, Heat Pump (kWh)	20,435	20,507	20,488	18,571	18,649	18,644
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	27,533	27,399	27,409	24,260	24,170	24,188

TABLE 7
Hybrid System Simulation Summary for Control Strategy 4c

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	5456	4909	4993	5542	5002	5088
Energy Consumption, Cooling Tower Pump (kWh)	330	297	302	335	302	308
Energy Consumption, Cooling Tower (kWh)	2018	1816	1847	2050	1850	1882
Energy Consumption, Heat Pump (kWh)	18,162	17,722	17,792	16,648	16,423	16,463
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	25,903	25,229	25,335	23,078	22,621	22,699

cooling tower from the first year to the 20th year of system simulation is observed for the three control substrategies. This is because on an annual basis more heat is extracted than rejected. The hybrid system with this control strategy cools the ground rather than heats it on a long-term basis. For case 4b, the cooling tower runs significantly less hours than for case 4a. An increase in the annual average time of operation is observed when the ExFT is used to establish the temperature differential for this control strategy (case 4c). This was to be expected since, in the cooling mode, the ExFT is greater than EFT.

Unlike case 3, in cases 4a, 4b, and 4c, significant savings in the heat pump electricity consumption are realized. For the best case, 4c, 27% average annual savings in heat pump electricity consumption are achieved in Houston and 17% in Tulsa. In Houston, the overall average annual savings in electricity consumption (37%) are significantly higher because of the reduced pumping requirements. In Tulsa, the overall savings are approximately the same as the heat pump savings because the reduced electricity consumption of the main circulating pump is offset by the electricity consumption of the cooling tower fan and secondary circulating pump.

Hourly entering fluid temperatures to the heat pump and the hourly heat rejection in the cooling tower of this control strategy are shown in Figure 7 for a two-year simulation using the control strategy 4c for Houston TMY. The maximum entering fluid temperature is 80.5°F (26.9°C) occurring in the first month of the 20-year simulation, while the minimum EFT is 40.5°F (4.7°C) and occurs in the 20th month. The EFTs to the heat pump for the 20-year simulation are shown is Figure 8.

For cases 4a and 4b, the peak entering fluid temperatures to the heat pump are relatively close to each other for both climatic regions (91.0°F [32.8°C] and 94.2°F [34.6°C] for Houston; 93.2°F [34.0°C] and 94.7°F [34.8°C] for Tulsa, respectively). However, when the temperature dead band is increased, as in case 4b, the system runs “hotter,” since the supplemental heat rejection runs less frequently.

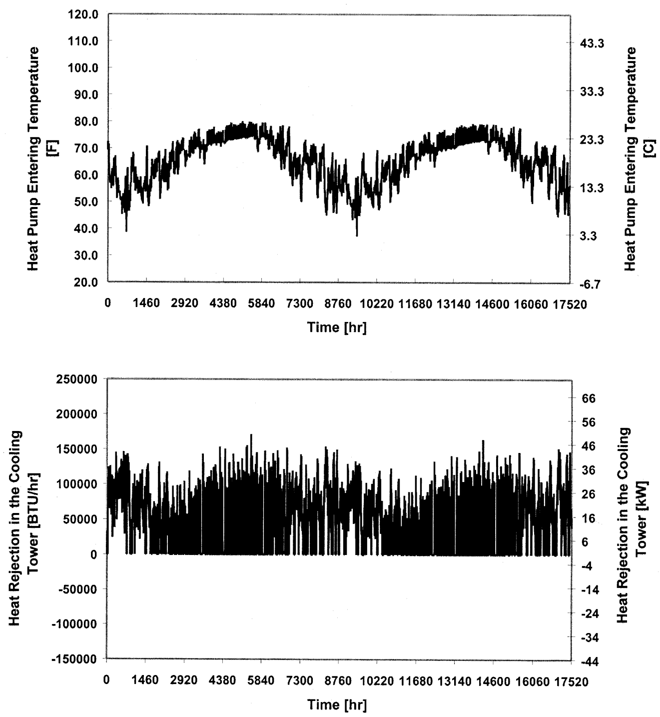

Figure 7 *Hourly entering fluid temperatures to the heat pump and heat rejection in the cooling tower for typical Houston, Texas, climatic conditions—two-year simulation, case 4c.*

Case 5 – Scheduled Recharge of the Borehole Field

In this case, excess heat is rejected by simply running the cooling tower and both circulating pumps at scheduled times (midnight to 6:00 a.m.) during the night. In addition, if the EFT to the heat pump exceeds 96.5°F (35.8°C), the supplemental heat rejection is turned on.

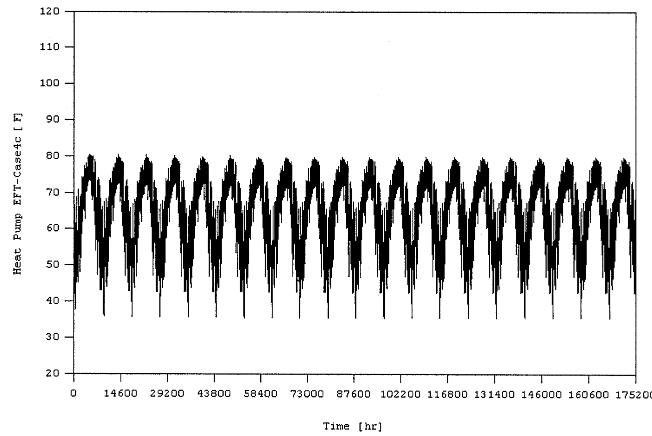

Figure 8 *Hourly entering fluid temperatures to the heat pump and heat rejection in the cooling tower for typical Houston, Texas, climatic conditions—20-year simulation, case 4c.*

Tables 8, 9, and 10 show the operating hours of the cooling tower and the energy consumption of the hybrid system for each control substrategy for Houston and Tulsa TMY conditions. Control strategies case 5b and 5c are designed to compare seasonal effects of cool storage. Case 5b considers the winter months (ground recharge starts in January and runs through March), and case 5c considers the summer months (ground recharge starts in June and runs through August).

The annual operating hours for the cooling tower in case 5a remains relatively stable throughout the 20-year simulation period. Most of the hours of operation are scheduled, so the slight increase in run time is due to the set point condition being reached more often as the fluid temperatures increases. The energy consumption due to the operation of the cooling tower fan and the secondary loop circulation pump account for

TABLE 8
Hybrid System Simulation Summary for Control Strategy 5a

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	2649	2740	2721	2619	2672	2660
Energy Consumption, Cooling Tower Pump (kWh)	160	165	164	158	161	161
Energy Consumption, Cooling Tower (kWh)	980	1013	1006	969	988	984
Energy Consumption, Heat Pump (kWh)	23,964	24,532	24,453	20,343	20,853	20,769
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	30,497	31,105	31,018	25,515	26,048	25,959

TABLE 9
Hybrid System Simulation Summary for Control Strategy 5b

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	815	1,081	1034	710	865	834
Energy Consumption, Cooling Tower Pump (kWh)	49	65	62	42	52	50
Energy Consumption, Cooling Tower (kWh)	301	399	382	262	320	308
Energy Consumption, Heat Pump (kWh)	24,587	25,886	25,696	20,735	21,876	21,664
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	30,330	31,744	31,534	28,085	26,293	26,067

TABLE 10
Hybrid System Simulation Summary for Control Strategy 5c

	Houston, Texas			Tulsa, Oklahoma		
	1st year	20th year	20-year average	1st year	20th year	20-year average
Operation of the Cooling Tower (h)	830	1148	1094	722	870	839
Energy Consumption, Cooling Tower Pump (kWh)	50	69	66	43	52	50
Energy Consumption, Cooling Tower (kWh)	307	424	405	267	321	310
Energy Consumption, Heat Pump (kWh)	24,899	26,324	26,108	20,915	21,995	21,800
Energy Consumption, Main Circulation Pump (kWh)	5392	5392	5392	4044	4044	4044
Total Energy Consumption (kWh)	30,649	32,221	31,972	25,270	26,414	26,205

3.8% of the total energy consumption for this strategy. The energy consumption of the heat pump accounts for 78.8% of the total energy consumption of the system and is about 1.3% less than the energy consumption as compared to the base case for Houston. Overall, the system uses about 25% less electricity than the base case, due substantially to the reduced pumping energy requirements. In Tulsa, the overall electricity savings are only about 6%, since the pumping energy requirements are not as strongly reduced.

Figure 9 shows the results of this operating strategy using case 5a for Houston TMY. The entering fluid temperatures to the heat pump appear to remain at relatively stable levels throughout a 20-year simulation period. The maximum EFT to the heat pump is 96.0°F (35.6°C) and the minimum is 54.1°F (12.3°C), both occurring in the first year of simulation. It may also be noted that there are a few hours when the cooling tower adds heat to the ground loop during the spring months. During this time, the ground loop is still relatively cold, while the ambient wet-bulb temperature is higher, but the cooling tower is being operated based only on the operating schedule.

The hourly entering fluid temperatures and cooling tower heat rejection plots for Tulsa are qualitatively very similar to the ones for Houston.

The maximum EFTs to the heat pump for cases 5b and 5c are only about 1.5°F (0.8°C) higher than for case 5a. As expected, the minimum EFT to the heat pump is significantly higher for case 5c than for case 5b. In addition, an increase of about 9% is observed in the cooling tower operation time for case 5c. Overall, the ground loop in case 5b runs hotter than in case 5a, and the ground loop in case 5c runs hotter than in case 5b.

Consequently, the savings in electricity consumption for case 5b compared to the base case are 1%-2% lower than for case 5a. The savings in electricity consumption for case 5c are 1%-2% lower than for case 5b. With this control strategy, it appears to increase in performance as more night run time is scheduled. However, only three schedules were considered and there may be a more optimal schedule that can be imple-

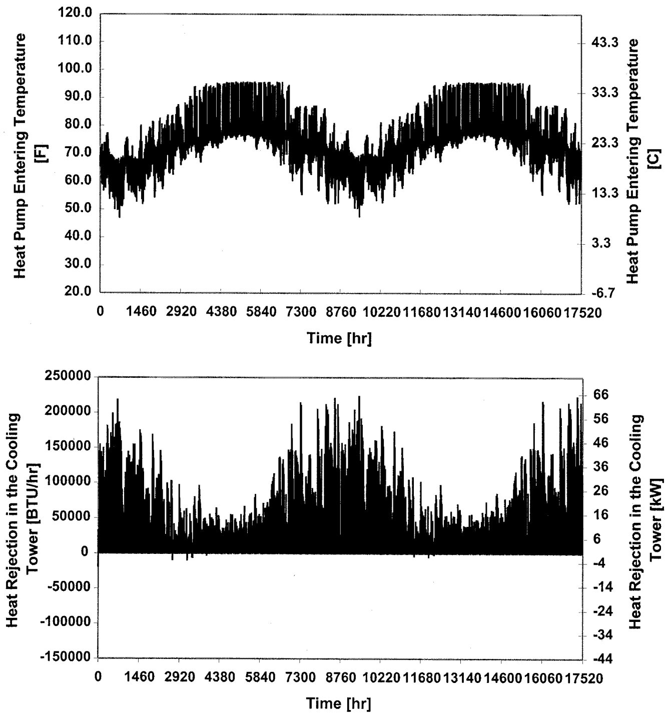

Figure 9 *Hourly heat pump entering fluid temperature and heat rejection in the cooling tower for Houston, Texas typical weather conditions and using control strategy 5a for the first two years of simulation.*

mented with a timer. In addition, the savings in electricity cost may be considerably different from the savings in electricity consumption if time-of-day utility rates apply.

Installation and Operating Cost Analysis

In order to compare the various cases, a cost analysis is conducted considering a system's first and operating costs for a 20-year design period. The present value of the predicted operating costs and the system's first costs are calculated based on series of assumptions.

a. The cost of the ground heat exchanger is calculated at $6.00 per foot of the borehole (Kavanaugh 1998). This amount includes the horizontal runs and connections.

b. The first cost of the cooling tower, including the isolation plate heat exchanger, is calculated at $350.00 per ton (3.52 kW) of cooling tower capacity (Means 1999). This amount includes other equipment and apparatus required for controls.

c. The cost of auxiliary equipment and materials for the cooling tower and the plate heat exchanger is estimated to be about 10% of the first cost.

d. The cost of electricity is assumed to be $0.07 per kWh.

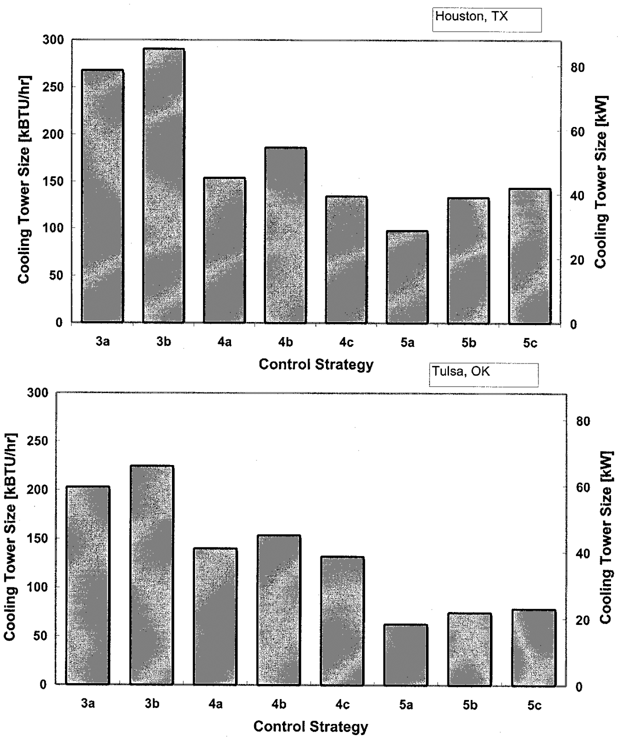

Figure 10 *Cooling tower size required for each control strategy.*

e. A 6% annual percentage rate is used for the present value analysis. Annual compounding is used for the 20-year analysis.

It should be emphasized here that this is a fairly simple approach, and it is no replacement for a detailed financial feasibility study of a specific building at a specific location with local climatic and ground conditions.

Consistent with the purpose of this paper of demonstrating the use and power of a short time step simulation model in building energy analysis, issues related to the maintenance of supplemental heat rejecters and related equipment were not included in this study. Nevertheless, it must be pointed out that failure to implement proper maintenance on supplemental heat rejecters (more so for cooling towers and fluid coolers than for surface heat rejecters) may at the end negate any economic benefits of hybrid ground-source heat pumps. A strict maintenance program as suggested by ASHRAE (1996) must be considered for the proper operation of hybrid systems.

The results of the cost analysis are summarized for Houston and Tulsa TMY conditions in Tables 11 and 12. Figure 10 shows the cooling tower size that was selected to implement each control strategy based on the rate of heat transfer in the cooling tower at the time of peak entering fluid temperatures to the heat pump. The cooling tower size, as discussed previously, is specified at the design wet-bulb condition. It can, of course, reject more heat when the wet-bulb temperature is

TABLE 11
Cost Analysis Summary for Each Control Strategy for Houston, Texas

	Base Case—“optimum design”	Case 2	Case 3a	Case 3b	Case 4a	Case 4b	Case 4c	Case 5a	Case 5b	Case 5c
Number of Boreholes	6 × 6	3 × 4	3 × 4	3 × 4	3 × 4	3 × 4	3 × 4	3 × 4	3 × 4	3 × 4
Total Length of Loop Installation (ft)	9000	2400	3000	3000	3000	3000	3000	3000	3000	3000
Total Cost of Loop Installation ($)*	$54,000	$18,000	$18,000	$18,000	$18,000	$18,000	$18,000	$18,000	$18,000	$18,000
Savings in Boreholes and Loop Installation ($)			$36,000	$36,000	$36,000	$36,000	$36,000	$36,000	$36,000	$36,000
Max. Heat Transfer in the Cooling Tower (Btu/h)			267,762	290,280	154,036	186,268	134,294	97,763	133,061	143,365
Max. Heat Transfer in the Cooling Tower (tons of cooling)			22.31	24.19	12.84	15.52	11.19	8.15	11.09	11.95
Max. Flow Rate (gpm)	108	36	36	36	36	36	36	36	36	36
EWT Max. during 20 Years of Operation (ºF)	96.6	126.6	96.3	97.6	90.9	94.3	80.5	96.0	97.8	97.6
EWT Min. during 20 Years of Operation (ºF)	71.3	67.3	67.3	67.3	39.9	41.3	40.5	54.1	52.5	67.2
Design Capacity of the Cooling Tower (tons of cooling)	n/a	n/a	22.5	24.5	13.0	15.5	11.5	8.5	11.5	12.0
First Cost of Cooling Tower + Plate Heat Exchanger incl. Controls ($)†	n/a	n/a	$7875	$8575	$4550	$5425	$4025	$2975	$4025	$4200
Cost of Auxiliary Equipment ($)‡	n/a	n/a	$787	$857	$455	$542	$402	$297	$402	$420
Total First Cost of Equipment ($)	n/a	n/a	$8662	$9432	$5005	$5967	$4427	$3272	$4427	$4620
Present Value of 20-Year-Operation (includes CT fan + Circ. Pump Elec. Cons. for Cases 2 through 5) ($)**	$32,062	n/a	$23,671	$24,841	$21,224	$22,013	$20,375	$24,874	$25,248	$25,592
Present Value of Total Cost($)	$86,062	n/a	$50,333	$52,274	$44,229	$45,980	$42,803	$46,146	$47,676	$48,212

* Estimated as $6.00 per ft of borehole, including horizontal runs and connections.
† Estimated as $350.00 per ton of cooling, including controls.
‡ Estimated as 10% of the first cost.
** $0.07 per kWh is assumed for cost of electricity. A 6% annual percentage rate is used for life-cycle cost analysis.

TABLE 12
Cost Analysis Summary for Each Control Strategy for Tulsa, Oklahoma

	Base Case—"optimum design"	Case 2	Case 3a	Case 3b	Case 4a	Case 4b	Case 4c	Case 5a	Case 5b	Case 5c
Number of Boreholes	4×4	3×3	3×3	3×3	3×3	3×3	3×3	3×3	3×3	3×3
Total Length of Loop Installation (ft)	3840.00	2160.00	2160.00	2160.00	2160.00	2160.00	2160.00	2160.00	2160.00	2160.00
Total Cost of Loop Installation ($)*	$23,040	$12,960	$12,960	$12,960	$12,960	$12,960	$12,960	$12,960	$12,960	$12,960
Savings in Boreholes and Loop Installation ($)			$10,080	$10,080	$10,080	$10,080	$10,080	$10,080	$10,080	$10,080
Max. Heat Transfer in the Cooling Tower (Btu/h)			202,942	224,423	139,962	153,646	131,825	62,580	74,631	78,509
Max. Heat Transfer in the Cooling Tower (tons of cooling)			16.91	18.70	11.66	12.80	10.99	5.22	6.22	6.54
Max. Flow Rate (gpm)	48	27	27	27	27	27	27	27	27	27
EWT Max. during 20 Years of Operation (ºF)	96.4	121.8	96.9	98.2	93.2	94.7	79.0	97.9	98.5	97.7
EWT Min. during 20 Years of Operation (ºF)	50.2	39.9	39.8	39.9	24.3	24.5	24.2	39.2	38.8	39.9
Design Capacity of the Cooling Tower (tons of cooling)	n/a	n/a	17.0	19.0	12.0	13.0	11.0	5.5	6.5	7.0
First Cost of Cooling Tower + Plate Heat Exchanger incl. Controls ($)†	n/a	n/a	$5950	$6650	$4200	$4550	$3850	$1925	$2275	$2450
Cost of Auxiliary Equipment ($)‡	n/a	n/a	$595	$665	$420	$455	$385	$193	$228	$245
Total First Cost of Equipment ($)	n/a	n/a	$6545	$7315	$4620	$5005	$4235	$2118	$2503	$2695
Present Value of 20-Year-Operation (includes CT fan + Circ. Pump Elec. Cons. for Cases 2 through 5) ($)**	$21,587	n/a	$19,254	$20,360	$19,003	$19,424	$18,248	$20,814	$20,863	$20,978
Present Value of Total Cost($)	$44,627	n/a	$38,759	$40,635	$36,583	$37,389	$35,443	$35,892	$36,325	$36,633

* Estimated as $6.00 per ft of borehole, including horizontal runs and connections.
† Estimated as $350.00 per ton of cooling, including controls.
‡ Estimated as 10% of the first cost.
** $0.07 per kWh is assumed for cost of electricity. A 6% annual percentage rate is used for life-cycle cost analysis.

lower. Hence, the cooling tower may reject twice its rated capacity during cold winter hours. It should be noted that the control strategy with the least average operating hours per year for the supplemental heat rejection system does not necessarily represent the economically most beneficial approach. Attention must be paid to the size of the cooling tower with which a control strategy can be optimally implemented.

Furthermore, this is a complex design problem with tradeoffs between ground-loop heat exchanger size, cooling tower size, and control strategy. We have not attempted to optimize the design but note that development of optimal design procedure is an excellent topic for further research.

CONCLUSIONS AND RECOMMENDATIONS

This paper provides a comparative study of several control strategies for the operation of a hybrid ground-source heat pump system used in a small office building. A simple cost analysis considering the first cost of the supplemental heat rejection, the first cost savings achieved through smaller ground heat exchangers, and the system operating costs is conducted based on a 20-year period. The three control strategies might be broadly characterized as follows. In case 3, the set point control runs the cooling tower only when necessary to avoid a high EFT to the heat pump. However, this generally occurs under the least advantageous weather conditions. In case 4, the differential control strategy operates the cooling tower under the most advantageous weather conditions. Under this strategy, the ground-loop temperatures are held to a much lower level, and, as a result, the cooling tower never needs to operate to avoid a high EFT under weather conditions that are not advantageous. In case 5, the cooling tower is merely operated on a schedule. This strategy does not take particular advantage of weather conditions and wastes some energy by running the cooling tower during hours when little or no heat rejection may be performed. Specific conclusions are summarized below.

1. For the example building, typical of small office buildings, a hybrid ground-source heat pump system appears to be beneficial on both a first cost and an annual operating cost basis for relatively hot climates, such as Houston, Texas, and for moderately warm climates, such as Tulsa, Oklahoma. The analyses suggest that the higher the building cooling loads relative to the building heating loads, the more first cost can be saved due to reduction in the ground heat exchanger size, and, consequently, the more beneficial the hybrid ground-source heat pump application. For the example building that is analyzed here, a hybrid application operated based on differential control scheme (case 4c) appears to be the most beneficial choice. However, compared to the base case, a hybrid system implemented with any of the control strategies investigated appears to have significant economic benefits based on first cost and 20-year operating cost (Tables 11 and 12).
2. For the small office building, the addition of a supplemental heat rejecter could not be justified for locations in relatively cold or moderately cold climates. However, buildings with different load profiles might be good candidates for hybrid ground-source heat pump systems.
3. Based on the limited study of control strategies investigated, the best control strategy investigated was 4c, which operated the cooling tower based on the difference between the fluid temperature exiting the heat pump and the outside wet-bulb temperature. This control strategy had the lowest first cost and the lowest operating cost. It takes advantage of the storage capacity of the ground heat exchanger by “storing cold” in the ground during the winter. It also rejects heat when conditions are advantageous in the spring, summer, and fall.
4. In general, the control strategies that operated the cooling tower more hours gave more benefit than those that operated the cooling tower fewer hours. This is particularly true when the cooling tower was operated under advantageous conditions, as in case 4. But it is also true that running the cooling tower at night in addition to running it when the EFT exceeds the set point (case 5) is better than running it only when the EFT exceeds the set point, as in case 3. Just comparing case 5 to case 3, the additional hours that the case 5 cooling tower runs allows the case 5 cooling tower to be smaller and, thus, have a lower first cost. However, because the case 5 cooling tower runs indiscriminately, it has a slightly higher overall operating cost than the case 3 cooling tower.
5. The use of a hybrid ground-source heat pump system resulted in significant land area savings. For the small office building in Houston, the surface area of the borehole field was reduced from 3906 ft^2 (363.1 m^2) to 937 ft^2 (87.1 m^2), a 76% savings. In Tulsa, the area was reduced from 1296 ft^2 (120.5 m^2) to 576 ft^2 (53.5 m^2), a 55% savings. For commercial buildings located in areas where property costs are high, the savings on land costs might be considerable. They were not accounted for in this study.
6. The pumping cost associated with the circulation of the heat transfer fluid through the borehole field accounts for a significant share in total system operating costs. A benefit of hybrid applications is that through the reduction in ground loop length, the operating cost associated with pumping of the heat transfer fluid can also be reduced significantly. The need for smaller capacity pumps also reduces the system first cost, although this was not accounted for in our analysis.

Finally, we believe that the use of a short time step ground-loop heat exchanger simulation model in a component modeling environment proves to be a very powerful tool in assessing the behavior and dynamics of hybrid ground-source heat pumps. It allows the implementation of sophisticated (based on hourly or less time intervals) operating and control strategies, previously not considered. Time-of-day electricity rates may also be considered, though we did not do so in this study.

This study leaves open a number of areas for future research. These include

a. Optimization of the design procedure and control strategy. Hybrid ground-source heat pump systems have many degrees of freedom; there are trade-offs between the reduction in size of the ground-loop heat exchanger, the size of the cooling tower, and the control strategy. This is a good candidate for development of an optimal design procedure that could simultaneously optimize all of the parameters of interest.

b. Additional validation of the model using data from a working system would be useful.

c. A similar analysis of other supplemental heat rejecters, such as shallow ponds and pavement heating systems, would be useful.

d. The interaction between the control strategies, design, and time-of-day electricity rates should also be considered. It is quite possible that the optimal solution in a case where electricity is much less expensive at night would involve running the cooling tower for the entire night.

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NOMENCLATURE

α = ground diffusivity in ft^2/h (m^2/h)

EFT = entering fluid temperature to the heat pump, °F (°C)

ExFT = exiting fluid temperature from the heat pump, °F (°C)

H = borehole depth, ft (m)

i = index to denote the end of a time step

k = ground thermal conductivity, Btu/h·ft·°F (W/m·°C)

Q = step heat rejection pulse, Btu/h·ft (W/m)

r_b = borehole radius, ft (m)

t = time (s)

t_s = time scale (h)

$T_{borehole}$ = average borehole temperature, °F (°C)

T_{ground} = undisturbed ground temperature, °F (°C)

$T_{WetBulb}$ = ambient air wet-bulb temperature, °F (°C)

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