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Comparison of Simulation-based Design Procedures for Hybrid Ground Source Heat Pump Systems

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1. ABSTRACT

Due to their higher efficiency, ground source heat pump (GSHP) systems offer an attractive alternative to conventional systems for residential and commercial heating and cooling applications. However, the higher first cost of GSHP systems has been a significant constraint for wider application of the technology. One way to reduce the first cost is to reduce the imbalance in the ground thermal loads by incorporating a supplemental heat source or sink into the system, which creates a hybrid ground-source heat pump (HGSHP) system. This work presents a new, computationally efficient method for sizing HGSHP systems based on a hybrid time step simulation. This design method utilizes an optimization routine to configure the ground heat exchanger (GHE) length and the size of the supplemental device so that the system meets both the desired maximum and minimum heat pump entering fluid temperature limits as closely as possible.

This method is compared to existing methods for a cooling-constrained HGSHP system serving a three-story office building in four U.S locations. The proposed method produces the closest life-cycle cost to a cost-optimized design method in three of four cases, while requiring substantially less computation time. The other methods occasionally, but inconsistently, provide close cost matches.

Keywords: ground source heat pump, ground heat exchanger, simulation

2. INTRODUCTION AND BACKGROUND

2.1 Overview of GSHP/HGSHP Systems

2.1.1 Background

Ground source heat pump (GSHP) systems are an attractive alternative to conventional systems for residential and commercial heating and cooling applications because of their higher efficiency. However, GSHP systems typically have as much as 20-40% higher first costs over conventional rooftop systems (Kavanaugh and Rafferty, 1997); this has been a significant constraint for the wider application of GSHP technology, especially for commercial and institutional locations. These commercial and institutional buildings typically have very high internal heat gains, and are consequently cooling-dominated. Depending on the degree of imbalance between annual heat rejection and heat extraction rates, the ground temperature around the ground heat exchanger (GHE) may rise over the system operation period, which will adversely affect system performance due to the variation in heat pump COP with temperature. This temperature drifting effect can be mitigated by increasing the size of the GHE, which of course will further increase the first cost of the system.

One way to reduce the size of the GHE, and therefore the first cost of the system, is to reduce the imbalance in the ground thermal loads by introducing a supplemental heat sink (or heat source, for heating-dominated cases) into the system. Such GSHP systems that incorporate a supplemental heat sink/source have been termed “hybrid ground source heat pump” (HGSHP) systems. The supplemental heat extraction may be accomplished with a cooling tower, fluid cooler, or pond heat exchanger, while supplemental rejection could be achieved by utilizing equipment such as a boiler or solar thermal collector with thermal energy storage.

2.1.2 The Design Problem

The design of a HGSHP system may be broken into the design of the base GSHP system and specification of a supplemental heat extraction or rejection device. The GSHP design problem involves specification of both the depth and configuration of boreholes. Typically, a specific configuration is selected up front, and if the resulting borehole length is either unnecessarily large or unreasonably small, a new configuration is selected. Multiple procedures are currently available for sizing vertical GHEs (Hellström and Sanner, 1994; Kavanaugh and Rafferty, 1997; Morrison, 2000; Spitler, 2000).

Designing a HGSHP, then, requires sizing not only the GHE but also the supplemental heat source or sink. One possibility for representing the domain of possible HGSHP solutions is with reference to the base GSHP system designed to meet all of the system heating and cooling loads, as shown in Figure 1. The vertical axis in Figure 1 represents the size of the HGSHP system GHE as compared to the GHE for the base GSHP system. For positive values, the horizontal axis represents the ratio of heat rejected by the supplemental heat sink to that rejected by the GHE of the base GSHP system; for negative values, it represents the ratio of heat extracted by the supplemental heat source to that extracted by the GHE of the base GSHP system. Thus, Point 1 in this figure represents the base GSHP system, with no supplemental heating or cooling. For a typical HGSHP system with a cooling tower, such as those designed for this comparison, the design might be represented by Point 2, where the GHE size is about 40% that of the base GSHP system and the cooling tower rejects about 60% of the total heat annually, while point 2’ would represent a HGSHP system where a boiler or solar collector could be used. Here, the GHE is about 70% as large as that of the base GSHP system while about 40% of the required annual heat transfer to the loop is provided by the supplemental heat source. In practice, there is a wide range of feasible solutions, ranging from a system with the entire load handled by the GHE, to a system with no GHE at all. This range is represented by the curves in Figure 1. These curves do not depict the actual shape of the solution domain, since this will vary depending on the system’s parameters; instead, the curves serve as an example of what the domain of feasible solutions might look like. The goal of the design procedure, then, is to find a feasible solution as near as possible to the optimum system design, and, for practicing engineers, to find that solution in an efficient manner.

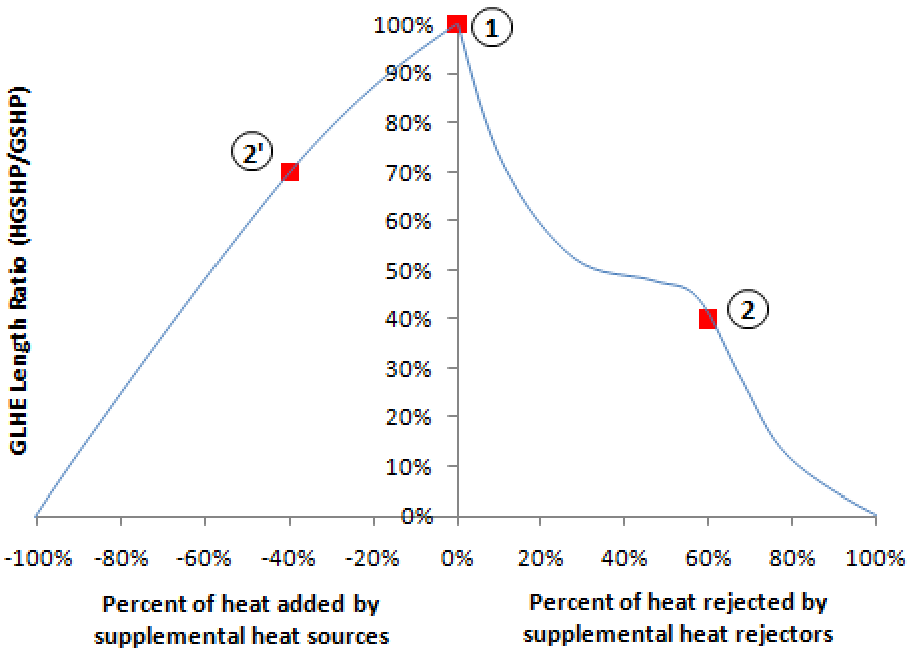

Figure 1: Conceptual diagram of HGSHP system design procedure

Existing methods for designing HGSHP systems are described in the next section, as is a new, computationally efficient procedure that has been integrated into an existing design tool.

2.1.3 "Dominated" or "Constrained"?

It is not necessarily the case that, for a cooling-dominated system, the required ground heat exchanger size will be determined by the cooling requirements. The required size depends not only on the system heat rejection/extraction demands, but also on allowable heat pump entering fluid temperatures (EFT) and the undisturbed ground temperature. It is entirely possible that, due to a small difference between the EFT limits on the heat pump and the undisturbed ground temperature, a GSHP system design may be constrained by one mode of operation (heating or cooling), while the "dominant" mode is the opposite. For example, a building with high cooling loads in a temperate or cooler climate would be cooling dominated, but the low ground temperature could cause the heating requirements to drive the size of the system. For this reason, two new terms are introduced: "heating-constrained" and "cooling-constrained"; these terms describe systems for which the designs are driven by the system heat extraction or rejection, respectively. A more complete example of this is given by Cullin (2008). For all of the cases described in this paper, the buildings are both cooling-dominated and cooling-constrained.

2.1.4 Goals

This work presents a comparison of existing HGSHP design procedures to a computationally efficient, design tool-based procedure, and to a full life cycle cost-optimization. Designs are analyzed based on their 20-year life cycle cost in order to determine if any one method performs markedly better or worse than others.

2.2 HGSHP System Design Methods

2.2.1 Kavanaugh (1998)

The general procedure given by Kavanaugh (1998) for sizing a regular GSHP system involves using an equation-based method to compute a required GHE length for cooling and another for heating, then using the larger of the two. For a cooling-constrained HGSHP system, the required length for heating is used, and a supplemental heat rejecter is utilized to meet the remainder of the heating loads unmet by the ground loop. Similarly, the cooling length is used and a supplemental heat extractor added for a heating-constrained system. For sizing the HGSHP system components, the peak block loads for the design day are required. Annual equivalent full load heating and cooling hours are required for calculation of the GHE system heat extraction and rejection rates, which are found via a spreadsheet procedure. The effects of annual and four-hour peak heat pulses are used to compute the required borehole lengths for heating and cooling. Next, the supplemental heat source or sink is sized from the difference between the two required borehole lengths. For a HGSHP system with a cooling tower as the supplemental heat rejecter, the capacity of the cooling tower can be specified in terms of the fluid flow rate by assuming that the fluid has a 5.6°C temperature change through the heat pump condenser and cooling tower.

2.2.2 Xu (2007)

The Xu (2007) HGSHP design method is based on Gentry's (2007) experimentally validated HGSHP simulation. Xu's method uses an optimization algorithm to minimize the 20-year life cycle cost of the system by adjusting the GHE length and cooling tower size. Although this algorithm would be ideal from a design perspective, since it uses a sub-hourly time step, it requires far too much computation time—many iterations of a simulation that requires about an hour to run once—to be a viable option for an engineer designing a HGSHP system. It is used here simply as a reference procedure to represent the absolute best possible design.

2.2.3 Hackel et al. (2009)

Hackel et al. (2009) created a detailed HGSHP system simulation in the TRNSYS environment. Then, using a parametric study of four building types and six locations, a set of first-order equation fits were generated to provide the ideal GHE size and cooling tower capacity in order to minimize the 20-year life cycle cost. The GHE size is directly proportional to the peak heating load and inversely to the difference between the heat pump minimum EFT constraint and the undisturbed ground temperature, while the cooling tower capacity is specified as 130% of the unmet cooling load, a figure determined from economic considerations. While the economic factors such as discount rate and electricity cost used in Hackel et al. are not exactly the same as those used in this work, Hackel et al. concluded that the optimal design does not vary substantially with the particular choice of economic parameters.

2.2.4 Chiasson and Yavuzturk (2009)

Chiasson and Yavuzturk (2009) take a somewhat different approach to HGSHP system design. They developed three dimensionless groups that describe the GHE parameters (such as ground thermal conductivity and borehole thermal resistance) as well as heating and cooling load data. Based on a 91-case parametric study using a TRNSYS simulation, Chiasson and Yavuzturk developed a correlation between the three dimensionless groups, the third of which includes the borehole length. For a hybrid system, the loads on the ground are equalized, and the change in the dimensionless groups creates a new borehole length, with the added heat rejection coming from a cooling tower. The cooling tower capacity is specified via a correlation between one of the dimensionless groups and the equivalent full load hours of the tower.

2.2.5 Proposed Method – Cullin (2008)

A new HGSHP design procedure has been developed by Cullin (2008) for implementation into a ground source heat pump design tool. This tool sizes the GHE using a hybrid monthly-hourly time step simulation of the GSHP system. Using an analogous simulation which incorporates the presumed behavior of the supplemental heat source or sink, combinations of the GHE size and supplemental device capacity can be evaluated. The procedure follows from to general observations of GSHP system design:

1. The GHE first cost tends to dominate the system life cycle cost. Therefore, designs that minimize the GHE size tend to have the lowest life cycle cost.
2. Systems that minimize the GHE size and make maximum usage of the supplemental heat source/sink tend to, as a result, reach both the maximum and minimum heat pump entering fluid temperature limits.

Accordingly, the procedure attempts to adjust both the GHE and supplemental heat source/sink sizes to come as close as possible to reaching both temperature limits over the lifespan of the system, without ever exceeding either limit. This requires some approximation regarding the behavior of the supplemental heat sink/source. Specifically, it is assumed that the portion of the heat rejection or extraction that is met by the supplemental heat sink/source can be reasonably approximated as a constant. Thus, the system heat extraction and rejection ratios illustrated in Figure 1 are assumed to apply to both the annual heat rejection/extraction rates and the peak heat rejection/extraction rates. Using a boiler, for example, as a supplemental heat source, may come very close to this approximation if it were operated correctly. For a cooling tower, this approximation would be less accurate, though it appears to be sufficiently accurate for design purposes. However, for a system with solar collectors as the supplemental heat source, one can imagine that the system might deviate significantly from this approximation, so the use of this method cannot at present be recommended for such an application.

The procedure is implemented as follows:

1. <u>Calculate the required borehole length of the base GSHP system.</u> Before any work can be done toward determining the optimum HGSHP system, the base GSHP system, without any supplemental device, must be sized. This is done to provide a starting point for the optimization step; this point is equivalent to point 1 in Figure 1. Additionally, this step determines whether the system is heating- or cooling-constrained, so that it is known whether a supplemental heat extraction or heat rejection device is needed.

   To size the GSHP system requires several inputs, including loads on the heat pump, ground and fluid thermal properties, and constraints (maximum and minimum) on the heat pump entering fluid temperature. An initial borehole length is guessed, and the system is simulated with a monthly time step; among the outputs of the simulation are the simulated maximum and minimum entering fluid temperatures to the heat pump. These temperatures are compared to the constraints, the borehole length is adjusted, and the process is repeated until both the simulated maximum and minimum temperatures fall inside the applied constraints. Since this is a univariate problem in borehole length, the technique of modified false position (Dowell and Jarratt, 1971) is used to determine the solution. More details on this GSHP simulation may be found in Spitler (2000).

2. Determine the optimal HGSHP system. The determination of the borehole length and supplemental device size for the HGSHP system is treated as an optimization problem, with the goal of meeting both the maximum and minimum heat pump entering fluid temperature constraints simultaneously and as closely as possible. To simplify the formulation of the optimization, the borehole length and supplemental device size are treated as ratios, as shown in Figure 1. The length ratio is defined as the ratio between the borehole length for the HGSHP system and the borehole length for the GSHP system. Thus, the length ratio can have any value between 0 and 1, with a value of 1 indicating that the HGSHP borehole length is equal to the GSHP borehole length. The load ratio is defined as the ratio of cooling or heating loads met by the supplemental device to the total cooling or heating loads on the system. Heating or cooling loads are used depending on whether the system is heating- or cooling-constrained, using the knowledge obtained from the first step. The load ratio can take on any value between -1 and 1, with a negative sign indicating heat addition.

   For any combination of length and load ratios that results in a physically possible system (that is, the borehole length, and thus the length ratio, must be non-negative), the system can be simulated using the same simulation as for the base GSHP system. The length ratio is multiplied by the GSHP length to get the HGSHP borehole length, and the original monthly loads—both average and peak—for the constraining mode are multiplied by the load ratio to obtain the new loads seen by the heat pump. Individual combinations of length and load ratios can be evaluated by comparing the simulated heat pump maximum and minimum entering fluid temperatures to the design limits. This is done via an objective function, given as

$$OF = (MaxEFT_{Cal} - MaxEFT_{Set})^2 + (MinEFT_{Cal} - MinEFT_{Set})^2 + Penalty \tag{1}$$

   where the *cal* subscript indicates the calculated (simulated) value, and the *set* subscript indicates the setpoint (constraint) value. The first two terms are the squares of the "errors" between the simulated and desired temperature extremes. The values are squared so that having one constraint being met while the other is exceeded does not produce a net zero effect. The purpose of the penalty term is twofold. First, as a consequence of squaring the temperature "errors" in the first two terms, exceeding the limits slightly will achieve the same objective function value as being slightly within the limits. The penalty adds to the objective function in order to push the solution away from the region where the constraints are violated. The second purpose of the penalty term is to prevent physically impossible or implausible systems from being considered. This is done by adding a large value to the objective function if the GLHE length is either negative or exceeds the size of the base-case GLHE, or if the size of the supplemental heat source/sink becomes negative. While negative lengths and capacities are obviously not physically possible, these values can potentially be encountered when searching numerically for a solution. Hence, this aspect of the penalty function seeks to avoid such values.

   Starting from an initial guess of the length and load ratios and initial step sizes, the simplex method (Nelder and Mead, 1965) is used to adjust the values of the length and load ratios until the objective function is minimized. A large penalty is assessed if the length ratio ever becomes negative, which would correspond to a physically impossible negative borehole length, and the system is not simulated; this is done to attempt to force the algorithm back into the plausible solution domain. After completion of the simplex method, the process is repeated with the solution as one of the initial simplex points to refine the solution. (Although this may not be necessary with respect to significant figures, it is a general recommendation for usage of the simplex method.) The result is a combination of length and load ratios that, when simulated, give maximum and minimum heat pump entering fluid temperatures as close to the constraint temperature limits as possible. This step, the main body of the procedure, requires under a minute of computation time on currently available computer hardware.

3. Compute the necessary capacity of the supplemental device. This step simply requires multiplying the optimum load ratio by the maximum load in the constraining mode (heating or cooling). As discussed at the beginning of this section, this is based on the assumption that the supplemental heat extraction/rejection device, when sized to meet some fraction of the peak load, will meet that fraction during the rest of the year.

4. For a cooling-constrained system, specify the cooling tower. The cooling tower is sized based on the local peak wet bulb temperature and the peak exiting fluid temperature of the heat pump. For this work, the peak wet bulb temperature is determined from the Typical Meteorological Year (DOE, 2007) weather data, while the peak heat pump exiting fluid temperature is assumed to be roughly coincident with the peak wet bulb temperature, and is set at 46.1°C. From these figures, the practical cooling capacity of the cooling tower can be calculated, which will be greater than the capacity at nominal design conditions. While the cooling tower capacity can be represented in multiple ways, it is convenient to use the effectiveness-NTU approach (Webb and Villacres, 1984) to pre-compute UA values for a range of cooling towers and place them in a database. Then, as a final step to the design procedure, the database can be searched to find the cooling tower with the smallest UA value that meets the required heat rejection rate.

For many systems, the method just described is equivalent to some of the other methods used in this study, in that for a cooling-constrained, cooling-dominated system, the GHE length will be that needed to satisfy the heating requirements, with a supplemental heat rejecter such as a cooling tower to handle the remainder of the loads. It should also be mentioned that this method does not attempt to balance the loads on the ground in order to achieve a steady-state behavior; as a result, the ground temperature may drift upward (or downward, for a heating system) over time so that extended operation after the design life cycle may result in equipment failure. For systems such as those presented by Cullin (2008), in which a building is cooling-dominated but heating-constrained due to a small temperature difference between the minimum EFT constraint and the undisturbed ground temperature, the design point will not be found by first satisfying the heating requirements. The design point will instead lie elsewhere in a location to be found by the optimization step, and it is in these situations where this method especially differs from others currently available.

3. COMPARISON APPROACH

3.1 Overview

The HGSHP systems designed with each method were simulated in the HVACSIM+ environment (Clark, 1985). Each design was modeled using the HGSHP system simulation done by Xu (2007), with the ground heat exchanger and cooling tower parameters changed to match each system design. The primary system loop consists of the ground heat exchanger, heat pumps, and a variable-speed circulating pump. The ground heat exchangers are designed so that the heat pump EFT never exceeds 43.3°C in cooling mode or falls below 2°C in heating mode. The secondary loop, isolated from the primary with a plate frame heat exchanger, consists of the cooling tower and a circulating pump. In the simulation, each loop also contains a thermal storage tank to account for the thermal mass of the system as well as transit delay. For all the systems designed, the cooling tower is operated under a temperature difference strategy: the cooling tower turns on when the difference between the heat pump exiting fluid temperature and ambient air wetbulb temperature is greater than 2°C, and shuts off when that difference falls below 1°C. A calendar signal governs the availability of the cooling tower, so that it does not run in winter months. Figure 2 shows the layout of the HGSHP system, with the thermal storage included. The primary (GHE) loop is on the top, while the secondary (cooling tower) loop is on the bottom of the diagram.

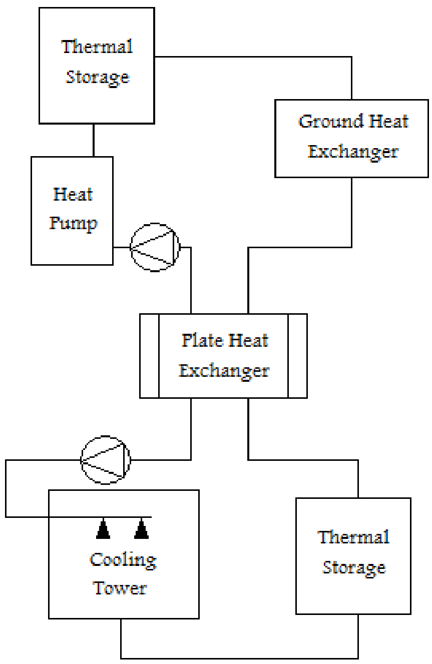

Figure 2: Layout of HGSHP system with thermal storage included

3.2 Building Description

The test building used in this study is based on an actual 52-story office building evaluated by Feng (1999), although only three stories are used for this study. Heating and cooling loads for the building were found for each location from the building model created by Gentry (2007). The office is designed to accommodate one person per 5 m^2, with combined lighting and internal gains of 23 W/m$^{2.}$ A minimum of 9.4 L/s of fresh air per person was specified, along with a 0.5 ACH infiltration rate. Glazing occupies about 65% of the building façade. The thermostat is controlled with a night and weekend setback schedule: from 7am-6pm, Monday-Friday, the setpoint is 20 °C for heating and 24 °C for cooling, with a night and weekend setback value of 5 °C in heating and 30 °C in cooling. The building was modeled in several locations as described in the next section.

3.3 Locations

For this comparison, a variety of climate types was desired to cover a range of cooling-dominated buildings. One method of classifying climate types is that of Briggs et al. (2002), which categorizes climates on the basis of heating and cooling degree-days as well as average relative humidity. Briggs et al. provide fifteen climate zones for the United States, with a representative city for each. Since only cooling-dominated locales are of interest for this study, the representative cities for four climate zones were chosen. The cities are Albuquerque, NM (mixed-dry climate), Baltimore, MD (mixed-humid climate), Houston, TX (hot-humid), and Memphis, TN (warm-humid). These locations encompass a building ranging from moderately cooling-dominated (Baltimore) to very heavily cooling-dominated (Houston). For each location, hourly building loads were calculated with EnergyPlus. Table 1 below shows the peak heating and cooling loads, as well as the equivalent full-load hours for both heating and cooling, for the office building at each location. Additionally, Table 1 lists the undisturbed ground temperature for each location.

Table 1: Building load summary

	Albuquerque, NM	**Baltimore, MD**	**Houston, TX**	**Memphis, TN**
$q_{peak, heating}$ [kW]	607	739	730	694
$q_{peak, cooling}$ [kW]	788	897	819	936
EFLH, heating	71	118	20	48
EFLH, cooling	898	599	1121	797
$T_{ground, undist}$ [°C]	13.9	13.9	23.3	15.5

3.4 System Parameters

Each ground heat exchanger is designed to use 100 boreholes in a square grid (120 in a 10x12 rectangular grid for Houston), on 6.1m spacing. Each borehole is 110mm in diameter and uses 1" SDR-11 piping. The thermal properties of the ground are assumed constant, with a conductivity of 3.5 W/m-K and a volumetric heat capacity of 2160 kJ/m^3-K. Each borehole is backfilled with thermally-enhanced grout so that the thermal resistance of the borehole is 0.209 K/(W/m).

3.5 Design Evaluation

Using each of the five design methods described above, a hybrid ground source heat pump system with a vertical ground heat exchanger and a cooling tower was designed for each location. Table 2 contains a summary of each design (total GHE length and cooling tower capacity).

To evaluate each of the designs, a 20-year life cycle cost analysis was performed. The present value of the first cost, plus annual operating costs for each of the 20 years of operation, was determined for each design using several assumptions:

- The first cost of the ground heat exchanger is calculated at $19.85 per meter of the borehole (Kavanaugh, 1998). This amount includes all horizontal runs and connections.
- The first cost of the cooling tower is calculated at $275 per ton (Means, 2006). This includes other equipment and apparatuses required for control of the tower.
- The first cost of any auxiliary equipment and materials for the cooling tower is estimated to be 10% of the first cost of the tower.
- Issues related to the maintenance of the cooling tower and associated equipment are neglected.

- Electricity is assumed to cost $0.07 per kilowatt-hour.
- A 6% annual percentage rate, compounded annually, is used for the cost analysis.

4. RESULTS

For each combination of location and design method, a 20-year simulation of the HGSHP system was run in the HVACSIM+ environment, based on Xu's (2007) HGSHP system simulation. The time step for each simulation was one hour. Table 2 shows the design for each combination of location and method and the average annual energy consumption, in kilowatt-hours, for each main system component: heat pump, cooling tower, and primary and secondary loop circulating pumps. Additionally, Table 2 also shows the maximum heat pump entering fluid temperature encountered during the 20 years of simulation and the average number of hours per year (if any) that the heat pump EFT exceeds the design constraint of 43.3°C.

Table 2: System energy consumption results

Location	Albuquerque					Baltimore				
Design Method	Kavanaugh	Hackel	Chiasson & Yavuzturk	Cullin	Xu	Kavanaugh	Hackel	Chiasson & Yavuzturk	Cullin	Xu
GHE Length [m]	7489	8061	8234	5769	5544	11658	9813	8604	7896	7774
CT Capacity [tons]	87	164	136	104	52	52	176	104	78	12
Avg. Annual Energy Consumption - Heat Pump [kWh]	172508	166809	166884	175306	189759	140274	138651	140293	143840	162132
AAEC - Primary Pump [kWh]	14855	16004	16004	16004	16037	15800	15431	15398	15537	15405
AAEC - Cooling Tower Fan [kWh]	4905	13771	10338	6261	3067	2857	6799	3348	3129	729
AAEC - Secondary Pump [kWh]	11766	22607	16546	14524	6117	5699	12104	8374	6476	1009
Total AAEC [kWh]	204034	219191	209772	212095	214980	164630	172985	167413	168982	179275
Max Heat Pump EFT	28.0	25.0	24.9	29.7	36.1	25.8	25.8	27.2	29.2	36.4
Hours EFT out of range/year	0	0	0	0	0	0	0	0	0	0

Location	Houston					Memphis				
Design Method	Kavanaugh	Hackel	Chiasson & Yavuzturk	Cullin	Xu	Kavanaugh	Hackel	Chiasson & Yavuzturk	Cullin	Xu
GHE Length [m]	6576	4285	4598	5231	4078	7483	7532	6417	6283	5644
CT Capacity [tons]	132	251	161	87	70	132	233	139	128	60
Avg. Annual Energy Consumption - Heat Pump [kWh]	253473	264578	262174	264311	279515	193988	195263	198289	201298	212239
AAEC - Primary Pump [kWh]	20406	16108	16649	15952	16027	18298	16833	16647	17091	16394
AAEC - Cooling Tower Fan [kWh]	11184	24219	15601	7384	6024	6784	9669	6876	6554	3312
AAEC - Secondary Pump [kWh]	29218	46777	34195	17689	11905	17734	27433	18283	17373	5693
Total AAEC [kWh]	314281	351682	328619	305336	313471	236804	249198	240095	242316	237638
Max Heat Pump EFT	32.6	44.6	42.9	37.8	42.6	30.2	34.6	36.9	37.9	38.4
Hours EFT out of range/year	0	0.8	0	0	0	0	0	0	0	0

In each instance, the Xu design method produced both the smallest GHE and lowest-capacity cooling tower. The Kavanaugh method gave comparatively large GHE lengths in each case (though not always the largest), while the Hackel method consistently gave the largest cooling tower. The Chiasson and Yavuzturk method also produced large cooling towers, although these were smaller than those from the Hackel method, while giving somewhat large GHE lengths. The proposed Cullin method produced GHE lengths very close to the cost-optimized Xu method in two of the four cases.

The hourly simulation of each design shows that only one case—the system designed with the Hackel method for Houston—produces a maximum EFT larger than the constraint of 43.3°C. This system exceeds that limit for 15 hours in 20 years. Especially for the Albuquerque and Baltimore cases, where the undisturbed ground temperature is fairly low and thus there is a large temperature difference to work with, the maximum EFTs were typically not very close to the upper limit, except with the Xu method. The Xu method produced the EFT closest to the maximum limit in each case. Thus, the other methods all produce designs that have excess capacity to some degree.

For each location, Table 2 shows that there are a set of five fairly different HGSHP designs. By plotting the designs as GHE length versus cooling tower capacity, some insight might be gained into the behavior of HGSHP systems in general. Figure 3 below shows two such plots, for Albuquerque (left) and Baltimore (right). The points indicate each of the five designs, while the solid line indicates HGSHP systems created using a GSHP design tool (Spitler, 2000), operating under the assumption that the cooling tower will handle the same percentage of the load throughout the year. As the plots show, the GHE length is linearly dependent on the tower capacity, until the point at which further decreasing the GHE length would fail to meet the system's heating requirements (this is the point when the solid curves in Figure 3 flatten out). The design point at the elbow of the solid line is the proposed Cullin method, while the Xu method is below the curve in each case and the other three methods fall above the curve. Since each of the system designs is viable in that the maximum allowable heat pump EFT is not exceeded (except for one case), it would appear that the assumption that the cooling tower handles a constant fraction of the loads over the course of the year is acceptable. From the scatter in the design points, however, it is clear that this is not the only possible approach.

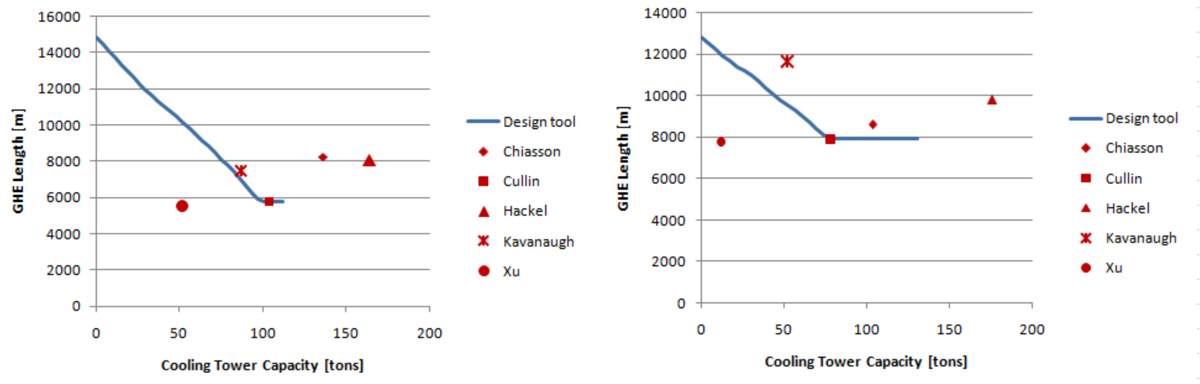

Figure 3: GHE length vs. cooling tower capacity for Albuquerque (L) and Baltimore (R)

4.1 Energy Consumption

For each combination of location and design method, Figure 4 shows the average annual energy consumption for each system component. Overall, the heat pump accounts for about 85% of the total energy consumption, the primary circulating pump another 5-10%, and the remaining 5-10% from the combination of the cooling tower fan and secondary circulating pump. The designs with larger ground heat exchanger sizes and cooling tower capacities—those designed with the Kavanaugh, Hackel, and Chiasson and Yavuzturk methods—tended to have the lowest heat pump energy consumptions. Those systems designed with the Xu method, on the other hand, had the largest heat pump energy consumption for each location; however, this is balanced out by much lower energy consumption for both components of the secondary loop.

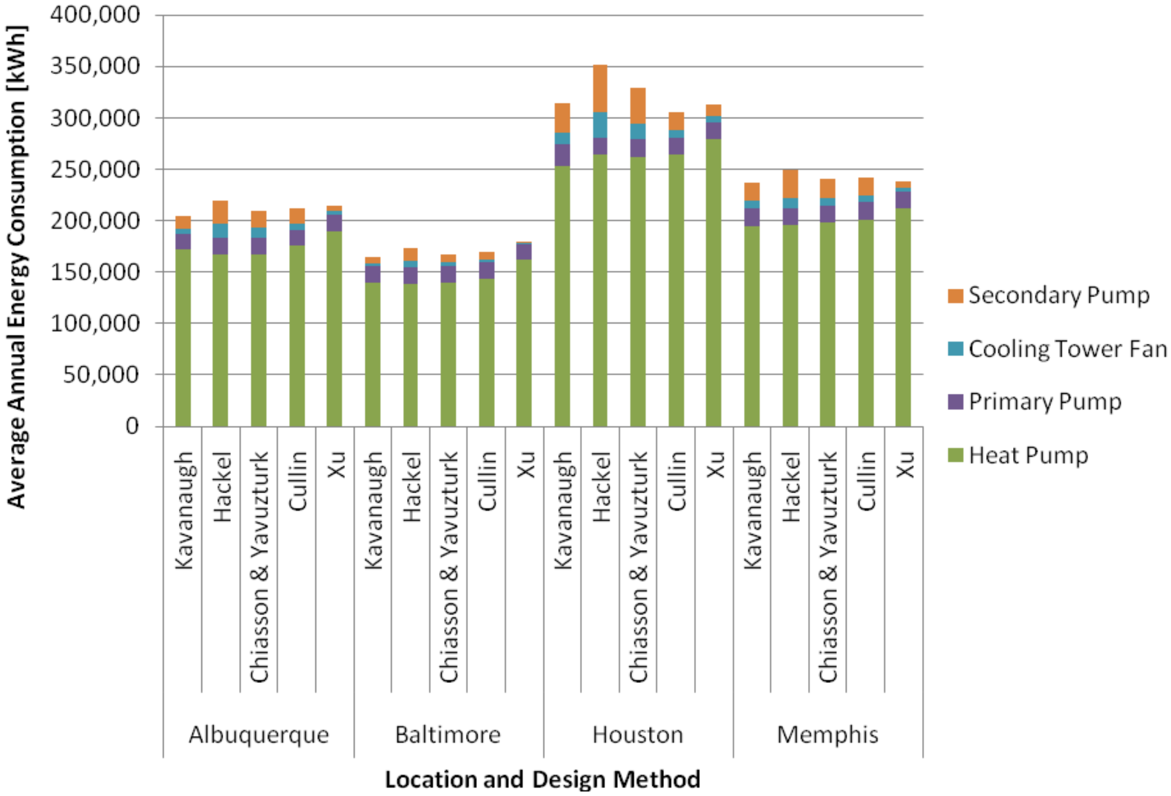

Figure 4: Average annual energy consumption for each design

4.2 Life Cycle Cost

Figure 5 shows the first cost and total operating cost over a 20-year life cycle for each system design. As should be expected from a life cycle cost-optimized design method, the systems designed with the Xu method have the lowest life cycle cost for each location. This backs up the first observation from Section 2.2.5, that minimizing the GHE length (since the Xu method produced the smallest GHEs for every location) will result in a low life cycle cost. After the Xu method, the next smallest life cycle costs were produced by the proposed Cullin method in three of the four locations, with the Kavanaugh method giving a slightly lower cost (about $3,000 less) for Houston. The Cullin method gives a life cycle cost within 12% of the optimum for all four cases, while the Chiasson and Yavuzturk method and the Kavanaugh method fall within that range once each out of four locations. The Hackel method, with costs driven upward by very large cooling towers, does not produce a life cycle cost within 12% of the optimum in any of the test cases.

4.3 Computation Time

Computation time for the Kavanaugh, Hackel, and Chiasson and Yavuzturk methods are trivial, as they can all be performed via a spreadsheet procedure. The proposed Cullin method, which has been implemented into a GSHP design tool, requires less than a minute to size a HGSHP system for a 20-year life cycle. Any of these methods would be acceptable from a computational efficiency perspective for a design engineer. The Xu method, on the other hand, utilizes multiple iterations of a sub-hourly simulation, taking sometimes many hours to determine one optimum design.

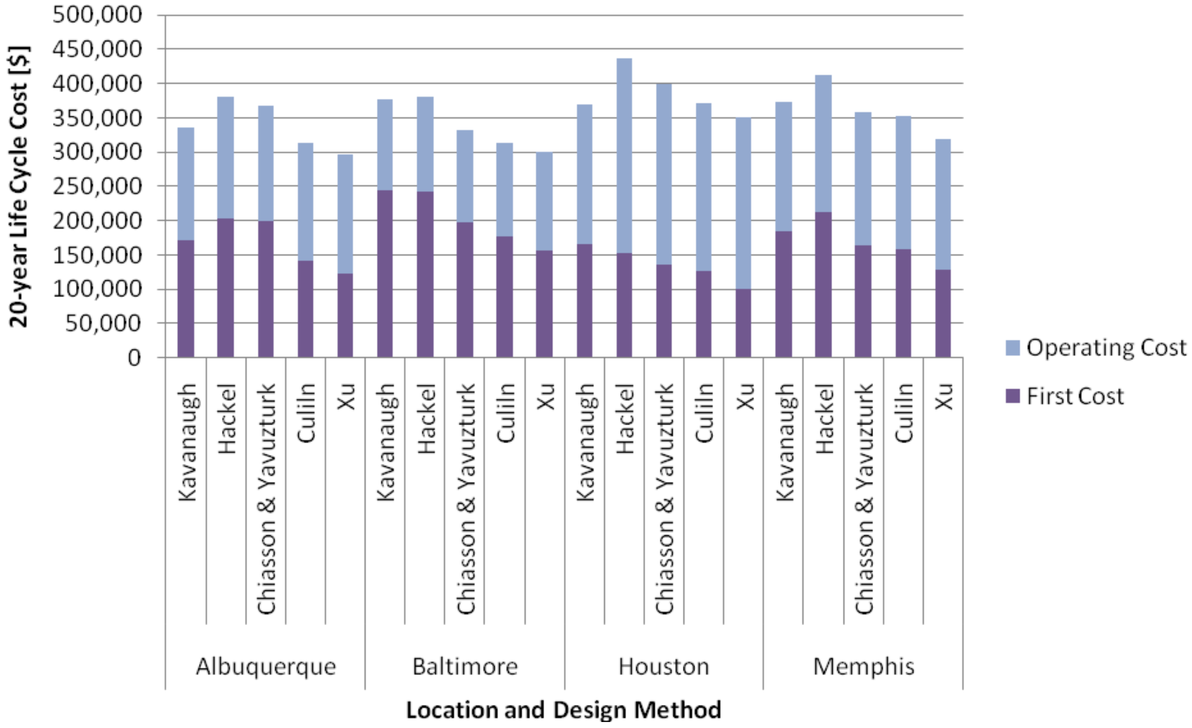

Figure 5: Life cycle cost for each design

5. CONCLUSIONS

In this work, a method for sizing hybrid ground source heat pump systems using an optimizing system simulation approach was proposed. This method, which has been integrated into a current ground source heat pump design tool, was then compared to three existing HGSHP design methods from the literature, as well as a cost-optimization scheme based on other previous work. For a three-story, cooling-dominated and cooling-constrained office building in four locations, the average annual energy consumption and 20-year life cycle cost of a hybrid system, consisting of a vertical ground heat exchanger and a cooling tower, designed with each of the five methods was determined. Some specific conclusions are as follows:

- The cost-optimized Xu method consistently produced designs that used smaller GHEs and cooling towers. Energy costs for this method tended to be higher than for the other methods, although the life cycle cost was (as expected) always the lowest. However, this method is far too computationally expensive to be useful as a design method for a practicing engineer.
- The proposed Cullin method gave designs that resulted in life cycle costs within 6% of the Xu method for three of the four cases, and within 12% for the fourth location. While the assumption that the cooling tower would handle a constant percentage of the total building cooling loads throughout the year is not necessarily accurate, it serves sufficiently well for design purposes here.

- The Chiasson and Yavuzturk method, as well as the Kavanaugh method, returned results somewhat close to the Xu method—within 12% once each and within 20% in two of the remaining three cases. The Kavanaugh method tended to produce larger GHE sizes, while the Chiasson and Yavuzturk method gave GHE sizes similar to the Cullin method, but with significantly larger cooling towers.
- The Hackel method gave the largest cooling towers, by far, of any of the design methods, and as a result produced systems that had the highest 20-year life cycle cost in each of the four cases. All of the systems designed with this method gave a life cycle cost 24-30% higher than the Xu method.
- For all but one case, all of the design methods produced acceptable (although often oversized) designs, when checked via a system simulation. However, some aspects of the system simulation, such as control strategies, were not considered as part of this work. Additionally, all of the cases used in this work were cooling-constrained, so further comparison of these HGSHP system design methods for heating-constrained systems is needed. Of particular interest might be systems that are, for example, cooling-dominated but heating-constrained, as such systems could provide a useful test for the robustness of each of the methods.

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