OPTIMIZATION OF HYBRID GEOTHERMAL HEAT PUMP SYSTEMS

Scott Hackel, Graduate Research Assistant; Gregory Nellis, Professor; Sanford Klein, Professor; all from the University of Wisconsin Solar Energy Lab, Madison, WI, United States

Abstract: Hybrid ground-coupled heat pump systems (HyGCHPs) couple conventional ground-coupled heat pump (GCHP) equipment with supplemental heat rejection or extraction systems. In unbalanced climates, the use of a supplemental heat rejection/generation device has been shown to significantly improve the economics of the system. However, the design and operation of HyGCHPs are more complex than GCHPs, and there is relatively little information available in this regard that is accessible to the practicing engineer. This paper describes the development of a simulation tool that integrates physics-based models of the HyGCHP components using the TRNSYS simulation program. The simulation model was used to complete a parametric study of optimal HyGCHP designs over a range of scenarios varying in climate, building type, and economic and physical assumptions. The results of the parametric study suggest a set of general design guidelines that can be used to select an equipment configuration, size equipment, and control the equipment for a typical HyGCHP system.

Key Words: ***hybrid, geothermal, heat pump, design, simulation, TRNSYS***

1 INTRODUCTION AND OBJECTIVE

Geothermal heat pump systems have sustained extensive growth due to their energy savings potential; a geothermal heat pump system can save up to 50% of the energy that would be used by conventional heating and cooling systems (EERE 2007a). However, geothermal heat pumps are still a secondary choice in most design scenarios. In order to allow geothermal heat pump technology to capture an even larger portion of the heating and cooling market, innovations are needed to improve the economics of the technology in some markets. One such innovation is the hybrid ground-coupled heat pump (HyGCHP) system. HyGCHPs interface conventional ground-coupled heat pump (GCHP) equipment with supplemental heat rejection or extraction systems. In a cooling-dominated climate (e.g., the southern U.S.) for example, a supplemental heat rejection device is operated during the cooling season in order to reduce the cooling load on the ground heat exchanger (GHX). The HyGCHP system results in a smaller, less expensive GHX that experiences a more balanced annual load - leading to more stable ground temperatures and therefore more efficient heat pump operation over time. This paper shows that the savings associated with the lower first cost of the GHX and more efficient heat pump operation can more than offset the cost of buying and operating the supplemental device, allowing hybrid systems to be a more attractive choice than geothermal-only systems.

HyGCHP systems add complexity to the common GSHP system; therefore, the primary objective of this research project is the development of tools that will allow engineers to better design and understand HyGCHP systems. A detailed simulation tool has been developed using the TRNSYS simulation program and integrated with an optimization package to allow design and control parameters to be globally optimized in order to minimize the system's life cycle cost. This computer tool has been used to determine the optimal design of HyGCHP systems over a range of climates and building types, resulting in a set of approximate design guidelines that can be applied generally. Specifically, these guidelines show the optimal size and control of cooling-dominated hybrid systems, as well as demonstrating where these systems are most beneficial. General conclusions are also provided for hybrid systems in heating-dominated climates.

1.1 Hybrid Systems in Literature

The ground heat exchanger model used in this project is referred to as the duct storage model ('DST model') and was developed at the University of Lund, Sweden (Hellström 1989). The DST model builds on previous Swedish research in ground heat storage and was implemented in TRNSYS by Pahud et al. (1996). Validations of the model have since been done using experimental data, including work by Shonder et al. (1999) and McDowell and Thornton (2008).

Research directed at understanding hybrid geothermal heat pump systems represents only a small fraction of the total body of geothermal research. Several papers have presented the details about actual, installed hybrid systems (for example, Phetteplace and Sullivan 1998). Research has also been done relative to hybrid systems for heating-dominated climates/buildings; primarily case studies using solar collectors as the supplemental device (e.g. Chiasson and Yavuzturk, 2003). Some studies have also used simulation tools to model (Ramamoorthy et al. 2000) or even optimize hybrid systems; one example is the optimization of the hybrid geothermal system installed at Fort Polk (TESS 2005). The model that is used in this project is based in part on the Fort Polk case study, in which a HyGCHP system was optimized for an administrative building in Louisiana. The HyGCHP model developed for this project is more general than the models used in the case studies above because its use is required over a wide range of buildings and climates. Kavanaugh developed one general methodology for sizing hybrid geothermal systems (Kavanaugh 1998) for cooling dominated climates. Two studies have also focused on identifying effective control strategies for hybrid geothermal systems. A study by Yavuzturk and Spitler (2000) examined several general hybrid control strategies in order to identify the optimal choice. The lowest-cost control strategy in that study was also identified as being lowest-cost in the Fort Polk study (TESS 2005).

2 HYBRID MODEL

The objective of this project is to identify the optimal hardware and control methodology for a specific building/climate/economic situation. The effects of changing building loads, ambient conditions, and even energy cost throughout a day suggest that an energy simulation with sub-hourly time resolution is required in order to obtain meaningful results. Additionally, the annual unbalance in the load and its effect on the ground temperature will substantially affect the long-term performance of the system; therefore, a multi-year simulation covering the life of the system is required. The TRNSYS (Klein 2006) modeling software was selected as the most appropriate simulation tool to meet these needs because TRNSYS has the ability to run multi-year, sub-hourly geothermal system simulations (with validated geothermal components).

2.1 Simulation Strategy

The HyGCHP model developed for this project utilizes several simplifying assumptions that are necessary in order to allow the consideration of a wide range of equipment size, buildings and climates while remaining computationally efficient; these characteristics are necessary to facilitate the optimization exercises that are the focus of this project. First, the current hybrid study utilizes building models that are independent of the simulation of the HyGCHP system itself. This methodology of de-coupling the building simulation from the heating and cooling equipment was previously adopted and justified in the Fort Polk case study (TESS 2005). This simplification necessarily assumes that all heating and cooling equipment is well designed so that peak loads can be met and indoor air conditions do not "drift" at any time during the day.

The hybrid configuration used in the cooling-dominated model places the cooling tower upstream of, and in series with, the GHX (see Figure 1). The Fort Polk study examined several configurations as well as different control strategies and supplemental devices (for cooling-dominated buildings) and found that the series configurations always resulted in a lower life-cycle cost than the parallel configuration (TESS 2005). Based on this observation, the HyGCHP model is always configured with the supplemental device in series with the ground coupled heat exchanger. The tower is placed upstream of the GHX based on the fact that the tower is the more expensive piece of equipment to operate due to the energy cost associated with fans and, to a lesser extent, the spray pump. (An exception to this rule occurs in areas with extremely high ground temperature and low wet bulb temperatures.) Therefore, when the tower is operating it should be doing so with the largest possible temperature difference between the fluid and the ambient air in order to maximize its performance.

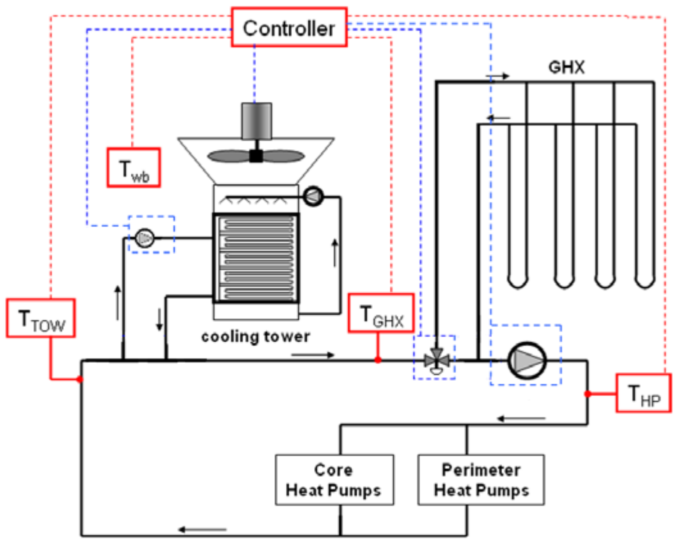

Figure 1. Configuration of hybrid system in the HyGCHP model; temperature measurements are shown in red, flow controls are shown in blue.

Two studies were found that investigated the most cost effective method of controlling the loop pump and cooling tower for cooling dominated systems. First, Yavuzturk and Spitler (2000) studied the hybrid system control strategy for an office that resulted in the lowest life cycle cost; in the most effective option the difference between the heat pump exiting fluid temperature and the ambient wet bulb temperature is used to control the operation of the cooling tower. The ground heat exchanger flow is then controlled using one setpoint for cooling and a different one for heating. The same conclusion relative to control methodology was also reached in the other study on hybrid control (the Fort Polk study; TESS 2005). This lowest-cost, general control strategy is used in the current HyGCHP model; its sequence is explained in more detail below.

For heating-dominated systems, a similar methodology was used, but the supplemental heat generation device is placed downstream of the GHX. This is done because the temperature associated with the boiler or solar system used in heating dominated systems is likely to increase above the ground temperature, which would render the GHX useless from a heating standpoint if it were placed downstream of these supplemental devices.

2.2 Components

Heat Pump. In this model, the building loads are divided into heating and cooling loads; one heat pump model then meets the total heating load and one meets the total cooling load. There are therefore only two heat pump components in the model, representing the entire system, and each heat pump is scalable to meet the respective peak load for any scenario. This simplification of lumping individual heat pumps into a single unit is appropriate (as shown below) from the standpoint of power consumption and energy flows. Additionally the scalable heat pump model is more generally applicable to the wide variety of buildings and climates studied in this project, without requiring detailed design of a heat pump system in each scenario.

In modeling an entire system of heat pumps, the component model needs to reject or absorb a quantity of heat, q_{tot}, to the fluid loop during each timestep. Note that q_{tot} is a function of the time step duration as well as both the load and the power consumption of the heat pump:

$$q_{tot} = \dot{m}_{fl}\, c_p\, \Delta T\, \tau_{on} \tag{1}$$

where $\dot{m}_{fl}$ is the mass flow of the fluid, c_p is the specific heat of the fluid, ΔT is the change in fluid temperature from inlet to outlet, and τ_{on} is the length of time that the heat pump operates during the time step. Although actual heat pumps operate intermittently (i.e., with varying τ_{on}), it is not necessary to model the process of turning individual heat pumps on and off during a time step. The actual, intermittent part-load operation of many heat pumps can be modelled using an equivalent steady-state operating condition, taking care to conserve energy in Equation (1).

The model employed here simulates this operation by varying the fluid flow rate, $\dot{m}_{fl}$, while maintaining a fixed temperature difference from inlet to outlet and keeping τ_{on} equal to a full timestep. The strategy of adjusting the fluid flow rate in order to represent part-load operation more closely approximates real HyGCHP operation as the number of actual heat pumps that are represented by the single, modeled heat pump increases. In an actual building with multiple zones that are served by individual heat pumps, each unit will cycle on and off as loads are met in each zone. As they cycle, the net effect during the timestep is approximately the same as with a single heat pump operating at a partial flow rate.

Modeling the performance of many individual heat pumps with one single component model is valid provided that the performance characteristics of individual heat pumps are linear. The performance specifications for several different manufacturers and models were studied for this purpose. It was found that, for a given entering fluid temperature, the ΔT across the heat pumps varied by only ±10% (independent of capacity) over all models. The volumetric flow rate of the fluid was then investigated as a function of heat pump capacity. For both cooling and heating capacity, the fluid flow rate was found to be directly proportional to the capacity of the heat pumps; this linear relationship justifies the assumption that the total fluid flow rate associated with many small heat pumps is consistent with the fluid flow rate required by a single large heat pump and allows the fluid flow rate to be computed according to the total heat pump capacity using only a simple, linear curve fit. Finally, by plotting efficiency (for both heating and cooling) as a function of heat pump capacity, efficiency is found to be independent of capacity.

During simulation, the performane data are used calculate the energy consumption of the heat pumps as well as the energy transfer between the heat pump and the HyGCHP model. The performance data are represented by equations to improve computational speed. These equations relate heat pump capacity, fluid flow rate, and efficiency. Both capacity and efficiency are also functions of entering fluid temperature ($T_{fl,in}$) and indoor air conditions; additional equations modify the capacity and efficiency to account for changes in these conditions.

GHX. The ground heat exchanger considered in this analysis is a vertical ground heat exchanger with U-tube piping, the most common configuration currently used in commercial/institutional buildings. The thermal interaction between the fluid and the ground is simulated using the duct storage (DST) model of a vertical GHX field (Hellström 1989). The DST model calculates the transient temperature distribution as the superposition of a global temperature solution, a local solution, and a steady-flux solution (accounting for interaction between the other scales). Many parameters are required to specify the GHX model. In this study, typical values of 91.4 m and 7 m are assumed for drilling depth and spacing, respectively. Ground conductivity of 2.4 W/m-K and ground thermal diffusivity of 0.1 m^2/day are assumed. Initial ground temperature is varied based on the geographic location of each scenario. The parameters that dictate the overall size of the GHX (i.e., the total bore length, number of bores, etc.) are controlled by the optimizer and varied in order to arrive at the most optimal GHX design. An algorithm based on Kavanaugh (1997) is used to layout the borefield to assure turbulent flow during peak load periods while minimizing pumping power. The piping is assumed to be 25 mm SDR-11 PE pipe; The bore diameter is assumed to be 0.11 m and is assumed to be filled with enhanced grout having a thermal conductivity of 1.4 W/m-K.

Supplemental Devices. This study has considered both a closed circuit cooling tower (CCCT) and a dry fluid cooler (DFC) to provide heat rejection to supplement the GHX. The overall size of these devices is determined by optimization, therefore all performance and economic parameters must be scalable with device size. The CCCT component model is based on a simulation method developed by Zweifel et al (1995). Cooling tower performance data used for baseline conditions are taken from manufacturers. The DFC (sometimes referred to as an air fluid cooler) component model assumes that the DFC device acts as a simple cross-flow heat exchanger (TESS 2005) with properties taken from manufacturer specifications. A boiler component model is required for the boiler/tower simulation (used in comparison to geothermal configurations) as well as modeling heating dominated hybrid systems. The boiler model assumes a constant 85% efficiency; its capacity is chosen by optimization. Additionally, for heating dominated cases a solar thermal collection system with flat plate collectors is considered as a supplemental device.

Controller. One general control strategy has been shown to be the most cost effective for cooling-dominated hybrid systems. Within this general strategy, several set points can be changed by the optimizer; therefore different buildings may have very different optimal control sequences even though they use the same general control strategy. The hybrid configuration places the supplemental device in series with, and upstream of, the GHX. The control strategy operates the supplemental device when ambient temperatures are favorable. Temperature measurements and flow controls are required in the locations shown in Figure 1 in order to operate the controller according to the sequence laid out below. This sequence assumes a cooling tower is used as a supplemental device (note that all temperature set points (i.e., T_{Cool1}, etc.) are constant during simulation and specified only once at the start of simulation):

1. The controller determines whether heating is needed based on the temperatures entering and leaving the heat pump. If heating is required, then the fluid is diverted through the GHX whenever the temperature falls below some set point control temperature, T_{Heat1} (note that all set points are operated with a dead band temperature difference, set to 2.5°F in this study).
2. If cooling is required, the controller follows the remaining steps. The cooling tower is turned on (at low fan speed and flow rate) if the fluid temperature leaving the heat pumps (T_{TOW}) is above the ambient temperature plus some set point temperature difference, ΔT_1.
3. If the fluid temperature leaving the heat pumps is also above some higher set point temperature, T_{Cool1}, then the cooling tower is switched to high fan speed and flow rate.
4. If the fluid temperature leaving the cooling tower remains above some set point temperature T_{Cool2}, then the fluid is diverted through the GHX as well.

For heating dominated systems, the boiler or solar system is downstream of the GHX. A single additional setpoint, T_{Heat2}, modulates the use of the boiler. When the solar system is used, fluid in the solar array is circulated through the storage tank when the collector is significantly warmer than the tank. A pump then circulates the fluid through a heat exchanger that interfaces with the main fluid loop when the storage temperature is greater than the temperature in the main fluid loop plus a setpoint ΔT_S. Limits are set on the flow rate through the heat exchanger to prevent the fluid from heating beyond the high temperature limit of the heat pumps.

Pump. A variable speed pump is modeled because the pump that circulates fluid through must operate over a range of speeds due to the changing mass flow rate in the system. The main role of the pump model is to calculate pumping power. Power is based on a constant efficiency, assuming good pump selection (60% efficiency is used based on average operation). Pressure drops are summed individually in each component (including the individual pipes in the GHX).

Economics. For a building energy system, such as the heat pump systems considered in this project, the life cycle cost (*LCC*) associated with owning and operating the equipment is the most appropriate figure of merit for optimization because *LCC* includes the size (i.e., capital cost) and efficiency (i.e., operating cost) of the equipment, as well as the effects of inflation rate and discount factor on these costs. The economic life is equal to the time span of the simulation, which is 20 years in this study. Equipment and maintenance costs are computed based on R.S. Means (2002 and 2006). Other economic parameters, such as discount rate, fuel cost, and GHX cost, were selected based on surveys of literature and current market costs. A summary of the key parameters assumed for the parametric study (quantities that are not optimized) is shown in Table 1; sensitivities to some of these are discussed in the results below.

Table 1. Summary of input parameters for the parametric study.

Category	Parameter	Baseline value
Bore field	Ground conductivity	2.4 W/m-K
	Ground diffusivity	0.1 m^2/day
	Grout conductivity	1.4 W/m-K
	Initial ground temperature	varied according to climate
	Maximum drilling depth	90 m
Other equipment	Pump efficiency	60%
	Range of allowable entering heat pump fluid temp.	1.7 - 35°C
	Heat Pump COP, cool / heat (ARI 13256-1)	4.7 / 3.6
Economic	Life span	20 years
	Discount rate	8.5%
	Loan	6.0% for 20 years, 30% down
	Tax rate	35%
	Electricity rate, 10am – 9pm / other times	0.101 $/kWh / 0.063 $/kWh
	Electricity demand charge	6.22$/kW, 15 minutes
	Gas price (for boiler)	9.4 $/GJ
	Water price	1.4 $/m^3
	Bore field cost	32.8 $/m

3 OPTIMIZATION

The design variables listed below (assuming a closed-circuit cooling tower as the supplemental device) have been selected for optimization:

- Ground heat exchanger length
- Cooling tower size
- Cooling tower control set points (ΔT_1 and T_{Cool1} – for high speed)

• GHX control set points (T_{Cool2} – for cooling, T_{Heat1} – for heating)

Due to the large number of design variables and the wide array of scenarios that must be studied, the use of a computerized optimization algorithm is the only practical way to accomplish the objectives of this project. Optimization was accomplished using the GENOPT software (Wetter 2007). The GENOPT optimization algorithm was set to find the HyGCHP system with the design parameters (from the list above) that result in minimum life cycle cost (LCC).

4 PARAMETRIC STUDY AND RESULTS

In order to develop guidelines for the design of hybrid ground source heat pump systems, simulations were run across a range of building load scenarios that included various climates (from warm and dry to cold and humid) and building types of interest (including retail, office, continuous-use, and school buildings). For each building/climate combination, three main equipment configurations were studied, all with identical heat pump systems and using the same economic and other assumptions: boiler/tower, geothermal-only, and hybrid geothermal (with cooling tower). A hybrid geothermal system with a dry fluid cooler was also studied for a subset of the building/climate combinations. Building loads were created using an existing building model created by the ASHRAE-sponsored research project TRP-1120.

4.1 Optimal Design – Cooling Dominated Systems

The optimal design and cost results were compiled for each equipment configuration in all 24 building/climate combinations. These results suggest the design guidelines detailed in the sections below; these design guidelines should be used applied with some caution as they are based on the economic assumptions and equipment parameters summarized in Table 1.

a) GHX Sizing: size the ground heat exchanger (GHX) so that it is just capable of meeting the peak heating load. Figure 2 illustrates the optimal size of the ground heat exchanger (sum of all bore depths) as a function of the peak heating load (each point represents a different climate/building combination from the parametric study); notice that the optimal GHX size is, to first order, proportional to the peak heating load. This result is consistent with some methods currently used in industry (e.g. GCHPCalc; Kavanaugh and Rafferty, 1997).

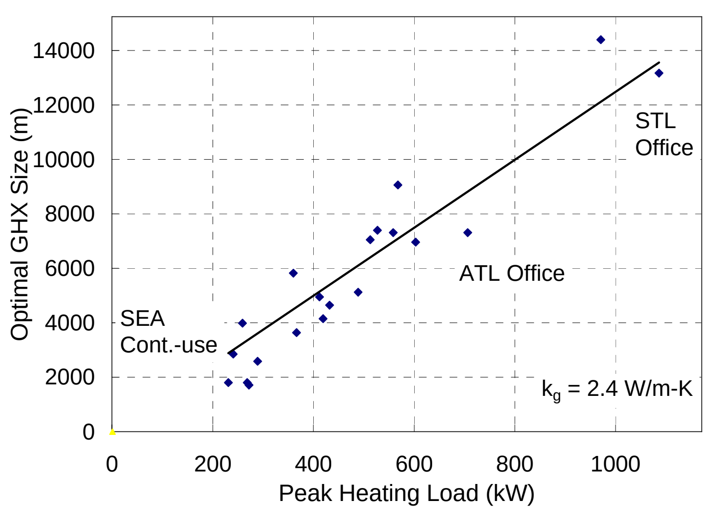

Figure 2. Optimal GHX size as a function of peak heating load for each scenario.

The scatter in Figure 2 is partly due to variation in the ground temperature associated with different climates. The same relationship was therefore plotted with the x-axis divided by a ground temperature term (the ΔT between ground temperature and minimum fluid temperature). The regression equation resulting from that plot is described by Equation (2):

$$L_{tot} = C_1 \frac{\dot{q}_{peak,heat}}{\Delta T_{ground}} \quad \text{for } T_{ground} = 10\text{-}24^{o}\text{C} \tag{2}$$

where C_1=147 m-^{o}C/kW, L_{tot} is the GHX length in m, $\dot{q}_{peak,heat}$ is in kW, and ΔT_{ground} (in ^{o}C) is the difference between initial ground temperature (T_g) and minimum fluid temperature, the main driver for heat transfer in the GHX. For example, this is equivalent to 13 m/kW at a ground temperature of 13^{o}C and 7.5 m/kW at a ground temperature of 20^{o}C. The assumed value of the ground conductivity (k_g) used for these results is 2.4 W/m-K. Sensitivity studies were carried out on k_g (as well as several other parameters in Table 1); results suggest that every 0.2 W/m-K decrease in k_g will result in a 5% increase in the optimal GHX size. The sensitivity of the optimal design to the economic parameters and operating temperatures is discussed below.

b) Supplemental Device Size: size the supplemental cooling device based on the peak cooling load that is not met by the GHX.

- The rated capacity of the optimally sized closed-circuit cooling tower (C_{CCCT}, in tons), is 2.1x the peak cooling load that is not met by the GHX ($q_{unmet,cool}$, in tons); the unmet load should be calculated according to Equation (3):

$$C_{CCCT} = 2.1\dot{q}_{unmet,cool} = 2.1\left(\dot{q}_{peak,cool} - \overbrace{\frac{L_{tot}}{C_1(T_{ground} - T_o)}}^{q_{GHX,cool}} \right) \quad \text{for } T_{ground} = 10\text{-}25^{o}\text{C} \tag{3}$$

C_1=0.476 m/kW-^{o}C and T_o=-18^{o}C

where T_{ground} is in °C and L_{tot} is in m. The 2nd term in Eq. (3), $\dot{q}_{GHX,cool}$, is the cooling capacity of the GHX; notice that the cooling capacity is proportional to length of the GHX (a longer *GHX* provides more cooling) and inversely proportional to the initial ground temperature.

For the cooling tower characteristics and economic conditions considered in the parametric study, it is economically attractive to oversize the tower and then almost always operate it at low speed; low speed in this project is 50%. (The setpoint for high speed operation of the tower, T_{Cool1}, can therefore be set to 3-5^{o}C above the maximum entering heat pump temperature, implying that the tower is only operated at high speed in extreme conditions.) If a single-speed tower must be used, the tower is 1.3x the unmet cooling load; this load should then be calculated according to Equation (3) (modified from Equation (4)).

$$C_{CCCT} = 1.3\dot{q}_{unmet,cool} = 1.3\left(\dot{q}_{peak,cool} - \overbrace{\frac{L_{tot}}{C_1(T_{ground} - T_o)}}^{q_{GHX,cool}} \right) \quad \text{for } T_{ground} = 10\text{-}25^{o}\text{C} \tag{4}$$

C_1=0.736 m/kW-^{o}C and T_o=-18^{o}C

- The optimal cooling tower size can also be estimated based on the unbalance in the building load, with similar results to Equation (3). Figure 3 illustrates the optimal cooling tower size (normalized by building size) as a function of the ratio of the total annual heating load for the building to the total annual cooling load for the building, for each scenario.

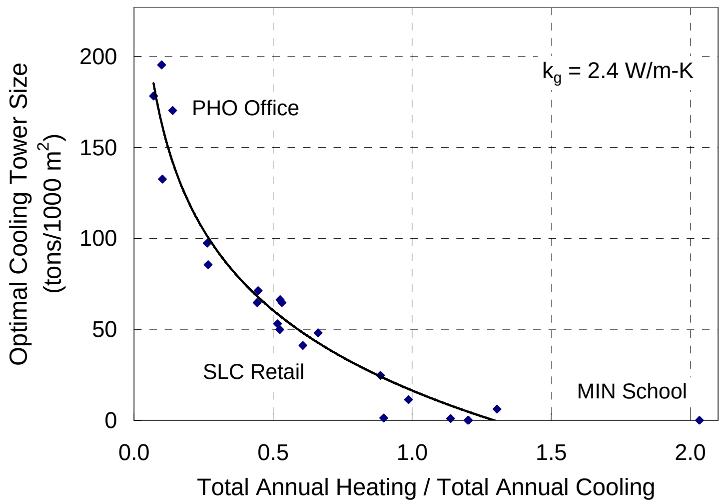

Figure 3. Normalized optimal cooling tower size as a function of the ratio of total annual heating load to peak cooling load. A few data points are labeled for reference.

- When using a dry fluid cooler rather than a cooling tower as the supplemental device, the optimal sizes follow trends that are similar to those described above.

c) The optimal sizes and control setpoints identified here never balance the load on the ground. Therefore, the ground temperature always increases over time by an amount that depends on the difference between the total annual heating and cooling loads. The timespan selected for the simulation (20 years here) can have a significant impact on the results.

d) The optimal design of the system does not depend substantially on the economic parameters used in the model (although the life cycle costs are substantially affected by the economic parameters); the equipment is sized almost entirely based on meeting the specified loads (as discussed above) which implies that it is rarely economically attractive to purchase larger equipment in order to improve the system efficiency. The design guidelines for sizing the equipment therefore remain valid over a large range of economic parameters.

e) Control setpoints: choose optimal control setpoints as indicated below.

- Supplemental cooling device: operate this device when conditions are favorable; that is, when the fluid temperature entering the device is greater than the ambient wet bulb (dry bulb for dry fluid cooler) + ΔT_1, where:

  ΔT_1 = 15°C for a cooling tower, if $T_{wb,July}$ < 21°C,
  13°C for a cooling tower, if $T_{wb,July}$ 21 to 24°C
  11°C for a cooling tower, if $T_{wb,July}$ > 24°C
  6.7°C for all dry fluid cooler scenarios

  where T_{wbJuly} is the ASHRAE 1% design wet bulb temperature for July.

- GHX, cooling setpoint (T_{Cool2}): the GHX is bypassed only occasionally. The optimal value of T_{Cool2} (in °C) increases with more cooling dominated buildings, as described in Equation (5).

$$T_{Cool2} = 8.5\left(\frac{q_{peak,cool}}{q_{peak,heat}}\right) \tag{5}$$

- GHX, heating setpoint (T_{Heat1}): the GHX is never bypassed in heating mode, therefore T_{Heat1} should be set to a high temperature that is never reached in practice.

f) Operating temperature sensitivity: the lowest *LCC* for the HyGCHP model generally occurs at a minimum operating temperature below 2°C. The limits on the temperature of the fluid entering the heat pump strongly drive the optimization; the equipment is sized in order to keep the entering fluid temperature within the specified limits (see Table 1). The entering fluid temperature is not allowed to go below 2°C or above 35°C in the base cases. These base case temperature limits reflect "typical" design values. However, in a separate study when the temperature limits are allowed to relax to between -7 and 43°C (more typical of the allowable operating range of a heat pump) then the optimizer will typically choose an optimal minimum operating temperature that is lower than 2°C, trading off heat pump efficiency for reduced first cost. The size of the GHX is reduced by up to 50%.

4.2 Cost Comparisons – Cooling Dominated Systems

The parametric study optimized a geothermal-only system, a boiler/tower system, and the hybrid geothermal system options for each building/climate combination. Therefore, meaningful comparisons can be made between these options based on life cycle costs.

a) In most moderate and southern climates, hybrid geothermal systems have a lower life cycle cost (LCC) than other options.

- The life cycle savings (*LCS*) of hybrid systems compared to geothermal-only systems is proportional to how unbalanced the climate is. Figure 4 illustrates the life cycle savings associated with a hybrid system compared to a geothermal-only system (normalized by the building size) as a function of the ratio of the total annual heating load to the total annual cooling load for each building/climate scenario. Just the savings from reduction in GHX cost is $75/m^2 in a climate like Atlanta, and decreases to $32/m^2 in a climate like St. Louis.

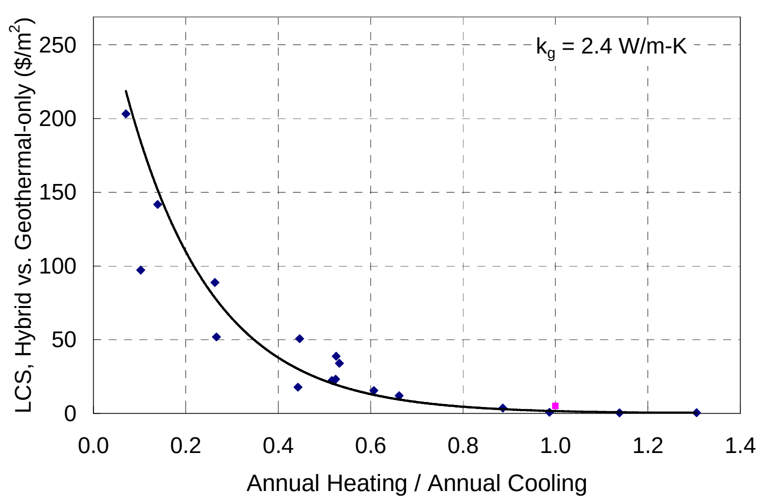

Figure 4. Life-cycle savings of hybrid systems over geothermal-only systems as a function of ratio of total annual heating load to total annual cooling load.

- The LCS of a hybrid system compared to a boiler/tower system is only \$11-22/m^2; this savings is smaller than the LCS of a hybrid system compared to a geothermal only system. The LCS of the hybrid system relative to the boiler/tower system increases with peak heating load. The savings is negligible when the peak heating load is near zero.

- The life cycle cost (LCC) generally changed very little from the hybrid that used a closed-circuit cooling tower. In some warmer climates, the LCC was slightly higher with a DFC.

b) For northern climates (like Minneapolis or colder) geothermal-only systems have a lower LCC than (cooling-dominated) hybrid or boiler/tower systems.

c) In warm dry climates (like Phoenix), buildings with low heating loads have almost the same LCC for a hybrid and a boiler/tower system.

d) Unlike the optimal design parameters, the observed costs and therefore *LCS* are sensitive to economic parameters. For example, when fuel inflation is increased to 7.5%, the *LCS* of hybrid systems as compared to boiler/tower systems *doubles*. The effect of GHX cost is also studied; increases in GHX cost significantly *increase* the savings of hybrid systems as compared to geothermal-only and *decrease* the savings as compared to boiler/tower systems.

4.3 Cost Comparison and Observations – Heating Dominated Systems

The heating-dominated hybrid systems were studied for climates represented by Minneapolis and Edmonton (northern Alberta, Canada). For the assumptions listed in Table 1, the results of the heating-dominated study suggest the design guidelines detailed below.

- With the assumptions used here, a solar/geothermal hybrid is never economically viable; this result does not apply to direct solar heating systems that bypass the heat pumps.

- The boiler/geothermal hybrid option is marginally beneficial in Minneapolis, but increasingly attractive going north from there. The GHX in an optimal boiler/geothermal system is sized to meet the peak cooling load and the boiler is sized to meet the unmet heating load. (69% of load in Minneapolis).

5 CONCLUSIONS

A HyGCHP model was created using TRNSYS, as shown in Figure 1. Parametric studies were carried out using the HyGCHP model, optimizing the system (to lowest life-cycle cost) for a range of building/climate load scenarios. The results from these studies were compiled into a set of general design guidelines/observations. Firstly, the conclusions refine (but generally agree with) the common idea that the GHX in a cooling dominated hybrid system should be sized to just meet the peak heating load. Consequently, the supplemental cooling device should be sized to meet the remaining cooling load; equations were given for determining these sizes. The guidelines also indicate how to implement an optimal control strategy for such a system, based on operating the supplemental device whenever conditions are favorable. Relationships between loads and the cost savings of hybrid systems were also established. Finally, a few general conclusions were made regarding the limited effectiveness of hybrid systems in heating-dominated climates.

6 ACKNOWLEDGEMENT

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