Misconceptions Regarding Design of Ground-source Heat Pump Systems

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1 Introduction

Ground-source heat pump (GSHP) systems are perhaps the most widely used "green" heating and cooling systems, with an estimated 1.7 million installed units with total installed heating capacity on the order of 18 GW. Despite some research and installations earlier in the 20th century, the large number of current installations can be traced back to research beginning in the late 1970s after the oil crisis. Despite the widespread availability of research results encapsulated in handbooks and design software, misconceptions persist among prospective system designers. The most problematic widely-held misconception is the idea that ground heat exchangers may be sized based on rules-of-thumb which relate ground heat exchanger length to building peak heating or cooling loads or to installed capacity of equipment, typically expressed as W/m or ft/ton.

Although such a rule-of-thumb is highly desirable, and there have been a number of such rules-of-thumb promulgated, there is little possibility of a meaningful rule-of-thumb that will cover a wide range of commercial buildings. The purpose of this paper is to explain why this is the case and quantitatively demonstrate this. But, in short, it is primarily related to the long time constant of the ground surrounding the ground heat exchanger as well as the highly variable relationship between peak heating and cooling loads and annual heating and cooling loads. The time constant of the ground surrounding a commercial ground heat exchanger is typically on the order of years. Practically speaking, this may mean that the heat pump entering fluid temperatures (EFT) rise (or fall) over time so that each year's peak EFT is greater that the last, with the greatest change coming in the first three or four years. This rise or fall over time has nothing to do with the peak load and everything to do with the annual loads. Therefore, rules-of-thumb that try to relate the size of ground heat exchanger to the peak loads are destined to fail, unless certain special conditions are present. These special conditions include:

- A strong relationship between peak loads and annual loads that is typically only the case for envelope-dominated buildings in a given location.
- Reasonably constant ground thermal properties between locations for which the rule-of-thumb would be applied.

The main example of where these special conditions would be present are residential buildings in areas with very similar climates and geology. It is possible that rules-of-thumb could be developed for other envelope-dominated buildings, like warehouses, or possibly, precisely identical buildings that meet the second condition, for example, chain restaurants in a small geographical area. However, it cannot be expected that reasonably accurate and general rules-of-thumb for ground heat exchanger design suitable for use with commercial/institutional buildings will ever be available.

Fortunately, a positive alternative to rules-of-thumb—the use of simulation-based design, coupled with in situ measurement of ground thermal properties—is available.

2 Previous Work

Results from one previous study are directly related to the current question and complementary to the results presented in this paper. Underwood and Spitler (2007) reported on a parametric study of GSHP systems in the UK. A typical four story UK office building was used as a baseline building. To this baseline building, a number of modifications were made to the orientation, envelope insulation, thermal capacitance, internal heat gains and building configuration in order to span the range between high and low extremes of expected heating and cooling loads for a single UK location - London. From all of these options, three sets of simulation results were selected to represent “typical”, “low” and “high” energy demand cases. Then, four different options for plant were considered: perimeter heating with and without chilled ceilings, as well 4-pipe fan coil units, and an all-air heating and cooling system. In the paper, peak instantaneous borehole heat transfer rates (W/m) are given for each system option as averages of the combinations of three energy demand cases and three soil diffusivity/ working fluid combinations. These range between 14 and 22.5 W/m. If the results are disaggregated, so that the individual cases may be examined, a more complex picture emerges. As shown in Table 1, the peak instantaneous heat transfer rate for heating conditions varies widely, between 7.2 and 40.7 W/m.

Table 1: Peak Instantaneous Borehole Heat Transfer Rates for Heating Conditions (W/m)

Energy Demand Case	Soil diffusivity/ working fluid	Htg. Only	Htg. w/ Chilled Ceilings	Fan Coil Units	All-Air
High	Low/water	7.2	10.0	7.4	12.7
Typical	Low/water	8.1	11.1	14.3	14.5
Low	Low/water	12.5	11.4	18.4	20.0
High	High/water	13.3	16.1	13.3	18.2
Typical	High/water	13.5	16.5	18.8	17.0
Low	High/water	16.1	17.8	21.1	22.4
High	Low/A.F.	15.8	22.4	15.9	27.7
Typical	Low/A.F.	19.4	10.4	22.8	29.8
Low	Low/A.F.	27.2	10.7	30.8	40.7

This wide range does convey some of the difficulty involved in trying to create a rule-of-thumb. Furthermore, the reasons for the variations may or may not be easy to discern. For example, it is fairly clear that going from a low soil diffusivity to a high diffusivity allows a higher W/m peak heat transfer rate. Going from a high diffusivity to low diffusivity, but changing working fluids to an antifreeze mixture allows even higher heat transfer rates, because the minimum EFT is 4ºC lower. Other variations may be harder to understand. E.g. changing from a heating only system to a system with both heating and chilled ceilings can either increase or decrease the allowable W/m, depending on whether the chilled ceilings predominantly serve to recharge the ground, or become a limiting factor on the cooling side and require an increase in ground heat exchanger size, as is the case with antifreeze. The building envelope can also make a substantial difference. Why? For this building, improving the envelope in the UK climate reduces the annual heating load significantly, while slightly increasing the annual cooling load. This change in ratio of loads changes the ratio of heat rejection/extraction. As the imbalance increases, there is a tendency for the annual peak fluid temperatures to drift upward or downward. Depending on the design temperature constraints, this drift may simply increase the required ground heat exchanger size or also switch which design temperature limit constrains the design.

If the temperature drift is upward with time, the GSHP system may be described as “cooling dominated”, which is usually taken to mean that the ground loop heat exchanger of the GSHP system will reject significantly more heat to the ground than it extracts on an annual basis. As a result, the ground temperature surrounding the heat exchanger will rise over the system operation period. A “heating dominated” system has the ground loop heat exchangers extracting significantly more heat from the ground than it rejects on an annual basis. As a result, the ground temperature surrounding the heat exchanger will fall over the system operation period.

However, 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 temperature limits on the heat pump and the 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 this reason, two new terms may be introduced: “heating constrained” and “cooling constrained”; these terms describe systems for which the designs are driven by the system heat extraction or rejection, respectively.

3 Methodology

Proving that reasonably accurate and general rules-of-thumb for ground heat exchanger design suitable for use with commercial/ institutional buildings are not possible is (perhaps) a difficult task. Certainly, it is highly undesirable to demonstrate this experimentally! However, experimentally-validated simulations should suffice to demonstrate, at the least, the problems associated with presuming upon rules of thumb.

Here, we rely on two types of simulations – simulations of commercial and institutional buildings which provide hourly heating and cooling loads and simulation of the ground source heat pump system.

Three different commercial/institutional buildings were simulated, each in a variety of locations. The first building is a three-story office building of floor area 7100 m^2, 9.1 m tall, with about 60% of the exterior facade glazed. This building has internal heat gains of 1 person per 5 m^2, 10 W/m^2 equipment and 13 W/m^2 lighting, all on office schedules. Further details of the building are given by Xu (2007).

The second building is a school building consisting of multiple classrooms and several larger common areas, such as a cafeteria. It has a total floor area of 4,925 m^2, typical occupant heat gains for classrooms, lighting heat gains of 10.8 W/m^2 and equipment heat gains varying with space type. The occupancy, lighting, and equipment gains occur from September-June, and are zero during July and August. Further details of the building are given by Chiasson, et al. (2004).

The third and final building is a hotel complex consisting of three identical buildings, each 10 stories tall with total floor area of 27,600 m^2, The complex has internal heat gains of 1 person per 36 m^2, 3 W/m^2 equipment and 8 W/m^2 lighting, operating on a hotel occupancy schedule and no setback. Further details of the building are given by Xu (2007).

Fourteen locations were chosen to represent fourteen of the fifteen U.S. climate zones (Briggs, et al. 2003). For each combination of building and location, the heating and cooling loads for every hour of the year were found by simulating the building in the EnergyPlus program.

Then, with the hourly heating and cooling loads in hand, a second simulation of the GSHP system was performed using the HVACSIM+ modular simulation environment [Clark 1985]. The system in HVACSIM+ consists of a ground loop heat exchanger model and a simple heat pump model. The ground loop heat exchanger model uses the g-function approach originated by Eskilson [1987], and consists of both long time step and short time step values [Xu and Spitler, 2006]. This model was validated for a ground heat exchanger with three vertical boreholes, each approximately 76 m based on twelve months of experimental data [Gentry et al. 2006, Xu 2007]. The complete design of the hybrid ground source heat pump test facility is discussed in more detail by Hern [2002]. An experimental validation of an earlier version of the model was reported by Yavuzturk and Spitler [2001] for a system with 120 vertical boreholes.

Additionally, the HVACSIM+ simulation utilizes a simple equation-fit heat pump model. The coefficients for the model as used in this work were based on catalogue values for a commercially available water-to-air heat pump

The simulations were used iteratively to determine lengths for the ground loop heat exchanger so that the fluid temperature entering the heat pump stayed within prescribed bounds. There are several design tools that automate this procedure, but for purposes of this work, we wished to establish the need for the design tool independently of a particular design tool.

The ground loop heat exchangers were sized so that the fluid temperature entering the heat pump stayed between 40°C and -5°C. To prevent freezing of the fluid in the ground loop, a 20% solution of ethylene glycol was selected as the working fluid. Each borehole was 127mm (5 in.) in diameter, with 1" Sch.40 pipe; boreholes were spaced 7.62m (25 ft) apart in soil with a conductivity of 3.5 W/m-K and heat capacity of 2160 kJ/m^3-K. Bentonite based grout wih a thermal conductivity of 0.74 W/mK was assumed. The number of boreholes for each system was selected to assure an individual borehole depth between 70m and 100m. The reader should note that the temperature limits, ground thermal properties, and borehole spacing are all at the favorable end of the range, in the sense that they allow minimally-sized ground heat exchangers. The reader should also note that the temperature limits and use of antifreeze were chosen to apply to all locations. In practice, locations such as Miami would be much more likely to use pure water.

4 Results

Using the simulation-based ground heat exchanger sizing procedure described in the previous section, required design lengths were determined for each of the three buildings, such that the heat pump entering fluid temperature never exceeds 40ºC and never falls below -5ºC. The peak heat rejection and peak heat extraction rates are divided by the total ground heat exchanger length to give the peak heat transfer in watts per meter of ground heat exchanger. These results are summarized in Table 2. For every combination of building and location, the design is constrained by either the upper temperature limit (40ºC) and can be said to be "cooling-constrained" or the lower temperature limit (-5ºC.) and can be said to be "heating-constrained." Whether each case in Table 2 is cooling-constrained or heating-constrained are indicated with bold numbers in Table 2. E.g., if the peak heat rejection rate is emboldened, the design is cooling-constrained and vice-versa.

From Table 2, a number of observations may be made. First, consider Duluth, one of the two colder climates where all three buildings are heating constrained. In Duluth, with all other aspects of the design being held constant – climate, ground thermal properties, undisturbed ground temperature, design temperature limits—the peak instantaneous heat extraction rate varies significantly. In Duluth, for example, the office ground heat exchanger can be sized to 48.2 W/m but the hotel can only be sized to 26.3 W/m. This raises the question of what an appropriate rule-of-thumb would be. If, say, 25 W/m were used as a rule-of-thumb, a ground heat exchanger for the office would be moderately oversized and more expensive than required. As the ground heat exchanger is often the largest part of the GSHP system cost and approximately represents the additional capital cost of a GSHP system over a conventional system, use of a conservative, "always works" rule of thumb will result in grossly-oversized systems and decisions to use some other type of less expensive system. Conversely, a rule-of-thumb based on, say, 50 W/m would give ground heat exchangers that are too small for some buildings.

Figure 1 demonstrates what would happen if a 50 W/m rule-of-thumb were used for the Duluth hotel. A horizontal line has been drawn in at the freezing point of the water/ethylene glycol mixture. As can be seen, the monthly minimum temperatures entering the heat pump quickly fall below the freezing point, although freezing will occur within the heat pump before it reaches that point. The 25 W/m design is adequate for the hotel.

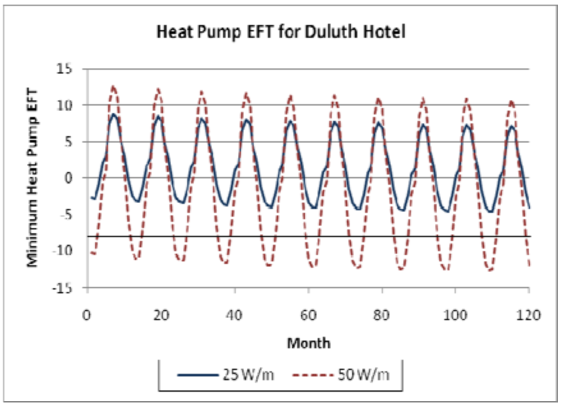

Figure 1: Monthly minimum heat pump EFT

Likewise, consider Houston, one of the five cases where the ground heat exchanger is cooling constrained for all three buildings. Figure 2 shows the monthly maximum heat pump EFT for two rule-of-thumb scenarios: 25 W/m, which is just a little lower than the 25.9 W/m maximum heat rejection rate that occurs when the ground heat exchanger for the hotel is sized to not exceed 40ºC heat pump EFT, and 67 W/m, which is acceptable for the Houston office building. For the hotel, the 67 W/m design will lead to temperatures in excess of 55ºC and heat pump failure due to excessive head pressure.

Another observation regarding Table 2 is that some very high W/m values can be found. Again, these are not design recommendations, but are only feasible because of a favorable combination of design temperature limits,

Table 2: Peak Heat Transfer Rates (W/m)

Location	Bldg.	Peak Ht. Rej. Rate (W/m)	Peak Ht. Extr. Rate (W/m)
Albuquerque, New Mexico	Office	**82.3**	27.5
	School	34.4	**73.2**
	Hotel	**57.8**	12.9
Baltimore, Maryland	Office	**78.0**	31.3
	School	24.1	**68.0**
	Hotel	**67.7**	23.5
Boise, Idaho	Office	93.6	**33.8**
	School	18.4	**56.5**
	Hotel	**71.9**	23.7
Burlington, Vermont	Office	115.0	**48.3**
	School	14.6	**52.9**
	Hotel	**85.0**	39.7
Chicago, Illinois	Office	**78.9**	36.0
	School	16.4	**52.4**
	Hotel	78.7	**37.2**
Duluth, Minnesota	Office	77.6	**48.2**
	School	7.9	**37.6**
	Hotel	41.6	**26.3**
El Paso, Texas	Office	**66.7**	17.5
	School	**52.8**	88.2
	Hotel	**36.3**	4.8
Helena, Montana	Office	116.8	**53.4**
	School	18.2	**57.1**
	Hotel	81.5	**42.5**
Houston, Texas	Office	**67.4**	8.7
	School	**45.6**	99.1
	Hotel	**25.9**	6.6
Miami, Florida	Office	**29.7**	2.8
	School	**31.7**	65.3
	Hotel	**41.0**	0.0
Phoenix, Arizona	Office	**48.9**	8.0
	School	**51.3**	52.7
	Hotel	**60.8**	5.1
Salem, Oregon	Office	**97.8**	42.6
	School	15.8	**60.0**
	Hotel	**75.6**	22.1
San Francisco California	Office	**92.0**	37.5
	School	24.7	**84.7**
	Hotel	**67.5**	9.4
Tulsa, Oklahoma	Office	**77.9**	25.7
	School	42.5	**96.1**
	Hotel	**55.5**	20.5

undisturbed ground temperatures, building load profiles, ground thermal properties, etc.

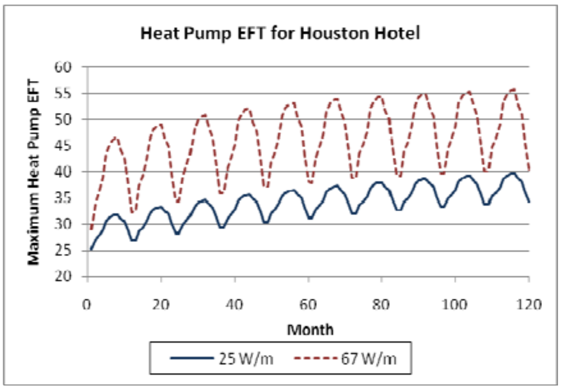

Figure 2: Monthly maximum heat pump EFT

The variability of the peak heat transfer rate is by no means a function only of building load profile. Consider Figure 3, which shows the peak heat extraction rate computed with a simulation-based design tool, showing the effects of the lower design temperature limit on the ground heat exchanger for the Chicago office building. Below about 2 °C, the peak heat extraction rate is essentially fixed because the design is cooling constrained. At about 2 °C, the design transitions from being cooling constrained to being heating constrained, and above 2 °C the required ground heat exchanger size increases rapidly and the peak heat extraction rate drops rapidly. Accordingly, any rule-of-thumb for W/m would somehow have to take this design limit into account.

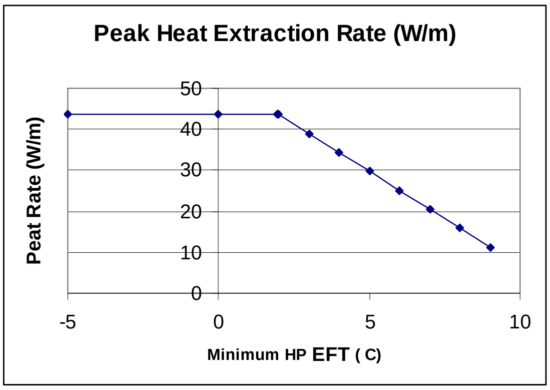

Figure 3: Effect of Design Temperature Limit on Chicago Office Building GHE Design

Or, consider the effect of grout thermal conductivity on the design. The assumed ground thermal conductivity is quite high for this design; as a result, the design is fairly sensitive to borehole thermal resistance. The grout thermal conductivity assumed for this study is based on a typical value for bentonite grout (0.74 W/mK); commercially-available thermally-enhanced grouts are available with thermal conductivities as high as 2.4 W/m K. Figure 3 shows the sensitivity of the design to the grout thermal conductivity. Again, to be useful, a rule-of-thumb would need to take this sensitivity into account.

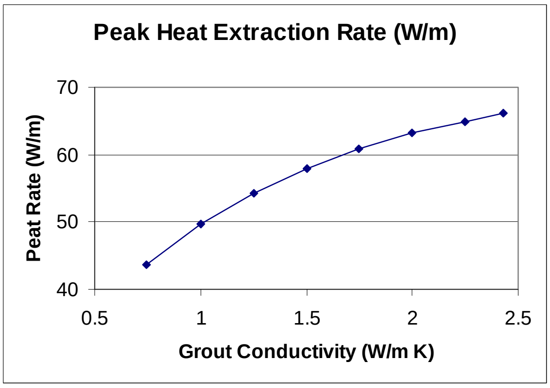

Figure 4: Effect of Grout Conductivity on Chicago Office Building GHE Design

To summarize, Table 2 contains a wide range of values; Figures 3 and 4 shows the sensitivity of the design to just a couple of the design variables. Other design parameters are also important, but space precludes a complete sensitivity study or even analysis of the interaction between design variables. From this, the astute reader will infer that there is a high degree of variability. Only the irresponsible reader will take these as recommendations! They most assuredly are not recommendations.

5 Conclusions

The use of rules-of-thumb for design length remains common in practice, and often leads to oversized, expensive systems or undersized failures. In reality, there are no generally-applicable rules-of-thumb that cover the diverse range of buildings and ground heat exchanger scenarios. Procedures based on building and ground heat exchanger simulation, accompanied by measurement of ground thermal properties will lead to successful designs. Though these procedures are more time-consuming in the design phase, they are a necessary prerequisite to successful, efficient GSHP systems.

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