This paper was originally published as an *ASHRAE conference paper* and may be cited as:

Gagné-Boisvert L. and M. Bernier. 2016. Accounting for borehole thermal capacity when designing vertical geothermal heat exchangers. ASHRAE summer conference, St-Louis, Missouri, June 2016. Paper ST-16-C027.

Accounting for Borehole Thermal Capacity when Designing Vertical Geothermal Heat Exchangers

Laurent Gagné-Boisvert
Student Member ASHRAE

Michel Bernier, PhD, PE
Member ASHRAE

ABSTRACT HEADING

Steady-state heat transfer inside boreholes is usually assumed when sizing geothermal boreholes and a constant borehole thermal resistance is used to calculate the temperature difference from the fluid to the borehole wall. Thus, heat rejected into the fluid is assumed to be transferred immediately at the borehole wall. In reality, rejected heat will heat the fluid and the grout first before reaching the borehole wall and be transferred to the ground. These transient effects, caused by the fluid and grout thermal capacities, are beneficial as they reduce the peak ground loads and, consequently, the required borehole length. In the first part of this study, simulations are performed on a residential ground-source heat pump system to quantify borehole transient effects. Then, correction factors for the current ASHRAE sizing equation are proposed to consider borehole thermal capacity. Results show that neglecting borehole transient effects leads to oversized boreholes and may overestimate heat pump energy consumption by about 5 %. A parametric study performed on several types of boreholes shows that the correction factor to the ASHRAE sizing equation varies from 0.69 to 1.24 for those particular cases. Correction factors below one are typically associated with oversized heat pumps which lead to intermittent heat pump operation and maximize the use of the borehole thermal capacity.

INTRODUCTION

The magnitude and duration of peak loads are important when sizing vertical geothermal boreholes. In the ASHRAE sizing equation for vertical geothermal boreholes, represented in a simplified form in Equation 1 (ASHRAE, 2015), the required length L is determined based on a sum of loads multiplied by equivalent thermal resistances divided by a temperature difference (mean fluid temperature in the borehole minus the undisturbed ground temperature). The second and third terms in this equation represent the contribution of the annual mean ground load, q_y, and peak monthly load, q_m, multiplied by their respective equivalent thermal resistance, R_{10y} and R_{1m}. The peak ground load, q_h, is present in the first and fourth terms in the numerator. In the first term, q_h multiplies R_b, an equivalent borehole thermal resistance from the fluid to the borehole wall. In the fourth term, q_h multiplies an equivalent ground thermal resistance, R_{6h}, which is typically based on a peak duration of 6 hours.

$$L = \frac{q_h R_b + q_y R_{10y} + q_m R_{1m} + q_h R_{6h}}{\frac{T_{inHP} + T_{outHP}}{2} - T_g} \tag{1}$$

For a residential system such as the one depicted in Figure 1, the peak ground load is determined based on the amount of heat rejected (or collected) by the heat pump for a given inlet fluid temperature, T_{inHP}. In ASHRAE's sizing equation, the peak ground load is assumed to be transferred immediately at the borehole wall. With reference to Figure 1, this means that q_b is equal to q_h. The temperature difference between the mean fluid temperature in the borehole and the borehole wall temperature is obtained using a steady-sate borehole thermal resistance, R_b. In reality, when there is a change in the value of q_h, there is a time lag before q_b reaches q_h. During that period, $q_b < q_h$ and R_b has not yet reached a steady-state value. Heat is stored in the borehole, due to the thermal capacity of the fluid and

Laurent Gagné-Boisvert is a M.A.Sc. student in the Department of Mechanical Engineering, Polytechnique Montreal, Montreal, Quebec, Canada. Michel Bernier is a professor in the Department of Mechanical Engineering, Polytechnique Montreal, Montreal, Quebec, Canada.

grout, before being released at the borehole wall. This transient period can last several hours depending on the characteristics of the borehole and cycling behavior of the system. The effects of the borehole thermal capacity are intimately linked to the operation of the heat pump. The heat pump will reject (or collect) heat as long as it operates. The period of operation will depend on the degree of oversizing of the heat pump. If the heat pump is undersized, it may run for more than the 6 hours suggested in Eq. 1. Conversely, an oversized heat pump will operate intermittently and will run for less than 6 consective hours.

In summary, borehole thermal capacity and heat pump sizing have an effect on the value of q_h and R_b in Equation 1. The objective of this paper is to quantify the combined effect of borehole thermal capacity and heat pump cycling on boreholes sizing. The paper is organized as follows. A literature review on the importance of borehole thermal capacity and some basic principles of borehole transient effects are presented first. Then, annual simulations of a residential ground-source heat pump (GSHP) system are performed with and without fluid and grout thermal capacities to determine the required length and heat pump energy consumption differences. Finally, simulations for various heat pump oversizing and several types of boreholes are performed. A borehole thermal capacity correction factor for the ASHRAE borehole sizing equation is then presented.

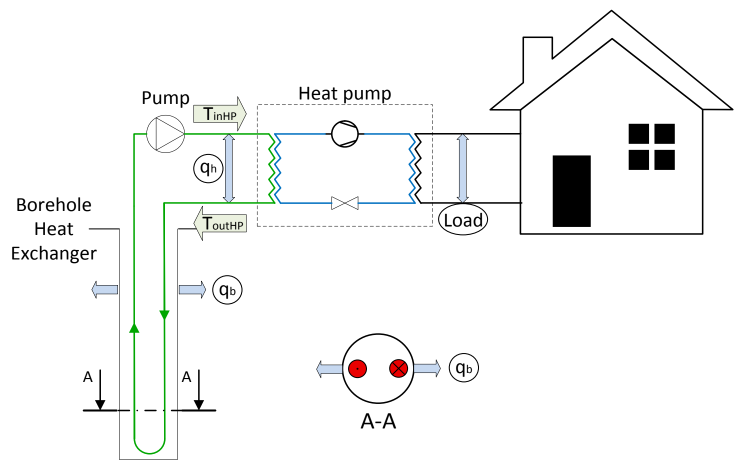

Figure 1 Schematic representation of the simulation setup and heat transfer rates.

LITERATURE REVIEW

Hellström (1991) was perhaps the first author to discuss borehole thermal capacity. He explained that fluid thermal capacity plays an important role under a time t_b equivalent to 5 r_b^2/α , where r_b is the borehole radius and α is the ground thermal diffusivity. This corresponds to about 2-3 hours for typical boreholes and longer for boreholes with small thermal diffusivities and large bore radius. During that time, a relatively large part of the thermal load entering the borehole is absorbed by the fluid instead of being instantly transferred to the ground.

Kavanaugh and Rafferty (1997), in their GSHP design guide, considered a constant equivalent bore thermal resistance R_b. They stated that the thermal mass of the grout, pipes and liquid are too small when compared to the adjacent ground to be considered. This formed the basis of the steady-state borehole heat transfer assumption used in the current ASHRAE sizing equation. The alternative sizing equation introduced in the 2015 handbook also assumes a steady-state borehole thermal resistance.

Young (2004) developed the borehole fluid thermal mass model (BFTM), a borehole model accounting for fluid and grout thermal mass. He compared the results of simulations using a classic steady-state model and his dynamic model. Relatively large differences in the outlet fluid temperature are obtained (1.3 °C/2.3 °F), especially with short peak load cases. He also concluded that yearly energy performance is correctly estimated by the use of a steady-state model. Xu and Spitler (2006) developed a one-dimensional borehole model accounting for fluid thermal mass. They noted that fluid thermal capacity has a tendency to damp outlet fluid temperature peaks. Their work used short time-step response factors, based on a model accounting for borehole thermal mass (Yavuzturk and Spitler, 1999).

Salim-Shirazi and Bernier (2013) also developed a one-dimensional transient ground heat exchanger model accounting for fluid and grout thermal capacities. They replaced the two-pipe geometry with an equivalent cylinder but kept the thermal mass of the fluid and grout intact. It has been used to perform short-time and annual simulations. They stated that neglecting borehole thermal capacity always leads to lower outlet fluid temperature in heating mode. The difference is more important as heat pump operates intermittently. Their results show that the predicted annual heat pump Coefficient of Performance (COP) is underestimated by an average of 4.5%.

Parisch et al. (2015) developed a pre-pipe model accounting for the fluid and grout thermal capacities. This pipe is located upstream of the borehole. It is coupled with a conventional steady-state borehole model to simulate its dynamic behavior. Residential simulations performed with the pre-pipe increased the seasonal performance factor of the GSHP system from 2.5 up to 3.5 and reduced the required length from 110 to 90 m (361 to 295 ft).

Ma et al. (2015) used their quasi-3D model, which is based on effective overall thermal resistances, to modify the resistance terms in the ASHRAE sizing equation. This model considers the grout heat capacity and evaluates ground heat transfer with the full-scale g-functions. This approach leads to slightly shorter boreholes, especially when the daily peak pulse has a short duration (1-3 hours). This method is also more accurate to size short boreholes.

Godefroy and Bernier (2014) developed a thermal resistance and capacity model (TRCM) and implemented it in the TRNSYS environment. This model considers the fluid and grout thermal capacities. It has been experimentally validated (Godefroy et al., 2016). Annual energy simulations showed that heat pump energy consumption can be overestimated by 3 % when thermal capacity is neglected. They also concluded that the grout thermal capacity has a minor effect and that the fluid thermal capacity is the dominant parameter. Finally, it should be noted that the most commonly used model for simulating bore fields, the so-called Duct ground Storage (DST) model (Hellström et al., 1996), assumes a steady-state condition in the borehole and uses a constant borehole thermal resistance.

THERMAL CAPACITY EFFECTS

The thermal capacity of a typical borehole with a 15 cm (6 in) diameter and a length of 150 m (492 ft) is in the order of 9 MJ/K (4.7 kBTU/°F). Thus, a power injection of 2.5 kW (8.5 kBTU/hr) during one hour will increase the average borehole temperature by 1 K (1.8 °F).

In this section, two simple cases are examined to quantify the difference between q_h and q_b when boreholes are subjected to varying conditions. The main characteristics of the borehole are given in Table 1.

Table 1. Main Characteristics of the Borehole

Parameter	S.I. Value	S.I. Unit	I.P. Value	I.P. Unit
Depth	180	m	591	ft
Borehole diameter	0.15	m	6	in
Inside pipe radius	0.013	m	0.51	in
Outside pipe radius	0.016	m	0.63	in
Borehole thermal resistance	0.182	m.K/W	0.315	hr.ft.°F/BTU
Grout conductivity	0.83	W/m.K	0.48	BTU/hr.ft.°F
Grout thermal capacity	3000	kJ/m^3.K	44.7	BTU/ft^3.°F
Ground thermal conductivity	2.2	W/m.K	1.27	BTU/hr.ft.°F
Ground thermal diffusivity	0.096	m^2/day	0.94	ft^2/day
Fluid thermal capacity	3.87	kJ/kg.K	0.924	BTU/lbm.°F
Pipe conductivity	0.42	W/m.K	0.24	BTU/hr.ft.°F
Flowrate	0.56	L/s	9	gpm

Simulations are performed using TRNSYS and the TRCM model of Godefroy (2014). This model discretizes the borehole in a series of thermal resistances and capacitances much like in a finite difference approach but with a relatively coarse grid so as to make computational cost reasonable when performing annual simulations. This model has been verified with other borehole models that include thermal capacity and validated against experimental results (Godefroy, 2014 and Godefroy et al., 2016). Furthermore, simulations were also performed with the DST model

(Hellström et al., 1996) for cases when thermal capacity is neglected. The results obtained with the DST model were in excellent agreement with the TRCM when the fluid and grout thermal capacities are artificially set to a very small value. A root-mean-square error (RMSE) of 0.083 K (0.15 °F) for the outlet fluid temperature and of 0.078 kW (0.27 kBTU/hr) for the heat transfer rate were obtained for the data presented in Figure 2.

In the first case, the borehole is subjected to three hourly heat collection rates, q_h, of 2, 4, and 8 kW (6.8, 13.6 and 27.3 kBTU/hr) followed by a three hour recovery phase where $q_h = 0$. This results in a total extraction of 14 kWh (48 kBTU) over a 6 hour period. Simulation results, performed with a 36 second time step (0.01 hr), are shown in Figure 2. The top curve shows that q_b is lower than q_h when thermal capacity is accounted for. For example, at t = 3 hours, $q_b \approx 4.4$ kW (15 kBTU/hr) while $q_h = 8$ kW (27.3 kBTU/hr). Thus, for this condition, 4.4 kW (15 kBTU/hr) are taken from the ground and 3.6 kW (12.3 kBTU/hr) are extracted from stored energy in the borehole. During the first three hours, about 9 kWh (31 kBTU) have been retrieved from the borehole. For t > 3 hours, $q_h = 0$, but q_b is non zero (for example, at t = 6 hours, $q_b \approx 0.7$ kW (2.4 kBTU/hr)). Heat is thus being transferred from the ground to thermally "regenerate" the borehole.

As shown on the bottom graph of Figure 2, the outlet fluid temperature decreases at a much slower rate with borehole thermal capacity. At t = 3 hours, $T_{inHP} \approx 4$ °C (39 °F) when borehole thermal capacity is considered. The corresponding value without borehole thermal capacity is ≈ 1 °C (34 °F). These lower outlet fluid temperature predictions when borehole thermal capacity is neglected have an impact on the heat pump COP and on the required borehole length as it will now be shown.

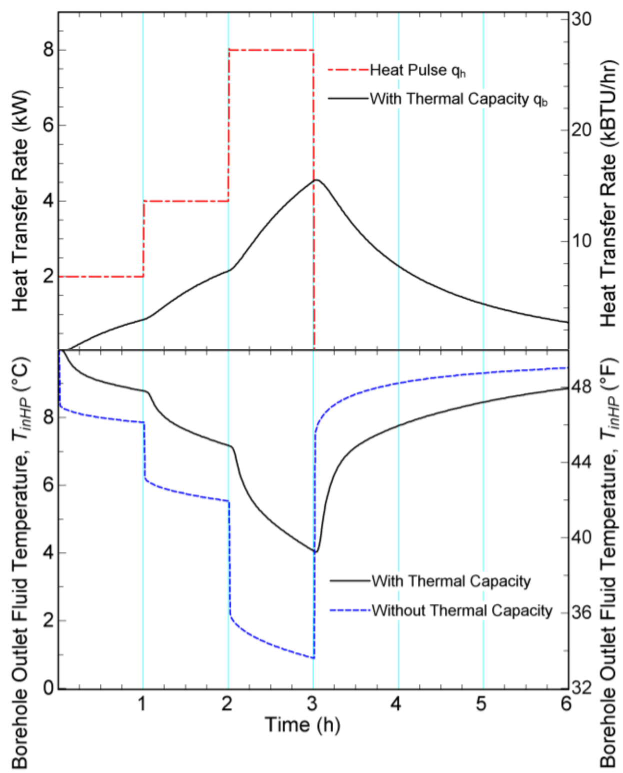

Figure 2 Heat transfer rate (top) and borehole outlet fluid temperature (bottom) for 3 hourly heat extraction pulses and a 3 hour recovery period for borehole models with and without thermal capacity.

To evaluate the required borehole length with and without borehole thermal capacity, dynamic simulations using TRNSYS v17 (Klein et al. 2010) are performed with a 6 minute time step, small enough to capture transient effects in the borehole. The system under study is the one presented in Figure 1 and consists of a 3 ton (10.5 kW) water-to-air ground-source heat pump providing space heating for a single-family house. The house is simulated in a heating dominated climate (Montreal, Canada). For the present case, the heat pump is oversized by 50 % giving an effect coverage of 150 %. The effect coverage is defined here as the heat pump capacity at peak load conditions divided by the building peak load. The house loss coefficient is set to have a peak load of 5.4 kW (kBTU/hr) and an annual heating energy requirement of 14 500 kWh. The energy performance and temperatures of the system are obtained by running 10 years + 1 month + 6 hours simulations (88350 hours) in accordance with the duration of the three ground heat pulses of the ASHRAE sizing equation (Eq. 1). Typical models found in TRNSYS for the house, heat pump and thermostat are used. The main characteristics of the borehole are given in Table 1. The TRCM borehole model from Godefroy and Bernier (2014) is used to model the single U-tube borehole. The intent of these simulations is to find the required length which gives a minimum heat pump inlet temperature equal to 0 °C (32 °F).

Table 2. Sizing With and Without Thermal Capacity

Calculation method	Thermal capacity	Minimum value of T_{inHP}	Required length (m/ft)	Annual Heat Pump Energy Consumption (kWh)	Relative heat pump energy consumption
Simulations	no	0 °C (32 °F)	156/512	5 853	1.0
Simulations	yes	1.5 °C (34.7 °F)	156/512	5 460	0.93
Simulations	yes	0 °C (32 °F)	131/430	5 586	0.95
Eq. 1 (ASHRAE)	no	0 °C (32 °F)	180/591	-	-

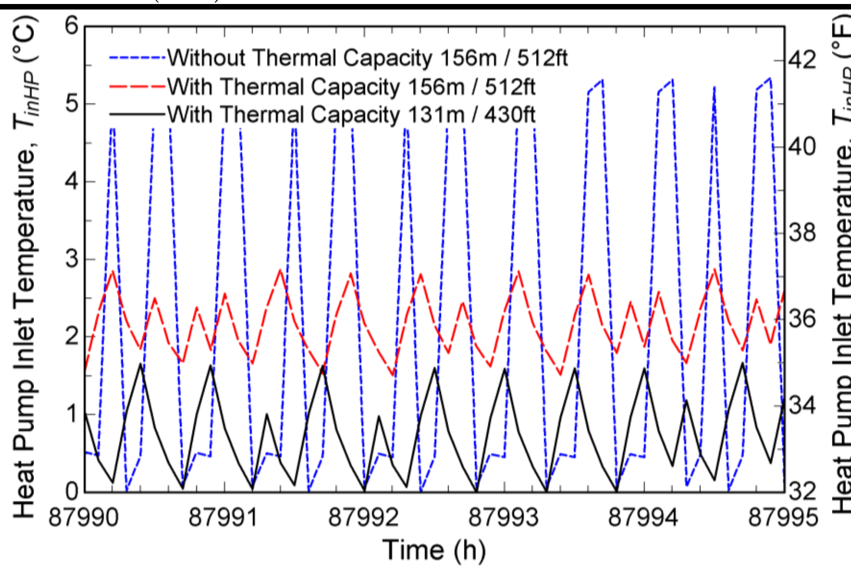

Figure 3 Heat pump inlet temperature for models with and without thermal capacity.

Results are summarized in Table 2 for four cases. The three first cases are also presented in Figure 3 where only the hours near the peak ground loads are shown, i.e. from t = 87990 to 87995 hours, with approximately two on-off heat pump cycles per hour. In the first simulation, borehole thermal capacity is neglected and a borehole length of 156 m (512 ft) is required to reach the minimum setpoint of 0 °C (32 °F) for the inlet temperature. With each heat pump cycle, the inlet fluid temperature oscillates over a ≈ 5.5 °C (10 °F) range. For the second case, the borehole length is kept at 156 m (512 ft) and simulations are run with borehole thermal capacity. Temperature oscillations are reduced down to about ≈ 1 °C (1.8 °F) and the borehole is clearly oversized as the minimum heat pump inlet temperature is ≈ 1.5 °C (≈ 34.7 °F). Finally, if the heat pump inlet temperature is allowed to reach a value of 0 °C (32 °F) then the required borehole length is 131 m (430 ft). This constitutes a length reduction of 16 % when compared to the first case. On the heat pump side, there is a 5 % difference in the energy consumption between the first and third cases. Thus, properly sizing a borehole by considering its thermal capacity can reduce the required length and the predicted energy consumption. As noted on the last line of Table 1, the ASHRAE sizing equation (Eq. 1) gives a length of 180 m (592 ft), which is longer than the dynamic sizing neglecting thermal capacity (156 m/512 ft). This is because the peak hourly load q_h lasts less than the 6 hours considered in the ASHRAE equation.

CORRECTIONS TO THE ASHRAE SIZING EQUATION

The actual sizing equation presented by ASHRAE (Eq. 1) doesn't consider the energy stored in the borehole (fluid and grout). Also, the sizing equation assumes that the peak load is constant for 6 consecutive hours. This case only happens if the heat pump is undersized and runs continuously at peak conditions. Most often, the heat pump will operate intermittently during the 6 hour period and borehole thermal capacity will be beneficial in that it will reduce the length requirement.

Several simulations with different operating conditions are performed to quantify the sizing error made when neglecting borehole thermal capacity and heat pump size. Much like results presented in Figure 3, simulations are performed over 10 years, 1 month and 6 hours, ending with the day with the peak conditions. In the simulations, the length is adjusted until T_{inHP} reached 0 °C (32 °F). The required length according to the ASHRAE equation is also determined for each case. The hourly, monthly and annual ground pulses are obtained using the results of the dynamic simulations. The annual and monthly ground pulses are the average of the first ten years and of the last month of operation, respectively. The hourly peak is determined as the highest heat pump extraction rate, occuring during the last month of operation.

Three main parameters are varied: i) Effect coverage (i.e. heat pump undersizing/oversizing); ii) Flowrate; iii) High/low ground and grout thermal conductivities and bore diameter. The effect coverage is varied from 50 % to 150 %. To modify the effect coverage, the peak load is changed by adapting the building heat loss coefficient. Thus, the same 3 ton heat pump unit is used in all simulations. For undersized heat pumps (effect coverage < 100 %), a two stages auxiliary electrical system is added to the heat pump, each with a capacity equal to half the difference between peak load and heat pump capacity. Two flowrates are used, 0.28 and 0.56 L/s (4.5 and 9 gpm), representing flowrates of 1.5 and 3.0 gpm/ton. Finally, two types of boreholes are simulated. The first borehole, referred to as borehole B, is the one described in Table 1. It is considered to have low thermal conductivites for both the borehole and the ground. Borehole A has the same characteristics as the one presented in Table 1 except that it is located in a more conductive ground (3.5 W/m.K or 2 BTU/hr.ft.°F), it has a smaller borehole diameter of 0.1 m (4 in) and a grout conductivity of 2.1 W/m.K (1.2 BTU/hr.ft.°F) with a corresponding steady-state borehole thermal resistance, R_b, of 0.074 m.K/W (0.13 hr.ft.°F/BTU). Borehole B' is also examined with a 1.5 in diameter pipe (0.019 m inside pipe radius) and a R_b value of 0.22 m.K/W (0.39 hr.ft.°F/BTU). Finally, two B boreholes are evaluated in parallel and in series configurations for cases where it is cheaper to drill two shallow holes instead of a deeper one. For the parallel setup, the 4.5 and 9 gpm flowrates (0.28 and 0.56 L/s) are split equally in both boreholes.

Results are presented in Figure 4 in a form of a correction factor, F_{corr}, defined as the ratio between the length determined by dynamic simulations and the length calculated using the ASHRAE equation.

$$F_{corr} = \frac{L_{dynamic\ simulation}}{L_{ASHRAE}} \tag{2}$$

As shown on Figure 4, the correction factor varies from 0.69 to 1.24 for the four combinations studied here. In all cases, the value of F_{corr} decreases as the effect coverage increases. When the heat pump is 25-50 % oversized (i.e. effect coverage of 125-150 %), the ASHRAE equation overestimates the length. This is because the heat pump operates intermittently, even during the peak load period. Consequently, the hourly ground load, q_h, does not last 6 hours as is typically the case in Eq. 1 and the thermal capacity effects are more pronounced. For an effect coverage of 50 to 100 %, the heat pump is operating continuously for 5 to 100 hours during peak load. This is illustrated in Figure 5 which shows the house load during the last month of the simulation for the 50 % effect coverage case. Considering that the 3 ton heat pump has an 8 kW (27.3 kBTU/hr) capacity at peak conditions, it is clear that the heat pump will work continuously for more than 6 hours. For example, the house load is above the heat pump capacity for more than 100 hours around t = 88000 h. For those cases, the duration of the hourly ground load in the ASHRAE equation is too short, which underestimates the required length. The equation also neglects thermal capacity effects, which overestimate required length. These effects influence the correction factor in opposite directions.

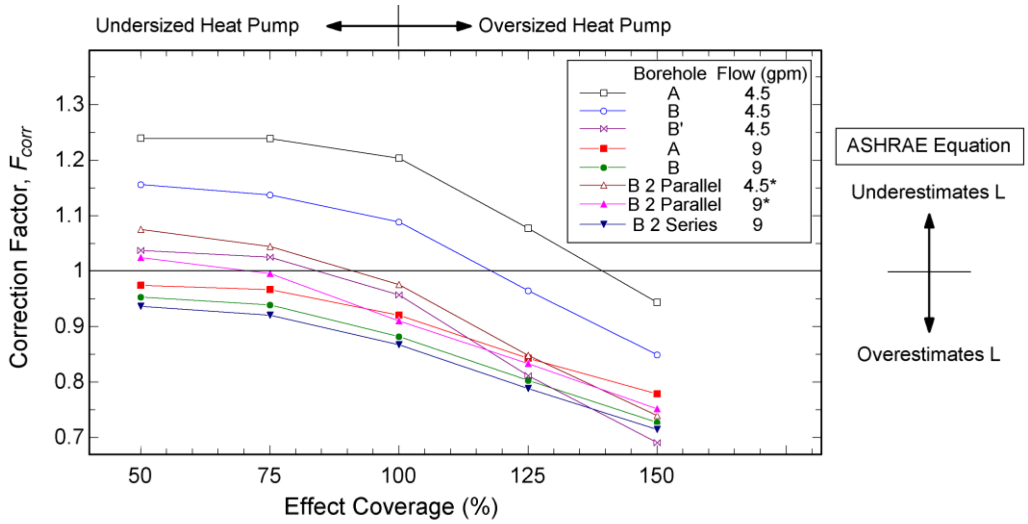

Figure 4 Correction factor to modify the ASHRAE sizing equation for boreholes A (4 in. diameter, high conductivities), B (6 in. diameter, low conductivities) and B' (1.5 in pipe). Note: 4.5 and 9 gpm stand for 0.28 and 0.56 L/s. For the two boreholes in parallel, * stands for the total flow in both boreholes.

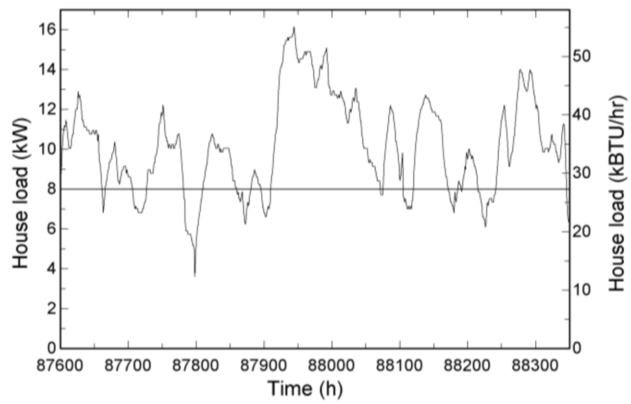

Figure 5 House peak load for the 50 % effect coverage case during the last month after 10 years of simulation. The horizontal line is the heat pump capacity.

The flowrate has also a significant impact on F_{corr}, which decreases as the flowrate increases. A larger flowrate (more gpm/ton) leads to a higher average fluid temperature in the borehole and a higher temperature difference between the fluid and the ground, which, according to Eq. 1, leads to smaller lengths. For the parallel configurations, two scenarios are observed. The 9 gpm (0.56 L/s) parallel case leads to higher correction factors because only 4.5 gpm (0.28 L/s) are flowing in each borehole. The 4.5 gpm case gives 2.25 gpm (0.14 L/s) in each borehole, leading to a laminar flow. A laminar flow reduces heat transfer and leads to thermal capacity effects lasting longer, reducing F_{corr} when compared to the one borehole 4.5 gpm case. Figure 4 then shows that placing two B boreholes in series has little impact on thermal capacity effects as F_{corr} is slightly smaller.

Finally, the borehole characteristics affect F_{corr} by introducing more or less thermal capacity effects. Borehole B has lower correction factors than borehole A because the effects relative to the borehole thermal capacity last longer. Those effects improve the borehole performances and have a tendency to reduce borehole length. Borehole B' also has lower correction factors then borehole B. This is because its larger pipes store more fluid allowing more energy storage and longer thermal capacity effects. The lowest value of F_{corr} (0.69) is observed for B' and an effect coverage of 150%.

In summary, these results show that when sizing a residential borehole using the ASHRAE sizing equation, designers should be aware that borehole thermal capacity and heat pump oversizing affect the required borehole length. Heat pump oversizing leads to intermittent heat pump operation and to lower required length because of thermal capacity effects. At the other end of the spectrum, heat pump undersizing leads to continuous heat pump operation which increases the duration of the peak load and increases the required length.

CONCLUSION

Borehole thermal capacity and heat pump cycling, which are not included in the ASHRAE borehole sizing equation, can change the required borehole length. Borehole transient effects, caused by the fluid and grout thermal capacities, are beneficial as they reduce the peak ground loads and, consequently, the required borehole length. Correction factors for the current ASHRAE sizing equation are proposed to consider borehole thermal capacity. Annual simulation results on a typical residential system show that neglecting borehole transient effects leads to oversized boreholes. Furthermore, heat pump energy consumption is overestimated by about 5 %. The correction factor can reach 0.69, a length reduction of 31 % compared to ASHRAE's equation. The largest reductions in the required length occur when heat pumps operate intermittently with an oversized heat pump. The ASHRAE equation can also underestimate the length, by up to 24 % in the present case (correction factor of 1.24), when an undersized heat pump and low thermal capacity boreholes are used. The results of this study indicate that dynamic simulations are required to properly size the borehole to account for borehole thermal capacity and heat pump cycling.

ACKNOWLEDGMENTS

The authors would like to express their sincere gratitude to ASHRAE for the Grant-In-Aid awarded to the first author. Also, Hydro-Quebec and the Natural Sciences and Engineering Research Council of Canada provided scholarships to the first author. Their support is gratefully acknowledged. Finally, this work was performed with funds provided by NSERC's Smart Net-Zero Energy Buildings Strategic Research Network.

REFERENCES

ASHRAE. (2015). Geothermal energy, *ASHRAE Handbook, HVAC applications*. Atlanta, GA: ASHRAE.

Godefroy, V., and M. Bernier. 2014. A simple model to account for thermal capacity in boreholes. *Proceedings of the 11th IEA Heat Pump Conference*, Montreal, Qc, Canada. *Paper #P.4.8*.

Godefroy, V., Lecomte, C., Bernier, M., Douglas, M., and M. Armstrong. 2016. Experimental Validation of a Thermal Resistance and Capacity Model for Geothermal Boreholes. *ASHRAE Winter Meeting*, Orlando, Fl. OR-16-C047.

Hellström, G. 1991. *Ground Heat Storage - Thermal Analyses of Duct Storage Systems - I. Theory*. Department of Mathematical Physics, University of Lund, Sweden.

Hellström, G., Mazzarella, L., and D. Pahud. 1996. *Duct ground storage model–TRNSYS version*. Department of Mathematical Physics, University of Lund, Sweden.

Kavanaugh, S., and K. Rafferty. 1997. *Ground-Source Heat Pumps: Design of geothermal systems for commercial and institutional buildings*. Atlanta, GA: ASHRAE

Ma, W., Li, M., Li, P., and A. Lai. 2015. New quasi-3D model for heat transfer in U-shaped ground heat exchangers: Effective overall thermal resistance. *Elsevier Energy*. doi: 10.1016/j.energy.2015.07.098

Pärisch, P., Mercker, O., Oberdorfer, P., Bertram, E., Tepe, R., and G. Rockendorf. 2015. Short-term experiments with borehole heat exchangers and model validation in TRNSYS. *Renewable Energy, 74*, 471-477.

Klein, S. A. 2010. *TRNSYS, a transient simulation program.* Solar Energy Laboratory, University of Wisconsin-Madison, USA.

Shirazi, A. S., and M. Bernier. 2013. Thermal capacity effects in borehole ground heat exchangers. *Energy and Buildings, 67* 352-364.

Yavuzturk, C., and J.D. Spitler. 1999. A Short Time Step Response Factor Model for Vertical Ground Loop Heat Exchangers. *ASHRAE Transactions*. 105(2): 475-485.

Young, T. 2004. *Development, Verification, and Design Analysis of the Borehole Fluid Thermal Mass model for Approximating Short Term Borehole Thermal Response.* (Masters of Science), Oklahoma State University.

Xu, X., and J.D. Spitler. 2006. Modeling of Vertical Ground Loop Heat Exchangers with Variable Convective Resistance and Thermal Mass of the Fluid. *Proceedings of the 10th International Conference on Thermal Energy Storage*. Ecostock 2006, NJ.