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Design of Residential Ground Source Heat Pump Systems for Heating Dominated Climates - Trade-Offs Between Ground Heat Exchanger Design and Supplementary Electric Resistance Heating

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ABSTRACT

Residential ground source heat pump systems in heating-dominated climates often incorporate auxiliary electric resistance heating. This is often utilized for some portion of the winter heating – it could be intended only for rare, emergency use, or it could be intended to provide a significant fraction of the annual winter heating. In Sweden, where a typical ground source heat pump provides both heating via radiant panels and domestic hot water, the auxiliary electric resistance heating may be activated either when there are high domestic hot water loads or when ground heat exchanger temperatures fall to low values. In cases where the electric resistance heating provides a significant fraction of the annual winter heating, several advantages are gained: a smaller and less expensive heat pump is required; a smaller and less expensive ground heat exchanger is required. But this comes at the expense of a lower system seasonal coefficient of performance and higher operation costs due to increased electricity usage.

This paper evaluates the tradeoffs between higher reliance on auxiliary electric resistance heating (with corresponding lower investment cost) and lower reliance on auxiliary electric resistance heating (with corresponding lower operating cost) for a typical Swedish house - a renovated 1940s-era house in Stockholm. The hourly building heating loads and water heating loads are estimated with a building simulation program. A range of system designs are developed with different heat pump capacities and different ground heat exchanger sizes. The different systems are simulated to compare operation and actual electricity consumption. The simulation evaluates the performance of the ground heat exchanger, heat pump, and auxiliary electric resistance heating for a ten-year period, the 5th year taken as typical. First costs and operating costs are determined based on current costs in Sweden for ground source heat pump systems and electricity. The optimal design is determined and a range of near-optimal designs are considered.

INTRODUCTION

About a third of all single-family houses in Sweden are heated by ground source heat pumps. The typical ground source heat pump is a 5-10 kW (17,060-34,120 Btu/hr) capacity heat pump connected to a 100-200 m (328-656 ft.) deep vertical groundwater filled borehole in hard rock. The heat pump is integrated with a hydronic panel radiator system for space heating, and also provides domestic hot water heating. In the 1980's when ground source heat pumps became popular in Sweden, a strategy of sizing the heat pump for 55-60 % capacity coverage, which will cover 90 % of the energy demand, was introduced, and is still used today (Karlsson et al. 2003). The remaining capacity demand is met by an electric resistance immersion heater. This strategy is said to be motivated by the last 10 % capacity coverage requiring twice the size of the heat pump, which would be very costly. However, the result is a lower system seasonal coefficient of performance and higher operation costs due to increased electricity usage. From a life cycle cost perspective, a larger heat pump and a

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deeper borehole may still be preferable.

This paper describes a parametric study of residential ground source heat pump system design for a house in Stockholm, Sweden. Specifically, we investigate the effects of varying borehole depth and heat pump size on system performance and life cycle cost.

METHODOLOGY

The study described here involved determining heating loads (house and domestic hot water) for a renovated 1940s era house in middle Sweden; these loads were then used as inputs to a model of a Swedish heat pump coupled to a ground heat exchanger model. This model was then used to simulate 121 different combinations of heat pump size and borehole depth and an economic analysis was performed to determine first costs, annual electricity costs and life cycle costs normalized on a per kWh basis. Each aspect is described below, with special emphasis given to the description of the heat pump and heat pump model, because it differs in several respects from those commonly used in North America.

House Description and Calculation of House Heating Loads

The building used in this study is a typical Swedish single family house – a renovated 1940s-era house in Stockholm. It is a 125 sqm (1345 sq.ft.) building with hydronic radiator panel heating. The building data is taken from the TABULA database (www.building-typology.eu). The hourly building heating loads and water heating loads (**Fig. 1**) are estimated with the building simulation program EnergyPlus. Typical daily domestic hot water data is taken from the Swedish Energy Agency (2009).

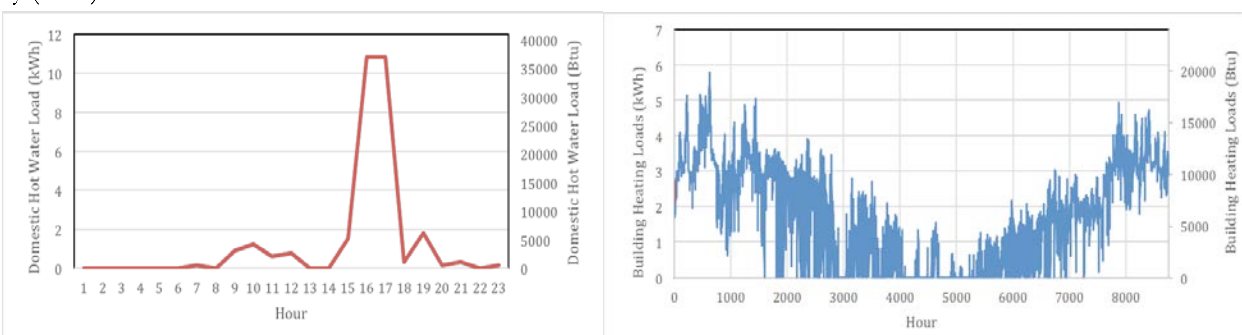

Figure 1 a) Daily Domestic Hot Water Load b) Annual Building Heating Loads

Heat Pump Model

Residential ground source heat pump units utilized in Scandinavia commonly have several features that differentiate them from those most commonly used in North America:

- Not reversible – used for heating only.
- Fluid-to-water – heat on the load side of the heat pump is distributed to the building by a hydronic system with either radiators or under-floor heating.
- Integrated domestic hot water heating and storage – a double-walled hot water storage tank within the heat pump unit stores around 160-200 liters (42-53 gallons) of hot water. The water within the tank is heated with hot water coming from the heat pump. A three-way valve directs the hot water heated by the heat pump through either the outer shell of the hot water storage tank or to the radiators. Typically, no desuperheater is used.
- When heating capacity beyond what the heat pump can provide is needed, an electric-resistance immersion water heater is used to raise the temperature of hot water going to the outer shell of the hot water storage tank or to the radiators.

- The internal controls are partly proprietary, but they have a control curve that controls the setpoint for the hot water going to the radiators based on outdoor air temperature. The temperature in the domestic hot water tank is controlled to a user-specified setpoint. The control curve is initially set according to manufacturer's recommendations but may be changed by the end-user. The control curve has the effect of lowering the setpoint when the outdoor air temperature is higher. As the setpoint is lowered, the heat pump COP is increased. Domestic hot water heating is typically given priority. Switching between demands with different temperature requirements is referred to as "floating condensation."

The heat pump model used here was developed for this investigation. The heat pump capacity and COP are fit with two polynomial functions of source-side entering fluid temperature and load-side exiting fluid temperature. It uses two key approximations:

- It is a quasi-steady-state model and thus assumes that the domestic water heating loads that occur in any hour are met that hour.
- The capacity and COP equation fits are based on a typical unit with a nominal 7 kW (23,880 Btu/hr) capacity. For other capacity heat pumps, the same equations are used, but the capacity at any combination of temperatures is multiplied by the ratio of the desired nominal capacity to that of the base unit.

The model first determines the capacity of the heat pump to meet the domestic water heating load; if it can be met with the heat pump, excess capacity can be used for heating the house. If the capacity is not sufficient for heating the house or heating the domestic hot water the electric-resistance immersion water heater will be used. For purposes of this study, we have specified the immersion water heater to be large enough (15 kW (51,180 Btu/hr.)) to ensure that all loads are met.

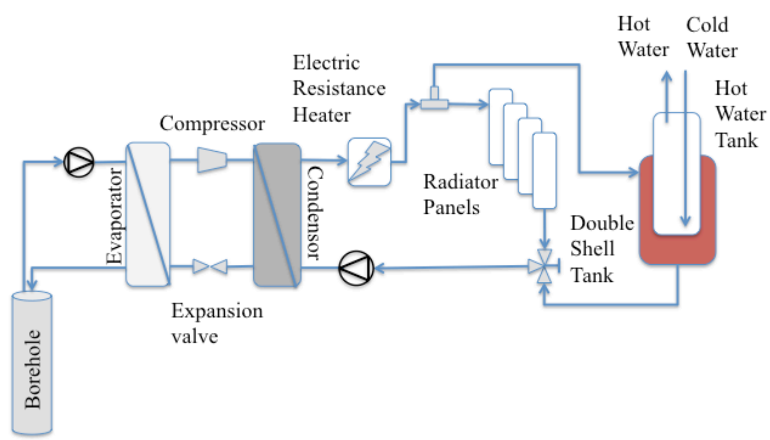

Figure 2 Typical Swedish Ground Source Heat Pump with Integrated Hot Water Storage Tank and Electric Resistance Heater.

Ground Heat Exchanger Model

Ground heat exchangers used in Scandinavia are commonly closed-loop, using a single U-tube but no grout. Rather, the hard-rock formation and high groundwater level allows the borehole to be groundwater-filled; the U-tube is suspended in the borehole. The uppermost 6 m (20 ft.) or more of the borehole is cased with a steel casing, sealed to the bedrock, to protect the groundwater from surface pollution (SGU, 2008).

The ground heat exchanger model used was originally described by Yavuzturk and Spitler (1999). Several validations have been described in the literature, including Gentry, et al. (2006). The model uses a response factor methodology; the response factors are determined from what is often referred to in the literature as the g-function. The g-functions for this single borehole case are determined with the method described by Claesson and Javed (2011). The thermal resistance of the groundwater filled borehole is taken as 0.07 K/(W/m) (0.121 °F/(Btu/hr·ft)) as a typical value (Gustafsson and

Gehlin, 2008) measured in groundwater-filled boreholes.

Description of Test Cases

For the analysis reported here, we analyzed 121 test cases with nominal heat pump capacities of 5 -15 kW heating (17,060-51,180 Btu/hr) in increments of 1 kW (3412 Btu/hr) and borehole depths of 100-300 m (328-984 ft.) in 20 m (65.3 ft.) increments. The eleven values of nominal heat pump capacity and eleven values of borehole depth combined to create 121 test cases. For each test case, the system was simulated for a 5-year period and results from the 5th year were used to determine typical operating costs. For some test cases a 10-year period was simulated, and this showed that after the 5th year of operation, the ground heat exchanger conditions had stabilized and become typical.

Economic Analysis

The electricity rates are calculated as an average monthly total electricity price based on data from a single-family house in the Stockholm area over a period of three years (2010-2012). Electricity rates vary over the year between 0.99 SEK/kWh (0.14 USD/kWh) and 1.32 SEK/kWh (0.19 USD/kWh) with an average of 1.12 SEK/kWh (0.16 USD/kWh), the lowest rate in July and the highest in February. (SEK is the Swedish currency unit, the crown. We have fixed the exchange rate at 7 SEK = 1 USD).

Heat pump first costs are calculated as a linear relation to the heat pump size, and are based on list prices from a Swedish heat pump manufacturer. The equation is given by (**Eq. 1**), where C_{HP} is the heat pump cost and Z_{HP} is the heat pump size (nominal heating capacity) in kW, and c_c is a currency factor that is 1 for SEK and 1/7 for USD:

$$C_{HP} = c_c \cdot (1730.1 \cdot Z_{HP} + 58186) \tag{1}$$

Ground heat exchanger cost is set as 230 SEK/m (10 USD/ft.) borehole, and includes drilling and collector, as verified by the Swedish Heat Pump Association as a good estimate of the average cost in Sweden (Forsén, 2013). The borehole is groundwater-filled; hence there is no cost for grouting. Installation cost for the heat pump is not included in the model as that cost is not dependent on heat pump size or borehole length. Cost for installation of radiators is not included as that is not specific for the ground source heat pump system. The total cost per kWh for the ground source heat pump system, c_{total}, is given by (**Eq. 2**), where q_i is the total hourly heating load including household heating and domestic hot water heating, $c_{el,i}$ is the electricity cost per kWh, e_i is the hourly electrical energy used by the system, C_{syst} is the first cost of the system, a is the annuity factor, set to 5.7%, and s is the maintenance cost factor, set to 1%:

$$c_{total} = \frac{c_c}{\sum_{i=1}^{8760} q_i} \left(\sum_{i=1}^{8760} c_{el,i} \cdot e_i + C_{syst}(a+s) \right) \tag{2}$$

This formulation of the life cycle cost normalized on a per kWh of heating provided is adapted from Nowacki (2013), though we used a more detailed analysis to obtain operating costs. In addition, as future values of electricity cost and interest rates, cannot be known with certainty, we have also adapted a simple payback analysis based on current investment costs and current electricity costs.

Figures of Merit

The trade-offs between borehole depth, heat pump capacity, and electricity consumption might be looked at from a number of different perspectives. From an economic perspective, first cost, payback time, and normalized costs per unit of heating provided are all of interest. Seasonal coefficient of performance (SCOP) (also called seasonal performance factor (SPF)) is another figure of merit that is useful when comparing system performance. As long as electrical costs are based on a simple constant rate structure or a relatively constant monthly rate structure, these figures of merit should suffice.

However, from an electrical utility perspective, the time at which the electricity is consumed is also important. If, the electrical utility's peak loads occur at low outdoor air temperatures, then having thousands of customers with ground source heat pumps that require electric resistance heating under these conditions can exacerbate peak demand problems and require bringing additional electrical generating capacity online and/or the purchase of electricity from the grid at high prices. Furthermore, this change in electricity sources can have implications for CO_2 and particulate emissions as the sources of electricity production change. In Sweden, where the electrical generation base within the country has relatively low CO_2 emissions due to significant hydropower, peak load conditions may require purchase of electricity from coal-fired power plants with high CO_2 and particulate emissions.

In this paper, we do not attempt to analyze the emissions, but introduce a figure of merit that partly addresses this– the fraction of the heating provided by the heat pump. The higher the fraction, the less use of electric resistance heating by the system. For utilities that have peak loads driven by house heating, peak loads will occur at times of low outdoor air temperature. For individual homes with ground source heat pumps, heating loads will be highest and heat pump capacity will be lowest at the same times, requiring the most electrical resistance heating. So, the fraction of heating provided by the heat pump will be related to the CO_2 and particulate emissions in situations where peak heating loads translate into peak electrical demands that must be met by electricity generation sources with higher emissions.

RESULTS

As discussed in the Figure of Merit section above, there can be multiple perspectives when judging the system performance and we attempt to present the results in a way that allows some comparison of the results from different perspectives. **Figure 3** shows a comparison of all 121 designs on the basis of normalized cost (annualized life cycle cost on a per unit of heating basis) and the fraction of the heating load that is met by the heat pump.

Each dot represents one combination of borehole length and nominal heat pump capacity. We have a "win-win" situation in that the designs with the lowest normalized costs have the highest fraction met by the heat pump and so, depending on the utility scenario, are likely to have the lowest emissions. These designs are in the upper left hand portion of the data cloud on the plot, in the box. The "zoomed-in-view" shows that there are two solutions very near to each other – one (the farthest left) with the lowest normalized cost has the 15 kW (51,180 Btu/hr) heat pump combined with 280 m (919 ft.) borehole and another (the uppermost) with the highest fraction met by the heat pump; that dot represents the 15 kW (51,180 Btu/hr.) heat pump combined with the 300 m (984 ft.) borehole.

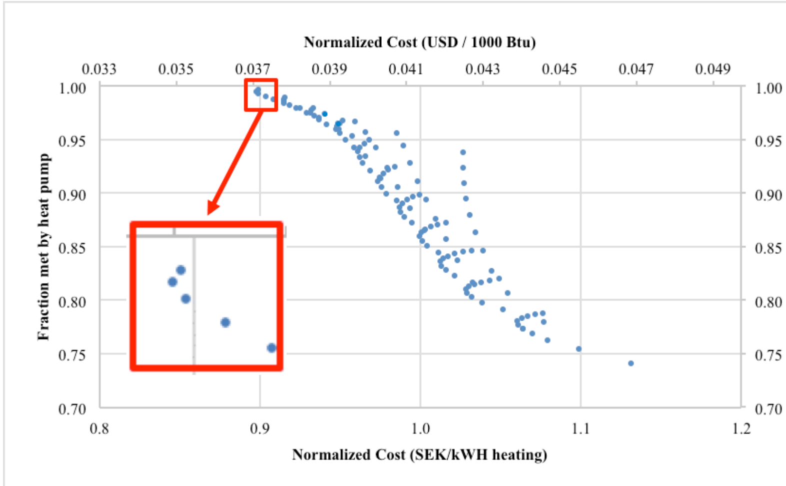

Figure 3 Fraction of heating load met by heat pump vs. normalized life cycle cost

Several observations may be made from this. First, these solutions with the lowest normalized life cycle cost are far from the typical Swedish design, which would more likely use a 5 kW (17,060 Btu/hr.) heat pump and a 100 m (328 ft.) deep borehole. The typical Swedish design is represented by the lowest dot, farthest to the right, though other nearby solutions may also be used. I.e. it has the lowest fraction of the heating provided by the heat pump and the highest normalized cost. Second, the fact that the 300 m (984 ft.) deep borehole has slightly higher normalized cost suggests that, from an economic perspective, the minimum cost has been reached at a lower depth and so increasing the borehole depth beyond 300 m (984 ft.) will not further improve the economics. However, this observation is made with the maximum heat pump capacity chosen for the study – a 15 kW (51,180 Btu/hr.) heat pump, and it is possible that lower costs might be reached with a larger heat pump.

These results raise an obvious question – why is the typical Swedish design so far from the apparent optimum? While the assumptions about future interest rates and electricity costs used to perform the economic analysis may not be shared by all homeowners, it seems more likely that many homeowners may simply not be prepared to invest in the "best" system. The strategy of sizing the heat pump for 55-60% capacity coverage, taken up in the 1980's, was based on economic considerations valid at that time, when electricity prices were low, and CO_2 emissions were not considered a problem. Since then electricity prices have more than doubled, but the contractors' rules of thumb have remained. The authors know of no recent study to show the effects of this sizing strategy.

Figure 4 illustrates the situation by comparing normalized life cycle cost to first cost for all 121 cases. This presentation of the data may be construed as a simple multiple criterion optimization using exhaustive search. The curve that has been added represents the Pareto front (Tye-Gingras and Gosselin 2012)– any point on the Pareto front has the characteristic that neither objective (minimizing normalized cost or minimizing first cost) can be improved without making the other objective worse. Starting at the right hand side of the Pareto front, the first eight points (from right to left) all have 15 kW nominal heat pump capacity and decreasing borehole depths. Beyond the eighth point, the nominal heat pump capacity also declines. The Pareto front may also be used to quickly understand the effect of a constrained first cost. In the likely event that there is a maximum first cost that the homeowner can accept, the Pareto front represents the best design at that cost.

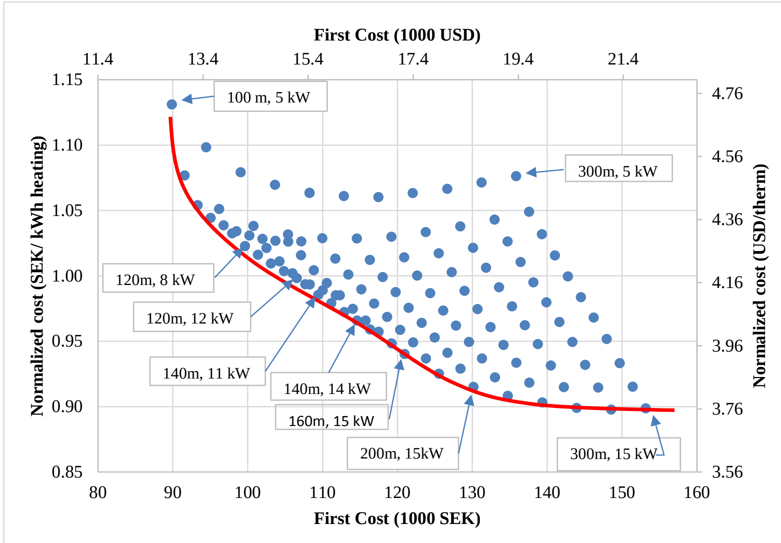

Figure 4 Normalized Life Cycle Cost vs. First Cost with Pareto Front Marked

Figure 5 shows the combinations of borehole depth and heat pump capacity that make up the Pareto front. It shows that as the borehole depth increases, the required heat pump capacity needed to minimize the life cycle cost increases. The occurrence of different heat pump capacities at a given borehole depth or different depths at a given capacity reflect the best (lowest life cycle cost) designs that are possible at different first costs.

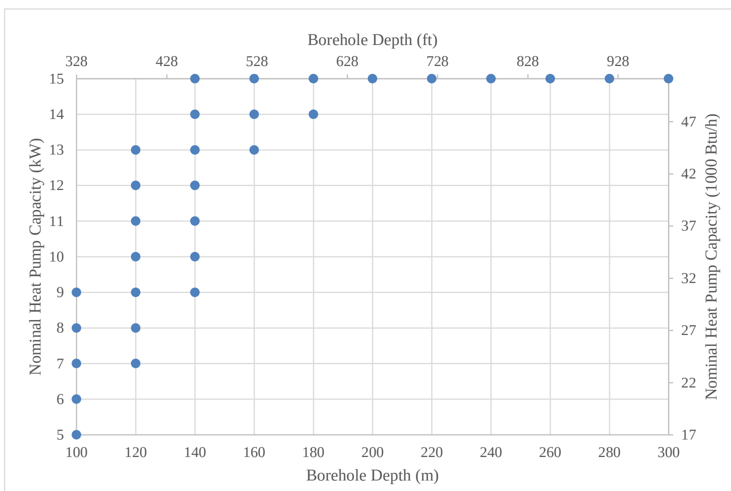

Figure 5 Heat Pump Capacity – Borehole Depth Combinations Corresponding to Pareto Front

To put things in perspective, comparing a typical Swedish design (5 kW (17,060 Btu/hr.) heat pump, 100 m (328 ft.) deep borehole) with the most costly system (15 kW (51,180 Btu/hr.) heat pump, 300 m (984 ft.) deep borehole), the most costly system has an additional investment cost of about 63,000 SEK (9,000 USD). However it saves about 9,900 SEK (1410 USD) per year in electricity cost, giving a simple payback on the additional investment of about 6.4 years. The more costly system also increases the fraction of the heating load met by the heat pump from 74.1% to 99.5%, and hence, will reduce emissions. There are numerous scenarios in between – taking just one more design on the Pareto front as an example, the 160 m (525 ft.) borehole/ 15 kW (51,180 Btu/hr.) heat pump has a 35% higher investment cost than the typical Swedish design, but saves 29% on annual electricity costs and has a simple payback of 4.5 years for the additional investment. At the same time, it increases the fraction of the load met by the heat pump to 97.4%.

CONCLUSIONS AND RECOMMENDATIONS

This paper has presented an investigation of a range of designs for a ground source heat pump system serving the house heating and domestic water heating demands of a renovated typical 1940s era house in Sweden. The typical Swedish design represents a lowest first cost ground source heat pump system design, but there is substantial room for improvement. From a life cycle cost viewpoint, the best system design is the most expensive first cost system investigated with three times the borehole depth and three times the heat pump capacity of the typical Swedish design. The question of why a more expensive, but more cost effective, ground source heat pump system design is not selected cannot be fully answered by this paper. However, to the extent that the initial investment required is too high, this study shows that there is a range of intermediate solutions that perform better than the typical design.

This study has been limited to a single house in a single location in Sweden and only two parameters – borehole length and heat pump capacity have been varied. Furthermore, the heat pump model used, being quasi-steady in nature cannot quantify the effects of thermal storage in the tank, which may allow some of the peak domestic water heating loads to be shifted. Therefore, we recommend that an improved model of the heat pump that includes dynamic behavior of the hot water storage tank be developed. Such a model could also incorporate some of the control modes that are available, such as switching the electric resistance heater off. With such an improved model of the heat pump, a more expansive study covering more house types and climates would be of use in further informing design decisions for residential ground source heat pump systems in Sweden.

REFERENCES

Claesson, J. and S. Javed. 2011. An Analytical Method to Calculate Borehole Fluid Temperatures for Time Scales from Minutes to Decades. ASHRAE Transactions 117(2):279-288.

Forsén, M. 2013. Personal communication, Swedish Heat Pump Association.Gentry, J.E., J.D. Spitler, D.E. Fisher, X. Xu. 2006. Simulation of Hybrid Ground Source Heat Pump Systems and Experimental Validation. Proceedings of The 7th International Conference on System Simulation in Buildings, Liège, Belgium. December 11-13, 2006

Gustafsson A.-M., S. Gehlin. (2008). Influence of natural convection in water-filled boreholes for GCHP. ASHRAE Transactions. 114(1):416-423.

Karlsson, F., M. Axell, P. Fahlén. 2003. Heat Pump Systems in Sweden – Country Report for IEA HPP Annex 28. SP AR 2003:01. Borås, Sweden.

Nowacki, J-E. 2013. Chapter 3. Värmepumpens storlek (och lite mer ekonomi). In: Björk, E., et al. 2013. Bergvärme på Djupet – Boken för dig som vill veta mer om bergvärmepumpar. Department of Energy Technology, KTH. Stockholm. In Swedish.

SGU, 2007. Normbrunn-07. Att borra brunn för energy och vatten – en vägledning. Normförfarande vid utförande av vatten och energibrunnar. (Guideline for design of wells for energy and water). The Swedish Geological Survey, Uppsala, Sweden. In Swedish.

Swedish Energy Agency. 2009. Mätning av kall- och varmvattenanvändning i 44 hushåll. (Measurement of cold and hot water usage in 44 households). ER 2009:26. In Swedish.

Tye-Gingras, M., L. Gosselin. 2012. Comfort and energy consumption of hydronic heating radiant ceilings and walls based on CFD analysis, Building and Environment, 54:1-13.

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