ISRN LUTMDN/TMHP—09/5192—SE
ISSN 0282-1990

Analysis of Geo-Energy System with Focus on Borehole Thermal Energy Storage

Henrik Holmberg

Thesis for the Degree of Master of Science

Division of Thermal Power Engineering
Department of Energy Sciences
Faculty of Engineering
Lund University
P.O. Box 118
SE-221 00 Lund
Sweden

Preface

This thesis is the final project to fulfill the Master of Science in Mechanical Engineering education at the faculty of engineering at Lund University.

The project has been carried out in cooperation with Malmberg Water AB. Malmberg Water is a contractor within geo-energy systems, biogas and water treatment. With 30 years of experience in building geo-energy systems the company is one of the main actors in the geo-energy sector in Sweden.

Firstly I would like to thank my supervisors at Malmberg Water AB, Peter Bäckström and Ola Svärd for showing great interest in my project and introducing me to the geo-energy sector. I would also like to thank my supervisor Magnus Genrup and my examiner Bengt Sundén at LTH.

A special thank should also be directed to Saqib Javed from Chalmers University of Technology, who joined the project as an assistant supervisor and has provided me with substantial support and many good ideas along the way.

Abstract

Ground source heat pump (GSHP) systems have become more popular lately both for residential space heating and for commercial applications. GSHP systems can use water, ground or the bedrock as a thermal source. The energy extracted is regarded as a renewable energy-source and it consists of both stored solar energy and a share of geothermal energy. A borehole thermal storage (BTES) consists of boreholes drilled in a specific pattern with a few meters in-between. A BTES can be used for both heating during winter and cooling during summer. The stable temperature of the bedrock provides good performance as long as the balance between extracted and injected heat is kept. With increasing size of GSHP-systems it is desirable to simulate the systems and to perform optimization and life cycle analysis to justify investments. There are several existing software that can simulate a BTES but they are either working with too large time steps or they are too complicated for industrial usage. Within this thesis a short time-step model of a BTES is constructed based on existing research. The model is validated using existing simulation software and then demonstrated within a simplified GSHP-system.

Table of Contents

1 Introduction

1.1 Background

Geothermal energy is a well-established concept referring to the heat production in the interior of the earth due to the breakdown of radioactive isotopes. Natural geothermal resources are geographically limited to geological active areas and often thought of as exotic. In areas where the thermal gradient is exceptionally high, steam can be extracted to use for electricity production. The waste heat can then also be supplied to a district-heating grid.

Most countries do not possess geothermal areas and active geysers. Sweden is one of these countries and has a thermal gradient of only 1.5-3 ^{o}C/100 m depth. Despite this Sweden has a significantly higher use of geothermal energy than the pioneers like Iceland. It might be argued that Iceland has a much lower population in relation to Sweden and therefore correspondingly lower energy consumption. However Sweden's use of geothermal energy does not only exceed Iceland's but also rest of Europe. Only China and USA has a higher total use of geothermal energy.

The reason for this is the abundance of low temperature energy sources in combination with the climate in Sweden. What might be experienced as significant seasonal temperature changes are quite insignificant in a thermodynamically perspective. With the use of a heat pump the energy from these low temperature sources can easily be upgraded to provide useful heating of buildings.

The use of heat pumps has increased rapidly in Sweden and with an estimated 350 000 installed heat pump systems Sweden stands second after USA. In 2006 heat pump units could be found in nearly 30 percent of the residential houses and accounted for the heating in 10 percent of the apartment buildings. Heat pumps typically deliver three times the supplied energy, thus the energy consumption in the residential sector has decreased with increased number of heat pumps (Swedish Energy Agency).

In residential houses the air-source heat pump has become popular as it is a simple and cheap alternative to electric heating. Another alternative is a heat pump system that uses ground, groundwater or surface water as a heat source. These systems are categorised under the common name Ground Source Heat Pumps (GSHP) (Kavanaugh and Rafferty 1997) and are recognized as being amongst the most energy efficient and cost effective heating and cooling systems for residential and commercial buildings (Lamarche and Beauchamp 2006). The energy these heat pumps absorb is called geo-energy and is considered as a renewable energy according to the Swedish Energy Agency (2008).

Lately GSHP systems have grown both in size and in popularity. From being small residential heating systems consisting of example one borehole and a heat pump, they now often utilize tens or hundreds of boreholes together with heat pump systems for both heating and cooling of commercial buildings. A borehole thermal storage (BTES) consists of boreholes drilled in a specific pattern with a few meters in-between. The stable temperature of the bedrock provides good performance as long as the balance between extracted and injected heat is kept. With increasing size of GSHP systems it is desirable to simulate the system to perform optimization and life cycle analysis to justify investments. There are several existing software that can simulate a BTES but they are either working with to large time steps or they are too complicated for industrial usage.

1.2 Company presentation

Malmberg Water AB is a contractor within Geo-energy systems, biogas and water treatment. Malmberg Water AB was one of the first Swedish companies to apply the energy storage method to generate heating and cooling. With 30 years of experience from building profitable and energy efficient geo-energy systems the company is now one of the main actors in the geo-energy sector in Sweden.

1.3 Aim of the thesis

The first aim of this thesis is to develop a model of BTES that can be used in simulation of GSHP-system with an hourly time step to perform optimization and life cycle analysis of the system. The second aim is to demonstrate how this model can be used in interaction with other components of a GSHP system.

1.4 Delimitations

Although the thesis describes different GSHP systems the BTES is the main focus. It is also mainly related to the climatic conditions in Sweden and written from a Swedish perspective. Because of the time limitations it has not been possible to conduct any measurements on existing BTES.

2 Ground source heat pump systems (GSHP)

2.1 Overview of GSHP systems

The different subcategories of GSHP systems will be briefly presented with emphasize on the vertical ground coupled heat pump system. GSHP systems can be divided into closed and open systems referring to if they use a closed loop with a heat carrier fluid, or if they circulate the water from an existing water resource. Closed systems can use antifreeze solutions as carrier fluid and are therefore more common in colder climates. The closed loop also has the advantage of keeping the circulating fluid clean from contaminations. In an open system the water quality is crucial and the primary heat exchangers can be affected by corrosion, fouling and blockage (Kavanaugh and Rafferty 1997).

2.1.1 Horizontal ground coupled heat pump system

The collector is buried in trenches in the uppermost soil on a depth between 0.9 and 1.5 meter (SVEP 2004). A carrier fluid is circulated in a closed loop. In colder climates an antifreeze solution is used. The main thermal recharge is provided by solar radiation. Therefore it is important not to cover the used land. This type of system cannot be used for cooling during summer since that would severely increase the ground temperature. A large land area is required, which often is a limitation in densely populated areas, see figure 1. An advantage is that the trenching cost is lower than the cost for drilling a well. It is also flexible in the sense of installing pattern. Apart from the large land use, another disadvantage is degradation of system performance with seasonal temperature changes in the ground. The temperature in the ground can also be affected by rainfall (Omer 2006).

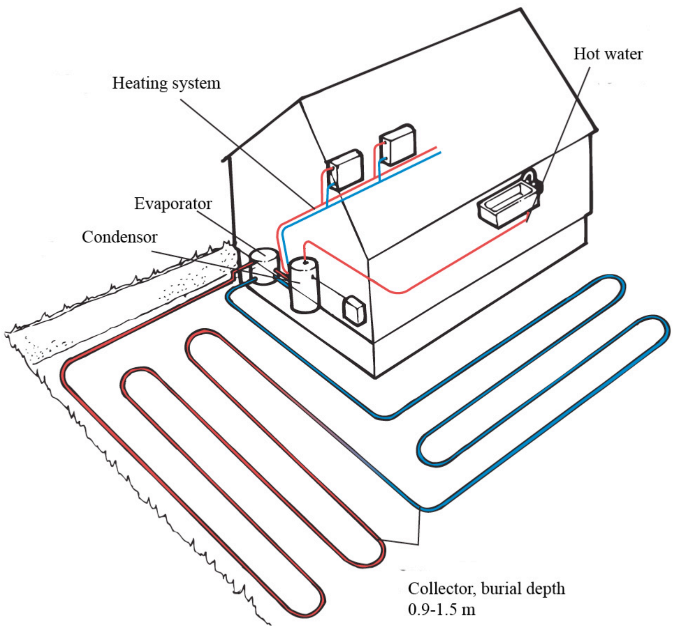


Figure 1. Horizontal ground coupled heat pump system (SVEP 2004).

2.1.2 Surface water heat pump system (SWHP)

The principle is the same as for the horizontal GSHP. Instead of being buried in the ground, the collector is submerged in a lake or a river and anchored to the bottom, see figure 2. The advantages are high reliability, low maintenance costs and low operating costs. Disadvantages are the possibility of carrier fluid leaking into the water resource from the collector and performance degradation by seasonal temperature changes in shallow lakes (Kavanaugh and Rafferty 1997).

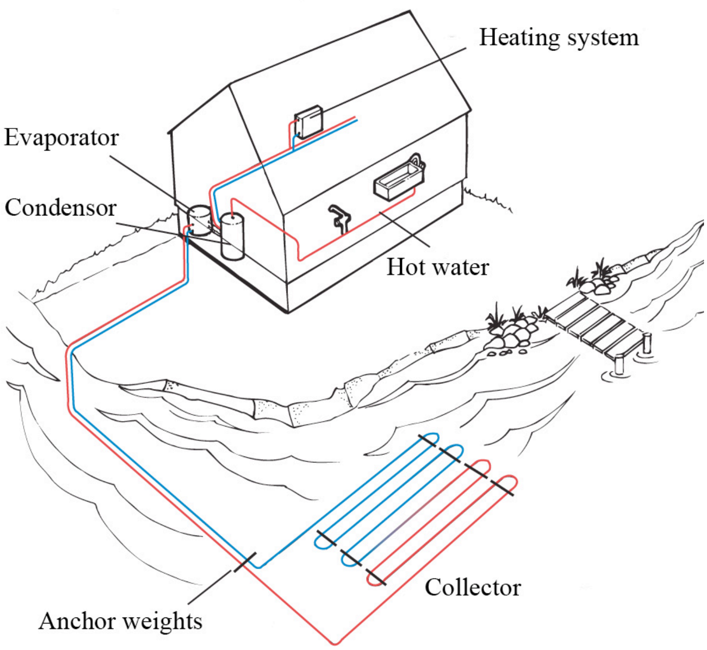


Figure 2. Surface water heat pump system (SVEP 2004).

2.1.3 Ground water heat pump system (GWHP)

The ground water heat pump system is simple and requires low drilling costs and a small area (Omer 2006). It is according to GEOTEC (2009) the most effective way of using geo-energy. These systems operate with an open loop and use either the groundwater or the surface water. Depending on the water quality the primary heat exchanger can be subject to corrosion, fouling and blockage. The availability of water is important since a significant amount might have to be circulated to meet heating and cooling loads. When water resources admit these system can reach a high capacity and are often used in large-scale systems (Kavanaugh and Rafferty 1997).

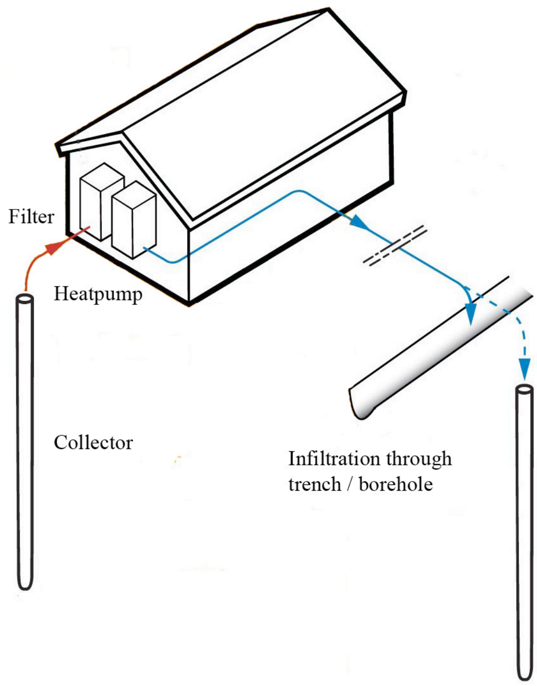


Figure 3. Ground water heat pump system (SVEP 2004).

2.1.4 Aquifer thermal energy storage (ATES)

ATES uses underground aquifers with water to store energy. The aquifers are hydraulically coupled underground and connected to the heating and cooling system with a couple of wells. In summertime cold water is extracted from one well for cooling purpose and then injected in the warm well. In winter the system is reversed and hot water is extracted. (Gao et al 2008). ATES-systems are geographically limited to areas where suitable aquifers can be found. Disadvantages are risk for corrosion, fouling and blockage of heat exchangers, wells and other components (GEOTEC 2009).

A large scale ATES system is currently under construction at Arlanda Airport in Stockholm. It uses aquifers in an esker, see figure 4. When the system will be operational it is estimated that 15 Gwh heat and 4 Gwh of electrical energy will be saved at the airport (Arlanda).

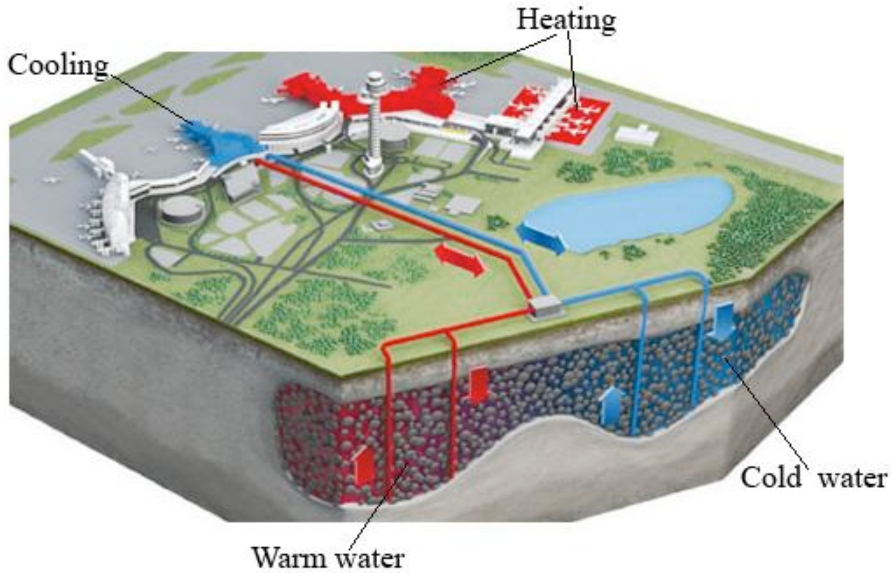


Figure 4. Aquifer thermal storage at Arlanda airport (Arlanda).

2.1.5 Borehole thermal energy storage (BTES)

The BTES systems are often referred to as vertical ground-coupled heat pump system. They consist of several boreholes, also called energy wells, at a depth ranging from 30m to more than 200m. The number of boreholes are often from one or a few for smaller system up to hundreds in large-scale systems. The largest BTES system is in Fort Polk, USA where 8000 boreholes supply the residents with hot and cold thermal energy. BTES require the smallest amount of pipes and pumping energy (Kavanaugh and Rafferty 1997). They also have the advantage of being space efficient and are often a valid alternative even in central cities (Omer 2006) where the BTES can be hidden under a parking lot or even beneath a building.

The temperature fluctuations in the ground decrease with depth, after a certain depth, the temperature remains constant throughout the year. Therefore the performance variations of a GSHP system decrease if the energy is taken from a greater depth.

There is a fundamental difference between a borehole configuration for heat extraction and storage of thermal energy. When storing energy the interaction between boreholes is favourable and therefore the boreholes should be placed in a compact pattern with low spacing. While for a system where either heat extraction or injection is the main purpose, interaction is undesirable and therefore prevented by a spread pattern with long distance between the boreholes.

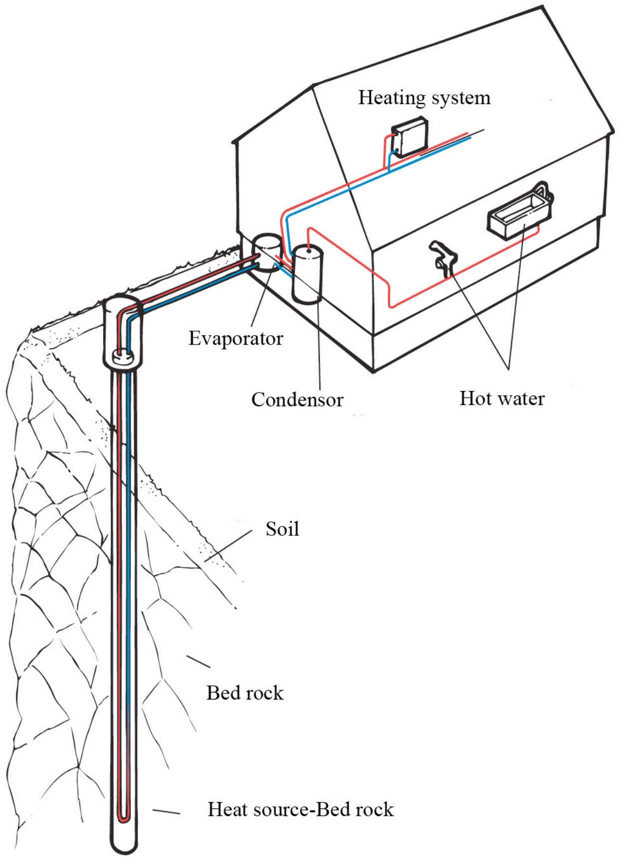


Figure 5. Vertical ground-coupled heat pump system (SVEP 2004).

The Geological Survey of Sweden (SGU) surveys and stores information about energy wells drilled in Sweden. According to SGU (2007) the number of boreholes drilled for GSHP system has increased rapidly. In 2006 alone about 40 000 energy wells were constructed, this can be compared with correspondingly 1000 wells teen years ago. Historically the majority of the wells have been used for heating residential buildings. The increase can be explained by the recent increase in large-scale systems containing from a few to hundreds of wells (SGU 2007).

2.2 Heat pump theory

The basic heat pump is a closed refrigerant circuit consisting of four components, the evaporator, a condenser, a compressor and an expansion valve, see figure 6. The refrigerant evaporates in the evaporator and absorbs energy from a low temperature source. Then the pressure and temperature of the refrigerant is increased in the compressor. After the compressor the refrigerant condenses and thereby rejects energy through the condenser before it passes through the expansion valve (Alvares 2006).

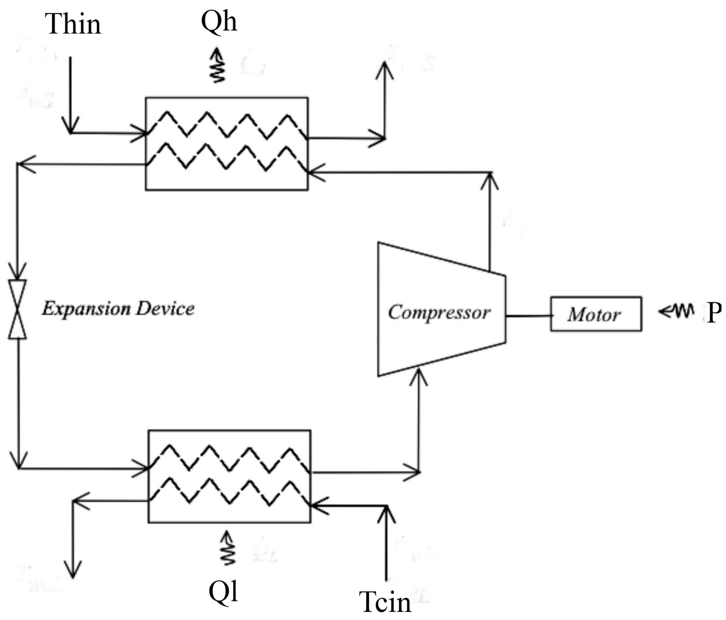


Figure 6. Heat pump (Jin 2002).

The thermodynamic process can also be viewed in a simplified temperature-entropy diagram (T-S diagram) see figure 7.

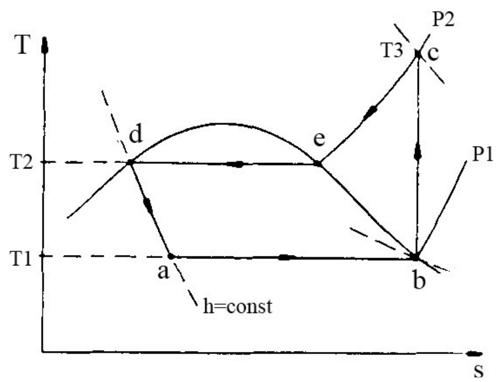


Figure 7. T-S diagram for the heat pump cycle (Alvares 2006).

Compared with the air-source heat pumps the GSHP-systems have an inherently higher efficiency since the ground temperature is higher than the air temperature during winter when heating is required and lower in the summer when cooling is required (Zeng et al 2002). The heat pump can usually be used for both heating and for cooling. Usually water-to-water heat pumps are used in GSHP-systems.

3 BTES and GSHP theory

There are a few parameters that are critical when sizing a BTES system. Two of them are the thermal conductivity and the initial undisturbed ground temperature.

3.1 Ground Heat transfer

Thermal energy can be transported through conduction, convection and radiation. In crystalline rock thermal conduction is dominating. While in porous rock thermal energy is also transported by natural or forced convection (Sundberg 1988).

The bedrock in Sweden consists mostly of the crystalline rock granite and gneiss. Granite and gneiss have a mean thermal conductivity of 3.47 (W/m K) with a standard deviation of 0.380 and 0.465 respective. The conductivity is correlated to the content of quartz in the rocks (Sundberg 1988).

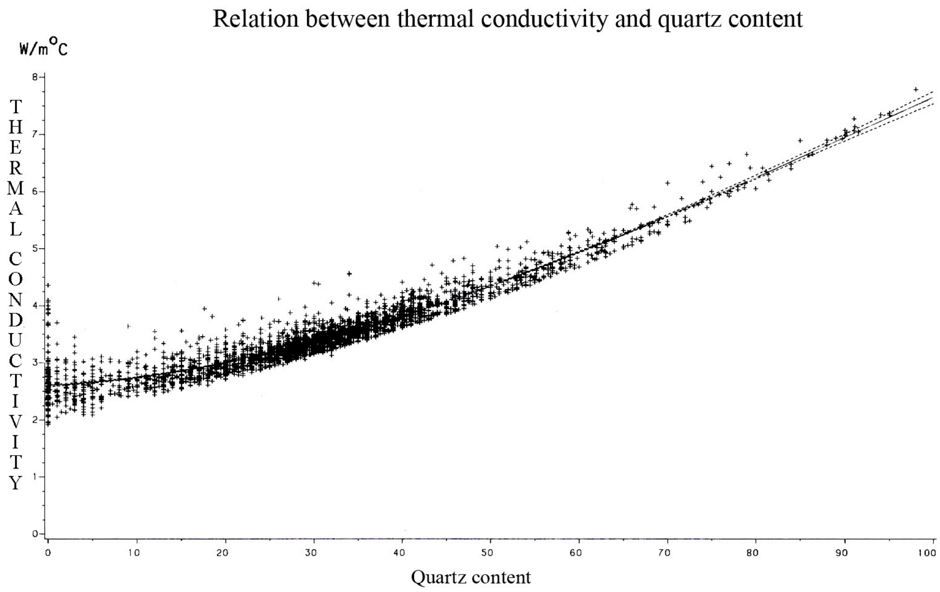
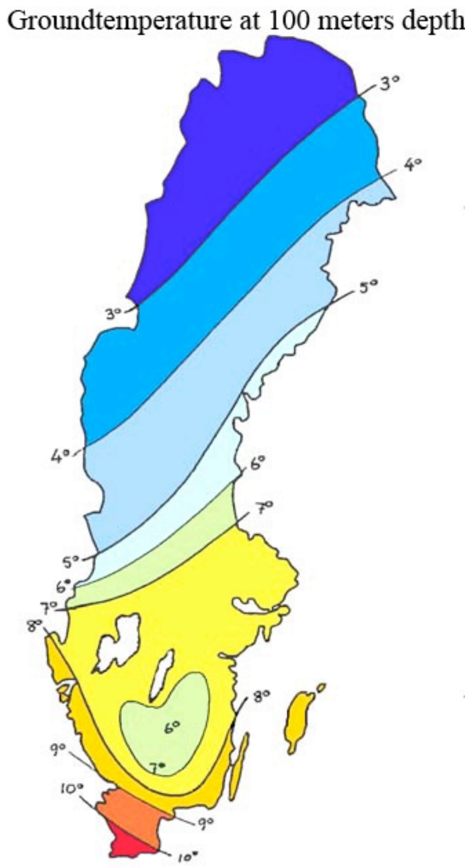


Figure 8. Relation between thermal conductivity and quartz content (Sundberg1988).

Other thermal properties of significance are thermal diffusivity (m^2/s) and specific heat capacity (J/kg, K). Thermal diffusivity is a measure of how quickly temperature differences are even out in body or between bodies. Specific heat capacity is a measure concerning the amount of energy a body can store (Sundberg 1988).

Decreasing the temperature of a cubic meter granite by one degree releases about 216 kW that can be used for spacial heating. A typical GSHP system with boreholes would use a storage volume exceeding 100 000 m^3 with 30-boreholes (GEOTEC 2009).



The undisturbed ground temperature is crucial when considering building a GSHP system. The temperature increases with depth which must be accounted for. Isotherms at 100 meters depth can be seen in figure 9. According to Eskilson (1987) the mean temperature along the borehole length is a good approximation of the undisturbed ground temperature.

Figure 9. Ground temperature at 100 meter depth (GEOTEC 2009).

3.2 Energy well and ground source heat exchanger

The energy well is constructed in two steps. First the drilling is done through the soil and down to solid rock, a casing is driven down into the rock to prevent the borehole from collapsing. The casing is then grouted towards the rock to ensure that there is no interaction between groundwater and the fluid in the borehole. In the second step the drilling reaches to the required depth for the energy well, see figure 10 (SGU 2007).

The heat exchanger usually consists of one or two polyethylene U-tubes that are inserted into the borehole. Spacers can be used to separate the collectors. If the ground water table is low some kind of filling material might be used to ensure thermal contact between the collector and the surrounding rock. This can also be done to prevent vertical migration of polluted water, drainage of near surface soil layers and or disturbance of artesian formations. This is not common in Sweden but in Germany and USA energy wells are always backfilled according to national regulations and recommendations (Gehlin 2002). Since grout has a poor thermal conductivity compared to water it is important to consider if grouting the upper part (3-10m) is adequate to prevent potential ground water contamination (Kavanaugh and Rafferty 1997). The length of the borehole where thermal interaction occurs is referred to as the active length. (Zeng et al. 2002)

In colder climate a mixture of water and alcohol is used for the ground loop. In Sweden brine solutions with 20-30 percent ethanol is most common and also recommended by SGU and the Swedish Environmental Protection Agency. Other solutions of glycol, salt solutions and vegetable oil are also used (SGU 2007).

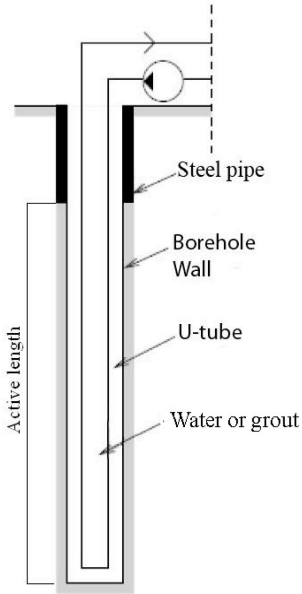


Figure 10. Energy well layout, the active length is defined as the length where thermal interaction occurs.

3.3 Thermal response test

A thermal response test is the common way of estimating the thermal conductivity of the bedrock, the initial undisturbed temperature of the ground and how well thermal energy is transferred between the carrier fluid and the borehole wall.

The borehole thermal response is the temperature change over time when a constant load is imposed. Firstly a borehole is drilled, then a u-tube collector is inserted and connected to a circulation pump and a resistance heater, the temperature of the carrier fluid entering and leaving the borehole is continuously measured, see figure 11.

To determine the undisturbed ground temperature the fluid is circulated without applying any heat load. This gives a value for the undisturbed ground temperature. It has to be considered that there will always be some heating of the water from the pump work. Another way of estimate the undisturbed temperature is through measuring the temperature every few meters of the collector. The collector has to be in equilibrium with the surrounding rock. A mean value for the ground temperature can then be calculated. This also gives a temperature profile of the ground (Gehlin 2002).

The temperature difference between the rock wall and the carrier fluid is proportional to the heat flow and the borehole thermal resistance. The thermal resistance is defined by the arrangement of the collector and the thermal properties of the collector, the filling medium and the surrounding rock (Heyi Zeng 2003).

Applying a constant load and measuring the change in temperature can estimate the values for thermal conductivity and borehole resistance. This assumes that the heat transfer in the rock is conductive without interaction from ground water flow. If there is a groundwater flow the thermal response is likely to induce a thermosiphon flow that enhances convective heat transfer and thus increases the estimated thermal conduction (Gehlin 2002). The difference between the thermal conductivity of the bedrock and the uppermost soil layer is ignored and has according to Eskilson (1987), little impact on the thermal performance.

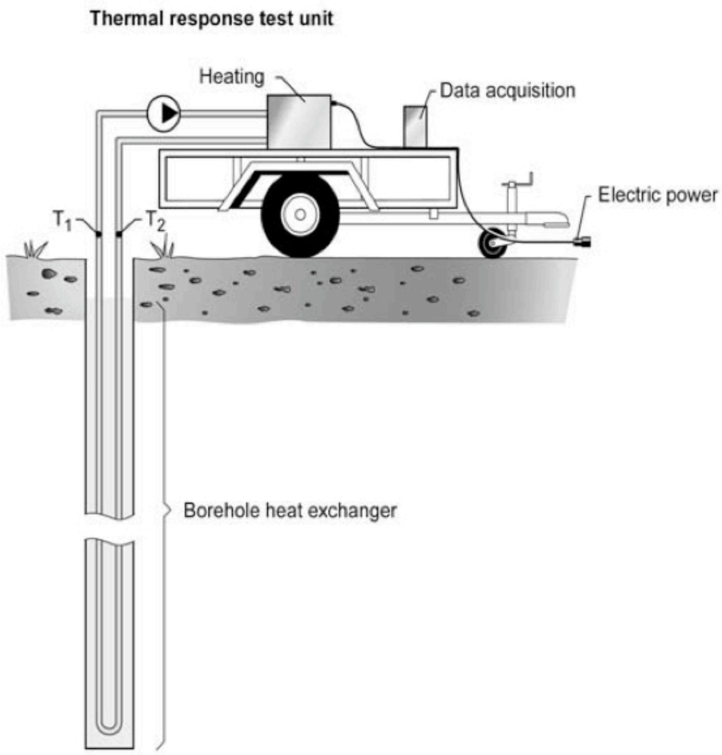


Figure 11. Rig for thermal response test, temperatures are measured at T_1 and T_2 (Gehlin 2002).

There are both analytical and numerical methods to evaluate thermal response tests. The analytical methods are the line source method and the cylindrical source method. These methods are discussed later in this thesis. The basic idea is to use simplified analytical expressions for the temperature change and then use a curve fitting method to evaluate the values for thermal conductivity and borehole resistance. The numerical methods can represent a higher level of detail of borehole geometry and the different physical properties of the carrier fluid, the collector, the filling material and the ground. The drawback is often that they require more extensive input and can therefore be time consuming compared to the analytical methods. A comparison of different methods to evaluate thermal response tests can be found in Gehlin (2002)

3.4 Balance of thermal loads

The load balance of a building varies depending on for example the geographic location and the utilization of the building. A shopping mall does not have the same load pattern as a domestic building. In a cold environment there can be a large difference between the amount of energy required for heating and for cooling. If the heat extracted during the winter can be compensated with heat injected during the summer the temperature surrounding the boreholes will change little and thereby also the performance of the system. However, if the injected heat is less than the extracted the temperature will subsequently sink in the ground and the performance of the system will deteriorate. If the system is operated enough long time with an imbalanced load the temperature distribution in the ground will reach a steady state. This might take 10 years or even longer (Zeng et al. 2002)

In Sweden the heating load during winter is dominant and therefore also determines the dimension of the system. One way of avoiding imbalance is to recharge the BTES during summer. This can be done in several ways. Some of the more common are listed below.

- Industrial excess heat
- Solar panels
- Ventilation air
- Outside air
- Surface water

An example of a BTES system with solar recharge is the Drake Landing Solar Community in Alberta, Canada. Where 52 homes are heated by solar energy that is gathered with collectors during the summer and then stored in 144 boreholes, see figure 12. It is estimated that 90 % of the required energy for space heating is met by solar energy (Sibbit et al 2007).

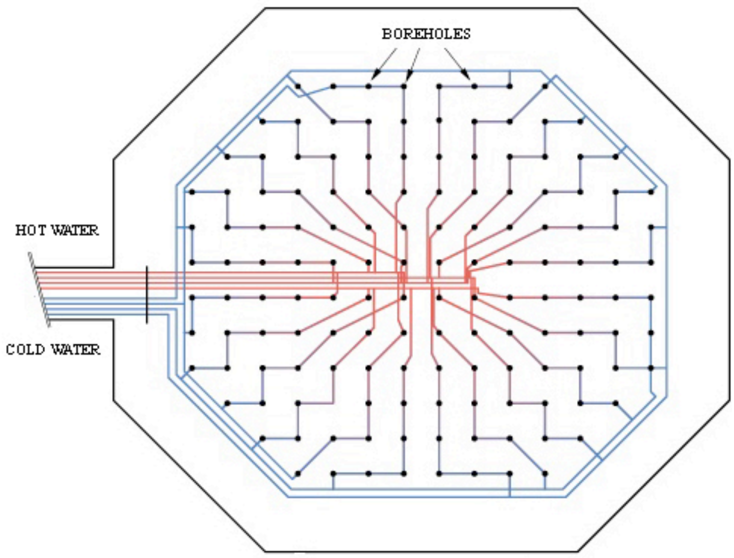


Figure 12. BTES at the Drake landing solar community (DLSC).

Balance can also be achieved by attaching extra cooling loads during summer. This is done at Lund University where a BTES consisting of 153 wells at a depth of 230m supply heating for 2 buildings and cooling for 3 buildings.

Usage of BTES system for direct cooling can also be referred to as free cooling. This means that the cold fluid from the ground loop is circulated through a heat exchanger that supplies the building with cooling. The benefits are obvious as cooling is provided without the interaction of a heat pump. Depending on the demands of cooling the mass flow in the loop can be varied.

If the cooling demand exceeds what the BTES can supply the carrier fluid can be cooled through the evaporator of the heat pump. This will cause the heat pump to produce heat on the condenser side. If there is no heating demand, the excess heat can either be fanned away or supplied to the BTES through a heat exchanger.

Depending on the load and the available land the boreholes can be placed in different patterns. Because of the thermal interaction different borehole patterns will give different thermal response. A closed pattern would have better performance as energy storage and also be more sensitive for unbalanced loads than an open pattern, see figure 13.

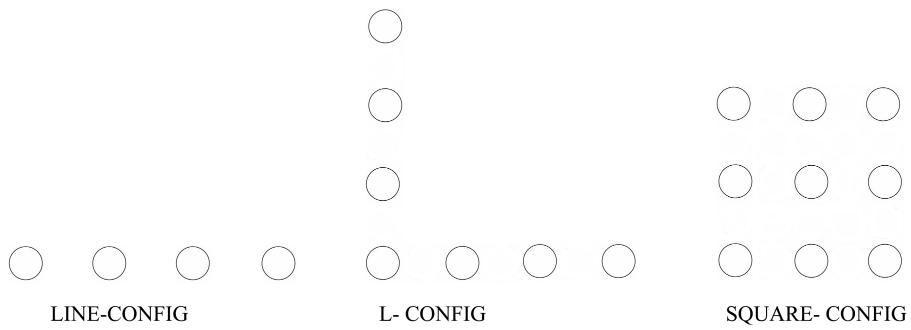


Figure 13. Different borehole field configurations.

3.5 Simulation of GSHP system

The main components in a GSHP system are a ground heat exchanger (GHE), a circulation pump, a heat pump and a heat exchanger. When a BTES is used the performance and running hours of these components are coupled with the temperature in the ground and the heat transfer between the carrier fluid and the ground. The heat transfer in the ground is a slow process while the rest of the system operates on a shorter timescale.

Simulations on hourly level have been performed by Bernier (2001, 2004) and by Nagano et al (2006) with the aim to perform life cycle analysis. An hourly timescale make it possibly to calculate COP more precisely and to determine running time of circulation pumps. As mentioned in Bernier (2001) the energy consumption by circulation pumps may account for a significant proportion of the buildings annual energy consumption. Hourly simulations can also be used to size the GSHP system more accurately (Bernier et al 2004).

Full system analysis has also been done in advanced simulation environments such as TRNSYS. Simulation environments such as TRNSYS are often difficult to learn and more suited for research than as an industrial tool (Bernier 2001).

There are several methods to simulate the heat pump performance. Bernier (2001) used a lookup table with interpolation while Nagano (2006) used an equation fit. Other methods will be discussed later in this chapter.

Since the rate of heat injection and extraction into the ground is coupled with the heat pump performance the temperatures and the load have to be solved for with an iterative process (Bernier 2001).

Both Bernier (2001) and Nagano (2006) used analytical solutions to simulate the ground source heat exchanger. Bernier (2004) used a combination of analytical and numerical methods. As mentioned in Nagano (2006) there are several software available for simulation of GSHP systems but these software are primarily used to determine length of ground source heat exchangers and have monthly values or peak values as input.

The main components of the GSHP system will be treated below with exception for the BTES that will be covered in the next chapter.

3.5.1 Heat pump unit

Several different approaches are available for simulation of heat pumps, from the simplest version of a look up table to sophisticated methods incorporating multivariable optimisation algorithms. Heat pump models can be made both very realistic and complicated. The more complicated methods might require insight beyond what is commonly listed in a manufacturer catalogue.

A heat pump model for simulation of GSHP systems would have to fulfil some criteria. It needs to be simple enough to work with listed data from a manufacturer and it needs to be both accurate and computationally efficient. The input would typically be the inlet temperatures and the mass flow in the condenser and the evaporator.

One method is the equation fit, the idea is to describe the performance of the heat pump by a set of coefficients that are determined by a least square curve fitting. This method must be handled with care when extrapolating the data beyond what is given in the catalogue (Jin 2002). Once the coefficients are determined the heat pump calculations are instantaneous and take no computational effort.

Another method is the parameter estimation method, where the thermodynamic relations and equations are used together with a multivariable optimization method. Initial guess values are given for the unknown parameters and then solved for with the Nielder-Mead Simplex method. The parameter estimation method is more precise than the equation fit and it also allows for extrapolation with reasonable results (Jin 2002).

Since a typical heat pump can use a scroll compressor, a reciprocating compressor or a rotary compressor the relations are somewhat different which must be considered in the parameter estimation method.

A disadvantage with the parameter estimation method is that it takes more computational time (Tang 2005). It does also require subroutines for refrigerant properties.

3.5.2 Heat exchanger theory and modelling

Heat exchangers are used to separate the different heat carrier fluids that might consist of different solutions of water and alcohol or pure water. Usually a counter flow plate heat exchanger is used. For analysis and simulation the Number of Transfer Units (NTU) method and the Logarithmic mean temperature difference (LMTD) are used. The effectiveness of the heat exchanger is defined as the ratio between the actual heat transfer rate and the thermodynamic maximum heat transfer rate. It can be expressed as a function of NTU and the heat capacity rate of the fluids (Sundén 2006)

$$\varepsilon = \frac{1-\exp\left[-(1-C_{\min}/C_{\max})NTU\right]}{1-(C_{\min}/C_{\max})\exp\left[1-(1-C_{\min}/C_{\max})NTU\right]} \tag{1}$$

$$NTU = \frac{-\dot{Q}/LMTD}{C_{\min}} \tag{2}$$

$$C_c = \dot{m}_c C_{pc} \text{ heat capacity rate of the cold fluid} \tag{3}$$

$$C_h = \dot{m}_h C_{ph} \text{ heat capacity rate of the hot fluid} \tag{4}$$

C_{min} is the minor of the heat capacity rates and C_{max} is the larger.

Values for LMTD and the heat transfer capacity ($\dot{Q}$) are typically listed from manufacturer.

Once the effectiveness is known the outlet temperatures can be calculated as.

$$T_{hout} = T_{hin} - \frac{\varepsilon C_{\min}(T_{hin}-T_{cin})}{C_h} \tag{5}$$

$$T_{cout} = T_{cin} + \frac{\varepsilon C_{\min}(T_{hin}-T_{cin})}{C_c} \tag{6}$$

3.5.3 Circulation pumps

A number of circulation pumps are used throughout the system. For the ground loop it is desirable to have a varying mass flow. With the affinity rules the characteristics of a circulation pump can be calculated from tabulated data for constant values of fluid density, viscosity (Alvarez 2006).

The circulation pumps can account for a significant part of the energy used in a GSHP system and therefore also affect the overall efficiency of the system (Kavanaugh and Rafferty 1997).

The affinity rules are:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \tag{7}$$

$$\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{8}$$

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{9}$$

Where n is the pump impeller speed. Q is the flow rate, H is the pressure and P is the power.

3.5.4 Pressure loss in pipes

The head loss in pipes can be divided into minor and major head losses. The minor refer losses in bends, valves and similar. The major losses are in the straight pipe sections and caused by the friction between the fluid and the pipe wall.

The largest pressure loss in a BTES system would be in the collectors. A system with 20 boreholes at 200 m depth would have a collector length of 8 km for just the borehole heat exchanger (Young et al. 2004).

4 BTES-Model

There are both numerical and analytical methods available for simulation of BTES. The numerical methods are typically the finite difference and the finite volume while the analytical are variations of the cylindrical source theory and the line source theory (Yavuzturk 1999). Combinations of numerical and analytical methods can also be found (Bernier 2004). Yavuzturk (1999) gives a review of different methods to simulate BTES.

The aim is to simulate BTES on a timescale suitable for lifecycle analysis of the total GSHP system. This means that running time of circulation pumps and performance of heat pumps should be possible to evaluate.

An hourly timescale has been chosen based on coherence in the literature (Yavuzturk 1999), (Nagano 2006), (Bernier 2001, 2004).

4.1 Governing equation for heat conduction

The governing equation for heat conduction is also known as Fourier's equation.

$$\frac{\partial T}{\partial t} = a_s \nabla^2 T \tag{10}$$

For the case of a borehole heat exchanger the temperature change is neglected in axial direction. And the heat conduction can be analysed in a radial plane. In radial coordinates the conduction equation becomes:

$$\frac{\partial T}{\partial t} = a_s \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) \tag{11}$$

The temperature will be a function of only the radial distance and time (Carslaw and Jaeger 1947).

4.2 Numerical methods

While numerical models of BTES have the advantage of being able to handle a high level of detail, the drawback is that they are computationally expensive and therefore not suited for short time steps.

One way of handling the long calculation time is to precalculate dimensionless temperature response factors for various BTES configurations. These response factors are commonly known as g-functions. Once the g-function is known they can be incorporated into dimensioning tools. This was initially done in the SBM-model by Eskilson (1987) with a two-dimensional finite difference scheme. Since the model doesn't account for the local borehole geometry the result was limited to time intervals larger than approximately 3-6 hours (Yavuzturk 1999).

Short time response g-functions have been developed by Yavuzturk (1999) with a 2-d fully implicit finite volume approach. The result was implemented in a TRNSYS component and validated by experimental data.

Hellström (1991) developed the duct storage model (DST). In this model the heat transfer problem is solved by superposition of a local and a global region. The immediate region surrounding the borehole is the local region while the global region is between the borehole storage volume and the far field. A combination of two-and one-dimesional finite difference schemes are used together with analytical solutions. In the DST the boreholes are assumed to be densely packed as a heat storage. The result has been implemented as a component in TRNSYS

The g-function approach by Eskilson is regarded as the state-of-the-art and has been implemented in several dimensioning software for GSHP systems.

A disadvantage with pre-calculated g-functions is that they are locked to specific configurations. According to Katusura et al. (2007) most of the available simulation programs utilize pre-calculated functions derived from numerical simulations for prepared layouts of multiple ground heat exchangers.

4.3 Analytical methods

The line source and the cylindrical source theory were proposed for analysis of BTES system by Ingersoll (1948). The line source is derived from Kelvin's line theory (Yavuzturk 1999), and the cylindrical source from Carslaw and Jaeger (1947).

"A line source might be thought of as a continuous series of point sources along an infinite straight line" (Ingersoll 1954)

The line source is often refereed to as the infinite line source due to the assumption of an infinite medium. Ingersoll (1954) mentions the line source as the simplest theory for BTES with a limitation to short time intervals. As an alternative the cylindrical heat source is proposed as a more robust solution.

The benefits of using analytical equations for BTES systems are primarily computational speed and flexibility. The disadvantages are that a number of assumptions have to be done to simplify the calculations, which might affect the result.

4.4 Nomenclature

α	ground thermal diffusivity (m^2/s)
c_p	specific heat (J/kg,K)
Fo	Fourier number
q	thermal load (Watt)
k	ground thermal conductivity (W/m,K)
L	active borehole length (m)
$\dot{m}$	massflow of heat carrier fluid (kg/s)
r	distance between boreholes (m)
r_b	borehole radius (m)
Rb	effective borehole thermal resistance (K/ (W/m))
t	time (hours)
T_f	fluid mean temperature (^oC)
T_w	borehole wall temperature (^oC)
T_g	far field undisturbed ground temperature (^oC)
T_p	temperature correction for thermal interference (^oC)
T_s	ground surface mean temperature (^oC)
w	heat flow per unit area (W/m^2)

4.5 Mean fluid temperature (T_f)

According to Bernier (2004) the mean fluid temperature for a BTES can be calculated as:

$$T_f = T_g - \frac{q}{L} R_b - \frac{q}{L} \frac{G(Fo)}{k} + T_p \tag{12}$$

Once T_f is known the temperature of the outlet fluid from the ground heat exchanger can be determined by applying an energy balance to ground loop (Bernier et al 2004)

$$q = \dot{m} c_p (T_{out} - T_{in}) \text{ and } T_f = \frac{T_{in} + T_{out}}{2} \text{ gives } T_{out} = T_f + \frac{q}{2\dot{m} c_p} \tag{13}$$

In this chapter equation (12) will be explained in detail. Methods to calculate T_p (temperature penalty) and the G-term (G-function) will be discussed then the cylindrical source, the infinite line source and the finite line source methods will be introduced.

4.5.1 Far field undisturbed ground temperature (T_g)

The far field undisturbed temperature (T_g) can be difficult to evaluate. Usually it can be measured when performing thermal response tests. A simplified way of estimating T_g is by using the thermal gradient or values of the ground heat flow and calculate the mean value for the borehole. The rate of flow of heat per unit area in the radial direction of a sphere is given by Ingersoll (1954)).

$$w = -k \frac{dT}{dr} \tag{14}$$

This gives the mean ground temperature:

$$T_g = T_s + \frac{w}{k} \frac{L}{2} \tag{15}$$

4.5.2 Borehole resistance (R_b)

The borehole resistance (R_b) defines the relationship between heat flow and the temperatures of the heat carrier fluid and the borehole wall (Zeng et al 2002).

$$T_w - T_f = qR_b \tag{16}$$

Equation 16 is better understood in relation to figure 14, the fluid temperature T_f is here the arithmetic mean of the inlet and outlet fluid temperatures.

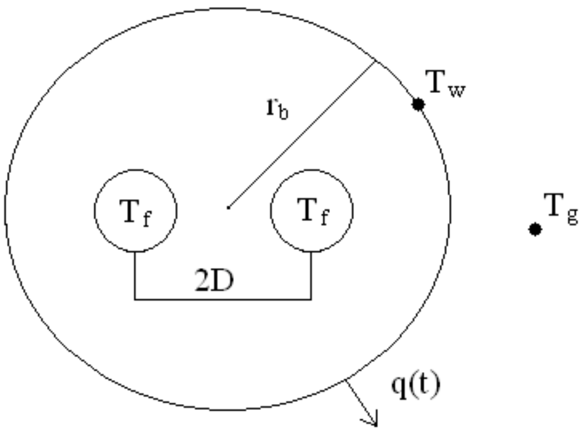


Figure 14. Borehole with single U-tube (Bernier 2004)

The borehole resistance is dependent on the borehole, U-tube configuration, the thermal properties of the materials and the mass flow. Different approaches to evaluate the resistance can be found in the literature. Nagano et al. (2006) describes an approach with the boundary element method (BEM) for calculation of the thermal resistance in the borehole. Kavanaugh and Rafferty (1997) treat the borehole resistance as a constant (steady state) value and display it in tables.

The thermal resistance can be lowered by using a double U-tube, or by using thermally enhanced grout as filling material when that is required. The distance between the collector pipes is called the shank spacing and has a natural impact on the thermal resistance. Thermal resistance decreases with increasing shank space since it favours conduction between the pipe and the borehole wall while interference between the pipes is lowered.

Zeng et al. (2003) describes a method that is derived from the line source equation as an extension of previous work by (Hellström 1991). The result is a function of the thermal properties of the construction materials, the physical properties of the borehole and the U-tube, the mass flow and specific heat of the carrier fluid. Zeng et al. (2003) provides results for both a single and a double U-tube in series and in parallel connection.

4.5.3 G-function

For a constant load the borehole wall temperature (T_w) can be determined with equation 17.

$$T_w = T_g - \frac{q}{L}\frac{G(Fo,p)}{k} \qquad (17)$$

Bernier (2004) uses this equation for the cylindrical source equation but it works equally well with the line source and the finite line source.

4.5.4 Thermal interference (T_p)

After some time of operation there will be thermal interaction between the boreholes in a borehole field. This is a slow process but it has a large influence on the performance of the system.

There are a number of different analytical approaches to calculate this interaction. In Bernier (2004) and Kavanaugh and Rafferty (1997) the thermal influence is accounted for as a temperature penalty (T_p) when calculating the mean fluid temperature.

$$T_p(t) = \sum_{j=1}^{n} \Delta T(r_j, t)) \qquad (18)$$

T_P is the sum of the thermal interference from the surrounding boreholes and **r** is the distance to the boreholes according to figure 15.

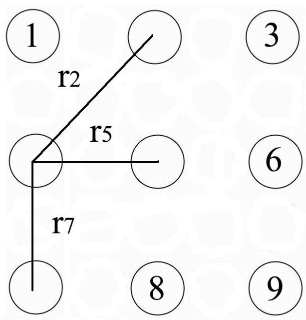


Figure 15. Borehole field

Equation 18 implies that calculating the interference for one borehole involves solving for ΔT for each of the surrounding boreholes. This becomes quickly computational heavy for larger borehole fields. Therefore the equations are instead solved for every unique distance and then summarized for the corresponding borehole.

The difference is 5 equations instead of 30 for a 3X3 configuration or 50 instead of 9900 for a 10X10 configuration

In Bernier (2004) a numerical approach with the finite volume is used to calculate the thermal interference. Because of the slow nature of the heat transfer process and a need to increase computational speed the thermal interference was not calculated for every time step. Bernier refers to a sensitivity analysis by (Pinel 2003) where it was concluded that evaluating the thermal interference every two weeks had a small impact on the accuracy.

4.5.5 Cylindrical source theory (CS)

Carslaw and Jaeger (1947) give the cylindrical source equation as the solution of a heat transfer problem where an infinite hollow cylinder in an infinite solid medium is transferring heat at a constant rate.

The surrounding medium is assumed to have uniform and constant properties a constant initial temperature and a perfect contact with the source (Ingersoll (1954)).

When the cylindrical source equation was presented for ground source heat exchangers by Ingersoll (1954) it was difficult to evaluate as it involves integration over Bessel-functions of first and second grade. Ingersoll presented some results that had been derived numerically and referred to the infinite line source for cases that were not covered.

$$\Delta T = \frac{q}{k} G(Fo, p) \tag{19}$$

$$G(Fo, p) = \frac{1}{\pi}\int_0^\infty \left(e^{-\beta^2 Fo} - 1\right)\frac{J_0(p\beta)Y_1(\beta) - J_1(\beta)Y_0(p\beta)}{J_1^2(\beta) + Y_1^2(\beta)}\frac{d\beta}{\beta^2} \tag{20}$$

Where J and Y are Bessel functions, p is the ratio between the radius of the borehole and the point where the temperature is sought. The value 1 corresponds to the surface wall of the borehole.

Fo is the Fourier number formulated as $Fo = \frac{\alpha t}{\mathrm{r_b}^2}$ (21)

In reality the load q will not be constant, so the equation must be able to handle variable loads. This can be done with a superposition technique that has been described in Bernier (2000). Figure 16 can help to illustrate the idea of the superposition technique in equation 22.

$$\Delta T = \frac{q_1}{L}\frac{\left(G\left(Fo_{t_3-0}\right) - G\left(Fo_{t_3-t_1}\right)\right)}{k} + \frac{q_2}{L}\frac{\left(G\left(Fo_{t_3-t_1}\right) - G\left(Fo_{t_3-t_2}\right)\right)}{k} + \frac{q_3}{L}\frac{G\left(Fo_{t_3-t_2}\right)}{k} \tag{22}$$

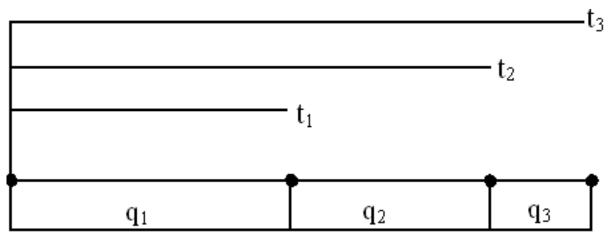


Figure 16. The superposition technique for variable loads.

4.5.6 Infinite line source (IFLS)

In the infinite line source the borehole is assumed to be of infinite length so that all the heat flow can be considered to be radial. As for the cylindrical source the surrounding medium is assumed to have uniform and constant properties a constant initial temperature and a perfect contact with the source (Ingersoll 1954).

As the infinite line source theory is only correct for a true line source Ingersoll gives a criteria for when the equation involves appreciable error $\frac{\alpha T}{r_b^{\ 2}} < 20$ (24)

For times longer than one day the line source gives results with an error not exceeding 2 percent. This limits the use of line source for shorter time periods (Ingersoll 1954).

The ground surface temperature is neglected with the assumption of an infinite line source, this assumption has the effect that the infinite line source will never reach a steady state solution. Clearly this is not realistic for a borehole heat exchanger and should be thought of when using the method for analysis of long-term performance. However the method is simple and has been used widely (Zeng et al 2002).

$$\Delta T = \frac{q}{2\pi k} \int_{\frac{r}{2\sqrt{\alpha t}}}^{\infty} \frac{e^{-\beta^2}}{\beta} d\beta \qquad (25)$$

For variable loads Ingersoll (1954) proposed that the equation could be split according to equation 26.

$$\Delta T = \frac{1}{2\pi k}\left(q_1 \int_{\frac{r}{2\sqrt{\alpha t_3}}}^{\frac{r}{2\sqrt{\alpha t_2}}} \frac{e^{-\beta^2}}{\beta} d\beta + q_2 \int_{\frac{r}{2\sqrt{\alpha t_2}}}^{\frac{r}{2\sqrt{\alpha t_1}}} \frac{e^{-\beta^2}}{\beta} d\beta + q_3 \int_{\frac{r}{2\sqrt{\alpha t_1}}}^{\infty} \frac{e^{-\beta^2}}{\beta} d\beta \right) \quad (26)$$

4.5.7 Finite line source (FLS)

The finite line source was presented in Zeng et al (2002) as an alternative to the infinite line source. The method uses a virtual sink to account for the ground surface. This solves the previous problem with the assumption of an infinite line source.

The method uses the same assumptions as the infinite line source regarding the surrounding medium with the addition of a constant ground surface temperature.

Zeng et al (2002) proposed an expression with a double integral that would be computationally heavy and unpractical for engineering applications. This was then made more efficient by Lamarche and Beauchamp (2006).

$$\Delta T = \frac{q_0'}{2\pi k} G\left(t^*, \beta\right) \quad (27)$$

$$G\left(t^*, \beta\right) = \frac{1}{2}\int_0^1 \partial\eta \int_0^1 \frac{erfc\left(3\tilde{r}^+ / \left(2\sqrt{t^*}\right)\right)}{\tilde{r}^+} \partial\xi - \frac{1}{2}\int_0^1 \partial\eta \int_0^1 \frac{erfc\left(3\tilde{r}^- / \left(2\sqrt{t^*}\right)\right)}{\tilde{r}^-} \partial\xi \quad (28)$$

$$\tilde{r}^+ = \sqrt{\beta^2 + (\eta - \xi)^2} \quad (29)$$

$$\tilde{r}^- = \sqrt{\beta^2 + (\eta + \xi)^2} \quad (30)$$

$t^* = t / t_s$ and $t_s = H^2 / 9\alpha$

$\beta = r / H$

H is the active borehole length, r either the distance between the boreholes or the radius of the borehole, and α is the thermal diffusivity.

Variable loads are treated in equation 31 with superposition technique analogue to the treatment of the cylindrical source equation with reference to figure 16.

$$\Delta T = \frac{q_1}{L}\frac{\left(G(t_3 - 0) - G(t_3 - t_1)\right)}{k} + \frac{q_2}{L}\frac{\left(G(t_3 - t_1) - G(t_3 - t_2)\right)}{k} + \frac{q_3}{L}\frac{G(t_3 - t_2)}{k} \quad (31)$$

4.5.8 Kavanaugh and Rafferty

In the approach proposed by (Kavanaugh and Rafferty 1997) it is assumed that heat which normally would have diffused away from a single borehole will be stored in a cylinder with a radius of half the borehole separation distance, see figure 17.

The average temperature change is calculated with the infinite line source for a series of hollow cylinders up to a radius of 25 feet (7.6 m), and then the related amount of energy is summed up and divided by the area of the inner circle.

For boreholes at the outer sides and at the corners the temperature change is multiplied with a correction of 0.5 and 0.25 respectively.

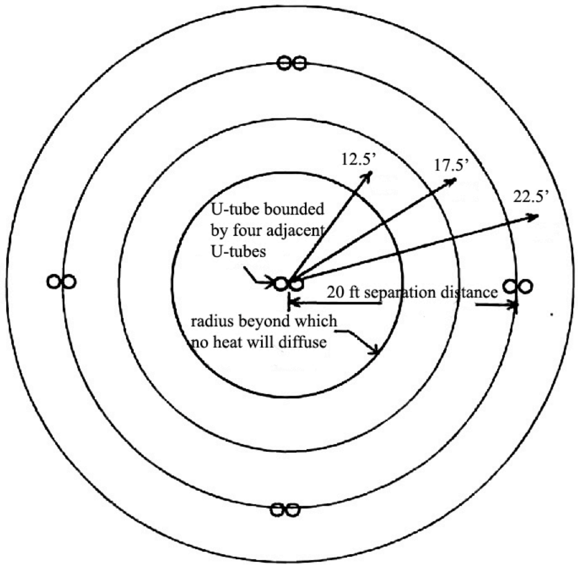


Figure 17. Calculation of thermal interference (Kavanaugh and Rafferty 1997).

4.5.9 Aggregation

The cylindrical source, the line source, and the finite line source equations are all dependent on the thermal history. This means that the solution at a given time step would have to take into account all the previous loads. For small time steps or a long period of time this becomes quickly computational heavy.

According to Bernier et al (2004), the immediate thermal history is more important than the past history. The past load history can therefore be aggregated to increase computational speed while the immediate history is kept unaggregated. Bernier et al (2004) and Yavuzturk and Spitler (1999) have proposed methods for aggregation of loads. In Bernier et al (2004) the past history were divided in four time intervals corresponding to a day, a week a month and a year.

5 Method and results

Results from simulations of BTES with different methods and combination of methods will be presented here and a demonstration of the possibilities with simulation on an hourly level. The simulations have been performed on a standard laptop.

The BTES-model and different components of the GSHP system has been simulated with Matlab. For evaluation of integrals the built in Matlab function quad.m has been used. When the BTES-model is interacting with other components in a GSHP system the coupled temperatures and loads has been solved for with Matlab fsolve.m

To achieve computational speed the cylindrical source equation has been evaluated and stored in cubic splines. The thermal interference (T_p) is solved for every unique distance in the borehole field and then summarized to the corresponding borehole. A mean value of T_p is then used when calculating the mean fluid temperature (T_f). For hourly simulations the load history is subsequently aggregated in 10 different periods and T_p is updated every second week (336 hours).

Simulations of the mean fluid temperature have been done with the cylindrical source, the infinite line source and the finite line source. The result has then been compared with a software using the state-of–the-art SBM. The best consistency was given with the cylindrical source method and the finite line source method. Since monthly load values were used as input to the software a monthly time step has been used in these comparisons.

The results for the cylindrical source and finite line source are almost identical with a monthly time step. However, the finite line source is restricted for shorter time steps just as the infinite line source and the SBM. This is because the borehole geometry is not accounted for.

Based on results from simulations and literature the cylindrical source together with the finite line source has been chosen as the best way to calculate the mean fluid temperature for shorter time steps. This combination has then been used to first demonstrate the influence of different load and BTES configurations. Then the interaction with a heat exchanger that uses the NTU-method is demonstrated. Finally a 10-year simulation of a simplified GSHP-system consisting of a BTES, a heat pump, a heat exchanger and a hot water accumulator is performed.

The BTES-model has not been validated on a short timescale within this thesis. However, good results have been presented for the cylindrical source by Bernier (2004), Katsura et al. (2007), Nagano et al. (2006) and Deerman and Kavanaugh (1991).

5.1 Thermal Interaction between boreholes

The influence of thermal interaction has been calculated individually for each borehole, and then an arithmetic average has been used for the whole borehole field. The mean influence calculated with infinite line source and finite line source can be seen in figure 18.

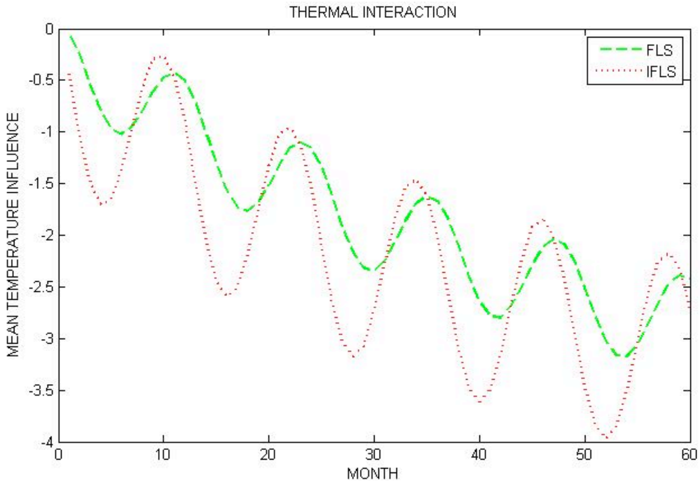


Figure 18. Interaction between boreholes with infinite line source and finite line source.

5.2 Aggregation

The concept of aggregation was presented in the background. It is a way of increasing the computational speed. In figure 19 the results for a five-year simulation is presented together with the result when the first 36 months are lumped together in yearly loads. A slight difference can be seen between the simulations at 4 and 5 years.

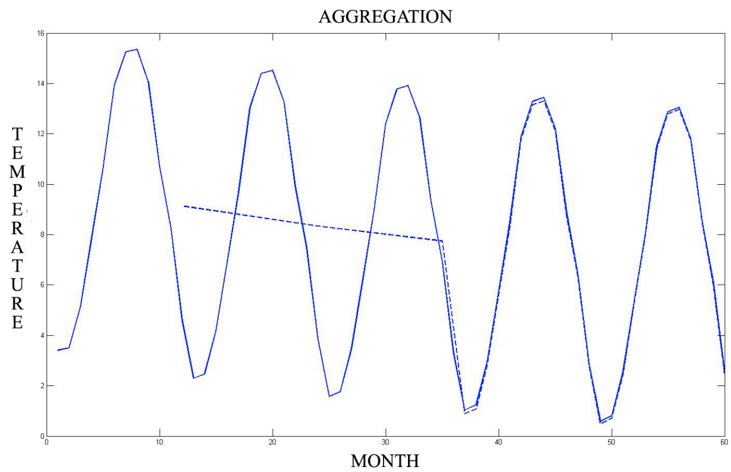


Figure 19. Aggregation of the first three years of a five-year period.

5.3 Validation of BTES-model

To validate a model for the BTES the results has been compared with results from SBM for various BTES configurations. In figure 20 five different cases are compared. With CS + IFLS it means that the cylindrical source has been used to evaluate the borehole wall temperature while the interaction has been calculated with Infinite line source. As can be seen the results for CS + FLS is consistent with the result for FLS for both interaction and borehole wall temperature.

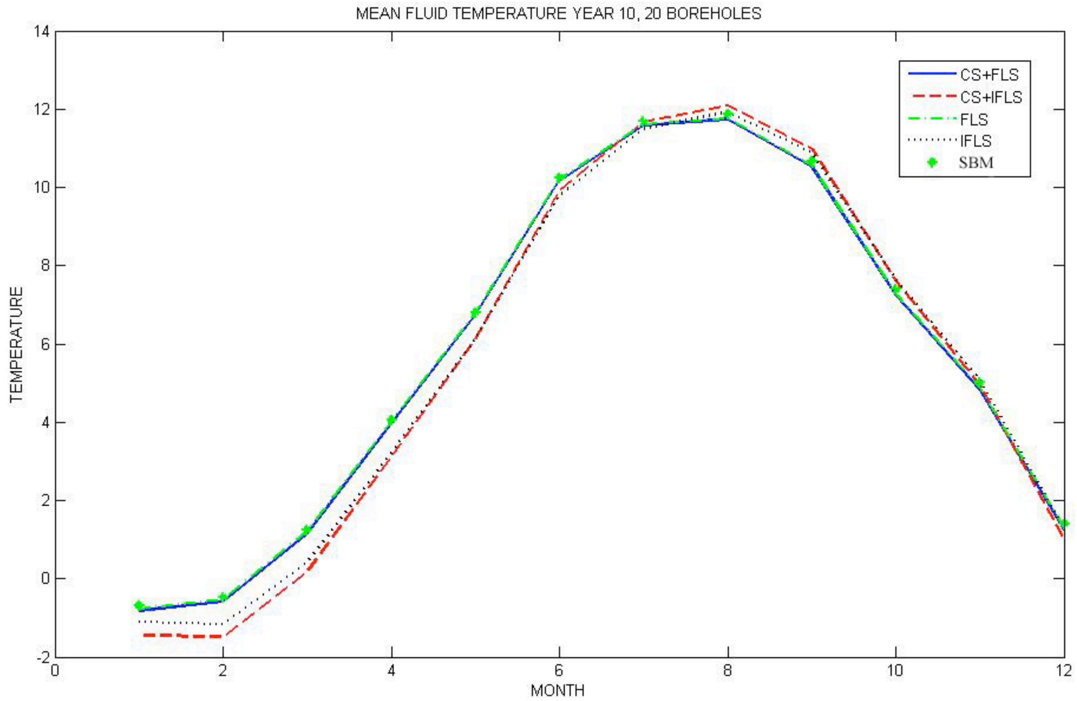


Figure 20. Comparison between the different analytical methods and SBM.

5.4 BTES configurations

An advantage of using analytical solutions in simulations is the flexibility. Simulations of BTES with a cylindrical borehole pattern or a varying distance between boreholes are easy to implement. As an example the mean fluid temperature for four different configurations are compared in figure 21.

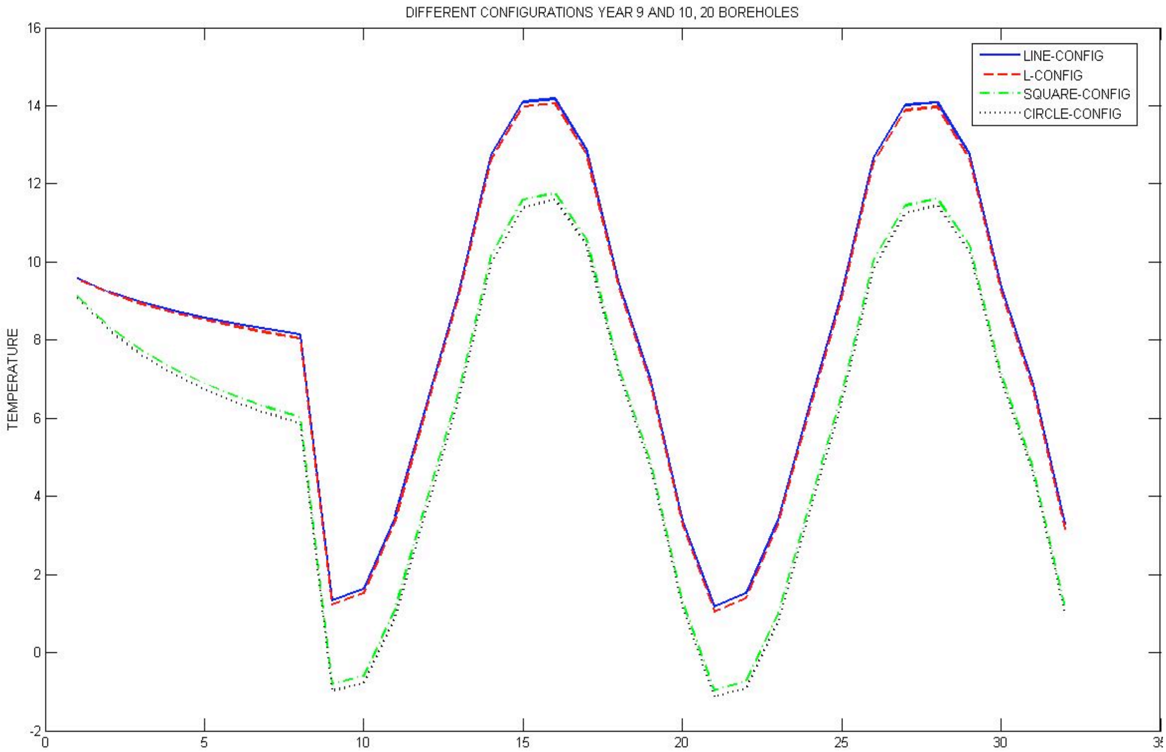


Figure 21. Mean fluid temperature as a consequence of different borehole patterns.

A distance of 6 m has been used in all configurations. The cylindrical pattern has been constructed by placing boreholes with a distance of roughly 6 m on a radius of 6 and 12 m with a borehole in the centre. The load is unbalanced and more heat is extracted than injected. The line- and L-configurations suffer the least from interference while the cylindrical-configuration gives the lowest temperatures. If instead more energy had injected the cylindrical storage would reach the highest temperatures.

5.5 Interaction with heat exchanger

A system where the BTES-model is interacting with a heat exchanger is demonstrated in figure 23. Initially there is no load but the temperature to the heat exchanger is 17 ^{o}C. After 24 hours a constant load is applied on the hot side of the heat exchanger.

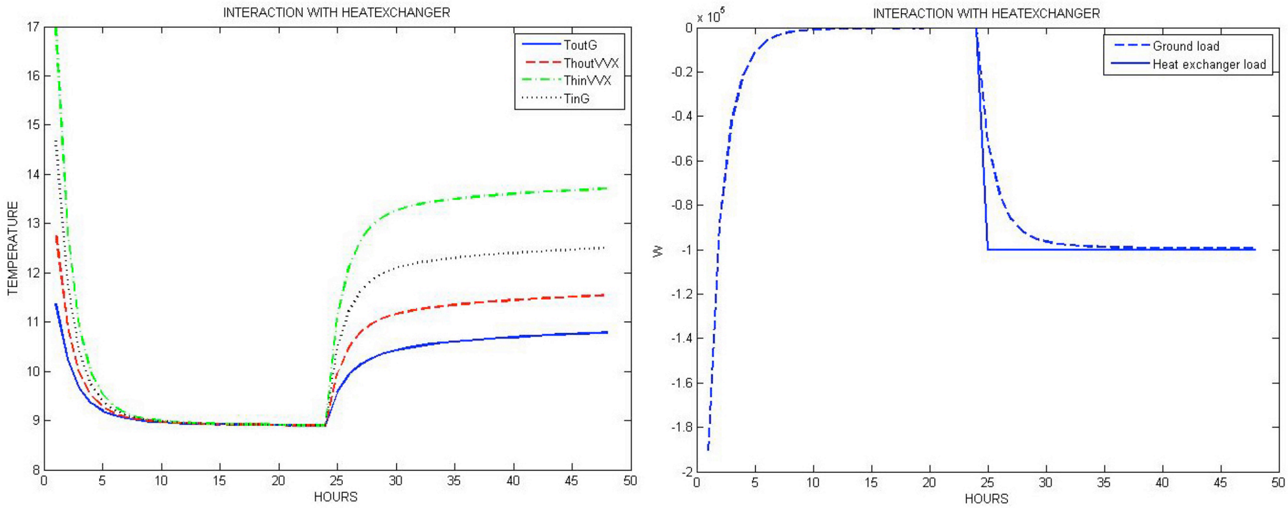


Figure 23. Interaction with heat exchanger, outlet and inlet temperatures to ground and heat exchanger are displayed.

5.6 GSHP-system simulation

A simplified GSHP system consisting of a BTES interacting with a heat pump, heat exchanger, and a hot water accumulator is simulated to further demonstrate the BTES-model. The components have been treated as:

- Equation fit heat pump model according to Tang (2005)
- Heat exchanger simulation with the NTU-method
- Hot water accumulator without thermal stratification
- BTES consisting of 49 boreholes with 10 meters spacing and 200m depth.

Because of the hourly time steps the volume of the accumulator is larger than it normally would be. A synthetic load profile has been applied to the accumulator and the heat exchanger circuit. The outlet temperature of the heat exchanger (ThoutVVX) and the outlet temperature of the accumulator (ToutACC) are together with the air temperature inputs to a function that determines whether the system will be operating in a cooling or a heating mode, see figure 24.

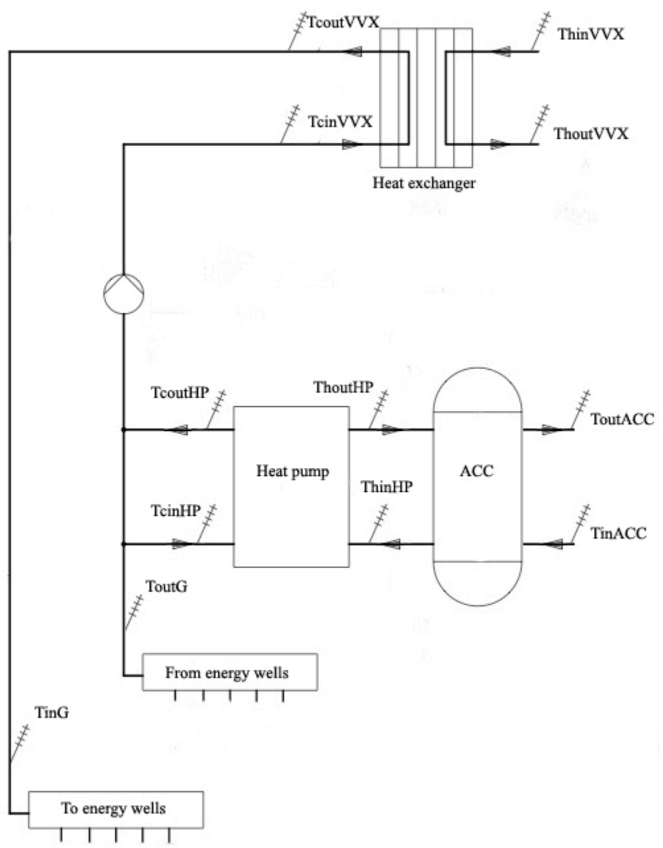


Figure 24. Flow schedule over simplified GSHP system.

In heating mode the heat exchanger is ignored and the heat pump operates with full mass flow. In cooling mode the fluid is circulated with a tree step variable mass flow through the heat exchanger without interaction from the heat pump. While the heat pump is off there is no circulation of fluid through it. If there is no cooling or heating load the mass flow in the ground circuit is sett to the lowest value. The heat carrier fluid is brine while the cold and hot circuits contain water.

It is desirable to be able to simulate the system for a long time to see the effects of thermal interaction between the boreholes. To achieve computational speed the temperature influence (T_p) has been evaluated every second week. In comparison to evaluating T_p for every time step for a one year simulation this reduces the simulation time from 2 hours to roughly 4 minutes. A 10 year simulation takes 70 minutes with this configuration.

The load profile is obtained from Bernier (2004) and has been modified to represent an imbalanced yearly load. The positive values in figure 25 are heat that is extracted from the ground and the negative are heat injected to the ground. A profile for the outside temperature has been constructed with values from the Swedish Meteorological and Hydrological Institute (SMHI). The profile is from a weather station that measures the temperature every third hour in Malmö, Sweden. Interpolation is then done to get the values for every hour, see figure 26.

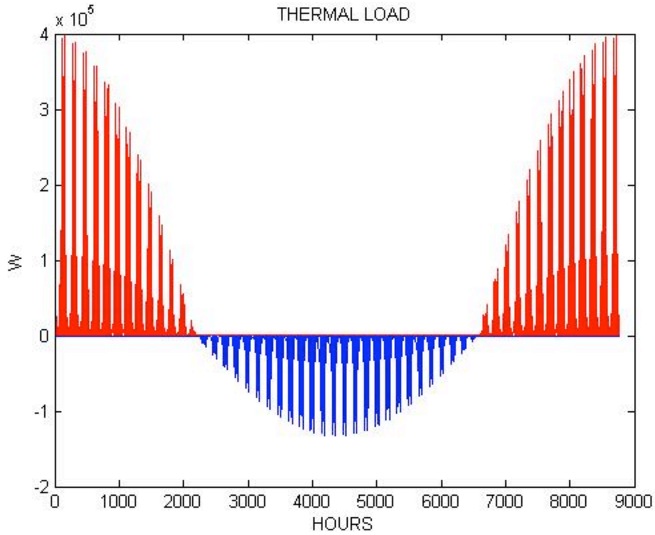


Figure 25. Synthetic load profile Bernier (2004).

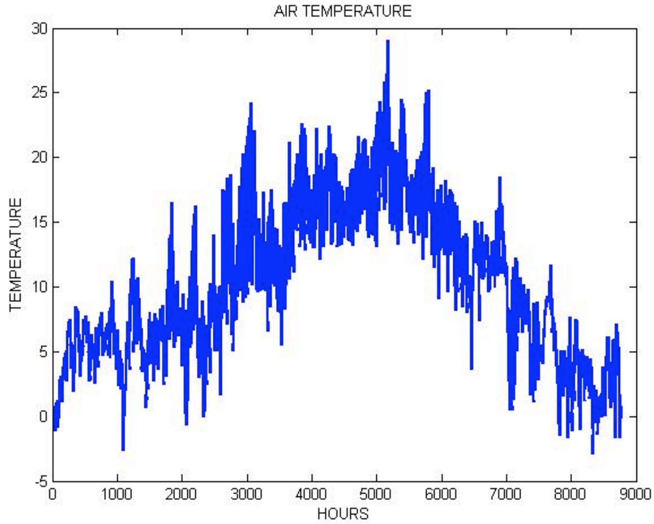


Figure 26. Outside Air temperature (SMHI).

The performance of the heat pump has been obtained through a software called ECAT2 and correspond to a CARRIER 30HXC090-PH. For more information on the heat pump model see appendix.

The results from this simulation can be discussed and should be thought of mainly as a demonstration of the possibility to perform hourly simulations of a GSHP-system with the BTES-model. As can be seen in figure 27 the temperature in the BTES will subsequently sink as a result of the imbalanced load. The outlet from the accumulator should be kept at a temperature around 50-55 ^{o}C. In figure 28 the temperature drops at peak load. This would in reality be treated with an additional heat source. The outlet temperature in this simulation is kept within 11-17^{o}C. Also here the temperature would in reality be lowered during summer through cooling with the heat pump.

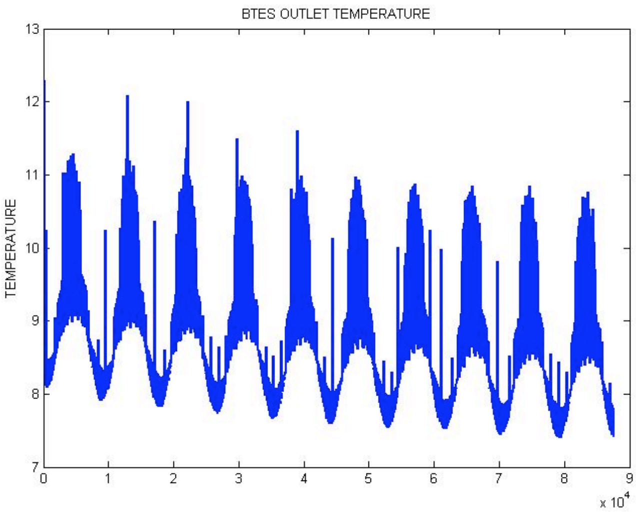


Figure 27. BTES outlet temperature.

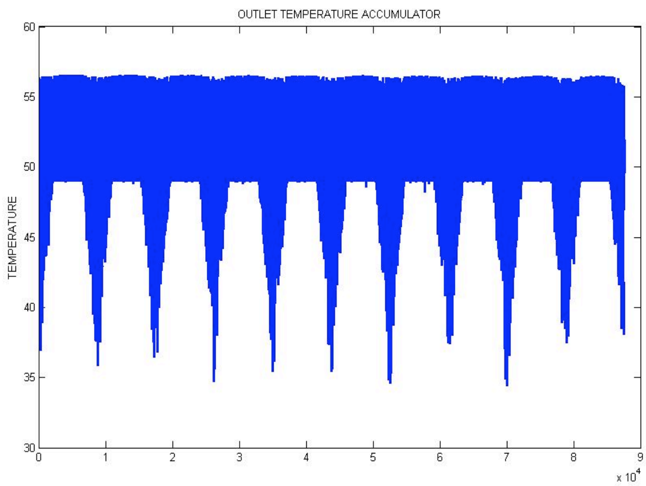


Figure 28. Accumulator outlet temperature.

6 Discussion

There are many things that come to mind when working with a thesis project like this, questions arise and solutions are found along the way. What might be simple at a glance can prove to be cumbersome and difficult and vice-versa. Some questions remain unanswered and these will here be summarized and discussed as a starting point for further research.

The basic GSHP system is a quite simple technology. It is reasonable to believe that there are room for improvements, especially in the design and operation strategy of the BTES. With an increased size and popularity of GSHP systems the demand for simulations and life cycle analysis will most likely continue.

A common problem in the literature has been cooping with insufficient computational speed. This is the main reason why lots of research has been done around analytical solutions for BTES systems. Even the analytical solutions have been proved to be cumbersome if they are not treated properly. For example aggregation and symmetry has to be considered. Some of the simplifications made to reduce the computational burden will undoubtedly have impact on the result. In the approach by Kavanaugh and Rafferty (1997) they assume that the surrounding boreholes lock in the heat. The method can be discussed but a benefit of this assumption is that the calculations can be performed quickly with only a few individual distances to consider. It would be very beneficial for the computational speed if some kind of average distance between the boreholes could be used for the calculations. As presented in the result there was a difference between the unaggregated and the aggregated temperatures, see figure 18. It was not expected to be any visible difference and it is uncertain if this difference is due to a computational error. It should be mentioned that increased computer speed would probably affect the way equations are treated.

Another reason for analytical solutions would be the simplicity and the flexibility. It is easy and quick to try configurations that are outside what existing software allow.

There are many things that would be interesting to try. For example simulations with both varying mass flow and a load that varies between the boreholes. If the purpose is to store excess heat from the summer it would be reasonable to believe that a higher performance would be achieved if more energy would be injected to the interior boreholes. This is being tested at the Drake Landing solar community in Alberta, Canada (Sibbit et al 2007).

In the simulations presented for monthly load values, a constant mass flow has been used when calculating the borehole resistance. In reality the mass flow would vary depending on the cooling or heating load as in the hourly simulations. It is also assumed that the collector shank spacing and the distance to the borehole wall are known. In reality these distances will vary along the borehole length. Only simulations with a single u-tube have been performed, it would as well be interesting to simulate a system with double U-tubes or with combinations of single and double.

It was earlier mentioned, in reference to the literature that the cylindrical source theory has proven to be valid on a short timescale. However it would be interesting to validate the model against short time step simulations. This could be done either against the duct-storage model or the short response model by Yavuzturk.

The 10-year GSHP-system simulation is simplified and should as mentioned be thought of mostly as a demonstration of the BTES-model for short time steps. The weakest link of this system would probably be the heat pump model. As described earlier there are several different methods to simulate a heat pump. The equation fit is simple but the physical connection is lost and the method can produce unrealistic results if not treated carefully. A more sophisticated model would increase the credibility of the GSHP-system.

For optimization and life cycle analysis of GSHP-systems this simplified model could be further improved and validated. Another way would be to incorporate the BTES-model into existing energy system simulation software.

To further validate the BTES-model measurements should be conducted on existing systems. Because of the nature of the heat transfer process these measurements would have to be made during a long time period. Large GSHP systems are usually equipped with temperature and flow gauges what for this should be a matter of just gathering and storing the information in an appropriate way.

One of the main assumptions is that the heat transfer is purely conductive. This is a widely used, but nevertheless it can be faulty if there is a noticeable ground water movement. It is also very important to have an appropriate estimate of the thermal conductivity in the ground and the initial ground temperature because these parameters have a great impact on the results.

Neither of the methods encountered in the literature account for recharge of the BTES by geothermal energy. This is probably reasonable during the conditions in Sweden but where higher geothermal heat flux is encountered the validity would need to be discussed.

Finally it is worth mentioning that the results from a simulation of a BTES will never be better than the data provided. With this it means that it is often difficult to estimate the real energy consumption of a building and even more when large domestic buildings are considered.

7 Conclusions

As described in this thesis the GSHP system consists of few main components. The ground source heat exchanger, a heat pump a heat exchanger and a circulation pump. It is a mature technology that has meet wide acceptance and that has been implemented in many countries. The basic idea is simple and has been used for a long time. The simplicity of GSHP systems together with the lack of good simulation tools could be thought of a sign that their potential could be radically improved.

In this thesis a BTES-model has been developed based on previous research and experimental simulations. It uses the cylindrical source theory and the finite line source, and has shown good consistency with existing dimensioning software and could be used for simulation and optimization of BTES.

To demonstrate the BTES-model in a realistic simulation a simplified large scale GSHP-system with a heating dominated load has been simulated with hourly time steps for 10 years. The simulation time is 70 minutes on a standard laptop, which should be acceptable.

To perform full system analysis and life cycle analysis of GSHP-systems the BTES-model could either be used in a more advanced GSHP-model in Matlab or be incorporated into other energy system simulation software.

References

Alvares, H (2006): Energiteknik, 3rd edition, Studentlitteratur, Lund Sweden

Arlanda, http://www.arlanda.se/sv/Information-om/Miljoarbete/Minskade-utslapp-av-koldioxid/Energi/Akvifaren/. Access date 2 June 2009

Bandyopadhyay,G., W.D. Gosnold., M. Mann (2006): Thermal study of a large ground heat exchanger in clay soil in the cold weather environment of northern USA: some initial findings. School of Engineering and Mines, University of North Dakota

Bernier, M (2000): A Review of the Cylindrical Heat Source Method for the Design and Analysis of Vertical Ground-Coupled Heat Pump Systems. 4th International Conference on Heat Pumps in Cold Climate, Aylmer, Québec

Bernier, M (2001): Ground-Coupled Heat Pump System Simulation, ASHRAE Transactions 107 (1): 605-6015

Bernier, M., P.Pinel, R, Labib and R.Paillot (2004): A multiple load aggregation for annula hourly simulations of GCHP systems, HVAC&R Research 10 (4):471-87

Carslaw, H.S., J.C. Jaeger (1947): Conduction of heat in solids, Oxford University Press, Oxford

Deerman, P.E. and S.P. Kavanaugh (1991): Simulation of vertical U-tube ground-coupled heat pump systems using the cylindrical heat source solution, ASHRAE Transactions 97 (1): 287–295

DLSC. Drake landing solar community. http://www.dlsc.ca/borehole.htm. Access date 2 June 2009

Eskilson, P (1987): Thermal analysis of heat extraction boreholes. Lund, Sweden, University of Lund. Department of Mathematical Physics

Gao, Q., Ming. Li., Ming. Yu., J.D. Spitler., Y.Y. Yan (2008): Review of development from GSHP to UTES in China and other countries. Renewable and Sustainable Energy Reviews 13 (2009) 1383-1394

Gehlin, S (2002): Thermal Response test: method development and evaluation. Ph.D dissertation Luleå University of Technology

Geoenergigruppen (2009): Geoenergi –Vägval Energi.Geotec. http://www.geotec.se/pressrum/Geoenergi%20-%20vägval%20energi.pdf. Access date 13 may 2009

H. Zeng., N. Diao., Z. Fang (2002): A finite line-source model for boreholes in geothermal heat exchangers, Heat Transfer Asian Research 31 (7) 558–567

Hellström, G (1991): Ground heat storage: thermal analysis of duct storage systems. Lund, Sweden, University of Lund. Department of Mathematical Physics

Ingersoll, L.R. and H.J. Plass (1948): Theory of the ground pipe heat-source for the heat pump, Heating, Piping, Air Conditioning Journal 20 (7) 119-122

Ingersoll, L.R., Zobel, O.J., Ingersoll, A.C (1954): Heat Conduction With Engineering, Geological, And Other Applications, University of Wisconsin Press

Jin, H (2002): Parameter estimation based models of water source heat pumps. Ph.D dissertation, Oklahoma State University

Katsura, T., K. Nagano., S. Takeda (2007): Method of calculation of the ground temperature for multiple ground heat exchangers, Applied Thermal Engineering 28 (14-15):1995-2004

Kavanaugh, S.P., K. Rafferty (1997): Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings. Atlanta: ASHRAE

Lamarche, L., and B. Beauchamp (2006): A new contribution to the finite line-source model for geothermal boreholes. Energy and Buildings 39 (2007) 188-198

Nagano, K., T. Katsura., S. Takeda (2006): Development of a design and performance prediction tool for the ground source heat pump system, Applied Thermal Engineering 26 (14-15):1578-1592

Omer, A.M (2006): Ground-source heat pump systems and applications. Renewable and Sustainable Energy Reviews 12 (2008): 344-371

SGU (2007): Normbrunn-07 : Att borra brunn för energi och vatten- en vägledning. Davidsons tryckeri AB, Växjö

SMHI. SMHI_3hours_clim_5235.http://data.smhi.se/met/climate/time_series/3hours/. Access date 10 may 2009

SVEP (2004): Fakta om värmepumpar och anläggningar http://www.geotec.se/upload/cms/file/svep_faktavarmepump.pdf. Access date 15 may 2009

Swedish Energy agency (2008): Energiläget 2008. http://www.swedishenergyagency.se/web/biblshop.nsf/FilAtkomst/2053.pdf/$FILE/2053.pdf?OpenElement. Access date 15 april 2009

Sundén, B (2006): Värmeöverföring. Studentlitteratur, Lund, Sweden

Sundberg, J (1988): Thermal properties of soils and rocks, Ph.D thesis. Publ. A57, Department of Geology, Chalmers University of Technology, S-412 96 Göteborg, Sweden

Sibbit, B., T. Onno., D. McCleanahan., J. Thornton., A. Brunger., J.Kokko., B.Wong (2007): The Drake landing solar community project – early results. CANMET Energy technology centre, Natural Resources Canada, Ottawa, Canada

Tang, C.C (2005): Modeling packaged heat pumps in a quasi-steady state energy simulation program. Master Thesis. Oklahoma State University

Yavuzturk, C (1999): Modeling of vertical ground loop heat exchangers for ground source heat pump systems. Ph.D dissertation, Oklahoma State University

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

Young, D.F., B.R. Munson., T.H. Okiishi (2004): A brief introduction to fluid mechanics: 3rd edition. John Wiley & Sons

Zeng, H., D. Nairen., F. Zhaohong (2003): Heat transfer analysis of boreholes in vertical ground heat exchangers, International Journal of Heat and Mass Transfer 46 (23): 4467-4481

Appendix

Borehole resistance.

Nomenclature:

```
kg     % ground thermal conductivity (w/m,K)
L      % Active length of the borehole (m)
rb     % borehole radius (m)
rp     % pipe outer radius (m)
rpi    % pipe inner radius (m)
D      % half spacing of U-tube shanks (m)
Kb     % grouth thermal conductivity (w/m,K)
kp     % pipe thermal conductivity (w/m,K)
m      % massflow heat carrier fluid kg/s
c      % specific heat of heat carrier fluid (J/kg,K)
```

Equations:

```
%-------------calculate borehole resistance -------------------------

Rp=log(rp/rpi)/(2*pi*kp); % Hellström (1991) Ground heat storage page
75 eq 8.2


% Zeng (2002), Heat transfer analysis of borehole in vertical ground
heat
% exchangers, equation number refers to Zeng (2002)



R11=(1/(2*pi*kb))*( log(rb/rp)-((kb-kg)/(kb+kg))*log((rb^2-
D^2)/(rb^2)))+ Rp;                                            %eq (5)

R13=(1/(2*pi*kb))*( log(rb/(2*D))-((kb-
kg)/(kb+kg))*log((rb^2+D^2)/(rb^2)));                         %eq (5)

S1=(m*c/L)*(R11+R13);                                         %eq (9)

S12=(m*c/L)*(R11^2-R13^2)/R13;                                %eq (9)

beta=sqrt(1/S1^2 + 2/(S1*S12));                               %eq (10)

f=(beta*S1*cosh(beta)-sinh(beta))/(beta*S1*cosh(beta)+sinh(beta));
%eq (10)

Rb=(L/(2*m*c))*(1+f)/(1-f)                                    %eq (27)
```

The full derivation of the equations can be found in Zeng (2002).

In figure A.1 the borehole resistance is plotted as a function of shank space

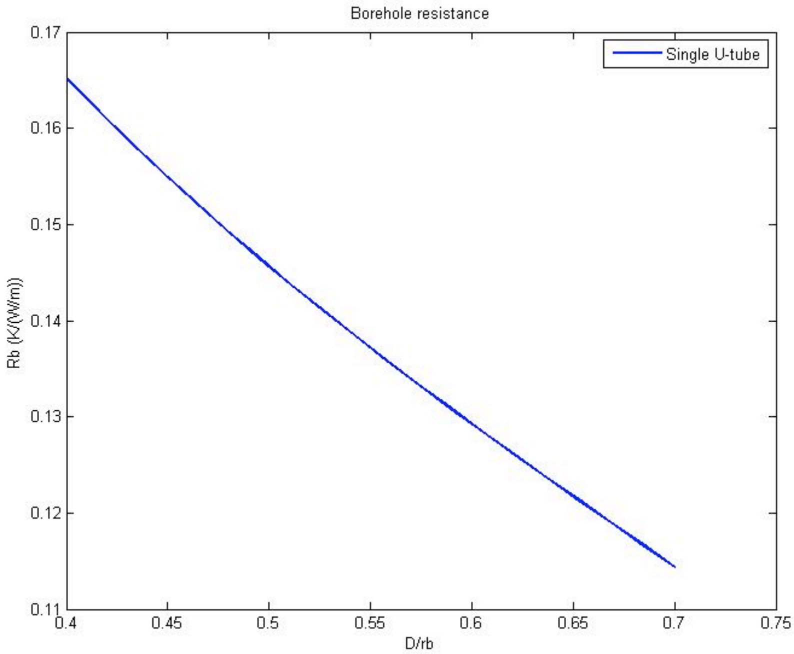


Figure A.1. Relations of borehole thermal resistance to the U-tube shank spacing for a single U-tube.

Simulation Parameters

Simulation parameters used when comparing with SBM for 20 boreholes.	
L=200 m	m=0.47 kg/s / borehole
k_g =2.8 W/m.K	T_s=9.5 ^oC
k_b=0.6 W/m.K	D=0.05 m
k_p=0.42 W/m.K	r_p=0.02 m
α= 0.1052 m^2 /day	r_{pi}=0.0163 m
d_b=0.16 m	c_{pf}= 4250 j/kg.K (heat carrier fluid)
W= 0.07 W/m^2	c_{pg}= 2300 j/kg.K

Table A.1 Simulation Parameters used when comparing with SBM for 20 boreholes in figure 19

Month	Heating (MWh)	Cooling (MWh)
January	65.233	0
February	59.533	0
March	46.55	3.41
April	27.233	7.285
May	13.933	15.0350
June	0	25.733
July	0	33.328
August	0	31.313
September	0	19.222
October	20.9	13.33
November	31.9833	6.355
December	51.3	0

Table A.2 Load profile that has been used when comparing with SBM in figure19.

Equation fit heat pump model

The equation fit model is constructed as a set of coefficients that are generated with a least square curve fitting to catalog data. More information of the heat pump model and its performance can be found in Tang (2005).

Cooling mode:

$$\frac{Q_{H\,c}}{Q_{H\,c}} A1 + A2\frac{T_{evap}}{T_{ref}} + A3\frac{T_{cond}}{T_{ref}} + A4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + A5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

$$\frac{Power_{c}}{Power_{c.ref}} B1 + B2\frac{T_{evap}}{T_{ref}} + B3\frac{T_{cond}}{T_{ref}} + B4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + B5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

$$\frac{Q_{Lc}}{Q_{Lc}} C1 + C2\frac{T_{evap}}{T_{ref}} + C3\frac{T_{cond}}{T_{ref}} + C4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + C5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

Heating mode:

$$\frac{Q_{H\,h}}{Q_{H\,h}} D1 + D2\frac{T_{evap}}{T_{ref}} + D3\frac{T_{cond}}{T_{ref}} + D4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + D5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

$$\frac{Power_{h}}{Power_{h.ref}} E1 + E2\frac{T_{evap}}{T_{ref}} + E3\frac{T_{cond}}{T_{ref}} + E4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + E5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

$$\frac{Q_{Lh}}{Q_{Lh}} F1 + F2\frac{T_{evap}}{T_{ref}} + F3\frac{T_{cond}}{T_{ref}} + F4\frac{\dot{m}_{evap}}{\dot{m}_{evap.ref}} + F5\frac{\dot{m}_{cond}}{\dot{m}_{cond.ref}}$$

The reference values for the loads the power and the mass flows are taken as the values when the heat pump is operating at its maximum performance in either heating or cooling mode. The temperatures are in Kelvin and T_{ref} is 273.