KTH Industrial Engineering and Management

Transient modeling of a high temperature borehole thermal energy storage coupled with a combined heat and power plant

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KTH Industrial Engineering and Management	**Master of Science Thesis EGI 2017: 0106 MSC** **Transient modeling of a high temperature borehole thermal energy storage coupled with a combined heat and power plant** Malin Malmberg	
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Abstract

Coupling High-Temperature Borehole Thermal Energy Storages (HT-BTES) with existing Combined Heat and Power (CHP) systems is a promising approach to increase energy efficiency of district energy systems through recovery of otherwise wasted heat. This solution is currently being discussed in Sweden by the company Tekniska Verken in Linköping AB, for storing waste heat from their CHP operation in summer in a HT-BTES and to utilize it during peaks in winter. This would increase the flexibility between energy supply and demand in one of their plants. The available supply temperature during charge of the BTES is around 95°C. There is, though, still limited experience of HT-BTES operation with just a few installations throughout the world.

The aim of this Master´s thesis has been to evaluate a potential system design configuration for effective extraction and storage of waste heat from the Gärstadverket CHP-plants in connection to a HT-BTES. Data from previous operation of the CHP-plants and an existing TRNSYS model, developed at KTH and Bengt Dahlgren AB based on the well-known DST approach (Duct Ground Heat Storage Model), was used as a starting point to the development of a new, more complete model that includes a heat pump. The heat pump model was developed from manufacturer's data for a non-standard 50 MW heat pump system using R717 as refrigerant. As an additional objective, design and operational experience of already existing HT-BTES installations has been compiled and analyzed.

The BTES design were simulated with varied number of boreholes and borehole depth. The system was furthermore simulated with two different borehole heat exchangers (BHEs): double U-pipes and coaxial. Based on the results three optimized designs were found: 1 400 boreholes with double U-pipes and a borehole depth of 300 m, 1 300 boreholes with coaxial BHEs and a borehole depth of 300 m, and a design with 1 500 boreholes and 275 m borehole depth – all three designs with a borehole spacing of 5 m and with loops of 3 boreholes connected in series. The three BTES designs showed similar results with a potential to store around 107 GWh/year and to extract around 93 GWh/year with the use of a GSHP. The resulting discharge temperature from the BTES ranges between 40-60°C, and up to 70°C in the initial discharge period in the tenth simulation year. Further investigation is though needed regarding if there are any coaxial BHE available on the market that can work with the high temperatures in the BTES.

Sammanfattning

Koppling av högtemperatur-borrhålslager (HT-BTES) med befintliga kraftvärmeverk (CHP) är ett lovande tillvägagångssätt för att öka energieffektiviteten i fjärrvärmesystem genom återvinning spillvärme. Denna lösning diskuteras för närvarande i Sverige av Tekniska Verken i Linköping AB, för att lagra spillvärme från kraftvärmeproduktion sommartid i en HT-BTES och utnyttja denna under effekttoppar på vintern. Detta skulle öka flexibiliteten mellan energiförsörjning och efterfrågan i en av deras anläggningar, Gärstadverket. Den tillgängliga framledningstemperaturen under laddning av borrhålslagret är ca 95 ℃. Det finns dock fortfarande begränsad erfarenhet av HT-BTES med bara några få installationer i drift över hela världen.

Syftet med detta masterexamensarbete har varit att utvärdera en potentiell systemkonfigurationskonfiguration för effektiv utvinning och lagring av spillvärme från Gärstadverkets kraftvärmeverk kopplat till ett HT-BTES. Data från tidigare drift av kraftvärmeverket och en befintlig TRNSYS-modell, utvecklad hos KTH och Bengt Dahlgren AB baserat på den välkända DST-metoden (Duct Ground Heat Storage Model), användes som utgångspunkt för utvecklingen av en ny, mer komplett modell som inkluderar en värmepumpsmodell. Värmepumpsmodellen utvecklades utifrån data från en värmepumpstillverkare för ett icke-standardiserat 50 MW värmepumpsystem, med R717 som kylmedium. Som ett ytterligare mål har designparametrar och erfarenheter från drift av redan befintliga HT-BTES installationer sammanställts och analyserats.

BTES-designen varierades genom simuleringar med olika antal borrhål och borrhålsdjup. Systemet simulerades fortsatt med två olika borrhålsvärmeväxlare (BHE): dubbla U-rör och koaxiala BHE. Baserat på resultaten hittades tre optimerade BTES-geometrier: 1 400 borrhål med dubbla U-rörs BHE och 300 m borrhålsdjup, 1 300 borrhål med koaxiala BHE och 300 m borrhålsdjup samt en design med 1 500 borrhål med dubbla U-rör och ett borrhålsdjup på 275 m – alla tre konfigurationer med ett borrhålsavstånd på 5 m och borrhålsloopar med tre borrhål kopplade i serie. De tre BTES-geometrierna visade liknande resultat med potential att lagra cirka 107 GWh / år och att extrahera runt 93 GWh / år med användning av en värmepump. Den resulterande urladdningstemperaturen från borrhålslagret varierar mellan 40-60 °C och upp till 70 °C i början av urladdningsperioden under det tionde simuleringsåret. Vidare studie krävs dock för att undersöka tillgängligheten av koaxiala BHE på marknaden som kan fungera med de höga temperaturerna i borrhålslagret.

Acknowledgement

This master thesis work has been performed as a collaboration between the Royal Institute of Technology (KTH), Bengt Dahlgren AB and Teknsika Verken in Linköping. I would firstly like to thank José Acuña from Bengt Dahlgren and Henrik Lindståhl from Tekniska Verken for making this collaboration possible. I would also like to direct a special thanks to Willem Mazzotti for all the support and good advisement during the project. I would furthermore like to thank my supervisor at KTH, Alberto Lazzarotto, for valuable discussions and advisement. Thanks also to Nicky Cowan at Star Renewable Energy.

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Abbreviations

BHE	Borehole Heat Exchanger
BH	Boreholes
BTES	Borehole Thermal Energy Storage
CHP	Combined Heat and Power
DH	District heating
DST	Duct Ground Heat Storage Model
DTRT	Distributed Thermal Response Test
GHP	Geothermal heat pump
GSHP	Ground Source Heat Pump
GWP	Global Warming Potential
HT	High Temperature
ODP	Ozone Depleting Potential
RPM	Revolutions per minute
SPF	Seasonal Performance Factor
TRNSYS	TRaNsient SYstems Simulation Program
TRT	Thermal Response Test
UTES	Underground Thermal Energy Storage

Nomenclature

$\dot{Q}_1$	Heating capacity on the condenser side of the heat pump [MW]
$\dot{Q}_2$	Cooling capacity on the evaporator side of the heat pump [MW]
$\dot{E}_k$	Compressor power (shaft] [MW]

$\dot{E}_{k,tot}$	Total compressor power [MW]
COP	Coefficient of Performance [-]
COP_1	Coefficient of Performance in heating mode [-]
COP_2	Coefficient of Performance in cooling mode [-]
c_p	Specific heat capacity [J/kgK]
ΔP	Pressure drop [Pa]
ΔPf	Pressure drop due to friction [Pa]
ΔT	Temeprature difference [K]
f	Friction factor [-]
Re	Reynolds number [-]
w	Velocity [m/s]
D_h	Hydraulic diameter [m]
ρ	Density [kg/m^3]
η	Efficiency [-]
$\dot{V}$	Volumetric flow rate [m^3/s]
T	Temperature [K]
μ	Dynamic viscosity [kg/ms]
v	Kinematic viscosity [m^2/s]
A	Area [m^2]
L	Pipe length [m]
D	Diameter [m]
N	Number of boreholes
d	Distance between boreholes [m]
h	Borehole depth [m]
ϑ_m	Logarithmic mean temperature difference [K]
$\dot{m}$	Mass flow rate [kg/s]
$\dot{E}_{pump}$	Pumping power [kW]
UA-$value$	Overall heat transfer coefficient [kW/K]

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1 Introduction

Borehole thermal energy storage (BTES) for industrial purpose has received increased attention in recent years. BTES may be used to store waste heat and/or solar heat, as well as increase flexibility between supply and demand in small to large energy systems. Introducing renewable hybrid systems working in synergy with geothermal energy storages is a promising approach in order to replace non-renewable energy sources and decrease greenhouse gas emissions. This can effectively be done for example by connecting geothermal energy storages to already existing CHP plants.

Tekniska Verken in Linköping AB (Sweden) is a municipally owned company with responsibility of the district heating production and distribution in the municipality of Linköping. A large part of the district heating production takes place in a combined heat and power (CHP) plant, Gärstadverket, mainly fed by municipal waste. Since Gärstadverket is a part of the waste management system in the area it is run on full capacity full time. Hence, cooling is required of surplus heat produced during summer when the district heating demand is low. Furthermore, due to uneven heating demand over the year the capacity of Gärstadverket is insufficient during winter peak heat demand. The heat production is therefore complemented by another CHP-plant, KV1, fueled by biofuels, coal, and oil. The operation of KV1 is costly and contributes to a larger carbon footprint. Solutions to phase out KV1 from the heat and power production in Linköping are currently being investigated.

Based on this background it has been questioned to study the possibility of shifting some of the surplus heat from summer to the winter operation. One suggested solution is to connect a seasonal HT-BTES (so called HEFAISTOS) to the Gärstadverket at Tekniska Verken in Linköping. Parts of the surplus heat produced by burning the municipal waste in the summer, which is around 260 GWh per year, could be stored in the BTES to be utilized in the district heating network during the peak heat demand in winter. This would increase the flexibility of the energy system at Tekniska Verken and furthermore be a step towards their goal of phasing out fossil fuels.

1.1 Aim and objectives

The aim of this master thesis is to evaluate a potential system design configuration for effective extraction and storage of waste heat from the Gärstadverket CHP-plants in connection to a high temperature borehole thermal energy storage. The objectives of the master thesis are:

1. Identification of feasible heat pumping technology for lifting the discharge power and temperature from the storage level to the demand level
2. Development of a heat pump model based on performance data from a heat pump manufacturer
3. Optimization of the size of the BTES including:
    - Evaluation of potential charge and discharge power with regard to the size of the BTES
    - Evaluation of the hydraulic implications (pumping power, pressure drop)
    - Estimation of investment and operational costs for the BTES and the heat pumps

As an additional objective of this thesis important design parameters and operational experience will be evaluated regarding already existing HT-BTES installations to find common denominators and useful experience within the field.

1.2 Methodology

This project was firstly conducted by performing a literature review of HT-BTES systems and different alternative heat pump technologies suitable for high temperature systems. The review included literature on existing HT-BTES systems regarding design and the best practices utilized in this area. This was done in order to find a factual basis regarding design, construction and operation of existing HT-BTES.

After identification of a feasible heat pump technology for the application of interest, a heat pump manufacturer was contacted to retrieve operational data for the heat pump. The results of discharge temperature and power levels from the pre-studies for connecting a HT-BTES (so called HEFAISTOS) to Gärstadverket in Linköping, as well as the desired output power levels and temperatures to be delivered to the CHP-plant, was used as a basis for the dimensioning of the heat pump. The pre-studies includes simulated results from a TRNSYS model developed at KTH and Bengt Dahlgren AB with the well-known DST model (Duct Ground Heat Storage Model). The software TRNSYS is a widely used transient modeling tool that provides the opportunity to simulate different types of systems in a flexible way, and has been used in this study to simulate the system performance.

A heat pump model was created in excel using the least square method to minimize the error (RMSD) between the heat pump manufacturer data and the model. This model was integrated in the already existing TRNSYS model developed at Bengt Dahlgren AB. The TRNSYS model was further developed with new control strategies for the BTES and the heat pump to simulate the performance of the system for a 10 year period. The size of HEFAISTOS was evaluated in TRNSYS through sensitivity analyses for different design parameters and with some economic analysis.

2 Theoretical background

Underground thermal energy storage (UTES) are sensible storage systems that can be used in a variety of applications, e.g. for actively store waste heat from industrial processes, CHP or solar thermal. The three main designs of UTES are aquifer storages, storage in rock caverns and borehole storages (Hellström, 1991). Depending on if the storage temperature is lower or higher than 40-50 °C UTES systems can furthermore be classified as low- respectively high temperature storages (Sang, 2013).

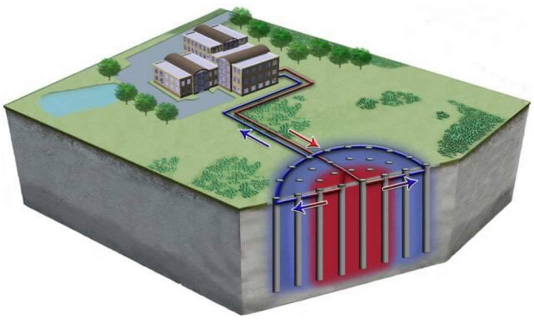
In this study, due to its flexibility and large applicably, a high temperature borehole thermal energy storage (HT-BTES) coupled to a ground source heat pump (GSHP) is considered to be connected to the combined heat and power plant (CHP-plant), Gärstadverket. The other two storage technologies, aquifers and rock caverns, have some restricting limitations. An aquifer storage could be limited both by size, legal or environmental concerns (Sang, 2013). Furthermore, suitable aquifers only exist on 10-15 % of the Swedish land area (Barth, et al., 2012). The construction for large underground rock caverns requires extremely specific site conditions and high quality rock which is usually not available and this is the reason why the actual application of such system is very limited (Nordell, 1994). Overall the design and principles for aquifer and rock cavern differs greatly from duct storage systems and will not be further discussed in this thesis. In the following sections of this chapter, a theoretical background of HT-BTES and GSHP-systems will be outlined.

2.1 Borehole systems

A BTES can be constructed in rocks, clays and soils and uses the ground as a heat storage medium through vertical heat exchangers. BTES is foremost used in the industry and for large properties having a demand for both heating and cooling (Barth, et al., 2012). BTES can be used in most locations and this is why it is one of the most applicable and popular form of UTES systems. The procedure for authority approvals is usually very simple, it is easy to construct and requires limited maintenance (Sang, 2013). It may however not deal with large load variations (Barth, et al., 2012).

The most favorable geometries of the BTES are cylindrical and cubical (Reuss, 2015). A cylindrical geometry, especially, allows for reduced heat losses at both the surface and the edges of the storage (Mangold & Deschaintre, 2015). For large commercial and industrial GSHP systems in Sweden the borehole depth is usually rather deep, commonly 200-300 m. Boreholes deeper than 300 m are uncommon and there is a lack of information of such systems (Gehlin, et al., 2016). BTES is constructed with a small distance between the boreholes, usually 4-6 m distance, for the boreholes to interact with each other as the storage volume is alternately heated and cooled (Barth, et al., 2012).

To decrease storage losses to the surrounding ground boreholes can furthermore be coupled in series in several parallel flow strings. Thereby horizontal thermal stratification can be induced by circulating the heat carrier fluid from the center to the outer parts of the storage during charging of the BTES, and in the contrary flow direction during discharge. In this way the temperature is kept higher at the center of the storage than at the edges (Reuss, 2015). This is illustrated in Figure 1 and Figure 2 below, which shows summer and winter operation respectively of a BTES system controlled with thermal stratification (Underground Energy, 2009).



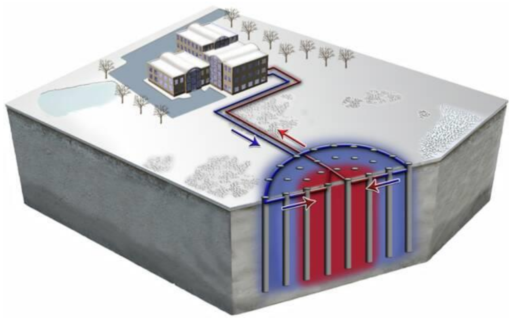


Figure 1 BTES summer operation
Source: Underground energy, LCC, 2009 (Underground Energy, 2009).

Figure 2 BTES winter operation
Source: Underground energy, LCC, 2009 (Underground Energy, 2009).

2.1.1 Collector designs

The heat carrier fluid exchange heat with the ground by circulating within a pipe installation inserted in the vertical borehole. Such system is called borehole heat exchanger or collector. Two fundamental collector designs exist, U-pipe and coaxial. The U-pipe design is the most common design for single BHE installations (Acuña, 2013). It dominates around 99 % of the Swedish market, despite its poorer thermal performance compared to the coaxial design (Björk, et al., 2013). The U-pipe consists of two pipes connected in the bottom and forms a closed system. In Figure 3 and Figure 4 the geometry of an U-pipe can be seen from top respectively side view. For a better heat exchange up to three U-pipes can be installed in the same borehole which decreases the thermal resistance and the head loss in the pipes (Gehlin, 2002).

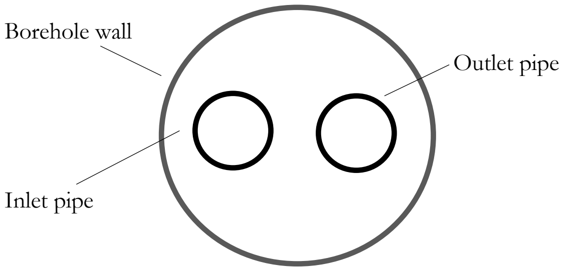


Figure 3 U-pipe from top view (Free from (Acuña, 2013))

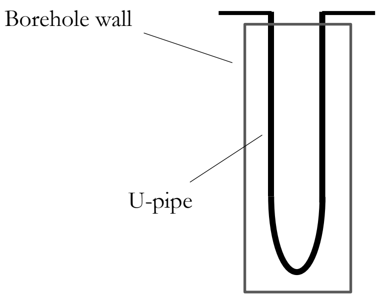


Figure 4 U-pipe from side view (Free from (Acuña, 2013))

For coaxial collectors there exist different geometries of which the tube-in-tube geometry is the simplest one. It consists of a central pipe inserted inside a larger tube, or liner, and an annular flow channel is formed between them. A coaxial BHE can though also be designed as an open system, i.e. without outer tube or liner (Acuña, 2013). In an open system a higher heat transfer rate can be obtained as the heat carrier fluid is in direct contact with the borehole wall. The drawback is that the local geochemical and geohydrological conditions often are unfavorable for open system installations (Hellström, 1989). In Figure 5 and Figure 6 the geometry of a coaxial tube-in-tube collector can be seen from above respectively from the side (Acuña, 2013). The coaxial design is characterized by that heat exchange only occurs in the downward or upward flow channel and two different flow directions may be used for charge respectively recharge of the storage (Gehlin, 2002).

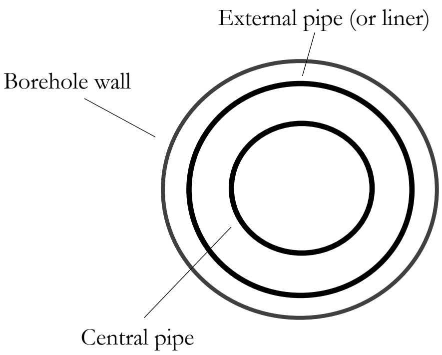


Figure 5 Coaxial collector from top view

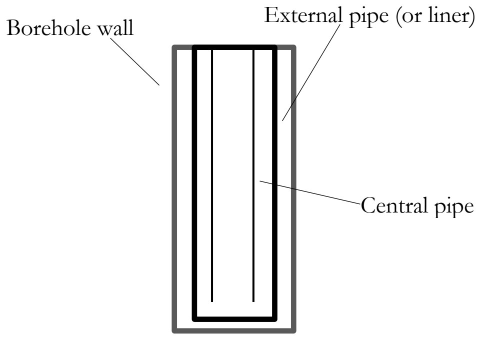


Figure 6 Coaxial collector from side view

Collector materials

Except the design of the collector the physical properties, such as maximum operating temperature and pressure, of the collector material may have a large influence on the applicability. Especially if designing a HT-BTES system. Plastic pipes have shown to be favorable for BTES installation due to its flexibility, high corrosion resistance and relatively low price. In Table 1 below the physical properties of different BHE pipes that is commonly used can be seen (Reuss, 2015).

Table 1 Physical properties of different BHE pipes. Source: M. Reuss, 2015 [14].

Material	Continuous operating temperatures (50 years)	Peak temperature (during 1 year)	Thermal conductivity (W/mK)
PE100	40°C at 11.6 bar	70°C at 6.2 bar	0.42
PE100-RC	40°C at 11.6 bar	70°C at 6.2 bar	0.42
PE-RT	70°C at 6.5 bar	95°C at 5.2 bar	0.42
PA	40°C	70°C	0.24
PB	70°C at 12.1 bar	95°C at 8.1 bar	0.22
PE-X	70°C at 8.5 bar	95°C at 6.8 bar	0.41

2.1.2 Hydrodynamic considerations

The information in this section is taken from the book *Introduction to Heat Transfer* by Incropera et al. (Incropera, et al., 2007). When circulating the heat carrier fluid in the boreholes it is desirable to have a turbulent flow which result in a lower thermal resistance between the heat carrier fluid and the collector pipes compared to laminar flow. Turbulent flow is achieved for Reynolds numbers (Re) above 2300, but for fully developed turbulent flow much higher numbers are required (>10 000). The Reynolds number can be calculated according to Eq. 1.

$$\mathrm{Re} = \frac{w \cdot D_h}{v} = \frac{\rho \cdot w \cdot D_h}{\mu \cdot} = \frac{\rho \cdot \dot{V} \cdot D_h}{\mu \cdot A} \qquad \textbf{Eq. 1}$$

When circulating the heat carrier fluid in the boreholes a circulation pump is used. It is desirable to obtain good heat transfer performance while using as little pumping power as possible and hence at the lowest energy cost. The required pumping power, $\dot{E}_{pump}$, is proportional to the volumetric flow rate and pressure drop in the pipes according to Eq. 2.

$$\dot{E}_{pump} = \frac{\Delta P \cdot \dot{V}}{\eta_{pump}} \quad \textbf{Eq. 2}$$

The pressure drop (ΔP) can be estimated from Eq. 3 and occurs mainly due to friction which increases with higher velocities. From Eq. 3 it may also be seen that the pressure drop is related to the configuration of the pipes.

$$\Delta P_f = f \frac{\rho \cdot w^2}{2} \cdot \frac{L}{D} \quad \textbf{Eq. 3}$$

The value on the friction factor *f* depends on whether the flow is laminar or turbulent. For laminar flows (Re ≤2300) Eq. 4 is used and for turbulent flow (Re≥ 2300) Eq. 5 is used for estimation of the friction factor.

$$f_{Re \leq 2300} = \frac{64}{Re} \quad \textbf{Eq. 4}$$

$$f_{Re \geq 2300} = \frac{1}{(0.79 \cdot lnRe - 1.64)^2} \quad \textbf{Eq. 5}$$

The pressure drop considered in Eq. 3 only accounts for friction losses in the pipes. However, also single losses occur over bends, valves etc. Single losses are assumed to be negligible in this study compared to the friction losses due to the length of the pipes.

Besides the increase in required pump capacity a high pressure drop decreases the maximum allowed temperature inside the collector due to material specifications. There is also risk for the collectors to buckle at high pressures. To install double U-tubes with larger diameter, or a coaxial design, may help the pressure drop to be acceptable. The common SDR 17 and SDR 11 polyethylene pipes can handle pressures of 1.3 and 5.7 bar respectively (Gehlin, et al., 2016).

2.1.3 Geological considerations

The city of Linköping is situated in the Östergötland County where the bedrock to 86% consists of granite (Sundberg, et al., 1985). Granite is classified as crystalline bedrock and is suitable for BTES systems due to its good geological properties and is the most common rock type in Sweden. The thermal conductivity for granite usually varies between 2.9-4.2 W/(m*K) with a mean value of 3.5 W/(m*K) (Nordell, 1994). Overall crystalline bedrock is especially suitable for BTES due to its high thermal conductivity and the relatively easy drilling procedure (Barth, et al., 2012). For energy storage systems also the volumetric heat capacity, C, of the ground is important. It is a measure of how much energy that may be stored in the ground and is presented in kWh/(m³*K) (Erlström, et al., 2016). The density and specific heat capacity for granite is 2700 kg/m³ and 830 kJ/(kg*K) respectively which gives a volumetric heat capacity of 2241 kJ/(m³*K) (or 0.62 kWh/(m³*K)) (Nordell, 1994).

The construction of large BTES requires preliminary invesitigation such as an in-situ thermal response test (TRT) or a distributed thermal response test (DTRT). A TRT gives information on the thermal response regarding effective thermal conductivity of the ground, thermal resistance of the borehole, and the undistributed ground temperature. A DTRT can furthermore give information about potential ground water flow affecting the thermal response around the borehole (Acuña, 2013). For thermal energy storages the influence of groundwater flow can have a bad effect on the storage capacity and heat losses from the storage can be large due to convective heat transport. Significant flow can take place in rock with fractures and cracks and in materials with high hydraulic conductivity, such as sand and gravel (Gehlin, 2002).

2.2 Ground source high temperature heat pump

A vapor compression heat pump has primary four essential components: evaporator, compressor (including motor), condenser and expansion valve. In Figure 7 the equipment diagram of a simple vapor compression heat pump can be seen. If designed as in the figure, a refrigerant is circulating in the heat pump in counter clockwise direction. Heat is collected in the evaporator, $\dot{Q}_2$, from the secondary loop circulating in the BHE. In the compressor the refrigerant is compressed to a higher pressure level, from liquid to gas, which also causes the temperature to increase. The heat is ejected in the condenser and is collected through a heat sink circulating in the secondary loop. After the refrigerant has passed the condenser the refrigerant is passing through the expansion valve and the pressure and temperature of the refrigerant is decreased to its original value and back to liquid before it returns to the evaporator (Granryd, et al., 2011).

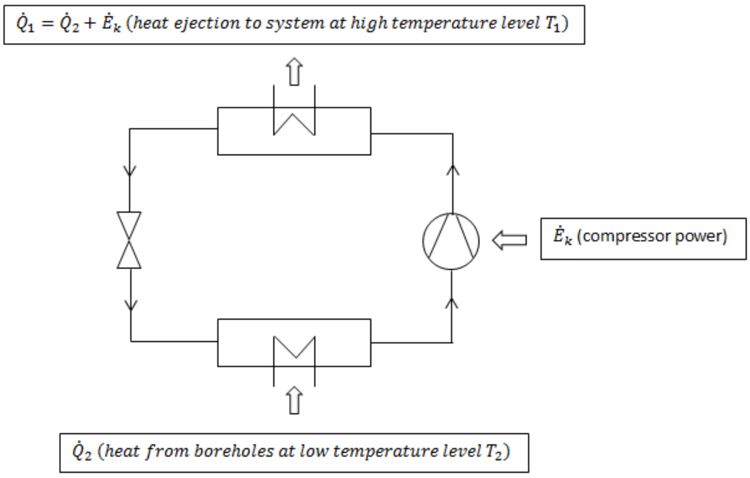


Figure 7 Equipment diagram of a simple ground source vapor compression heat pump.

2.2.1 Coefficient of performance (COP)

The information in this section is taken from the book *Refrigerating Engineering* written by Granryd et.al. (2011) and recall the concept of coefficient of performance (COP) of a heat pump.

The energy delivered from the heat pump, $\dot{Q}_1$, is the sum of its heat absorption ability, $\dot{Q}_2$, and the needed shaft power of the compressor, $\dot{E}_k$:

$$\dot{Q}_1 = \dot{Q}_2 + \dot{E}_k \qquad \textbf{Eq. 6}$$

The efficiency of a heat pump can be expressed by the COP_1, or sometimes referred to as COP_h, defined as the useful energy, $\dot{Q}_1$, in relation to the compressor work, $\dot{E}_k$:

$$COP_1 = \frac{\dot{Q}_1}{\dot{E}_k} \qquad \textbf{Eq. 7}$$

The cooling factor of the heat pump, COP_2, or also defined as the cooling effect of a cooling machine (the reverse of a heat pump), is in the same way defined as the heat absorption ability, $\dot{Q}_2$, in relation to the compressor work, $\dot{E}_k$:

$$COP_2 = \frac{\dot{Q}_2}{\dot{E}_k} \qquad \textbf{Eq. 8}$$

Hence a relation between the heat factor and the cooling factor of the heat pump can be defined as:

$$COP_1 = COP_2 + 1 \qquad \textbf{Eq. 9}$$

The coefficient of performance for a real heat pump cycle will though be lower due to different types of losses. One important parameter in order to achieve a high COP is, independently of losses, to have as small temperature difference as possible between the hot and the cold side of the heat pump. . This can easily be demonstrated from the definition of the COP of a Carnot heat pump cycle (COP_{1C}), or an ideal heat pump, which is calculated based on the condensing and evaporating temperatures according to Eq. 10:

$$COP_{1c} = \frac{T_1}{T_1 - T_2} \qquad \textbf{Eq. 10}$$

A high COP implies a decreased required compressor work as the pressure drop decreases in the system. Hence, with a constant condensing temperature, the higher the heat source temperature is the less compressor power is needed.

2.2.2 Choice of refrigerant

The choice of refrigerant may be the most important parameter for the performance of a HT-HP. In this thesis HT-HP applications will be referred to as systems with heat sink temperatures >80°C. The application of HP for heat sink temperatures below 80°C is widely applicable and developed (Bamigbetan, et al., 2017). When working at temperatures above 80°C the suitable number of refrigerants is currently limited (Eriksson, 2008). The technology and refrigerant for HT-HP in industrial processes needs to be evaluated using a holistic approach and both cost and energy efficiency needs to be included, but also environmental considerations. Due to environmental agreements synthetic refrigerants with high GWP (global warming potential) are phased out gradually. Furthermore, many synthetic refrigerants already are or will be phased out due to high ODP (ozone depleting potential). There are though new synthetic refrigerants available with low or even zero GWP and OPD. However, as synthetic refrigerants are known for having the drawback of unforeseen consequences on the environment, and furthermore often are flammable and/or toxic (Bamigbetan, et al., 2017), in this thesis only natural refrigerants solutions are considered.

Hydrocarbons (HC)

Hydrocarbons (HC), such as propane (R290), butane (R600) and isobutene (R600a), are suitable refrigerants for high temperature applications in district heating systems and industry with heat sink temperatures >80°C. They give high COP, have low saturation pressures and good transport properties regarding pressure drop and heat transfer rate. Isobutene is the hydrocarbon showing most promising thermodynamic properties for high temperature applications according to Ericson (2008), and has shown to give better or comparable performance when compared to synthetic refrigerants (Bamigbetan, et al., 2017). However, hydrocarbons are highly flammable and explosive when blended with air and careful security and leakage measures needs to be taken (Eriksson, 2008). These challenge is further complicated when operating hydrocarbons at high pressure and temperature levels (Bamigbetan, et al., 2017). In small systems the refrigerant charge is though usually too low to form an explosive mixture if a leakage would occur (Granryd, et al., 2011). Furthermore, there are several technical constraints concerning available compressors for working with hydrocarbons at high temperatures and further research and development is required (Bamigbetan, et al., 2017).

Carbon dioxide (CO_2)

Carbon dioxide has shown to be competitive in high temperature applications with good safety aspects. The critical temperature for carbon dioxide is very low, approximately at 31 °C. Above this point carbon dioxide works in its transcritical or supercritical region and the heat transfer occurs at sliding temperature without distinction between gas and liquid (Björk, et al., 2013). The advantage of the transcritical operation of CO_2

applications is shown for high heat sink temperatures, for example in applications of hot water production (Bamigbetan, et al., 2017).

Carbon dioxide though requires very high pressures and the available compressors at the market which can work in these conditions are limited (Eriksson, 2008). Furthermore, the heat source temperature is preferably low, around 10°C (Björk, et al., 2013). Due to the low critical temperature (31°C) the evaporating temperature always needs to be below this value, as the heat pump can't be run in fully transcritical mode (Bamigbetan, et al., 2017). Hence carbon dioxide is not a suitable choice for the application at *Gärstadverket*.

Ammonia (R717)

The thermodynamic properties of ammonia are superior for small scale applications compared to hydrocarbons (Palm, 2008). Ammonia has also been widely used in large capacity systems for both heating and cooling applications (Bamigbetan, et al., 2017). Both GWP and ODP for ammonia are equal to zero and its relatively cheap to manufacture. One disadvantage is though that ammonia is toxic, even at low concentrations. However, ammonia has a very characteristic smell and has hence a built-in warning system for even lower concentrations. Another disadvantage is that ammonia reacts with copper when in presence of water. Compatible materials have though been developed for applications with ammonia (Bamigbetan, et al., 2017).

Due to the very high required discharge pressure it may be difficult to find a compressor developed for HTHP applications working with ammonia (Eriksson, 2008). Recent improvements in compressor technology have though enabled pressure levels up to 60 bars (97.5°C) in ammonia compressors (Bamigbetan, et al., 2017). At a given compressor discharge pressure ammonia has a very high compressor discharge temperature compared to its saturation temperature, unlike other refrigerants. The high discharge temperature can be used to produce hot water of >90°C, if an intercooler/de-superheater is properly integrated in the design (Bamigbetan, et al., 2017). Furthermore two-stage compression or liquid injection is often used in ammonia heat pump systems due to the high discharge temperature (Granryd, et al., 2011).

2.3 Environmental considerations

Geothermal energy is classified as renewable by the Swedish Energy Agency. By the Eueropean Union (EU) it is also comparable with solar energy as it is passively stored in the ground. However, the electrical power used in the heat pump should be environmentally evaluated by the carbon dioxide emissions according to directives within every nation's borders, for example the Swedish or Nordic power mix (Barth, et al., 2012). The utilization of BTES has though shown to reduce emissions of CO_2, SO_2 and NO_x as it provides the possibility to replace fossil fuels with seasonally stored waste heat (Gehlin, 2016).

Even if geothermal energy is classified as a renewable energy source and generates an environmental benefit it can collide with other interests, such as protection and quality of groundwater assets as well as integrity between groundwater aquifers. In the design of boreholes, the thermal effects, pressure changes, water quality and short cut between different ground water aquifer needs to be accounted for. Furthermore, if there is risk of reaching depths with salt ground water this can affect nearby drinking wells. A measure that can be taken is to backfill, or grout, the boreholes with sealing materials whereby the risk for contamination of groundwater can be decreased significantly (Erlström, et al., 2016). The regulations of backfilling differ between countries. In Sweden no regulations for backfilling exists, except in water protected areas or in risk of salt water contamination, and groundwater filled boreholes are most common (Gehlin, 2002).

2.3.1 Effects on microbiology in groundwater

One subject regarding UTES where the information is very limited is the risk related to the temperature increase in the ground and how this can affect the microbiology in groundwater (Bonte, et al., 2011). Regarding low-temperature BTES the change in temperature in the ground is more or less in the same magnitude as what already has been caused by the urbanization (buildings, industries, district heating networks etc.) which may reach more than 100 meters depth. However, for BTES system operating at temperatures above 40°C the risk of causing geochemical disturbance and affecting the microbiological balance in the ground is increased and should be considered in more detail (Gehlin, 2016).

A study of the effects on microorganisms in aquifers due to seasonal temperature fluctuations (12°C to 28°C) in a geothermal well field of 130 m depths was performed by York et al at The Richard Stockton Collage of New Jersey in the 1990s. The study showed an increase in bacterial number related to the temperature increase in the BTES, but also that in aquifers where a change in total bacterial number wasn't observed the effects could be seen in the types of microorganism present (York, et al., 1998).

More recent studies have shown no evidence for increasing number of bacteria or growth of pathogens related to increased temperature in the ground, but in the composition of bacterial species. Possibly due to that different bacteria thrive in different temperature regions (Bonte, et al., 2011). Besides temperature the controlling factors for microorganism includes for example the access to oxygen, light, phosphorus, nitrogen and sulfur, where the access to nutrients may be the most important factor regarding growth in microbiological fauna. This could be an explanation why a bacterial growth not has been observed in relation to HT-UTES. It is furthermore believed that changes in the bacterial fauna will be recovered once a BTES are taken out of operation and the ground temperature returns to its original value (Andersson, 2008).

3 HT-BTES – existing projects and development

In this chapter both early designs and state of the art projects of HT-BTES and different hybrid systems are presented with the aim to build some background based on experience within design, construction and operation of existing HT-BTES. In total seven HT-BTES are included in the study: the two Swedish industrial heat storages in Luleå and Emmaboda, the Danish solar assisted HT-BTES in Brædstrup, the two German solar assisted HT-BTES in Necklarsum and Crailsheim, the Canadian solar assisted HT-BTES in Drake Landing Solar Community, Oktokos, and the experimental HT-BTES in Paskov, Czech Republic, charged with CHP. In Table 2 a summary including years of operation, storage volume and maximal storage temperature reached for the seven HT-BTES is given. A complete summary of important design and operation parameters regarding the seven HT-BTES are given in each respective section 3.1-3.7.

Table 2 Important design parameters of international HTBTES. (Sources: [1] (Hellström, 1991) [2] (Nordell, et al., 2016) [3] (PlanEnergi, 2013) [4] (Nußbicker, et al., 2003) [5] (Schneider, 2013) [6] (Sibbitt, et al., 2015) [7] (Sibbitt, et al., 2012))

HT-BTES	Years of operation	Storage volume [m³]	Storage temperature (max) [°C]	Source of charge
Luleå, Sweden[1]	1983-1990	115 000	65	Industrial waste heat
Emmaboda, Sweden[2]	2010-	323 000	45	Industrial waste heat
Brædstrup, Denmark[3]	2012-	19 000	60	Solar thermal
Necklarsum, Germany[4]	1999-	63 360	65	Solar thermal
Crailsheim, Germany[5]	2008-	37 500	65	Solar thermal
Oktokos, Canada[6]	2007-	34 000	74	Solar thermal
Paskov, Czech Republic[7]	2011-	Unknown	78	CHP

3.1 HT-BTES in Luleå – Sweden

The Luleå heat storage was built in 1982-83 and was the first HT-BTES constructed in bedrock. In Table 3 important design and operational data for the Luleå heat storage is presented. The storage was built for experimental and demonstration purpose for Luleå University of Technology located close to the campus (Nordell, 1994). The 120 boreholes were ordered in a rectangular shape divided in 24 parallel lines with 5 boreholes coupled in series in each. By connecting the boreholes in series thermal stratification of the storage was enabled, with the center of the storage kept at a higher temperature. Each borehole was equipped with open circulation collectors of coaxial design with a thermally insulated centric pipe (Nordell, 1994).

The storage was yearly charged with around 2 GWh recovered from waste heat from gas combustion in the steelwork of the Swedish Steel Company (SSAB) (Hellström, 1991). The heat was supplied to the storage at a temperature of around 70-80 °C through a district heating network separated from the storage by a heat exchanger (Nordell, 1994). During extraction phase around 1 GWh could yearly be recovered directly for space heating in one of the university buildings with a supply temperature between 35 - 55 °C. 20 % of the heat was recovered through two 200 kW heat pumps. Additional heat demand was covered by district heating. Over the year the storage temperature was kept between 30 - 60 °C, with a mean of 52.5 °C (Hellström, 1991).

During the first year of operation of the Luleå heat storage problems occurred with large amount of air in the pipes as a result of the open coaxial collector design. This was corrected by installing a vacuum pump. No problems occurred however with chemical reaction between the heat carrier fluid and the bedrock. Overall, the system was performing below expected values concerning energy storage and extraction

(Nordell, 1994). The storage was taken out of operation in 1990 due to unfavorable control strategies and mistakes during the construction leading to lower performance than expected. The storage though showed the potential of HT-BTES and gave experience of design, construction and operation of this type of systems. The interest for HT-BTES has however been limited until recently. 20 years after the Luleå heat storage was taken out of operation a new larger version of the storage was though constructed in Emmaboda, Sweden, taken into operation in the year of 2010 (Barth, et al., 2012).

Table 3 Design parameters for HT-BTES in Luleå, Sweden (Sources: (Hellström, 1991), (Nordell, 1994))

HT-BTES in Luleå			
Years of operation	1983-90	**Storage shape**	Rectangular
Storage volume [m3]	115 000	**Ground type**	Rock
Number of boreholes	120	**Thermal conductivity ground [W/(m*K)]**	3.42
Borehole depth [m]	65	**Volumetric heat capacity ground [MJ/(m3*K)]**	2.28
Total borehole length [m]	7800	**Insulated [Yes/No]**	No
Borehole spacing [m]	4	**Insulation material**	-
Borehole diameter [mm]	152	**Insulation thickness [mm]**	-
Collector type	Open Coaxial	**Source of charge**	Waste heat
Collector diameter [mm]	Unknown	**Injected energy/year (expected) [GWh]**	2.3 (2.8)
Collector material	Unknown	**Extracted energy/year (expected) [GWh]**	1.0 (1.6)
Collector temperature resistance	Unknown	**Charging temp [°C]**	70-85
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	65
Number of boreholes in series	5	**Storage minimum temp [°C]**	30
Total collector length in series [m]	325	**HP for extraction [Yes/No]**	Yes

3.2 HT-BTES in Emmaboda – Sweden

The design and operation of the HT-BTES in Emmaboda, a larger copy of the Luleå heat storage, have been evaluated by Nordell et al. and the results were presented in their report *Long-term Evaluation of Operation and Design of the Emmaboda BTES - Operation and Experiences 2010-2015*. The BTES in Emmaboda works both as an energy research project and as a system to increase the energy efficiency of the Xylem Water Solutions AB plant in Emmaboda. The BTES has been operated since 2010 and were initially designed to work without a heat pump and thereby rather high storage temperatures were required (Nordell, et al., 2016). Important design parameters for the HT-BTES in Emmaboda can be seen in Table 4.

The borehole field of 140 boreholes was divided into 7 individually controlled sections with 20 wells in each. The sections open or close individually depending on the storage temperature during injection, and extraction, which enables good thermal stratification in the storage volume with higher temperatures in the center. The flow in the BTES is frequency controlled and is varied in the range of 3-20 l/s. The collectors are of open coaxial design with double tube solution with inner and outer spacers every 10 m. Since no high temperature resistant collectors were available at the market by the time the storage was constructed a prototype were used developed by Pemtec AB. The collectors are designed with polypropylene (PPE PN10) (Nordell, et al., 2016), and is temperature resistant up to 80°C (Abrahamsson & Milesson, 2013).

The storage temperature has increased to around 45 °C as of September 2015 and it has been clarified that the targeted storage temperature of 55 °C will not be reached. The anticipated supply temperature to the storage of 60 °C, assumed during predesign, showed to be much lower in reality and rarely reaches above 50-55 °C. In total between July 2010 and September 2015 around 11 875 MWh of heat has been injected to the storage. The storage was first discharged in 2013 and in total approximately 198 MWh was recovered. between 2013 and 2014 (Nordell, et al., 2016).

The expected investment cost of the storage in 2009 was approximately 10 MSEK. In the end of construction in 2010 the cost had reached 11 MSEK, mainly due to unforeseen problems with drilling. The grand cost of the storage is estimated to be around 12 MSEK as the maintenance cost for the first two years was 1 MSEK/year due to operational problems. During construction approximately 65% of the boreholes were drilled without complications while 30% had to be grouted due to cracks, others had to be completely moved which resulted in unforeseen costs. Furthermore, during the first two years there were circulation problems due to gas formation in the pipes, just as in the Luleå heat storage, because of the open coaxial collector design. There were also problems with chemical reactions between the heat carrier fluid and the rock. First in May 2012 the system could run without complications. For future operation the maintenance cost is approximated to 0.1-0.2 MSEK/year (Nordell, et al., 2016).

Table 4 Design parameters for HT-BTES in Emmaboda, Sweden (Sources: (Nordell, et al., 2016), (Abrahamsson & Milesson, 2013))

HT-BTES in Emmaboda			
Start of operation	2010	**Storage shape**	Rectangular
Storage volume [m3]	323 000	**Ground type**	Rock
Number of boreholes	140	**Thermal conductivity [W/(m*K)]**	3
Borehole depth [m]	150	**Volumetric heat capacity [MJ/(m3*K)]**	2.34
Total borehole length [m]	21 000	**Insulated [Yes/No]**	Yes
Borehole spacing [m]	4	**Insulation material**	Foam glass
Borehole diameter [mm]	115	**Insulation thickness [mm]**	400
Collector type	Open Coaxial	**Source of charge**	Waste heat
Collector diameter [mm]	40/90	**Injected energy/year (planned) [MWh]**	3 271* (3 600)
Collector material	Polypropylene (PPE)	**Extracted energy/year (planned) [MWh]**	89* (2 700)
Collector temperature resistance	80	**Charging temp [°C]**	30-60
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	45
Number of boreholes in series	2	**Storage minimum temp [°C]**	Unknown
Total collector length in series [m]	300	**HP for extraction [Yes/No]**	Considered

* As of monitored operation in 2014 (Nordell, et al., 2016).

3.3 HT-BTES in Neckarsulm – Germany

The first German solar assisted district heating system was constructed in 1997 in Neckarsulm-Amorbach. The total annual heating load of the community consisting of around 700 flats, a school and a shopping center is around 3 GWh. The solar plant is connected to a duct heat storage which has been extended in two phases, first in 1999 and later in 2002, and the system is planned to be further extended. Important design parameters for the HT-BTES in Neckarsulm can be seen in Table 5. In 2016 the BTES consisted of 528 boreholes (Gehlin, 2016). This is about the half of the final size of what has been planned. A rectangular storage shape allows for future extension according to the expansion of the residential area which is planned to be increased to around 1300 flats. Each borehole is equipped with double U-pipes made of Polybutene (PB). This is one of the materials with best life expectancy (50 years) for temperatures of 85 °C at 10 bar. The boreholes are grouted with a mix of bentonite, sand cement and water. The thermal insulation is 200 mm thick of polystyrene covered with 2-3 m top soil (Nußbicker, et al., 2003).

The heat distribution net is directly coupled to the storage. Two buffer tanks, 100 m^3 each, is charging the storage by the solar collectors and works as short-term storage handling peaks. Depending on the temperature level heat is either supplied to the buildings through the duct storage or the buffer tanks, while peaks are covered by a gas condensing boiler. The goal has been to reach a solar fraction of 50% of the total space heating and hot water demand of the residential area (Nußbicker, et al., 2003). In 2007 a solar fraction of 44.8% were achieved with a total solar collector area of 5 670 m^2. The size of the community was though smaller by that time with around 300 flats. The approximate contribution from the solar collectors was in 2007 around 1 850 MWh (Nußbicker-Lux, et al., 2009). The fraction of this contribution that was charged to the BTES is though unspecified and the injection and extraction power rates of the BTES have not been found in literature. It has though been considered to install a heat pump to increase the possible temperature level of the heat sink (Nußbicker, et al., 2003).

Table 5 Design parameters for HT-BTES in Neckasulm, Germany (Sources: (Gehlin, 2016), (Nußbicker, et al., 2003), (Nußbicker-Lux, et al., 2009), (Mangold, et al., 2003), (Mangold, 2007))

HT-BTES in Neckarsulm			
Start of operation	1997/99/2002-	**Storage shape**	Rectangular
Storage volume [m3]	63 360	**Ground type**	Clay
Number of boreholes	528	**Thermal conductivity [W/(m*K)]**	2,2
Borehole depth [m]	30	**Volumetric heat capacity [MJ/(m3*K)]**	2,85
Total borehole length [m]	150 840	**Insulated [Yes/No]**	Yes
Borehole spacing [m]	2	**Insulation material**	Polystyrene
Borehole diameter [mm]	150	**Insulation thickness [mm]**	200
Collector type	Double U-tube	**Source of charge**	Solar thermal
Collector diameter [mm]	25	**Injection power [GJ]**	Not found
Collector material	Polybutene (PB)	**Extraction power [GJ]**	Not found
Collector temperature resistance	85	**Charging temp [°C]**	60 (66*)
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	65
Number of boreholes in series	6	**Storage minimum temp [°C]**	45
Total collector length in series [m]	360	**HP for extraction [Yes/No]**	No

* As of monitored operation in 2002 (Mangold, et al., 2003).

3.4 HT-BTES in Crailsheim – Germany

A new residential area connected to a solar collector system and duct heat storage was built in Crailsheim-Hirtenwiesen, Germany, in 2007. The heat storage, a HT-BTES installation consisting of 80 boreholes, was completed in 2008. Important design parameters for the HT-BTES in Crailsheim can be seen in Table 6. Double U-pipes of high-pressure cross-linked polyethylene PEX (PEXa) was installed in the boreholes delivered by the company RAUGEO. RAUGEO PEXa collectors have a temperature resistance of 95°C (Mangold & Schmidt, 2007). For product sheet of the RAUGEO PEXa collectors see Appendix 1. The storage is insulated with foam glass gravel and covered by water protecting foil and a drainage gravel layer. In the upper part (5 m) groundwater flow were detected. Hence the bottom of the boreholes was grouted with thermally enhanced material while the upper parts were grouted with a less conductive material (Mangold & Schmidt, 2007).

Simulations for long-term operation were performed in TRNSYS prior the construction of the residential area (Mangold, 2007). It was expected a coverage of 50 % of the heat demand in the residential area with solar thermal, which was almost achieved in the year of February 2012 - February 2013. The solar collectors are divided in two parts; one diurnal part and one seasonal part connected to the BTES. The BTES cannot directly handle the large capacity of the solar collectors, hence a buffer water tank works as a short storage of the solar collectors. Heat from the seasonal part can be transmitted through the district heating network to the diurnal part, either directly or via a 530 kW heat pump. The storage temperature can vary between 20-50 °C but the heat pump always supplies hot water of 60 °C. With the use of the heat pump the storage temperature can be kept lower and hence the heat losses can be decreased (Schneider, 2013).

Table 6 Design parameters for HT-BTES in Crailsheim, Germany (Sources: (Mangold & Schmidt, 2007), (Mangold, 2007), (Schneider, 2013))

HT-BTES in Crailsheim			
Start of operation	2008	**Storage shape**	Cylindrical
Storage volume [m3]	37 500	**Ground type**	Soil
Number of boreholes	80	**Thermal conductivity [W/(m*K)]**	2.46
Borehole depth [m]	55	**Volumetric heat capacity [MJ/(m3*K)]**	2.4
Total borehole length [m]	4 400	**Insulated [Yes/No]**	Yes
Borehole spacing [m]	3	**Insulation material**	Foam glass
Borehole diameter [mm]	130	**Insulation thickness [mm]**	400
Collector type	Double U-tube	**Source of charge**	Solar thermal
Collector diameter [mm]	32	**Charged heat [MWh]**	780*
Collector material	RAUGEO PEXa	**Charged heat [MWh]**	382**
Collector temperature resistance	95	**Charging temp [°C]**	90***
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	65***
Number of boreholes in series	2	**Storage minimum temp [°C]**	20***
Total collector length in series [m]	220	**HP for extraction [Yes/No]**	Yes

*Mean value between 2009-2012 (Schneider, 2013). **Value for year 2012 (Schneider, 2013). ***From long-term simulations performed in TRNSYS (Mangold, 2007).

3.5 HT-BTES in Brædstrup – Denmark

The Brædstrup District heating in Denmark is supplied by the Brædstrup Totalenergianlæg (Brædstrup Total Energy Plant), a natural gas fired CHP unit. Brædstrup Totalenergianlæg was in 2007 the first natural gas fired CHP plant to be combined with solar thermal. In 2011-2012 the solar plant was extended and a pilot borehole storage of 48 boreholes was added to the system. Important design parameters for the HT-BTES in Crailsheim can be seen in Table 7. The system was also equipped with a 5 000 m^3 buffer tank, a 1 MW heat pump and a 10 MW electric boiler. It is planned to extend the system further. The full-scale borehole storage will be placed around the pilot storage but with separate transmission lines, the two storages will hence be hydraulic separated, and is preliminary planned to consist of 432 boreholes This would allow for a flexible integration of fluctuating electricity production through UTES with an expected solar fraction of 50 % (PlanEnergi, 2013).

The borehole installation in Brædstrup Totalenergianlæg is based on the design of the borehole installation in Crailsheim, Germany. The borehole heat exchanger design was based on the ones used in Crailsheim with double U-tubes of the RAUGEO PEXa type (Appendix 1). The boreholes are connected in 16 parallel flow lines with 6 boreholes in series and a pressure drop of approximately 2.0 bars in each line. The boreholes are grouted with HDG Thermo HS with a thermal conductivity around 1.44 W/(m*K). The storage is insulated with cockles which is available in Denmark for large quantities (PlanEnergi, 2013).

A 1.2 MW ammonium heat pump of industrial SABROE® type (delivered by Johnson Controls) based on high pressure screw compressor is used in the system during heat extraction. The heat pump can deliver an outgoing temperature of up to 90 °C which may be used in the district heating network (PlanEnergi, 2013). Johnson Controls also deliver large non-standard SABROE® heat pumps of larger capacities up to 13 MW (Johnson Controls, 2017). The designed storage temperature of the BTES varies between 70°C during charging period (summer) and 10°C during extraction period (winter). In February 2013 the BTES delivered heat at a temperature of 20 °C to the heat pump which in turn delivered heat at a temperature of 80°C to the DH system (PlanEnergi, 2013).

Table 7 Design parameters for the HT-BTES in Brædstrup, Denmark (Sources: (Gehlin, 2016), (PlanEnergi, 2013))

HT-BTES in Brædstrup			
Start of operation	2012-	**Storage shape**	Cylindrical
Storage volume [m3]	19 000	**Ground type**	Clay
Number of boreholes	48	**Thermal conductivity [W/(m*K)]**	1.42
Borehole depth [m]	45	**Volumetric heat capacity [MJ/(m3*K)]**	1.9
Total borehole length [m]	2 160	**Insulated [Yes/No]**	Yes
Borehole spacing [m]	3	**Insulation material**	Cockles
Borehole diameter [mm]	150	**Insulation thickness [mm]**	500
Collector type	Double U-tube	**Source of charge**	Solar thermal
Collector diameter [mm]	32	**Energy charged [MWh]**	444*
Collector material	RAUGEO PEXa	**Energy discharged [MWh]**	163*
Collector temperature resistance	95	**Charging temp [°C]**	80
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	60
Number of boreholes in series	6	**Storage minimum temp [°C]**	20
Total collector length in series [m]	540	**HP for extraction [Yes/No]**	Yes

* As of operation between May 2012 to February 2013 (PlanEnergi, 2013). **Heat loss estimated as 24% (Gehlin, 2016).

3.6 HT-BTES in Okotoks – Canada

Drake Landing Solar Community (DLSC) in Oktokos, Canada is a community consisting of 52 energy efficient detached homes. A hybrid system utilizing solar thermal with seasonal storage in BTES is connected to a district heating system supplying space heat to the community. This is the first system delivering more than 90 % space heating from solar energy and also the first system of this type operating in such cold climate (Sibbitt, et al., 2012). The solar fraction reached 97% already in year 5 (Leidos Canada, 2014). Additionally, about 50 % of the domestic hot water load in the community is supplied by independent solar domestic hot water systems (Sibbitt, et al., 2012).

The BTES constituted of 144 boreholes, each 35 m deep. Important design parameters for the HT-BTES in Oktokos can be seen in Table 8. The boreholes are equipped with single u-tube collectors of cross-linked polyethylene material of the RAUGEO PEXa type. The boreholes are plumbed in 24 parallel circuits with 6 boreholes coupled in series in each from center and out which enables thermal stratification in the storage volume. The BTES field is covered with an extruded polystyrene (XPS) insulation. The system is also equipped with a short-term thermal storage (STTS) used as a buffer between the district loop, the BTES field and the collector loop. Compared to the BTES the STTS can accept and dispense heat at much higher rate, but with lower capacity, and is thereby a crucial part for the system to work properly (Sibbitt, et al., 2012).

Table 8 Design parameters for the HT-BTES in Oktokos, Canada (Sources: (Sibbitt, et al., 2012), (Catolico, et al., 2015), (Leidos Canada, 2014), (Ronglei Zhang, 2012))

HT-BTES in Okotoks			
Start of operation	2007	**Storage shape**	Cylindrical
Storage volume [m3]	34 000	**Ground type**	Soil
Number of boreholes	144	**Thermal conductivity [W/(m*K)]**	2.0
Borehole depth [m]	35	**Volumetric heat capacity [MJ/(m3*K)]**	2.3
Total borehole length [m]	5 040	**Insulated [Yes/No]**	Yes
Borehole spacing [m]	2.25	**Insulation material**	XPS
Borehole diameter [mm]	150	**Insulation thickness [mm]**	200
Collector type	Single U-tube	**Source of charge**	Solar thermal
Collector diameter [mm]	25	**Injection power [GJ]**	2 530*
Collector material	RAUGEO PEXa	**Extraction power [GJ]**	1 370*
Collector temperature resistance	95	**Charging temp [°C]**	~50-65
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	70**
Number of boreholes in series	6	**Storage minimum temp [°C]**	39**
Total collector length in series [m]	210	**HP for extraction [Yes/No]**	No

* Mean value between years 2007-2014 (Leidos Canada, 2014). **As of year 2012-2013 (Leidos Canada, 2014).

3.7 HT-BTES in Paskov– Czech Republic

In 2011 an experimental HT-BTES charged by waste heat from a CHP-unit was taken into operation in Paskov, Czech Republic. Important design parameters for the HT-BTES in Paskov can be seen in Table 9. The storage was built for experimental purpose with the aim to study the storage system and the rock environmental behavior during different operating states of charging and discharging. The use of a CHP unit gives a continuous supply of heat compared to storing of solar collector heat. The project was initiated to gain more knowledge of BTES with high heat carrier temperatures of 70-90°C (Grycz, et al., 2014).

The BTES consists of 16 boreholes of 60 m depth. In addition to this, 6 monitoring boreholes were drilled; four 15 m deep in the edges of the storage, one 60 m deep in 8.5 m distance outside the storage, and one 80 m deep placed in the center of the storage. The boreholes are divided in 8 loops with 2 boreholes coupled in series in each. Each borehole is equipped with double U-tubes made by high temperature resistance material – polyethylene PE-RT – and is grouted with a concrete-bentonite mixture. The circulating medium is pure water and is circulated so that the central part of the storage is kept warmer than the edges (Grycz, et al., 2014).

After the first charging period from February 2012 to February 2013 the storage had been charged with in total 2000 GJ. The maximum supply temperature from the CHP-unit was 97 °C. Charging stopped as the storage was approaching its maximum capacity. The initial temperature in the ground was 12 °C and the maximum temperature in the ground had now increased to approximately 75 °C in the central monitoring borehole. The storage had a resting phase of 189 days and the temperatures in the central borehole dropped to about 40 °C. The fluid temperature dropped from 60 °C to 39 °C (Grycz, et al., 2014). A second charging phase was then initiated and the maximum temperature in the center of the storage reached between 69 °C to 78.5 °C. In the outer region the temperature ranged between 30-45 °C depending on the depth and the direction of groundwater flow (Rapantova, et al., 2016).

After the field experiments the design of the BTES was further optimized by simulations. In the simulations a polystyrene insulation was added, and the results showed that this would be required to increase the performance of the BTES. The simulations furthermore indicated that as long as the BTES are not over exploited during discharge the length of the injection and extraction cycles have little or no impact. If the BTES would be overexploited the temperatures would though decrease below freezing point (Rapantova, et al., 2016).

Table 9 Design parameters for the HT-BTES in Paskov, Czech Republic (Sources: (Grycz, et al., 2014), (Rapantova, et al., 2016))

HT-BTES in Paskov			
Start of operation	2011	**Storage shape**	Cylindrical
Storage volume [m3]	Not specified	**Ground type**	Clay/rock
Number of boreholes	16	**Thermal conductivity [W/(m*K)]**	2.17
Borehole depth [m]	60	**Volumetric heat capacity [MJ/(m3*K)]**	2.3
Total borehole length [m]	960	**Insulated [Yes/No]**	No
Borehole spacing [m]	2.5	**Insulation material**	-
Borehole diameter [mm]	120	**Insulation thickness [mm]**	-
Collector type	Double U-tube	**Source of charge**	CHP
Collector diameter [mm]	32	**Injection power [GJ]**	2000
Collector material	PE-RT	**Extraction power [GJ]**	-
Collector temperature resistance	Not specified	**Charging temp [°C]**	70-97
Thermal stratification [Yes/No]	Yes	**Storage maximum temp [°C]**	78
Number of boreholes in series	2	**Storage minimum temp [°C]**	39**
Total collector length in series [m]	240	**HP for extraction [Yes/No]**	No

* Optimum long-term operation from simulation (Rapantova, et al., 2016). **After resting phase of 189 days (without energy extraction) (Grycz, et al., 2014).

3.8 Discussion

The construction and operation of HT-BTES systems are, compared to more conventual LT-BTES systems, rather unexplored. In the 1980s Sweden were the first to build a HT-BTES in bedrock, the Luleå heat storage. The only HT-BTES included in this study that are built with rock as storage medium is the two BTES in Sweden, in Luleå and Emmaboda. Hence, it is difficult to draw any conclusions from some of the storage design characteristics, such as borehole depth and spacing. In Sweden today it is, for larger borehole installations as mentioned before, common to drill with a borehole depth of 200-300 m and a borehole spacing between 4-6 m. The HT-BTES included in this study are shallower and some has even smaller borehole distance than 4 m. Overall rock has good characteristics for borehole installations, along with simple drilling procedures and high thermal conductivity. One important factor to consider is though the quality of the rock. When the heat storage in Emmaboda were built, unforeseen costs appeared as around 30% of the boreholes needed to be grouted or completely moved due to cracks and high groundwater flow.

One important design parameter, especially regarding HT-BTES, is the choice of collector design and material. The challenge with HT-BTES is to find a collector material that is resistant to high temperatures, and high pressures. One collector type that has been used in several HT-BTES is the PEXa double u-tube collector from RAUGEO, which is temperature resistant up to 95 °C. From a thermal performance and a hydrodynamic perspective, the coaxial collector design is though more favourable. Another advantage with coaxial collectors is that the head loss is lower compared to an U-tube collector. This in turn allows for higher operating temperatures.

Research is continuously made of new HT-resistance materials. One example is Triopipe Geotherm AB in collaboration with Kungsörs Plast AB – KPS Petrol Pipe System™, a producing and world leading factory of plastic pipes for usage with aggressive media. When operating with water in high temperatures (<100 °C) and pressures the aging of the pipes accelerates. Solutions exists on the market today but the material costs are too high. Triopipe Geotherm AB is considering as solution with multilayer plastic pipes using a combination of materials to contain the positive and mechanical characteristics of plastic pipes even at high temperatures. The knowledge is available at Triopipe Geotherm AB and production is possible at the factory in Kungsör, but theoretical testing is required. It is though estimated that the product could be available within 6 months after the project start (Andersson, 2017).

One issue that were experienced both in the early design in Luleå, and later in Emmaboda, was gas formation in the pipes. These difficulties have been linked to the fact that both storages were built with an open collector design where the heat carrier fluid was in direct contact with the borehole wall. In Emmaboda the open coaxial collector design also resulted in chemical reactions between the borehole wall (the bedrock) and the heat carrier fluid. One possible solution to these issues is to add a liner to the coaxial collector design. Using a liner gives almost the same favourable characteristics as of an open coaxial collector, with low borehole resistance and high heat transfer rate, but prevents from drawbacks with chemical reaction and gas formation as the heat carrier fluid is never in direct contact with the borehole wall.

Results from the literature study in this project, suggests that the most favourable storage geometry is the cylinder. Several of the large BTES-systems is though built with a rectangular geometry. These are however the older BTES such as the Luleå heat storage and the BTES in Necklarsum. Still, both designs are convenient if the storage is planned to be extended in the future, as the BTES in Necklarsum (rectangular design) and the BTES in Brædstrup (cylindric design).

In several of the HT-BTES included in this compilation heat pumps are used during discharge of the BTES. In this way a stable temperature and power delivery can be ensured, as for as an example the BTES in Crailsheim. Furthermore, the storage temperature can be kept lower in the BTES and hence the storage losses can be decreased. Another favourable design, which has been applied to the majority of the HT-BTES included in this compilation, is to couple boreholes in series. This allows for thermal stratification where the centre of the BTES is kept at a higher temperature. In this way the discharge temperature can be increased. It is furthermore common to connect HT-BTES to solar thermal, which also can be used as a complement to CHP. One example of a solar thermal system connected to a HT-BTES is the BTES in Okotoks, Canada.

Most of the existing BTES included in this study are built with an insulation layer on top of the storage. In the case of the BTES in Emmaboda insulation was added after the BTES had been built, and in the case of Paskov it is estimated that an insulation layer is required to improve the performance of the BTES. The addition of an insulation layer has proven to be efficient. Though, several of the BTES investigated have more shallow boreholes and are placed in clay. How efficient an insulation layer is for BTES built in rock with deeper boreholes is unknown. However, if an insulation layer is included in the design of a BTES, one way to decrease the environmental impact and the cost is to use natural insulation materials. One innovative example is the BTES in Braedstrup where cockles, or mussel shells that was first considered, were used as insulation material.

4 Project background

Tekniska Verken is a municipality owned company situated in Linköping, Sweden, that offers services within district heating and cooling, waste treatment, electricity, lighting, water, biogas and energy efficiency, among other (Tekniska Verken i Linköping AB, 2017). Tekniska Verken owns two CHP-plants in Linköping; the so called Gärstadverket and KV1. The main plant, Gärstadverket, is divided in three parts; the old part built in the 1980s with three boilers (boiler 1-3) situated in one common building. And two new plants taken into operation in 2004 and 2015 with one boiler each (boiler 4 and 5 respectively), situated in two separate buildings (Ericsson, 2017). The three parts of Gärstadverket production plants, boiler 1-3, boiler 4 and boiler 5, are divided as three blocks; 050, 061, 062 (Ericsson, 2016).

Gärstadverket is mainly fed by household and industrial waste and is a part of the waste treatment in Linköping. Boiler 5, also called the "Lion Boiler", can additionally be fueled by for example recycled wood (Ericsson, 2016). The heat from the incineration is used to produce steam; the walls of the boilers consist of water filled pipes. The produced steam is in turn used to generate electricity and remaining energy is used to produce district heating in steam condensers (KV50, KV61 and KV62). Additional heat, which is used to preheat the district heating return before the steam condensers, is recovered in flue gas condensers (FGC) where the moisture in the flue gases is condensed into water. During summer, district cooling can be produced by utilizing the surplus heat from the waste incineration to be used as fuel in the absorption chillers (Tekniska Verken i Linköping AB, 2014).

Since Boiler 5 ("Lion Boiler") was added to the plant in 2015 Gärstadverket has the capacity to cover almost the entire heat demand in the district heating network in Linköping. Still, due to uneven heat demand over the year the capacity isn't enough during peak heat demands in the winter why operation of the older CHP-plant KV1 is required. KV1 was built in 1964 and consists of three different boilers fed by biofuels, coal and oil (Ericsson, 2017). It is planned to phase out the operation of KV1 to decrease utilization of fossil fuels. The amount of heat and power produced by KV1 is thereby required to be replaced.

The total district heating demand in Linköping is around 1 450 GWh/year. Between May 2015 to April 2016 1 312 GWh of heat were produced at Gärstadverket, with the maximum capacity of 209 MW. In Figure 8 the available power surplus and the total power demand when KV1 would be taken out of operation is shown. As can be seen in Figure 8 large amounts of surplus heat is produced during summer in Gärstadverket while there are large peaks during the winter months when operation of Gärstadverket is insufficient. Between May 2015 to April 2016 the total surplus heat from combustion of the waste in Gärstadverket was 263 GWh. This surplus is in the current operation of Gärstadverket requiring cooling.

It has been estimated by Tekniska Verken that the district heating network can handle a power supply of maximum 50 MW from the BTES system, even if the power demand is higher. The model in this thesis has hence been limited to give 50 MW from the heat pumps at peak demand.

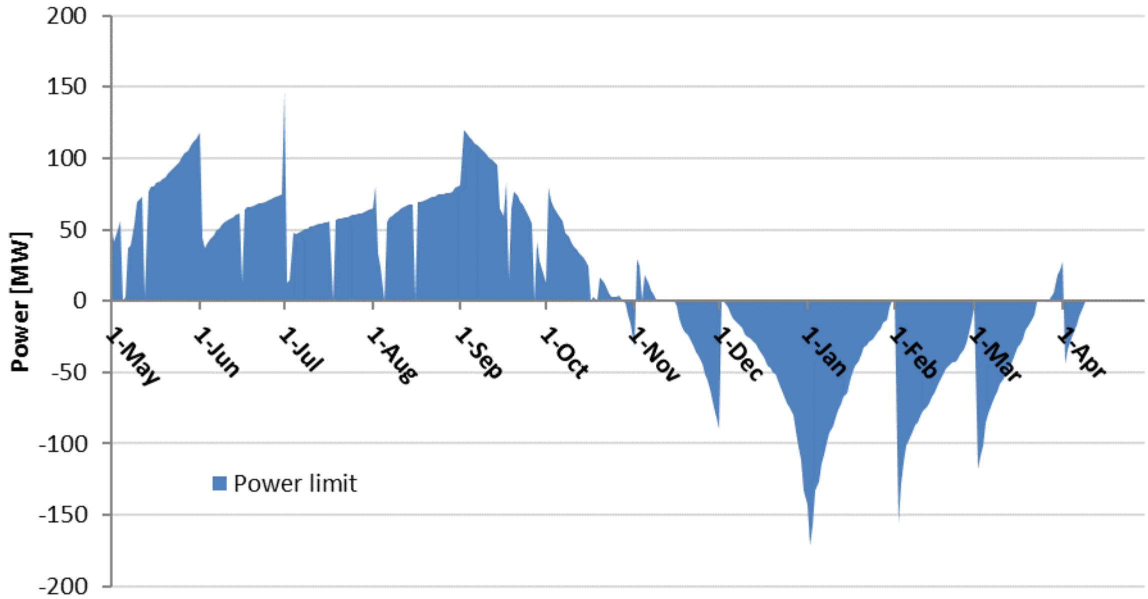


Figure 8 Available power levels at Gärstadverket constituting the power limit for charge (positive levels) and discharge (negative levels) of the BTES.

4.1 System description – system with HT-BTES

Tekniska Verken is currently investigating different solutions to cover the peak heat demand in winter when the production plant KV1 has been phased out, without utilizing fossil fuels. One possible solution that has been discussed is to seasonally store the surplus heat, which is produced during summer at Gärstadverket, in a HT-BTES. The stored heat can then be extracted to cover the winter peaks when the operation of Gärstadverket is insufficient. A HT-BTES could hence be one step towards phasing the production plant KV1, and could at the same time decrease the need for cooling of the surplus heat produced in summer. This system solution is investigated further in this master thesis.

In Figure 9 below the considered system layout for charge and discharge of the BTES can be seen. The interaction of the BTES and Gärstadverket would function differently depending on if the BTES are charged or discharged. During charging of the BTES surplus heat from Gärstadverket will be used to heat the storage and will be exchanged through heat exchangers between the district heating loop and the borehole loop. In winter operation the stored heat will be extracted from the storage and utilized in the district heating network. As the BTES itself would not be able to provide the demanded power and temperature levels a heat pump is required during discharge. The power levels shown in Figure 8 corresponds to the power limit for charge (positive values) and discharge (negative values) of the BTES with regard to Gärstadverket, between 1 May to 30 April.

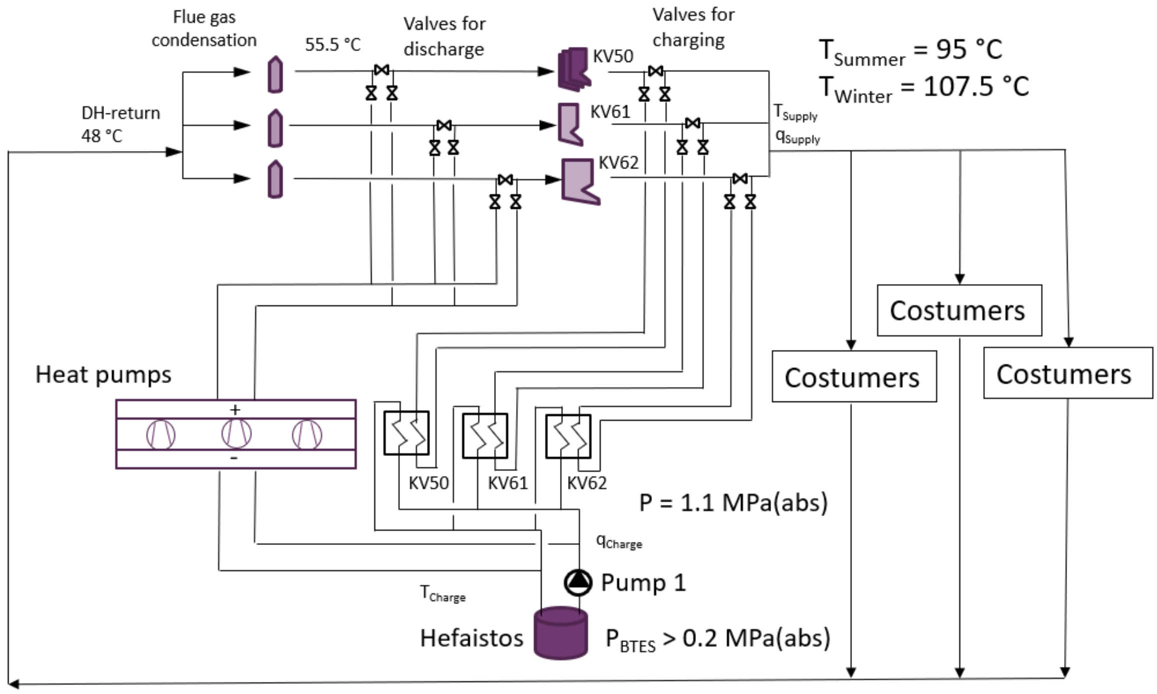


Figure 9 Flow chart for Gärstadverket including a BTES and heat pumps. The heat pump illustrates one or more heat pump stages that could be needed in the system. (Modified from diagram given from Tekniska Verken, 2017)

In the considered system layout (Figure 9) the valves for charging of the BTES is placed after the steam condensers where the temperature is approximately constant at 95 °C during summer operation of the CHP-plant. During discharge the BTES will be connected to Gärstadverket before the steam condensers, but after the flue gas condensers. In this way, the BTES preheats the district heating return without affecting the efficiency of the flue gas condensation. The district heating return temperature after passing the flue gas condensation is around 55.5 °C (varies between 54 °C to 57 °C). This temperature can during discharge of the BTES be used as input temperature for the heat sink at the condenser side of the heat pump. The temperature levels and flows available for charging respectively discharge of the BTES have been provided by Tekniska Verken based on operation of Gärstadverket in the period of May 2015 to April 2016. See Figure 10. The considered discharge period of the BTES is between end of October to the beginning of April.

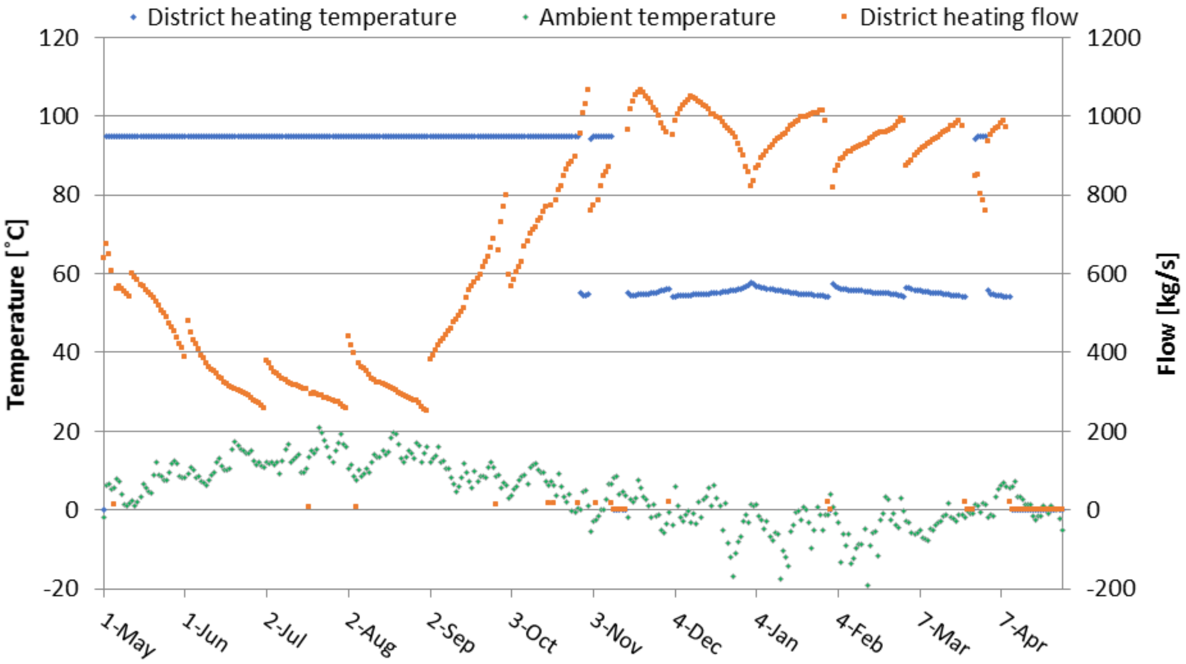


Figure 10 Input data for the simulations. District heating temperature and volumetric flow rate given from Tekniska Verken, and ambient temperature taken from daily mean outdoor temperature from SMHI.

4.1.1 System model - Previous work

Pre-studies of a HT-BTES connected to Gärstadverket at Tekniska Verken have recently been performed at KTH and Bengt Dahlgren AB including the development of a TRNSYS model based on the well-known DST approach. As a background to the system model in this master thesis, the already existing TRNSYS models and the earlier results has been used. In the pre-study at KTH the required size of the storage was initially estimated to 1 500 boreholes with 300 m depth and a distance between the boreholes of 5 m, which gives a (cylindrical) storage volume of approximately 9 740 000 m^3. The BTES were simulated with a serial connection of three boreholes. The model was based on a configuration without a heat pump during discharge and furthermore the interaction of the BTES and the rest of the system were not considered.

The TRNSYS model developed at Bengt Dahlgren AB includes the interaction between the BTES and Gärstadverket. The interaction is based on a configuration with connection of the BTES and the steam condensers (KV50, KV61 and KV62) with the water loops for the BTES and the district heating separated with heat exchangers (one per block), as in the system layout shown in Figure 9. The system was simulated with two different configurations, with and without a heat pump during discharge of the BTES. The heat pump was though simulated with a constant COP and the design of the BTES during these simulations was set constant based on the earlier results, with a geometry of 1 500 boreholes, 300 m borehole depth and 5 m borehole distance.

Results from earlier simulations performed at Bengt Dahlgren AB showed that a control for the amount of charged and discharged power is required, otherwise the power limit showed in Figure 8 were exceeded. The storage has, at non-peak periods, the capacity to store and discharge more power than is available in the district heating network to supply respectively accept. The consideration of a heat pump in the system design is required if the system is required to deliver the desired power and temperature level to the CHP-plant during discharge. The TRNSYS simulations performed at Bengt Dahlgren AB, Stockholm, also showed that when boreholes were coupled in series of three compared to all being coupled in parallel the available energy discharge increased. Hence, it is in this study considered a solution where a heat pump is included in the system and boreholes are coupled in series.

5 System model

In this chapter the heat pump model and the TRNSYS model will be described. The difference of the new TRNSYS model used in this project compared to the earlier TRNSYS model from Bengt Dahlgren AB is foremost the operation of the system during winter mode. The control strategy in this TRNSYS model differs between summer and winter operation of the BTES (charging respectively discharging of the BTES). The control strategy used for the summer mode are similar to the one used in the model from Bengt Dahlgren AB. The main strategy used in this model is as follows, with summer and winter operation separated with new control strategies for the winter mode:

- Summer mode (similar as the old model)
    - Heat exchangers used for charging the BTES
    - Volumetric flow in the BTES loop controlled so that the charged power does not exceed the power limit.
- Winter mode (different from the old model)
    - Heat pumps running for discharge of the BTES
    - A heat pump model is developed from manufacturer data.
    - The number of heat pumps running during discharge is controlled so that the delivered heating power from the heat pumps does not to exceed the power limit of 50 MW.
    - The flow in the BTES loop is controlled to give a temperature fall of 2 ˚K over each evaporator.
    - The inlet temperature to the BTES loop is the outlet temperature from the last evaporator.
    - The delivered temperature to the district heating loop is the outlet temperature from the last condenser.

The heat pump model and the TRNSYS model will be explained further in this chapter.

5.1 Heat pump model

As the BTES won't be able to deliver the desired temperature and power level to the CHP-plant without the use of a GSHP, the heat pump will be integrated in the design, as shown in in Figure 9. A heat pump manufacturer, Star Renewable Energy, were contacted for performance data for their heat pumps using ammonia as refrigerant. Ammonia was considered a suitable refrigerant for the installation at Gärstadverket since it is a natural refrigerant that is suitable in large scale high temperature heat pump installations. The heat pumps delivered by Star Renewable Energy uses two-stage compression with compressors delivered from Vilter (Emerson), which is developed for high pressures and enables variable speed control (Hoffmann & Pearson, 2011).

A 15 MW HT-HP delivered by Star Renewable Energy Heat Pumps (Neatpumps) using Ammonia as working fluid has in 2010 successfully been installed in Drammen, Norway, for hot water production for a district heating application (Bamigbetan, et al., 2017). The source for the heat pump is sea water at a constant water temperature of 8-9°C. The district heating return temperature is in turn constant between 60°C to 65°C all year around. Several heat pumps can be connected in series to reach a desired output power. The system was designed with 3 two-stage compressors systems with the heat exchangers coupled in series. After the main flow of hot water has left the third condenser it is divided in three parallel streams and the temperature is further raised through the high stage de-superheater of each system. Separate streams of water are also going through parallel sub coolers, intercoolers and high- and low stage oil coolers. With combination of de-superheating, intercooler and heat of condensation the delivery temperature could be brought up to 90°C (Hoffmann & Pearson, 2011).

A similar heat pump installation has been considered in this project. Star Renewable Energy constructs their heat pumps for specific installations why no standard data is available. Hence the performance of the heat pump was calculated by Star Renewable Energy for some specific cases of operation, with 2 to 9 heat pumps connected in series. Based on the heat pump performance data three models was developed in excel in the form of three polynoms. One polynom each was developed for the heating power $(\dot{Q}_1)$, the cooling power $(\dot{Q}_2)$ and the compressor power (P) of the heat pump. The general polynom used for the three models can be seen in Eq. 11, where $\dot{m}_{BH}$ is the volumetric mass flow rate in the borehole loop, $\dot{m}_{DH}$ is the volumetric mass flow rate in the district heating loop and T_{BH} is the temperature, in Kelvin, entering the evaporator on the secondary borehole side.

$$\dot{Q}_1, \dot{Q}_2, \dot{E}_k = C + a_1 * \dot{m}_{BH} + a_2 * \dot{m}_{BH}{}^2 + b_1 * \dot{m}_{DH} + b_2 * \dot{m}_{DH}{}^2 + c_1 * T_{BH} + c_2 * T_{BH}{}^2 \quad \textbf{Eq. 11}$$

The parameters $\dot{m}_{BH}$, $\dot{m}_{DH}$ and T_{BH} is the input variables used in the model. The temperature of the district heating return is assumed to be constant at 55 °C, as it only varies between 54 °C to 57 °C, and is assumed to have a minor effect on the heat pump performance, and this is the reason why it is not used as a variable in the model. The constants C, a_1, a_2, b_1, b_2, c_1 and c_2 are calculated in excel for the three models corresponding to $\dot{Q}_1, \dot{Q}_2$ and $\dot{E}_k$, using the least square method to minimize the error (RMSD) between the heat pump manufacturer data and the models.

As mentioned, the district heating return temperature is assumed constant for the heat pump model and it is not used as an input parameter in Eq. 11. The district heating return temperature is though used to calculate the district heating supply temperature delivered from the heat pump through the energy balance in Eq. 12, where cp is the heat capacity of water equal to 4.2 $kJ/(kg \cdot K)$.

$$T_{out,sink} = T_{in,sink} + \frac{\dot{Q}_1}{\dot{m}_{DH} \cdot cp} \quad \textbf{Eq. 12}$$

In a similar way, the return temperature of the heat source to the BTES after the heat pump is calculated through Eq. 13.

$$T_{out,source} = T_{in,source} - \frac{\dot{Q}_2}{\dot{m}_{BH} \cdot cp} \quad \textbf{Eq. 13}$$

The total compressor power is calculated according to Eq. 14, including 3 % losses in the motors and 3 % losses in the high stage compressors using inverters. The compressor shaft power, $\dot{E}_k$, is calculated from the heat pump model.

$$\dot{E}_{k,tot} = \frac{\dot{E}_k}{\eta_{electric\ motor} \cdot \eta_{diverters}} \quad \textbf{Eq. 14}$$

In turn the heat factor, COP_1, of the heat pump is calculated according to Eq. 15.

$$COP_1 = \frac{\dot{Q}_1}{\dot{E}_{k,tot}} \quad \textbf{Eq. 15}$$

Equations Eq. 11-Eq. 15 presented are used in TRNSYS to simulate the performance of the heat pump for each time step over a ten year simulation period.

5.2 TRNSYS model

When designing a BTES the dynamic thermal processes need to be accounted for. TRNSYS is a transient system simulation software tool used worldwide by researchers and consultant engineers which enables modeling of a wide range of thermal energy systems, with the possibility to combine a large variety of system component models (Simulation Studio components) (Pahud & Hellström, 1996). In TRNSYS different Simulation Studio components can be connected by creating a link between an outlet from one component to the inlet of another, giving the possibility to simulate flexible thermal energy systems.

Figure 11 below shows the layout of the TRNSYS model used for the simulations in this study, which is a modification of the model used at Bengt Dahlgren AB, Stockholm. The red connection lines represent the district heating loop while the blue lines represent the BTES loop. The pink lines represent the loops for the heat pump operation, while the purple lines are control strategies. The color scheme was made to make it easier to separate the different parts in the TRNSYS model. Looking at Figure 11 the pink connection lines and the heat pump component has been added to the model from Bengt Dahlgren, as well as the control strategies connected to the heat pump seen in the right side, and in the bottom of the figure.

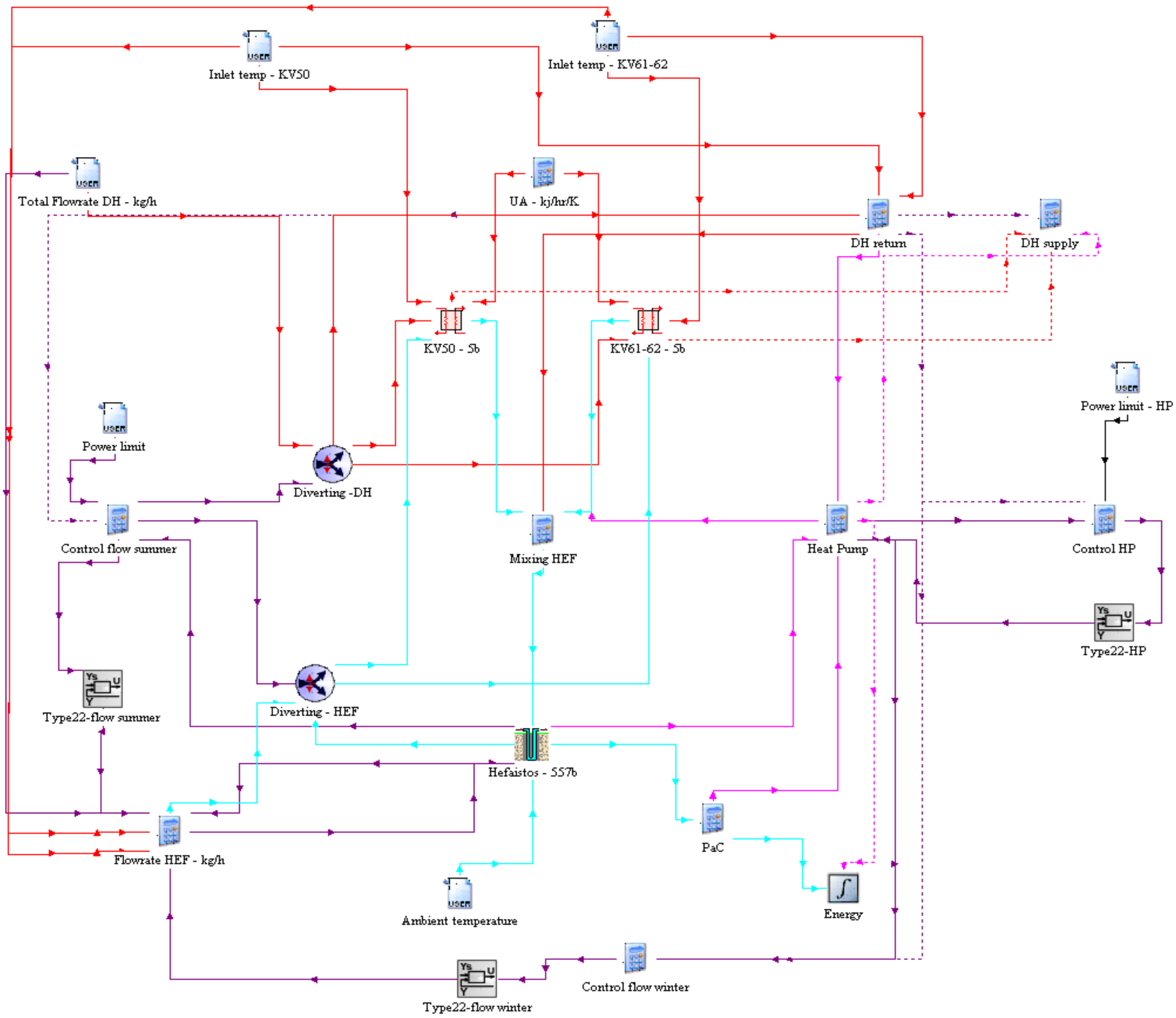


Figure 11 TRNSYS model of the interaction between *Gärstadverket*, the BTES and the heat pumps.

In Table 10 below the different simulation studio components used in the TRNSYS model are presented. In addition to these components it was also utilized the possibility to insert costum equations in a TRNSYS model. Furthermore, different data reader components are included in the model. These contains imported text files of district heating temperature and volumetric flow rate data and the power limit for HEFAISTOS, given from Tekniska Verken. The input data given from Tekniska Verken are daily mean values throughout 2016. However, the TRNSYS modeling is carried out with one-hour sample rate. Meaning that the 24 samples of one day will utilize the same mean value of input data. The simulation period starts in the first of May with a simulation period of 10 years. In the remaining part of this chapter the main components in the TRNSYS model will be explained further.

Table 10 List of the TRNSYS simulation components used in the model.

Name in TRNSYS model	Type	Description
Hefaistos	557b	Borehole heat exchanger
Type22	22	Iterative feedback controller
KV50	5b	Heat exchanger
KV61-62	5b	Heat exchanger
Diverting - DH	11f	Flow diverter
Diverting - HEF	11f	Flow diverter
Energy	24	Integrator

5.2.1 Borehole heat exchanger

The standard component library in TRNSYS doesn't provide any BHE types. An add-on can though be purchased from the TESS library called Geothermal Heat Pump (GHP) where a variety of BHE types are available. In this project the Type557b is used which is one of the models which is available for multiple BHE. Type557b is based on the duct ground heat storage model (DST) and is considered the state-of-the-art model for solution of ground heat exchangers in dynamic simulations.

The DST-model, developed by Göran Hellström at Lund Institute of Technology (LTH) in 1989, is a Simulation Studio component which enables simulations of ground heat storage systems formed by a duct or channel system. In DST the storage is simulated as uniformly placed ground heat exchangers with a hexagonal configuration in the shape of a cylinder (Pahud & Hellström, 1996), see Figure 12 for an illustration. The model has been widely used both for dimensioning of heat storages and evaluation of field projects and combines analytical solutions with numerical methods to solve transient systems iteratively during quasi-steady state conditions (Hellström, 1989). Further information can be found in the report "Duct Ground Heat Storage Model, Manual for Computer Code" from 1989 by Göran Hellström.

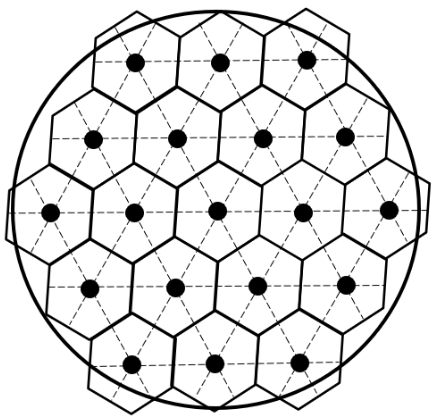


Figure 12 Illustration of the hexagonal borehole configuration simulated in the DST model.

In Table 11 fixed input values for Type557b in the TRNSYS model can be seen. The information in Table 11 is based on the theoretical background section of this thesis combined with experience from Bengt Dahlgren AB Stockholm of a variety of TRT and borehole installations. Furthermore, a climate file for the ambient temperature, in the form of a data reader component, is connected to the BTES as an input for the temperature on top of the storage as well as the temperature of the air surrounding the BTES. The fluid to ground resistance (borehole resistance) is set to 0.05 (m*K)/W which is a reasonable value for double U-pipes with regard to the high temperatures. Coaxial collectors are assumed to have half of the borehole resistance of double U-pipes and will be used for the optimized case.

Table 11 Input data for the borehole heat exchanger component (Type557b) in TRNSYS.

	Value	Unit
Borehole characteristics		
Borehole radius	0.055	m
Header depth	1	m
Number of boreholes in series	3	
Number of radial regions	1	
Number of vertical regions	10	
Distance between boreholes	5	m
Ground properties (rock)		
Fluid to Ground Resistance	0.05	(m*K)/W
Storage thermal conductivity	2.9	W/(m*K)
Storage heat capacity	2241	kJ/(m^3*K)
Ground properties (top soil)		
Number of ground layers	1	
Thermal conductivity layer	1	W/(m*K)
Heat capacity of layer	2241	kJ/(m^3*K)
Thickness of layer	5	m
Initial Surface Temperature of Storage Volume	8	˚C
Insulation		
Insulation thickness	0.4	m
Insulation thermal conductivity	0.13	W/(m*K)
Heat carrier fluid		
Fluid specific heat	4.2	kJ/(kg*K)
Fluid density	976	kg/m^3

The storage volume is in TRNSYS calculated according to Eq. 16, where *h* is the borehole depth, *N* is the number of boreholes and *d* is the distance between boreholes.

$$Storage\ volume\ = \pi \cdot h \cdot N \cdot (0.525 \cdot d)^2 \qquad \textbf{Eq. 16}$$

The distance between boreholes is set constant to 5 m in all simulations. The number of boreholes and the borehole depth will though be altered and hence the storage volume will be different between the simulations.

5.2.2 Heat exchangers

The heat exchangers used in the model between the district heating loop and the borehole loop during injection of heat is of Type5b which models a counter flow heat exchanger. For given inlet temperatures and flow rates on the hot and the cold side of the heat exchanger the effectiveness of the heat exchange is calculated for a fixed overall heat transfer coefficient (UA-value). The Type5b model was developed at the Solar Energy Laboratory, University of Wisconsin – Madison.

The three heat exchangers connected to the three parts of Gärstadverket, 050, 061 and 062, are in the TRNSYS model assumed to be of equal size, each with 1/3 of the total district heating flow respectively 1/3 of the flow in the BTES. The heat exchangers coupled to the sections 061 and 062 are though modeled as one larger heat exchanger taking 2/3 of the respective flows. There is a simple control in the model used to divert the total district heating flow so that 1/3 of the flow passes KV50 and 2/3 of the flow passes KV61-62. This control is used as input to the two flow diverters called "Diverting - DH" and "Diverting – HEF" in the TRNSYS model seen in Figure 11.

The total UA-value is calculated for the three heat exchangers as one large heat exchanger according to Eq. 17 (Granryd, et al., 2011), where $\dot{Q}$ is the heat transfer capacity [W], U is the overall heat transfer coefficient [W/m^2 K], A is the heat exchanger area [m^2] and ϑ_m is the logarithmic mean temperature difference between the fluids [K]. The dimensioning of the heat exchanger is based on the resulted maximum charging power of the BTES from the previous simulated values at Bengt Dahlgren AB, which reached 41 MW, and a pinch point temperature difference of 3 °K.

$$\dot{Q} = U \cdot A \cdot \vartheta_m \qquad \textbf{Eq. 17}$$

The logarithmic mean temperature difference between the hot, T_h, and the cold stream, T_c, flowing thorough the heat exchanger in counter wise directions, is calculated with Eq. 18, where $\vartheta_{in} = T_{h,in} - T_{c,in}$ and $\vartheta_{out} = T_{h,out} - T_{c,out}$ (Granryd, et al., 2011).

$$\vartheta_m = \frac{\vartheta_{in} - \vartheta_{out}}{\ln\left(\frac{\vartheta_{in}}{\vartheta_{out}}\right)} \qquad \textbf{Eq. 18}$$

The UA-value is assumed to be equal among the heat exchangers and is hence divided by 1/3 for KV50 and 2/3 for KV61-62. The resulting UA-value for KV50 and KV61-62 respectively were 2 600 kW/K and 5 200 kW/K.

5.2.3 Heat pump

There is no standard component in TRNSYS for heat pump models. The heat pump is thereby modeled by inserting costom equations in TRNSYS (found under the option *Assembly*). The equations inserted are based on the heat pump models for $\dot{Q}_1, \dot{Q}_2$ *and* P from Eq. 11, with corresponding constants. The input variables to the heat pump model is the temperature entering the evaporator from the borehole loop (the discharge temperature from the BTES), and the volumetric mass flow rate in the district heating and borehole loop. As will be explained in section 5.2.4 the volumetric mass flow rate in the borehole loop is controlled in TRNSYS. The output temperature from the boreholes is calculated iteratively in TRNSYS, while the volumetric mass flow rate and the district heating return temperature in the district heating loop is set by input values from Tekniska Verken, assumed to be equal for all 10 years of simulation. See Figure 10. Furthermore equations 5-10 presented in section 5.1 is included in the heat pump model in TRNSYS. How the operation of the heat pump, and the other components in the TRNSYS model, is controlled is explained in section 5.2.4 below.

5.2.4 Control

The main controls in the TRNSYS model will be further explained in this section. These includes control for winter and summer operation of the BTES, a flow-switch in the BTES, control of the mass flow and control of the heat pumps. The charging and discharging power of the BTES is limited by the available surplus power in summer respectively the available power shortage for discharge in winter. The results from earlier simulations at Bengt Dahlgren AB showed that without a control that limits the available effects for charge and discharge of the BTES it at times exceeds the power limit (Figure 8) for Gärstadverket.

5.2.4.1 Control mode – ONOFF

The summer and winter mode are controlled with respect to the district heating temperature. During summer operation, when the BTES are charged, the district heating temperature is around 95 ˚C, given in the input file from Tekniska Verken AB. The summer mode is hence set to be active when the district heating temperature is above 90 ˚C, see Eq. 19.

$$ONOFF_{summer}: \qquad T_{DH} > 90\ ^{\circ}\mathrm{C} = 1 \qquad \textbf{Eq. 19}$$

$$T_{DH} < 90\ ^{\circ}\mathrm{C} = 0$$

Winter mode is active in the reverse case, that is when the district heating return temperature is below 90˚C, see Eq. 20. During winter operation, when the BTES are discharged, the district heating return temperature mostly varies between 54˚C and 57˚C.

$$ONOFF_{winter}: \qquad T_{DH} < 90\ ^{\circ}\mathrm{C} = 1 \qquad \textbf{Eq. 20}$$

$$T_{DH} > 90\ ^{\circ}\mathrm{C} = 0$$

The discharge of the BTES is furthermore controlled so that no heat is discharge during the first year. In this way the storage is charged for two summer periods before it is discharged for the first time allowing for the temperature in the BTES to raise. This strategy has been applied to other HT-BTES, for example the one in Emmaboda.

5.2.4.2 Thermal stratification – flow switch

The BHE component, Type557b, includes a circulation switch that can be used to set the flow direction of the heat carrier fluid circulated in the boreholes, if these are to be coupled in series. The flow direction is determined by a value of 1 or -1 and is connected to the BHE component as an input parameter. When the circulation switch is equal to 1 the heat carrier fluid is circulated from the center to the border of the storage, and when the input for the circulation switch equals -1 the heat carrier fluid is circulated from the border of the storage to the center. In the TRNSYS model the value for the flow switch is controlled so that when the inlet temperature to the BTES is higher than the outlet temperature the value is equal to 1 (injection of heat), and when the outlet temperature of the BTES is higher than the inlet temperature the value for the flow switch is -1 (extraction of heat), see Eq. 21. In this way thermal stratification of the storage temperature can be simulated.

$$Control_{flow,switch}: \qquad T_{in,BTES} > T_{out,BTES} = 1 \qquad \textbf{Eq. 21}$$

$$T_{in,BTES} < T_{out,BTES} = -1$$

5.2.4.3 Control in summer

For the power limit to not be exceeded during charging of the BTES in summer the flow in either the BTES or the district heating loop can be controlled to limit the heat exchange. Both alternatives were tested in the work by Bengt Dahlgren AB. The alternative to control the charging power by varying the flow on the BTES side has in this project been assumed to be most realistic alternative and is the control strategy used in the simulations. Controlling the flow on the district heating side and as would require a by-pass of part of the flow on the district heating side and as a result two streams of different temperature levels would be mixed, one preheated from the BTES and one still at the district heating return temperature after passing the flue gas condenser. This could affect the control and the efficiency in Gärstadverket. The flow in the BTES does not directly affect the operation of Gärstadverket and is hence preferred.

The flow in the BTES is controlled by an iterative feedback controller in TRNSYS, the Type22. For a setpoint (yset) of a variable (y) the Type22 controller calculates the required control signal based on iterations in TRNSYS with accurate setpoint tracking. The tracking error (e=ySet-y) is minimized (or zeroed) using a secant method with fixed bounds for the control signal. For further information regarding the secant method see the TRNSYS 17 Mathematical Reference manual (SEL, et al., 2014).

In the simulations for HEFAISTOS and Gärstadverket the controlled variable (y) is simulated as the flow in HEFAISTOS and is controlled to vary between 0 and 1.1 m^3/s. The tracking error for the controlled variable, the flow in the BTES, in Type22 is calculated according to Eq. 22, where $abs(HEF_{power})$ is the average heat transfer rate in HEFAOISTOS at a specific time and $Power_{Limit}$ is the available power limit for charging of HEFAISTOS.

$$Control_{flow,summer} = y - ySet = abs(HEF_{power}) - Power_{Limit} \tag{Eq. 22}$$

Type 22 is set to control the tracking error to a value of 0 with a tolerance of 0.001. The output from the Type22 is a value of 0-1.1 and is the required flow in the BTES loop in m^3/s to not exceed the power limit. The model iteratively finds a flow in the BTES loop giving the proper heat exchange with regards to the tracking error $(y - ySet)$.

5.2.4.4 Control in winter

The control during winter operation is different compared to the summer operation. To increase the possible power and temperature levels to be delivered to Gärstadverket heat pumps are used during discharge of the BTES. The system is still required to be controlled so that the power limit is not exceeded, but now in relation to what power the heat pump is able to deliver to Gärstadverket. During discharge it is also a restriction that the district heating network is not able to accept more than 50 MW from the BTES. This is adjusted in the input file for the power limit.

The Type22 iterative feedback controller is used for control of the heat pump operation, now with the controlled variable (y) set to the equivalent number of heat pumps running. A 50 MW heat pump from *Star Renewable Energy* is constituted of approximately 5 heat pumps connected in series. Hence the maximum number of heat pumps is set to 5 and the minimum to zero. The tracking error, $y - ySet,$ for the controlled variable in Type 22 is calculated according to Eq. 23, where $abs(\dot{Q}_1)$ is the average heating power delivered from the heat pump at a specific time, and $Power_{Limit}$ is the available power limit for discharge of HEFAISTOS with a maximum of 50 MW.

$$Control_{HP} = y - ySet = abs(\dot{Q}_1) - Power_{Limit} \tag{Eq. 23}$$

Type 22 is set to control the tracking error to a value of 0 with a tolerance of 0.001. The output from the Type22 is a value of 0-5 and is the equivalent number of heat pumps running on full speed (3 600 *RPM*). It is assumed that the equivalent number of heat pumps in the model is not required to be an integer, but that it can take any value between 0 to 5. As an example, if the equivalent number of heat pumps running is 3.5 in the model it represents that in reality 4 heat pumps are running with the combined compressor speed on the second compressor equal to 3 600 *RPM* times 3.5 heat pumps, equally divided between the 4 heat pumps. This condition is assumed as the power of the heat pumps can be adjusted with the compressor speed but it is a simplification of the reality. The first compressor in each heat pump is always running on nominal speed of 2 950 *RPM*. The Type22 iteratively finds an equivalent number of heat pumps giving the proper heat exchange with regards to the tracking error $(y - ySet)$ so that the heating power delivered from the heat pump is not exceeding 50 MW.

There is a separate control of the flow in the BTES loop. It is controlled to give a proper temperature fall over each evaporator, set to a constant value of 2 ˚K. This control is hence depending on the number of heat pumps running in the previous time step. The tracking error $(y - ySet)$ is iteratively calculated according to Eq. 24.

$$Control_{flow,winter} = y - ySet = (T_{in\_evap} - T_{out\_evap}) - 2 \cdot N_{HP} \quad \textbf{Eq. 24}$$

The controlled variable, (y), is set to the flow in the BTES loop and the output from the Type22 controller is a value of 0 and 1.1 m^3/s giving the tracking error a value of 0 with a tolerance of 0.001. This control is similar to the one controlling the flow in the BTES in summer, with the same controlled variable but with another tracking error.

5.3 Simulation strategy

The BTES system is simulated with different geometries altering the number of boreholes between 1250 and 2000, and the borehole depth between 300 m and 250 m. In total 8 different geometries are modeled, see Table 12. The distance between boreholes is set to a constant value of 5 m, and the number of boreholes in series is set to 3, in all simulations. Simulations with a borehole depth of 200 m were also performed. These are though excluded from the results as the capacity of the BTES became too low. The simulations are run for a 10-year period, starting in 1 of May and ending in 30 of April each year. During the first year of operation no discharge is done. This gives two charging periods of the BTES before discharge starts.

Table 12 The input data for the borehole thermal energy storage geometry used in each case. All simulations are made with 5 m borehole distance and boreholes coupled in series of 3.

Case	Number of Boreholes	Borehole Depth [m]
1.1	2 000	300
1.2	2 000	250
2.1	1 250	300
2.2	1 250	250
3.1	1 500	300
3.2	1 500	250
4.1	1 250	300
4.2	1 250	250

Based on the results given from the simulations for the cases presented in Table 12, new cases optimizing the size of the BTES were run. The optimization is done with regard to energy, power and temperature levels in the BTES, as well as efficiency and costs for operation and investment for the different cases. One criterion is to size the BTES to reach a peak power of 50 MW when needed. It can though be considered that it is not required to fulfill this criterion in the first simulation years, especially in the end of the discharge period, since the capacity of the system increases over time. Some results are thereby only shown for the fifth simulation year.

6 Results and analysis

The results from the new simulations based on this project's model will be outlined in this chapter. The results are divided in three main sections. First some key results are presented for the reference case with 1 500 boreholes and 300 m borehole depth, case 3.1. See Table 12 in section 5.3 for a list of all cases. Case 3.1 is the geometry that was used in the simulations preformed at Bengt Dahlgren AB and that has been used as basis for this study. Secondly a more detailed comparison of some key parameters is presented for the cases in Table 12. These results are shown to give an overview of how the capacity of the system changes when the size of the BTES is varied. All simulations are done with double U-pipe collectors installed in the BHE. The final result for the optimized geometry is presented in the third part of this chapter. This geometry is further optimized with coaxial collectors.

If possible, the results are shown for the total simulation period of ten years. Some results are though presented only for the third and the tenth year of operation to ease the visualization. For some parameters the results are furthermore only shown for the discharge period.

6.1 Reference system

In this section results for the reference geometry, 1 500 boreholes with 300 m depth, is presented. In Figure 13 the system temperatures can be seen for the third simulated year. The diagram shows the inlet and outlet district heating temperature for each heat exchanger, KV50 and KV6162, used for charging the BTES during summer operation. Also, inlet and outlet temperatures of the BTES are presented over the year, as well as the outlet temperature from the condensers. This is the temperature that is supplied to the district heating system during winter operation and is related to the power limit and volumetric flow rate in the district heating system. Depending on whether the system is run on charge or discharge mode, the outlet temperature from the BTES is respectively the inlet temperature to the heat exchangers or the inlet temperature to the heat pump evaporators. In the fifth year of operation the discharge temperature from the BTES reaches around 46 ˚C in the end of the discharge period.

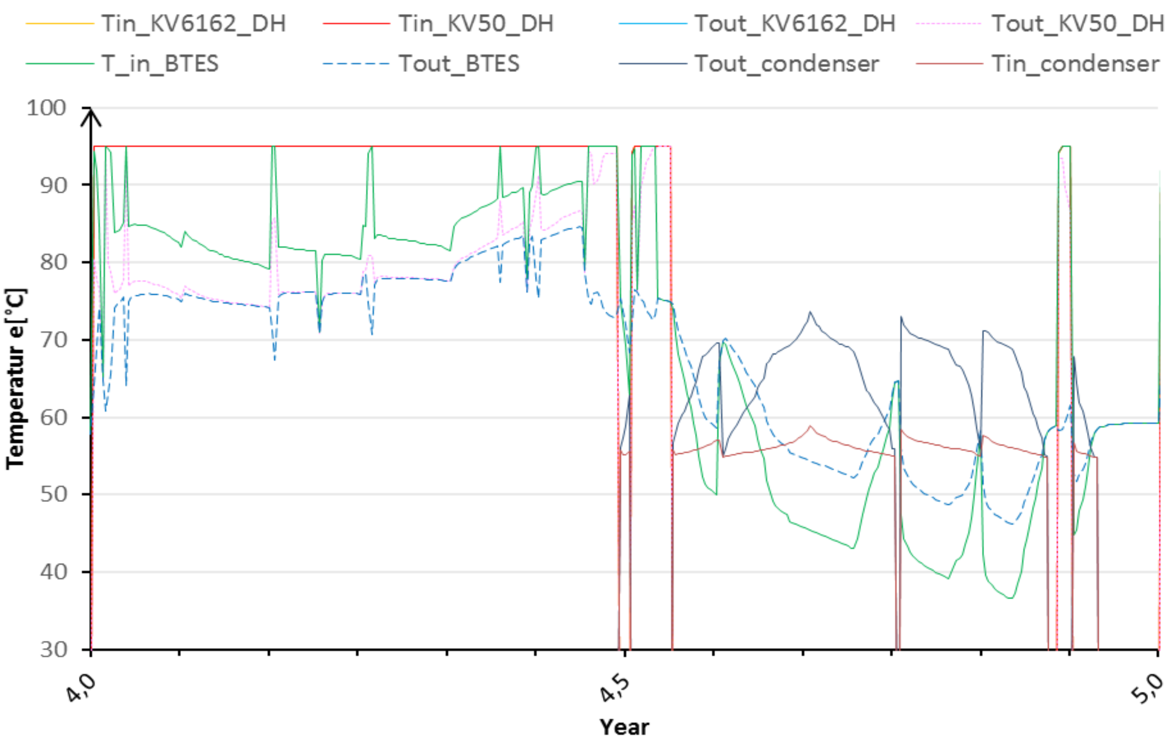


Figure 13 Results for the geometry of 1 500 boreholes with borehole depth 300 m regarding the temperatures in and out from the BTES, heat exchangers and heat pumps. Heat exchangers are used during charging of the BTES, and heat pumps are used during discharge of the BTES.

In Figure 14 the total amount of charged and discharged energy over the 10 simulation years can be seen. Also, the mean outlet temperature from the BTES during discharge, and the energy ratio regarding charged versus discharged energy is presented. No consideration has been taken to the storage temperature or power levels when calculating the energy ratio. The ratio could hence exceed 100 % if more energy would be discharged from the BTES than what have been charged. This would though result in a decreased storage temperature over time as the BTES would be overexploited.

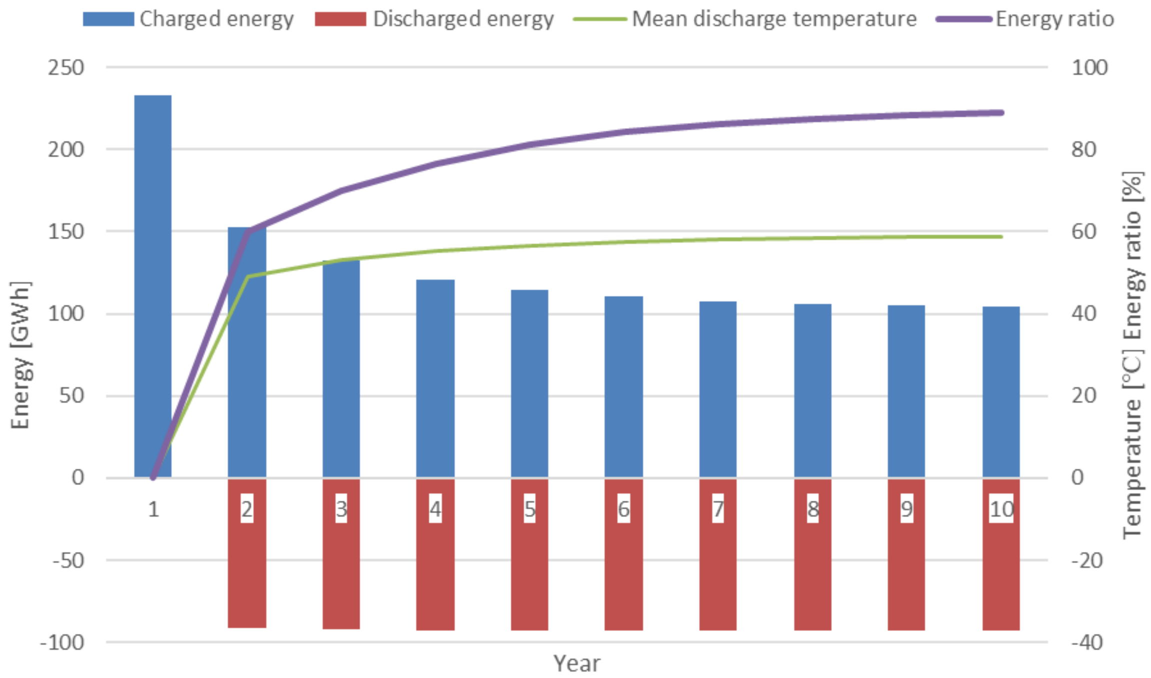


Figure 14 Results for the geometry of 1500 boreholes with 300 m depth regarding the amount of charged and discharged energy, the mean outlet temperature from the BTES during discharge and the energy efficiency of the BTES.

In Figure 15 the different power levels in the system is presented. The power limit shows the power surplus that is available in Gärstadverket to charge to the BTES (positive values) as well as the power demand in winter when the district heating production in Gärstadverket is insufficient (negative values). The goal is to cover the power demand in winter by discharging the BTES system, including heat pumps. During discharge the power limit is though set to maximum 50 MW in the model. As can be seen the BTES system, including heat pumps, can deliver a heating power of 50 MW when designed with 1 500 boreholes and 300 m borehole depth. The charged and discharged power furthermore adapts so that the power limit is never exceeded. The discharge power from the BTES and the cooling power, $\dot{Q}_2$, have the same values as heat losses in the pipes between the BTES and the heat pumps have not been taken into account.

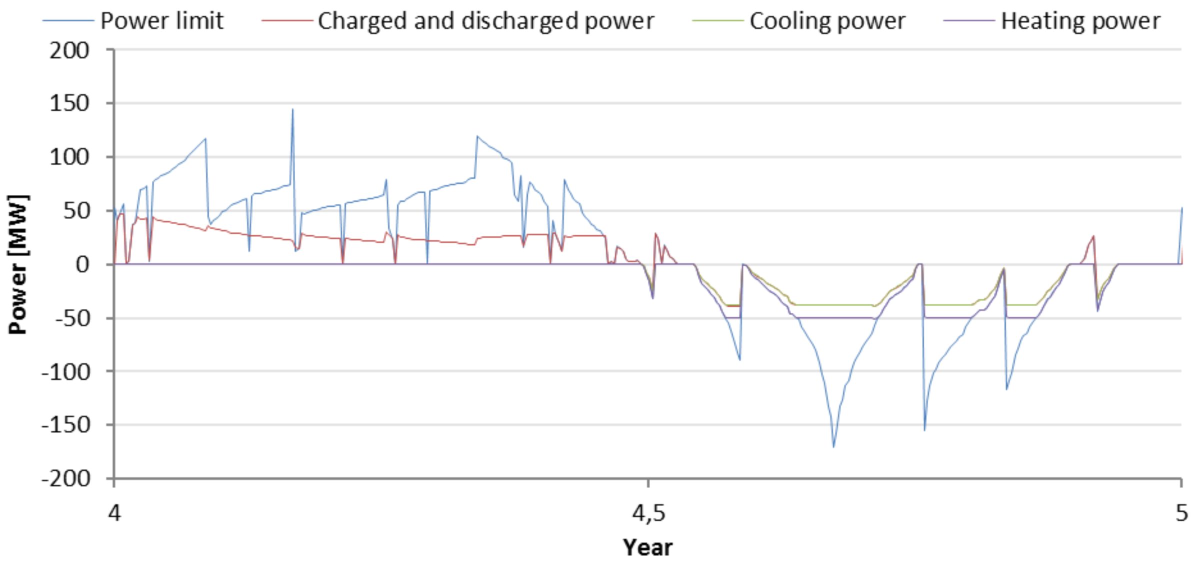


Figure 15 Results for the geometry of 1500 boreholes with 300 m depth regarding charged and discharged power of the BTES and the heating and cooling power of the heat pumps.

As previously mentioned, the number of heat pumps running in the model is not necessarily an integer, but it is representing an equivalent number of heat pumps running on full capacity. The actual number of heat pumps is always rounded up and works with variable compressor speed at the high stage compressor. The first compressor in each heat pump is always running on nominal speed (2950 RPM). See section 5.2.4.4 for a more detailed description of how the model works. In Figure 16 the heat pump model is illustrated for the reference case. The heat pump model is a simplification of the reality but is capturing the general behavior of the system.

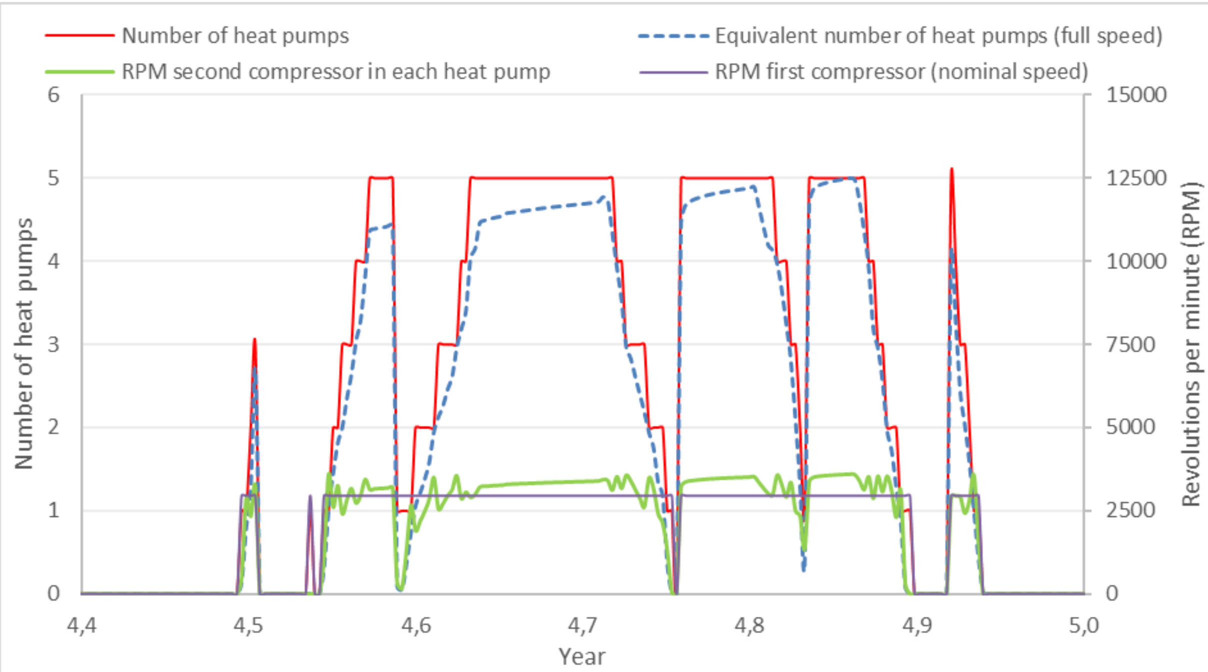


Figure 16 Visualization of how the heat pump model works regarding the number of heat pumps running and the compressor speed of the two stage compressors in each heat pump.

In Figure 17 some key indicators for the performance of the system is shown. The Figure shows the heating coefficient of performance (COP_h), the heating power ($\dot{Q}_1$), the compressor power (P_{tot}), and the number of heat pumps running at full speed (as in the model). The COP_h is calculated based on the relation between the heating power and the compressor power (including compressor efficiency). The high peaks in the COP_h occurs when both the compressor power and the heating power approaches zero and is an effect from the iterations made in TRNSYS.

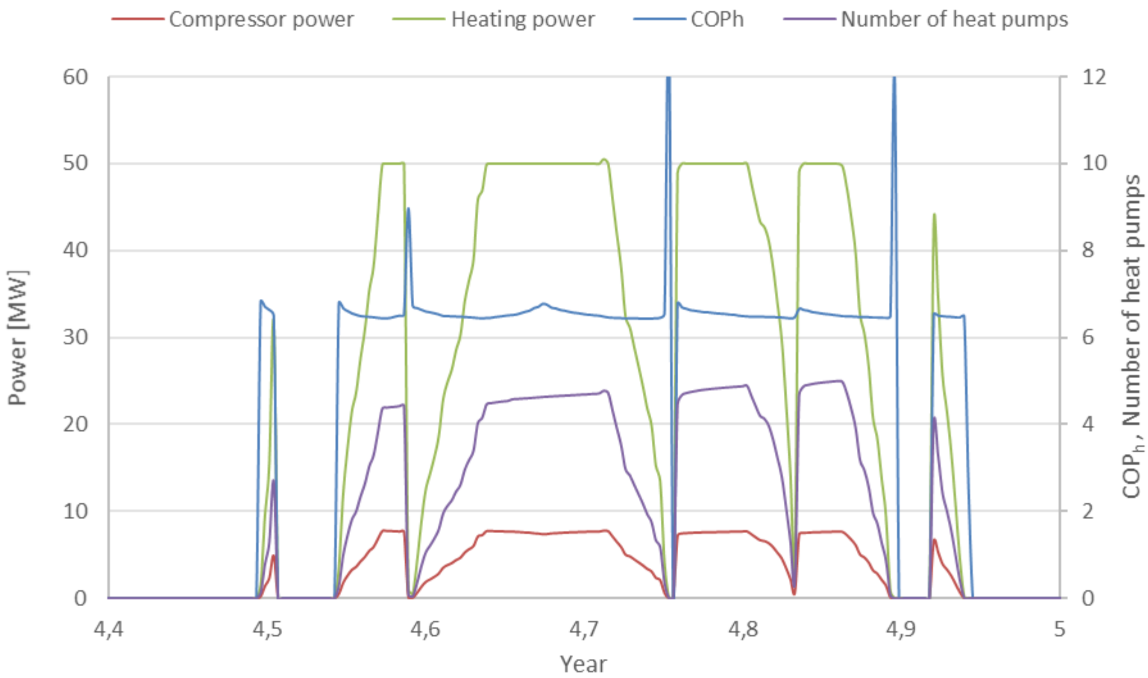


Figure 17 Results for the geometry with 1 500 boreholes and 300 borehole depths regarding the equivalent number of heat pumps (running on full capacity), the compressor power, the heating power and the heating coefficient of performance of the heat pumps.

In Figure 18 the seasonal performance factor (SPF) of the heat pumps is shown. The SPF only accounts for the delivered energy, $\dot{Q}_1$, from the heat pumps and the compressor power, P_{tot}. The energy used for the circulation pumps is not included. It can be seen that the performance of the system increases with time.

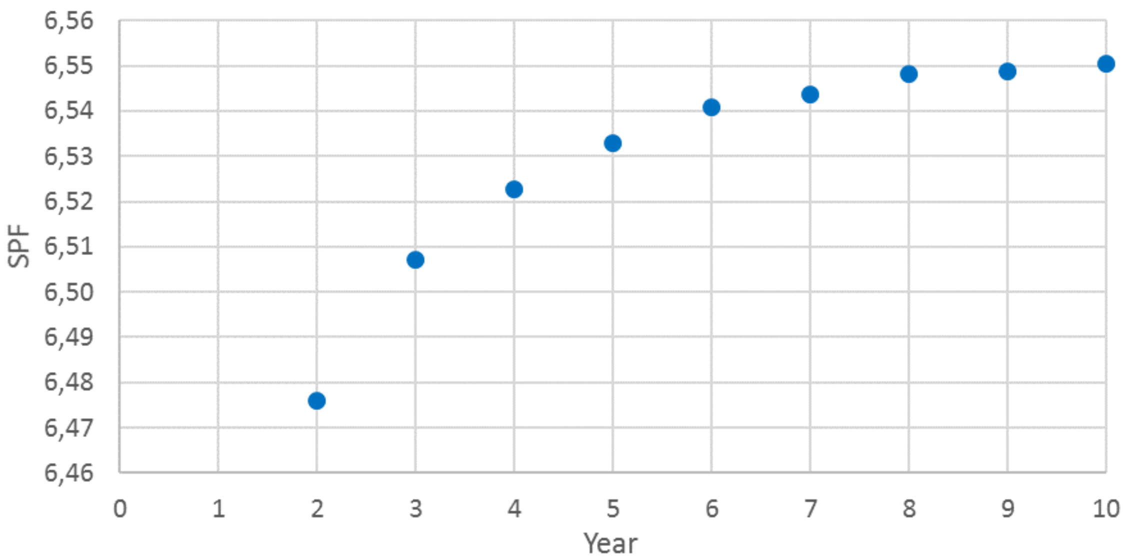


Figure 18 Seasonal performance factor of the heat pumps, excluding energy consumption of the circulation pumps. Reference system with 1 500 boreholes and 300 m borehole depth.

In Figure 19 the pressure drop in the collectors is shown for double U-pipe and coaxial design. The pressure drop is lower for coaxial collectors, as can be expected.

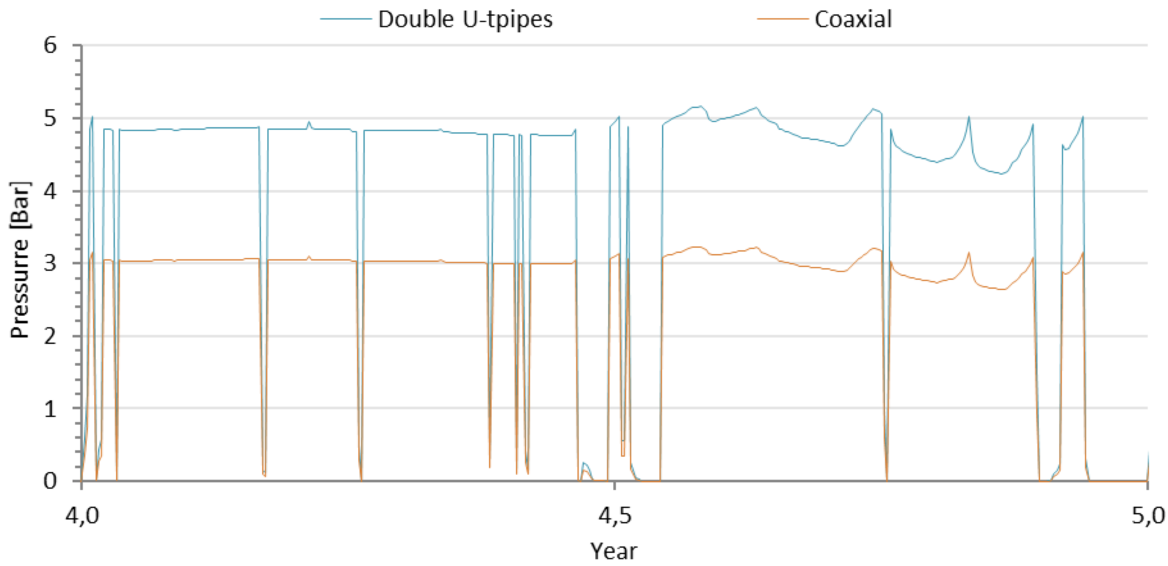


Figure 19 Pressure drop in the collectors for double U-pipes and coaxial design.

In Figure 20 the energy consumption of the system is shown for the circulation pumps in the BTES and the compressors in the heat pumps. When coupling boreholes in series the pressure drop in the system gets high, as seen in Figure 19 Pressure drop in the collectors for double U-pipes and coaxial design., resulting in rather high energy consumption for the circulation pumps.

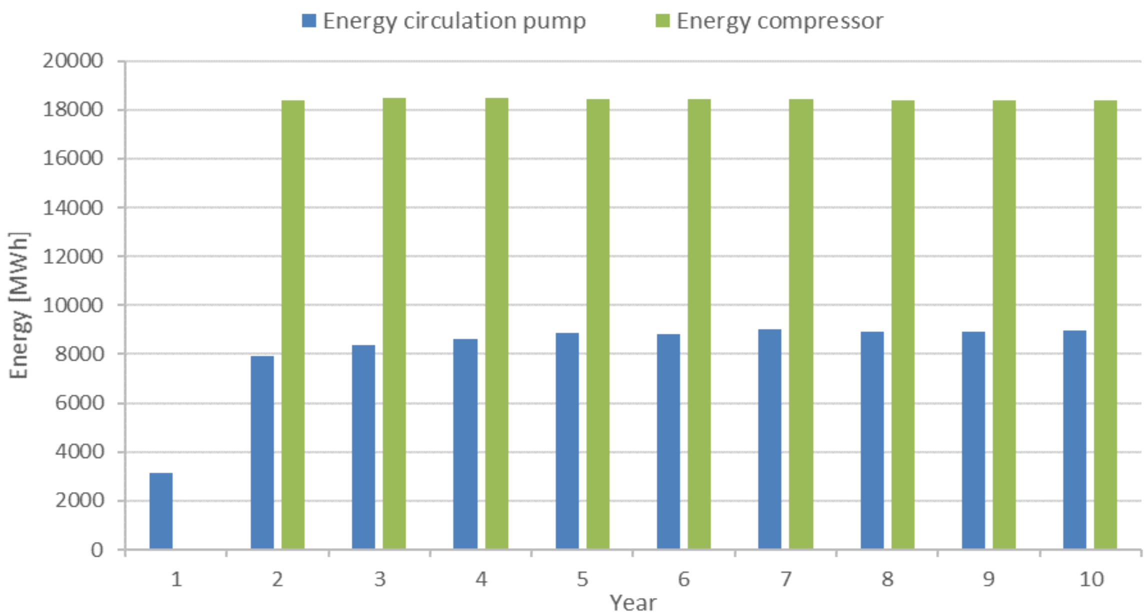


Figure 20 Energy consumption for the circulation pumps in the BTES and the compressors, simulated with double U-tube collectors.

6.2 Evaluation of the model

In this section results from the 10 cases presented in Table 12 will be evaluated. The comparison includes the heat power delivered to Gärstadverket ($\dot{Q}_1$), the number of heat pumps running, average storage temperature, storage outlet temperature during discharge and the pressure drop in the collectors.

6.2.1 Heating power delivered to *Gärstadverket*

In Figure 21 and Figure 22 the heating power delivered from the heat pumps during discharge period is shown for the geometries with borehole depth of 250 m. For a borehole depth of 250 m none of the simulated geometries can deliver a heating power of 50 MW in the end of the discharge period (Figure 21). For the tenth year of operation (Figure 22) both the geometries with 2000 and 1750 boreholes reaches the peak power limit also in the end of the discharge period. This shows that the capacity of the BTES increases over time.

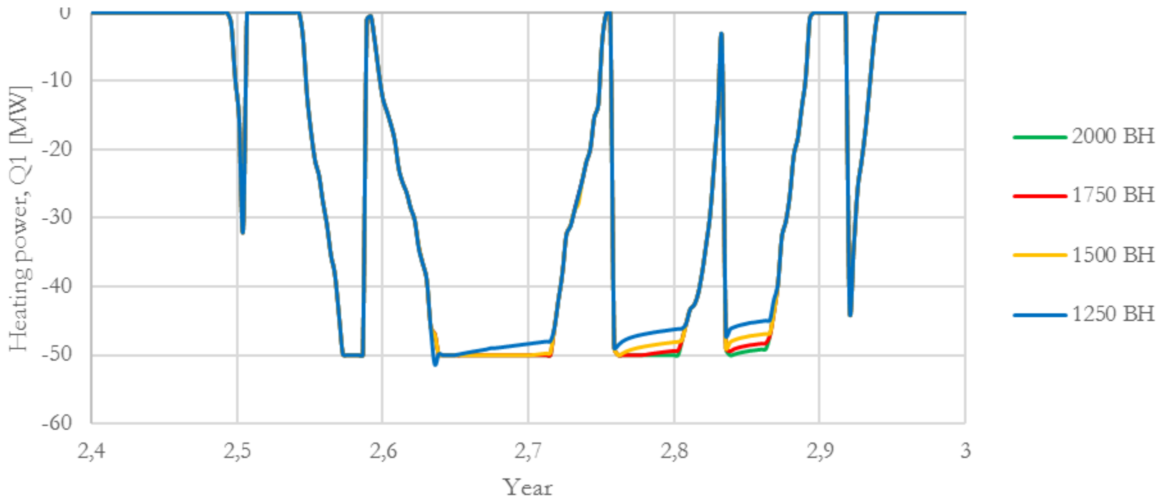


Figure 21 Results for the cases 1.2, 2.2, 3.3 and 4.2 with borehole depth 250 m regarding the heating power delivered to *Gärstadverke*t from the BTES system including heat pumps during discharge period in year 3.

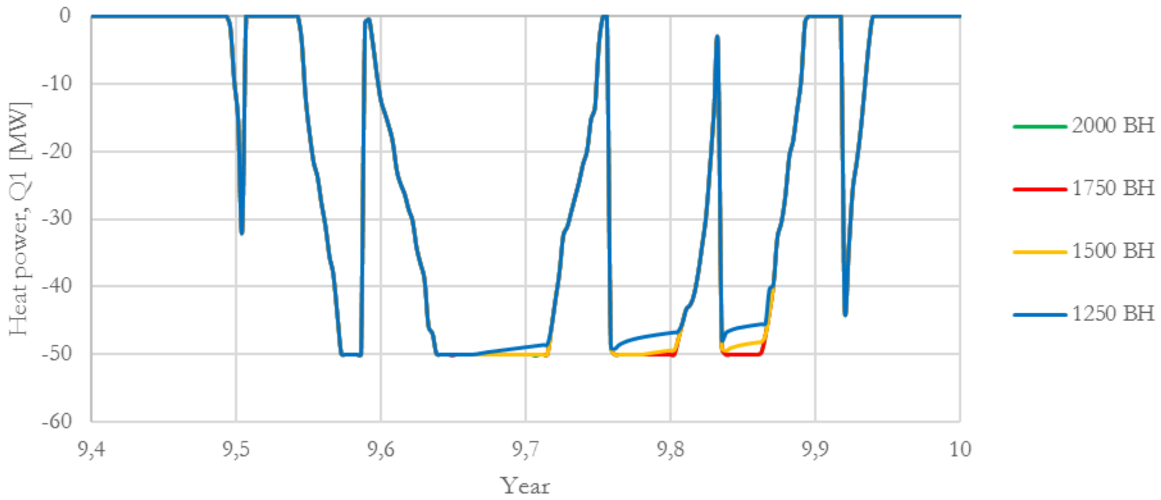


Figure 22 Results for the cases 1.2, 2.2, 3.3 and 4.2 with borehole depth 250 m regarding the heating power delivered to *Gärstadverket* from the BTES system including heat pumps during discharge period in year 10.

In Figure 23 results for the geometries with a borehole depth of 300 m is shown for the tenth simulation year. For a borehole depth of 300 m all geometries, except the one with 1250 boreholes, can deliver the 50 MW when required during the tenth simulation year.

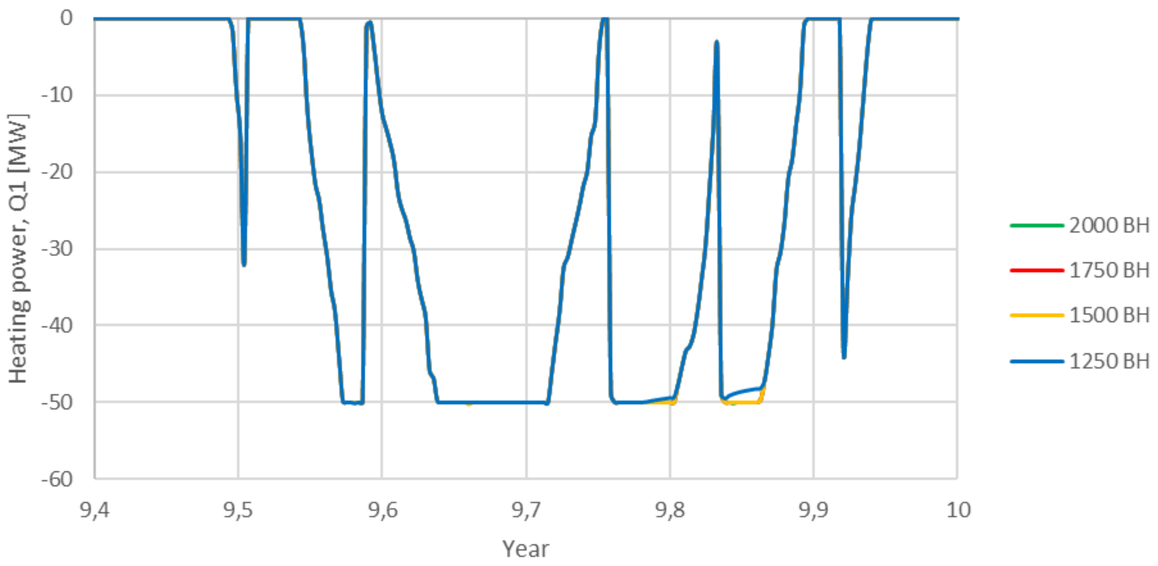


Figure 23 Results for the cases 1.1, 2.1, 3.1 and 4.1 with borehole depth 300 m regarding the heating power delivered to *Gärstadverket* from the BTES system including heat pumps during discharge period in year 10.

6.2.2 Heat pump

The number of heat pumps running for the geometries with a borehole depth of 300 m are presented in Figure 24 and Figure 25, for the third and tenth simulation year respectively. With the number of heat pumps, it is in this section referred to the equivalent number of heat pumps running on full speed, and is not necessarily an integer. See Figure 16 for an illustration of how the heat pump model works. It can, in Figure 24 and Figure 25 be seen that the required number of heat pumps is increasing with time of a discharge period. If comparing the two diagrams it can though be observed that the number of heat pumps required during peak heat demand overall has decreased for the tenth year of operation compared to the third. These results can be correlated with that the capacity of the system increases with time, as was shown in section 6.2.1 above.

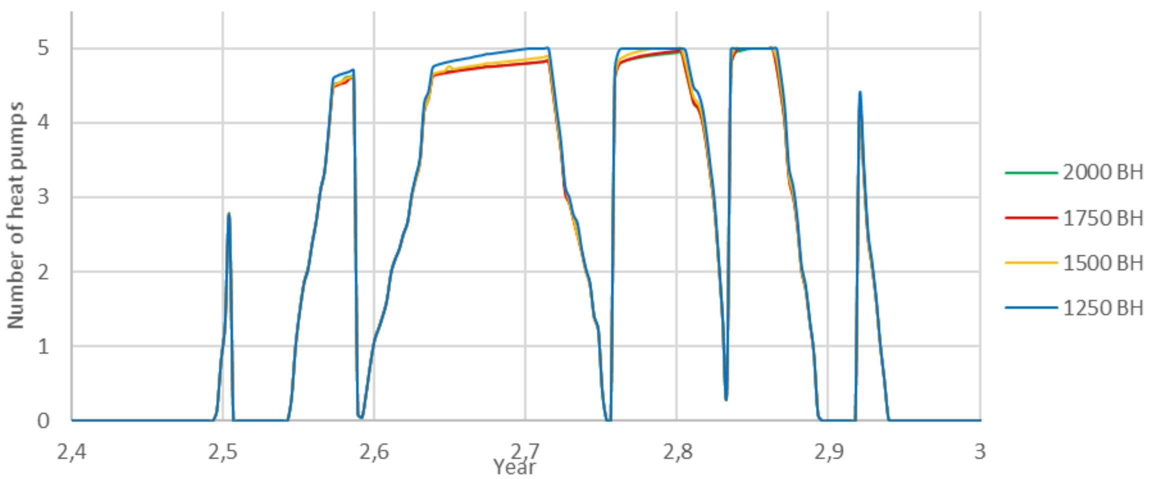


Figure 24 The equivalent number of heat pumps running on full speed during the discharge period in the third simulation year, for case 1.1, 2.1, 3.1 and 4.1 with borehole depth 300 m.

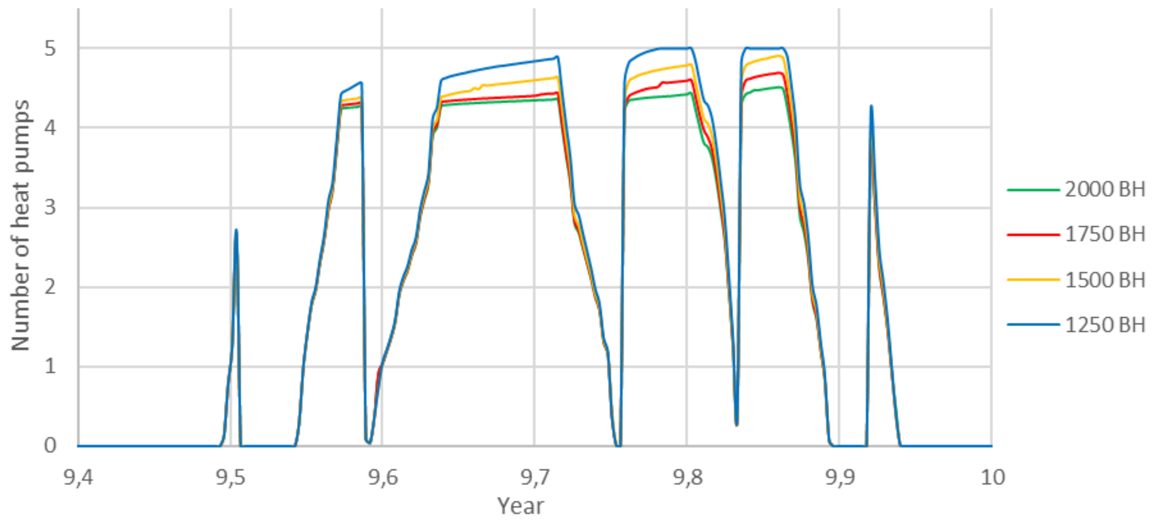


Figure 25 The equivalent number of heat pumps running on full speed during the discharge period in the tenth simulation year, for case 1.1, 2.1, 3.1 and 4.1 with borehole depth 300 m.

6.2.3 Average storage temperature

Figure 26 shows the average storage temperature for the different geometries with a borehole depth of 300 m. During the first simulation year, when the storage is only charged, the storage temperature increases faster for the BTES geometries with fewer boreholes. However, already in the end of the discharge period in year three the storage temperature is lowest for the BTES geometry with fewest number of boreholes (case 4.1 with 1250 boreholes). After a few simulation years the maximum average storage temperature, at the end of charging period, is rather similar for the different cases. The variation in storage temperature is hence larger for smaller sizes of the BTES, and the storage temperature is both increasing and decreasing faster than for a BTES with a larger storage volume. A larger storage volume can hence be said to have a higher thermal inertia. Overall, the average storage temperature is higher the larger the BTES is.

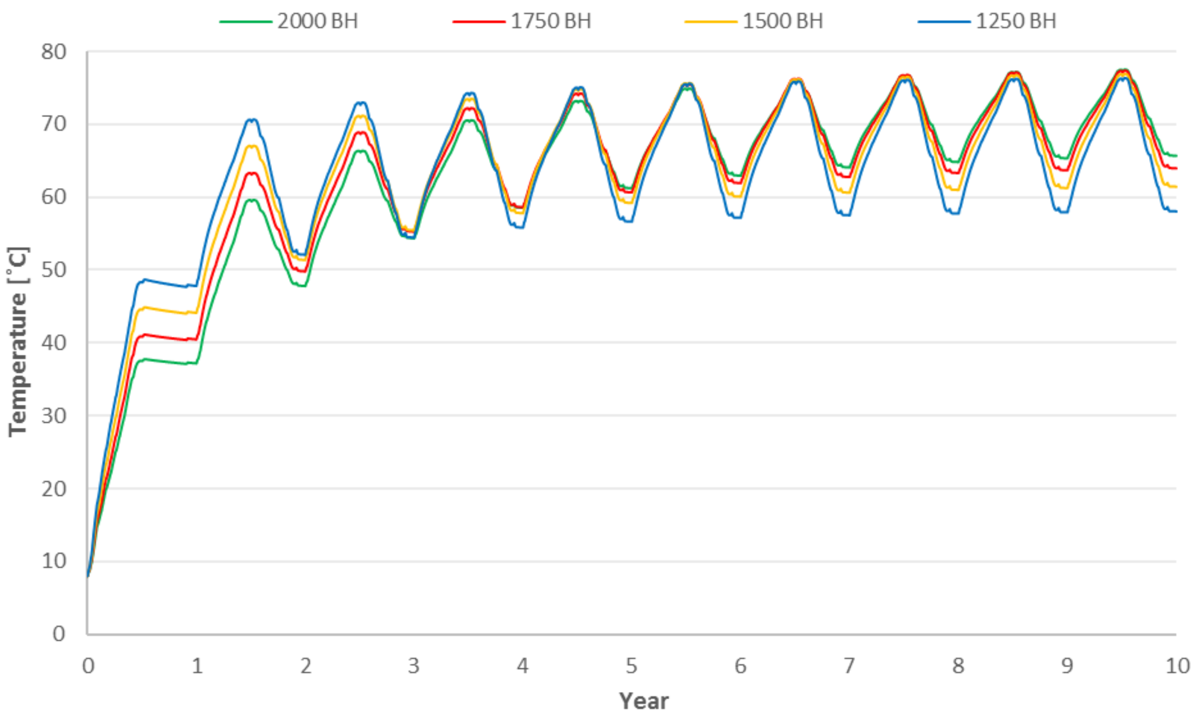


Figure 26 Results for the cases 1.1, 2.1, 3.1, 4.1 and 5.1 with 300 m borehole depth for the average storage temperature for the 10 year simulation period.

6.2.4 Storage outlet temperature during discharge

In Figure 27 and Figure 28 the outlet storage temperature can be seen for the geometries with 300 m borehole depth, for the third and the tenth simulation year respectively. The minimum storage outlet temperature is decreasing with decreasing size of the BTES (decreasing number of boreholes). If comparing the third year of operation (Figure 27) with the tenth year of operation (Figure 28) it shows that the storage outlet temperature is increasing with time, especially for geometries with higher number of boreholes. The trends are similar as for the average storage temperature shown in Figure 26.

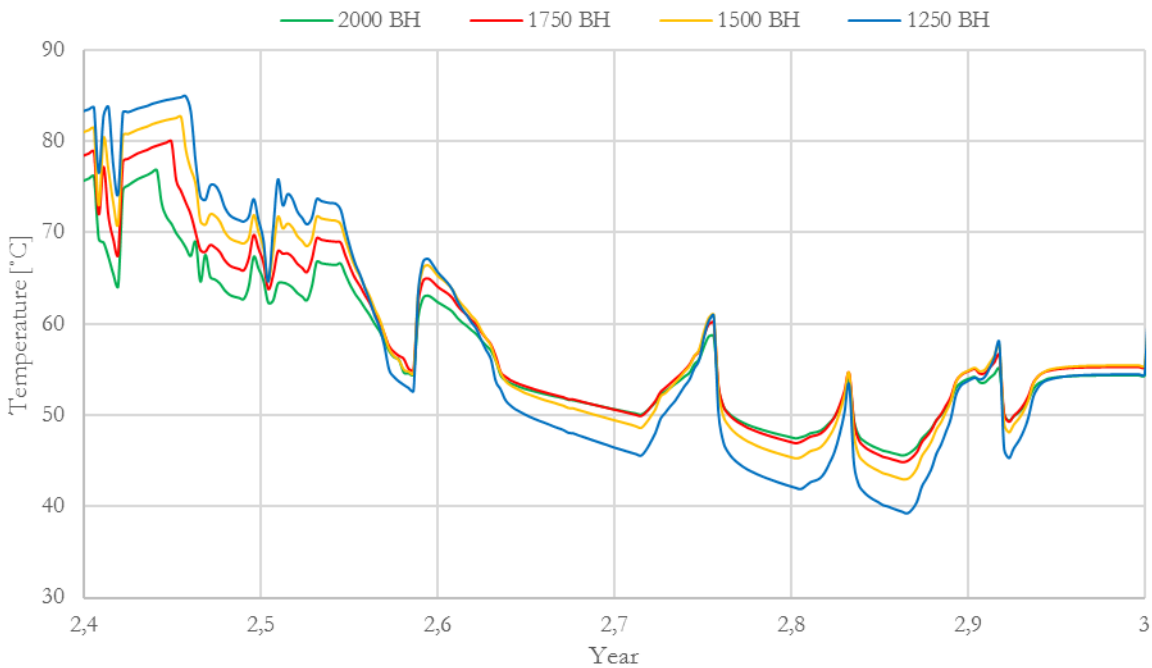


Figure 27 Results for the cases 1.1, 2.1, 3.1, 4.1 and 5.1, with 300 m borehole depth, for the outlet temperature of the BTES during the discharge period in the third simulation year.

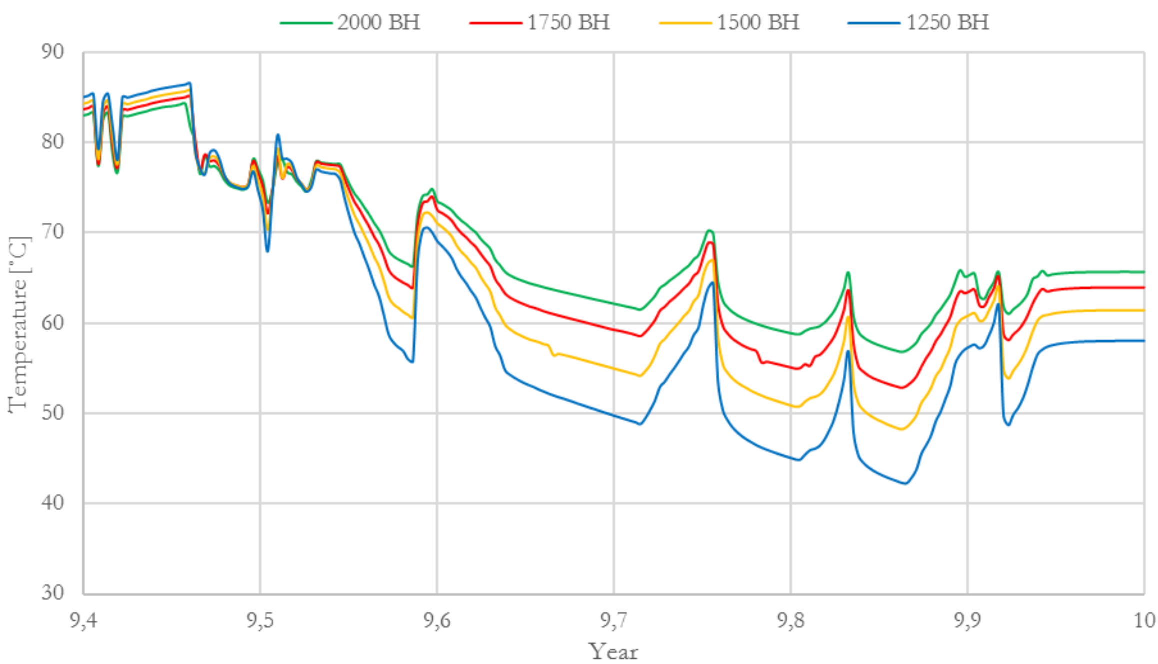


Figure 28 Results for the cases 1.1, 2.1, 3.1, 4.1 and 5.1, with 300 m borehole depth, for the outlet temperature of the BTES during the discharge period in the tenth simulation year.

6.2.5 Seasonal performance factor of the heat pumps

In Figure 29 the seasonal performance factor (SPF) of the heat pumps is presented for the geometries with borehole depth 300 m. It can be seen that the SPF increases for each season, but that it decreases with decreasing size of the BTES. Energy consumption of the circulation pumps has been disregarded in this calculation.

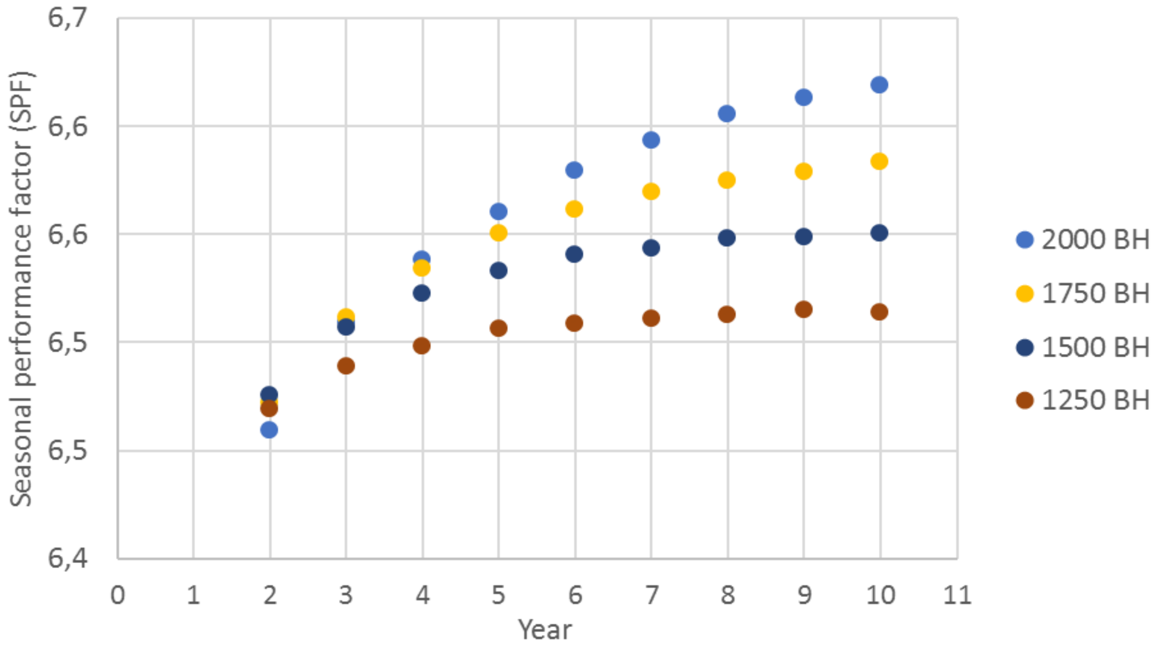


Figure 29 Seasonal performance factor for the heat pump over the ten-year simulation time, for the cases 1.1, 2.1, 3.1, 4.1 and 5.1, with 300 m borehole depth.

6.2.6 Pressure drop

In Figure 30 the maximum pressure drop is shown for the geometries with borehole depth 300 m. When decreasing the number of boreholes, the pressure drop increases. This applies for both double U-tube collectors and coaxial collectors since the range of the volumetric flow rate (0-1.1 m3/s) is the same in all cases. If comparing the pressure drop for double U-pipe and coaxial collectors it can furthermore be seen that the pressure drop overall is lower for coaxial collectors.

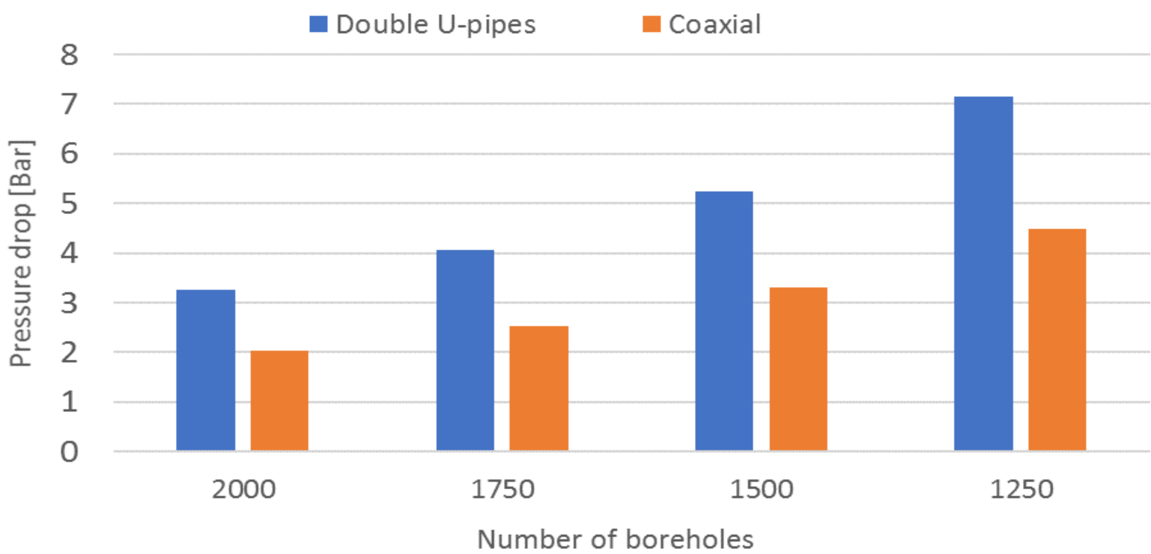


Figure 30 Pressure drop in the collectors for double U-pipe and coaxial design with borehole depth 300 m.

6.3 Optimizing the size of the BTES

From the results of the first simulations the criterion of 50 MW was reached for the BTES geometry with 1 500 boreholes with borehole depth of 300 m. With smaller size of the BTES, 1250 boreholes and borehole depth of 300 m as well as 1500 boreholes with borehole depth of 250 m, the criterion of 50 MW was not entirely reached. Hence, the optimization of the size of the BTES is done considering configurations of 1 500 and 1 250 boreholes with a borehole depth ranging between 300 m and 250 m. However, as seen in section 6.2.4 a decreasing size of the BTES results in a lower outlet temperature from the BTES during discharge. This affects the performance of the heat pumps and the seasonal performance factor negatively. Hence, it would not be feasible to build a too small BTES since it would result in higher operational costs. When considering the feasibility of the BTES system the marginal cost for saved energy in the total energy system is furthermore required to be accounted for. This is however not included in the scope of this project.

An economical estimation has been done for the different cases including costs for drilling, steel pipes (6 m per borehole), collector materials and collector mounting (double U-pipes), drilling lids, horizontal pipes and pipe parts as well as estimated labor costs. Costs for excavation and insulation on top of the storage insulation is not included. The installation cost for the geometry with 1 500 boreholes and 300 m borehole depth with double U-pipes is used as a reference. In Table 13 below, the difference in the installation cost for the BTES is shown in relation to the reference geometry. Furthermore, the difference in operational costs in relation to the reference geometry (1500 boreholes with 300 m borehole depth and double U-pipe collectors) can be seen. The operational costs are presented for the third simulation year and includes operational energy for the circulation pumps in the BTES and for the compressors.

Table 13 Difference in installation costs and operation al cost [%] of the BTES compared to the reference case of 1 500 boreholes, 300 m borehole depth and double U-pipes and the amount of saved energy for the different systems. The numbers are presented for the third year of operation.

Number of boreholes	Borehole depth	Collector type	Investment cost	Operational cost	Saved energy [GWh]
1 500	300	Double U-pipe	0%	0%	93.3
1 500	275	Double U-pipe	-4.6%	-1.9%	93.5
1 400	300	Double U-pipe	-6.6%	4.1%	92.0
1 300	300	Coaxial	-13.2%	-3.2%	94.3

Based on the results the cases 1 400 boreholes and 1 300 boreholes with double U-pipes and coaxial collectors respectively, both with borehole depth 300 m, and the case with 1 500 boreholes and 275 m borehole depth (double U-pipes) is considered most feasible. These designs are both cheaper in installation and operational cost than the reference case, except the higher operational cost for the case with 1 400 boreholes due to that the pressure drop in the system increases when the number of boreholes decreases with all other parameters kept the same. This evaluation does however not consider marginal costs from saved energy. The designs are furthermore fulfilling the desired goal of a 50 MW power supply during peak heat demand as will be shown in section 6.3.1. Hence it is considered good designs both in a thermal and economical perspective. The results for these three designs is presented in the next section. The design with 1 400 boreholes and double U-pipe collectors is here on referred to as Case A, and the design with 1 300 boreholes and coaxial collectors is referred to as Case B (both with 300 m borehole depth) and the design with 1 500 boreholes and double U-pipes with a borehole depth of 275 m is referred to as Case C.

6.3.1 Results for the optimized geometries

In Figure 31 below some key indicators regarding the charged and discharged energy can be seen for Case A (the geometry with 1 400 boreholes and 300 m borehole depth) simulated with a borehole resistance of 0.05 (assumed for double U-pipe collectors). The diagram is also showing the mean outlet temperature from the BTES during discharge and the energy ratio regarding charged versus discharged energy. As mentioned earlier, no consideration has been taken to the storage temperature or power levels when calculating the energy ratio. In Figure 32 the corresponding result can be seen for Case B (the geometry of 1 300 boreholes and a borehole depth of 300 m) simulated with a borehole resistance of 0.0275 (assumed for coaxial collectors). If comparing the two results it can be seen that the mean outlet temperature from the BTES during discharge is higher for Case B. This can be explained by the improved heat exchange between the heat transfer fluid and the borehole wall with a lower borehole resistance. Thereby the BTES has a larger capacity to exchange heat.

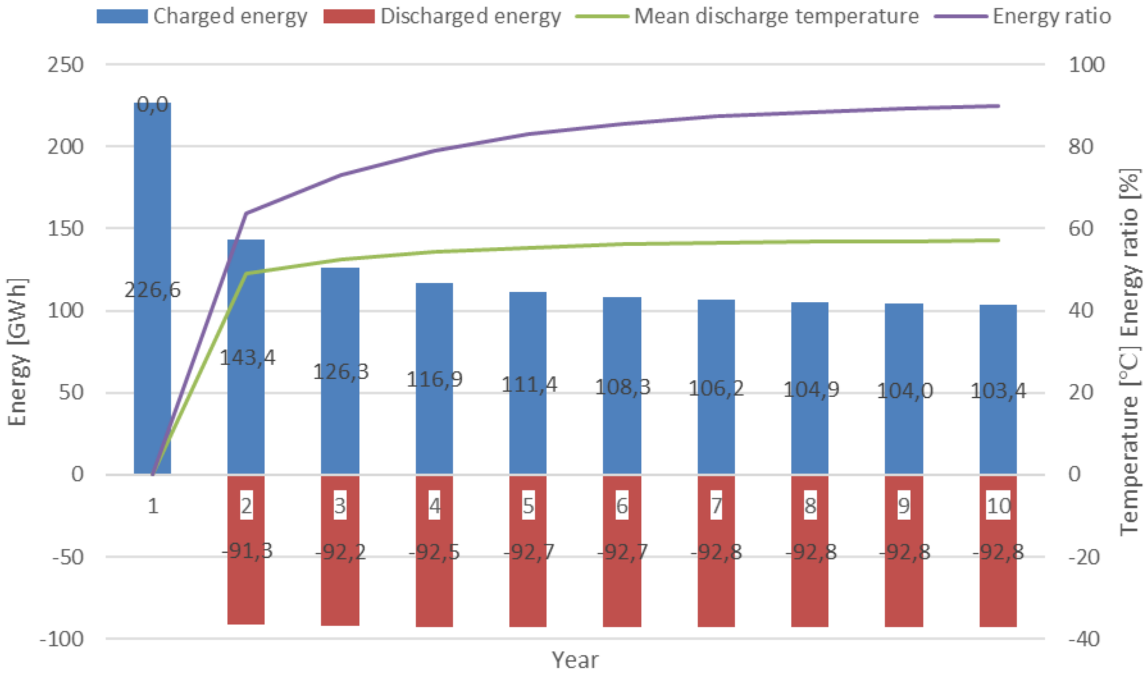


Figure 31 Results for the geometry of 1 400 boreholes with 300 m depth and double U-pipe collectors, regarding the amount of charged and discharged energy, the mean outlet temperature from the BTES during discharge and the energy efficiency of the BTES.

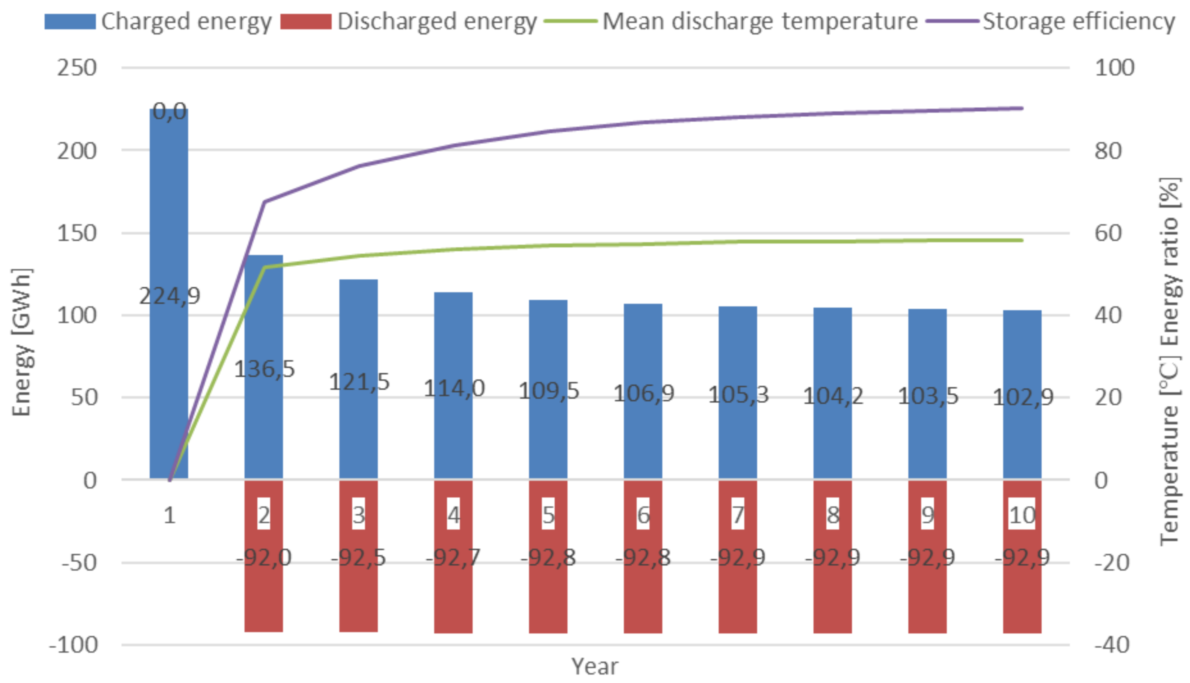


Figure 32 Results for the geometry of 1 300 boreholes with 300 m depth and coaxial collectors, regarding the amount of charged and discharged energy, the mean outlet temperature from the BTES during discharge and the energy efficiency of the BTES.

In Figure 33 the corresponding result is shown for Case C with 1 500 boreholes, 275 m depth and double U-pipe collectors. The results are very similar for the different geometries with comparable system capacity.

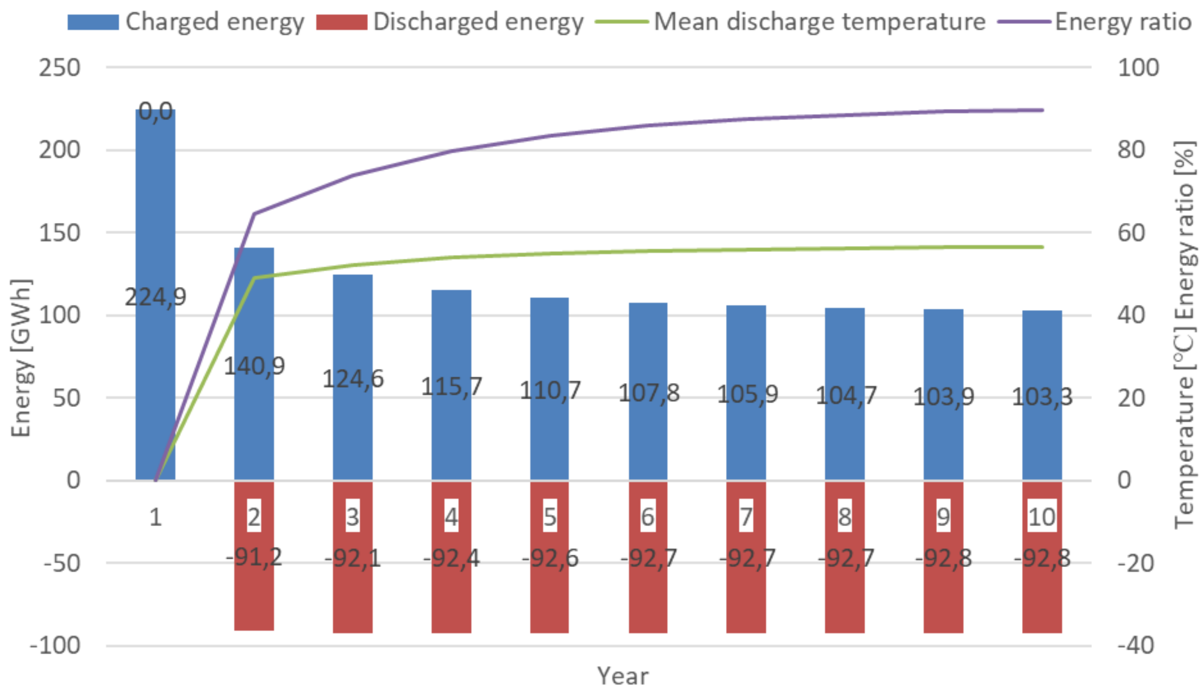


Figure 33 Results for the geometry of 1 500 boreholes with 275 m depth and double U-pipe collectors, regarding the amount of charged and discharged energy, the mean outlet temperature from the BTES during discharge and the energy efficiency of the BTES.

In Figure 34 the result for Case A can be seen regarding charged and discharged power and the delivered heating power to Gärstadverket. As can be seen this geometry fulfills the criterion of 50 MW heating power at peak heat demand, with an exception for the very last part of the discharge period when the heating power is around 49 MW. Similar result is gained for Case B and Case C, as can be seen in Figure 35 and Figure 36.

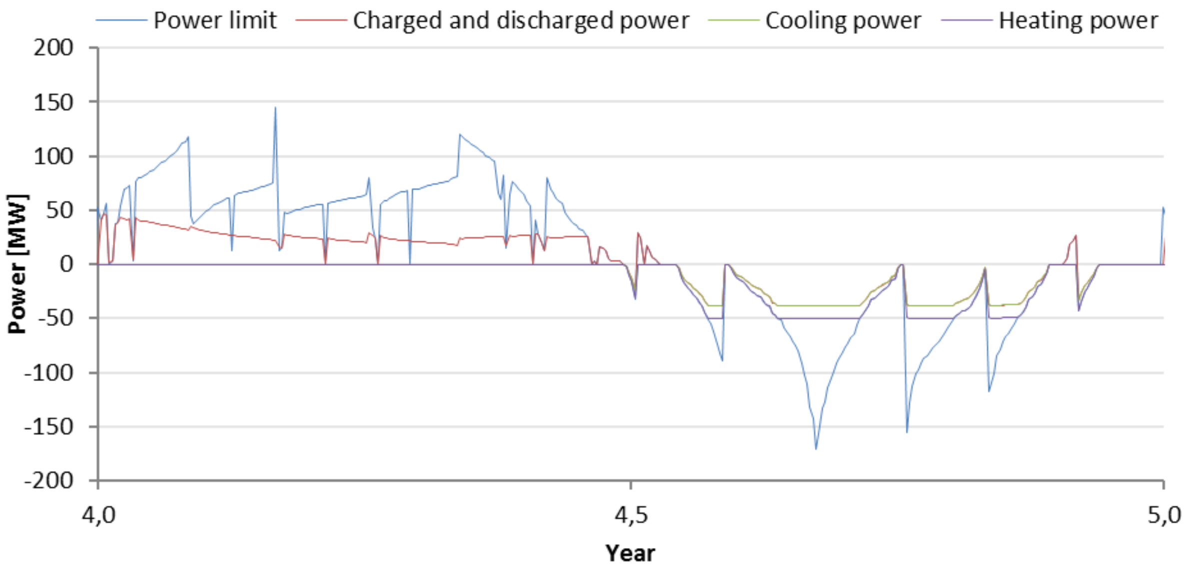


Figure 34 Results for the geometry of 1 400 boreholes with 300 m depth (double U-pipes) regarding charged and discharged power of the BTES and the heating and cooling power of the heat pumps.

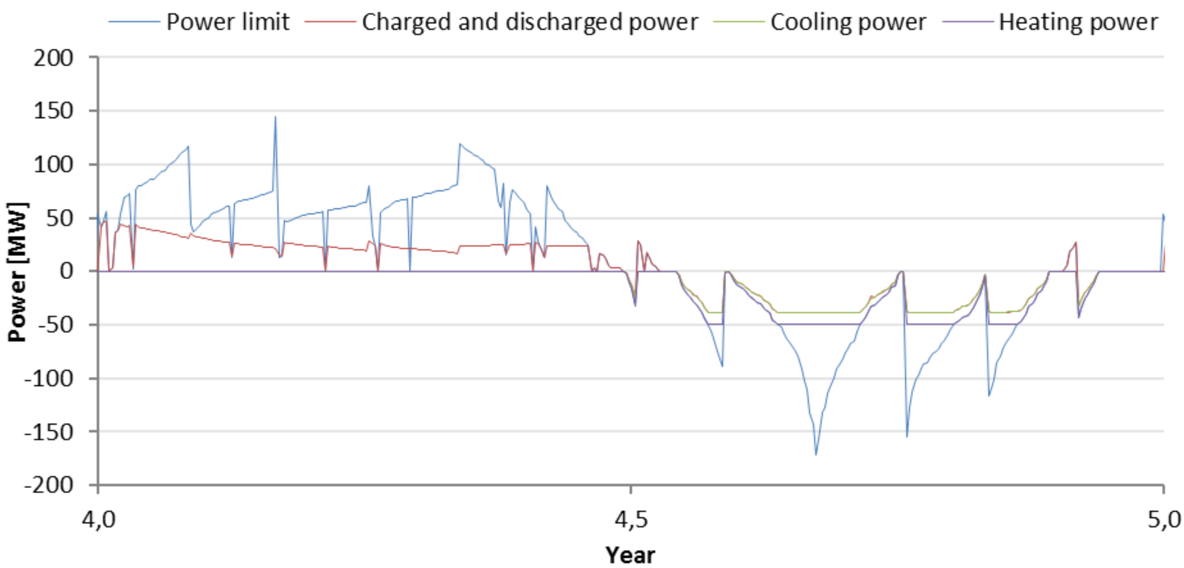


Figure 35 Results for the geometry of 1 300 boreholes with 300 m depth (coaxial) regarding charged and discharged power of the BTES and the heating and cooling power of the heat pumps.

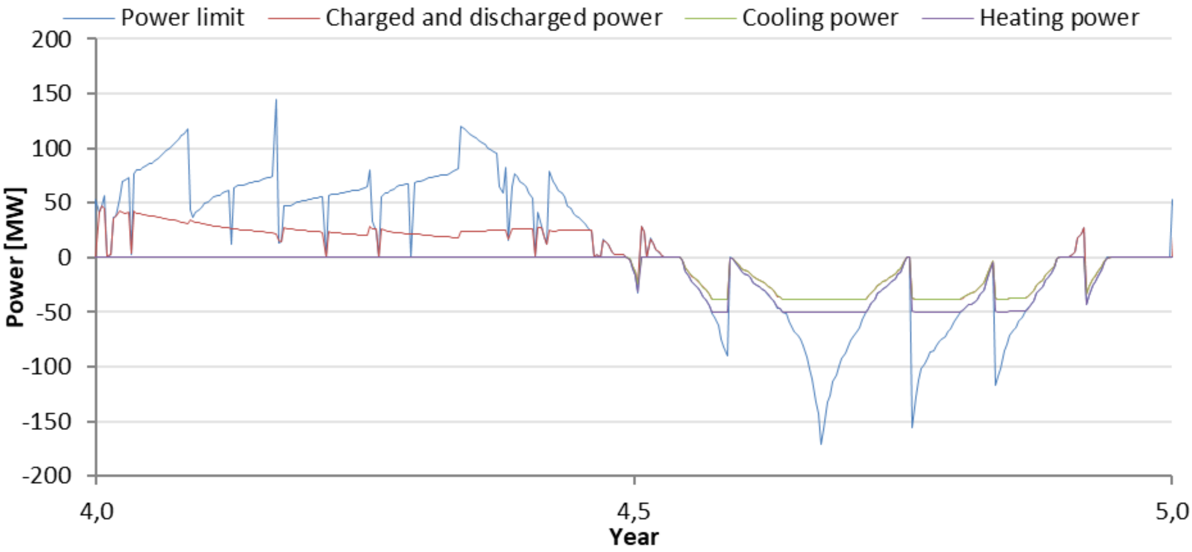


Figure 36 Results for the geometry of 1 500 boreholes with 275 m depth (double U-pipes) regarding charged and discharged power of the BTES and the heating and cooling power of the heat pumps.

If comparing the delivered heating power from the heat pumps between the three cases, Figure 37, it can though be seen that Case B performs slightly better than Case A and Case C in the end of the discharge period.

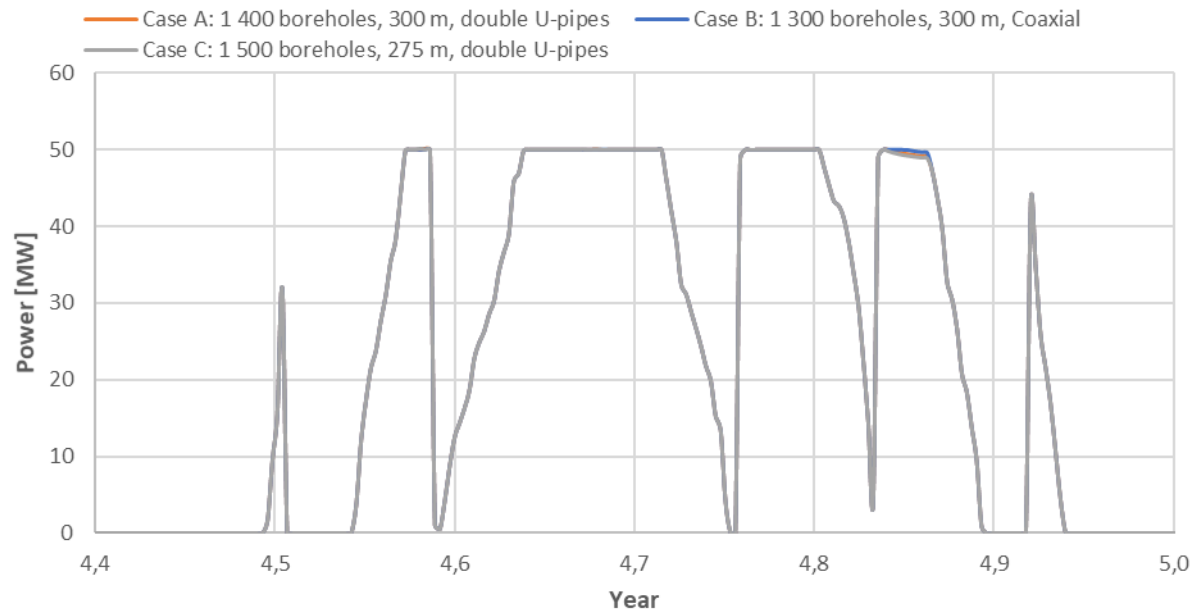


Figure 37 Heating power delivered from the heat pumps to Gärstadverket in the fifth simulation year for Case A, B and C.

In Figure 38, Figure 39 and Figure 40 some system temperatures can be seen for the fifth simulation year for Case A, Case B, and Case C respectively. The diagrams show the inlet and outlet district heating temperature in heat exchangers, KV50 and KV6162, used for charging the BTES during summer operation, as well as the inlet and outlet temperatures of the BTES and the outlet temperature from the condensers. The system temperatures are similar between the three cases.

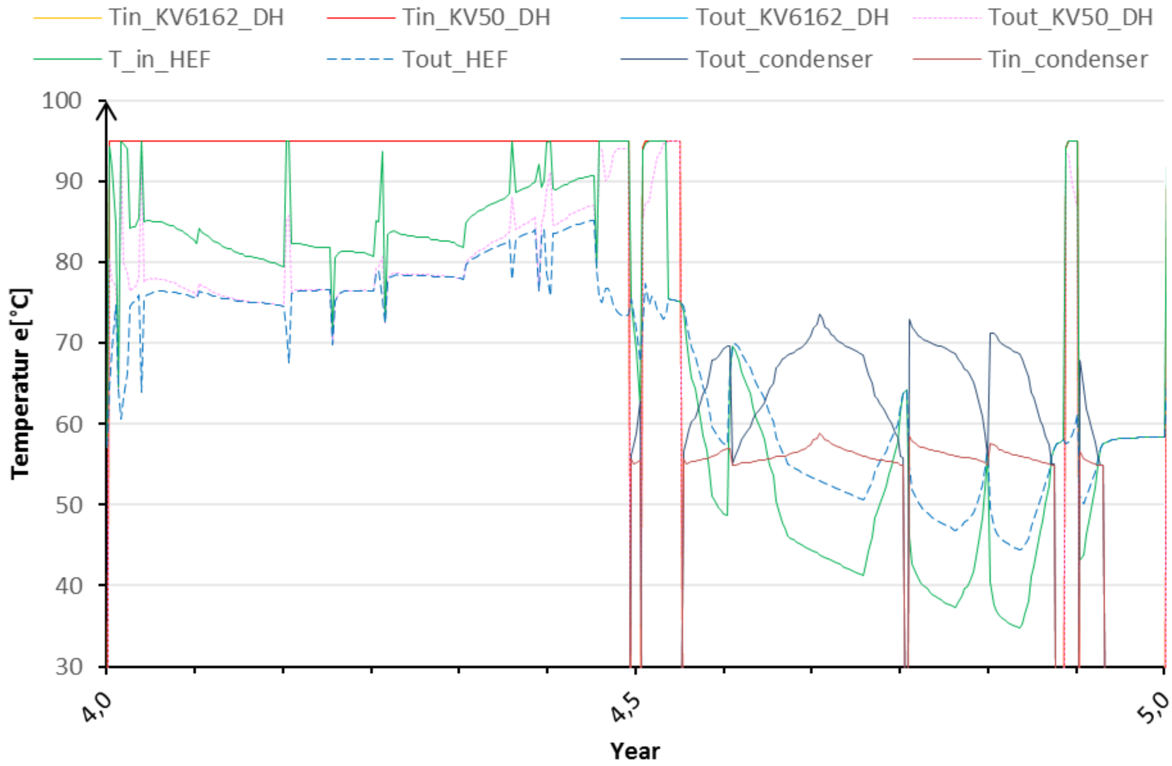


Figure 38 Results for the geometry of 1 400 boreholes with borehole depth 300 m (double U-pipes) regarding the temperatures in and out from the BTES, heat exchangers and heat pumps.

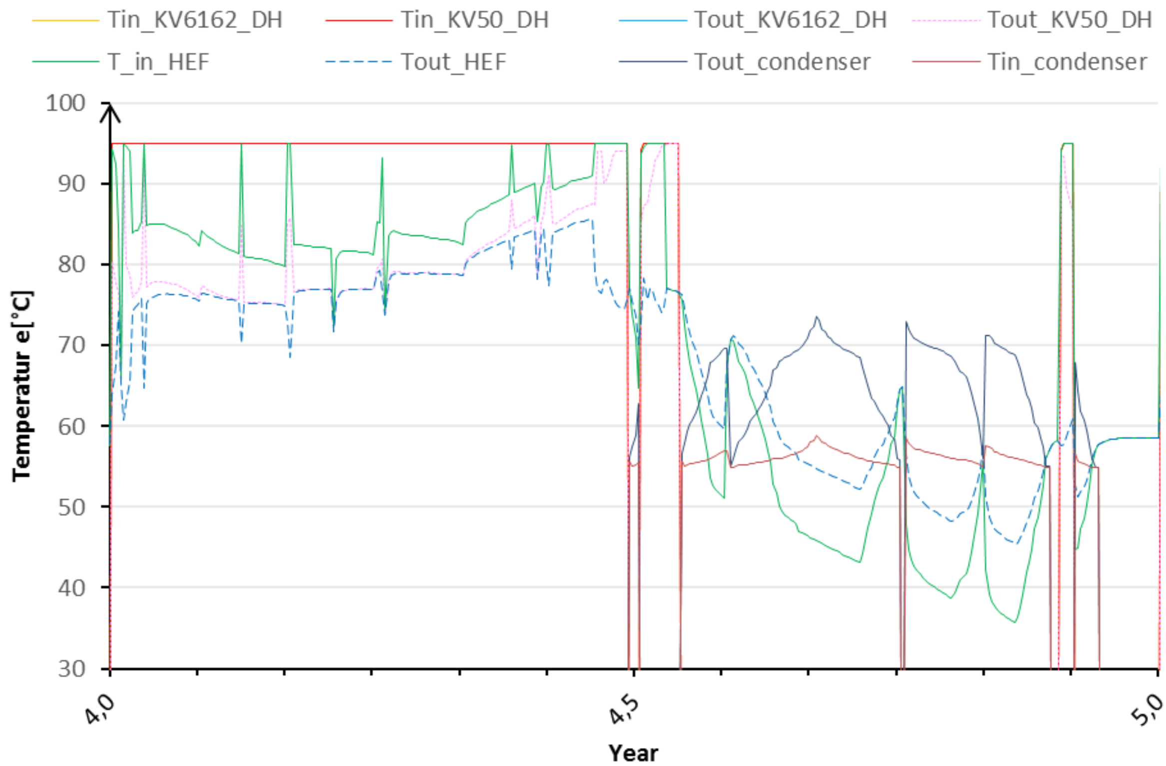


Figure 39 Results for the geometry of 1 300 boreholes with borehole depth 300 m (coaxial) regarding the temperatures in and out from the BTES, heat exchangers and heat pumps.

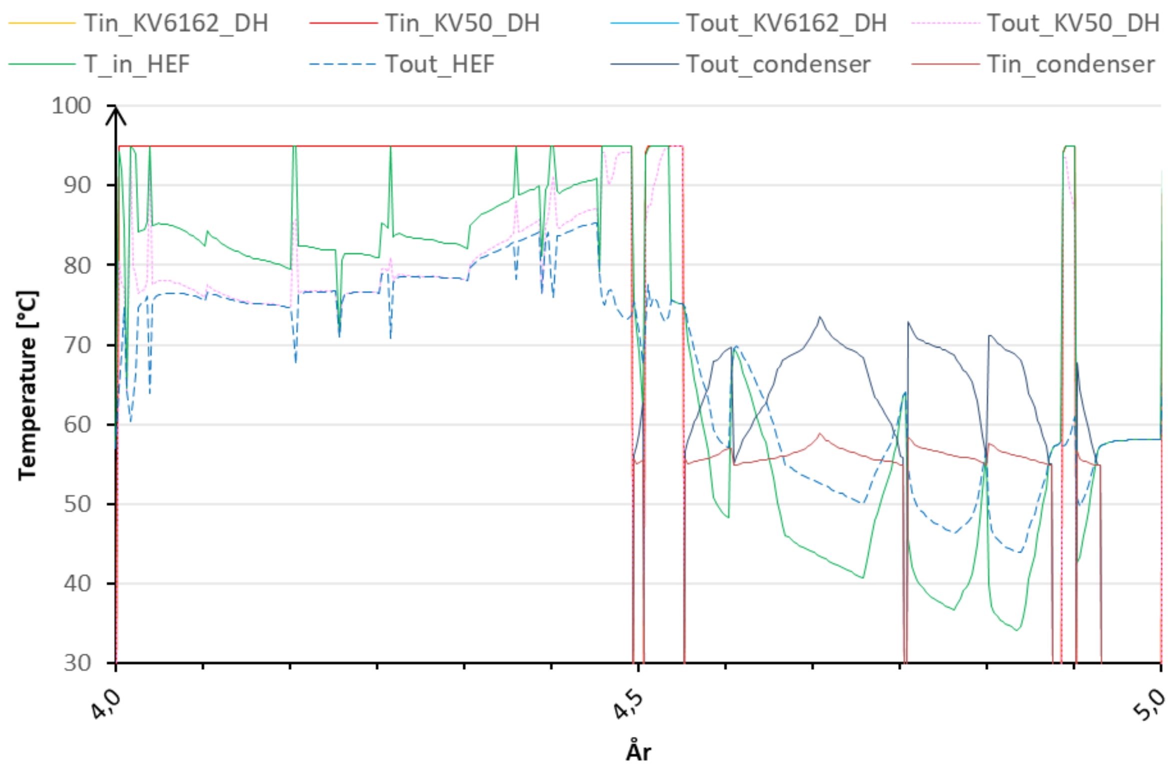


Figure 40 Results for the geometry of 1 500 boreholes with borehole depth 275 m (double U-pipes) regarding the temperatures in and out from the BTES, heat exchangers and heat pumps.

In Figure 41 the average storage temperature can be seen over the 10-year simulation period for the three systems, Case A, B and C. With coaxial collectors, Case B, the storage temperature increases faster and reaches overall a higher maximum. In Figure 42 it can also be seen that the outlet temperature during discharge is marginally higher for Case B compared to Case A and C.

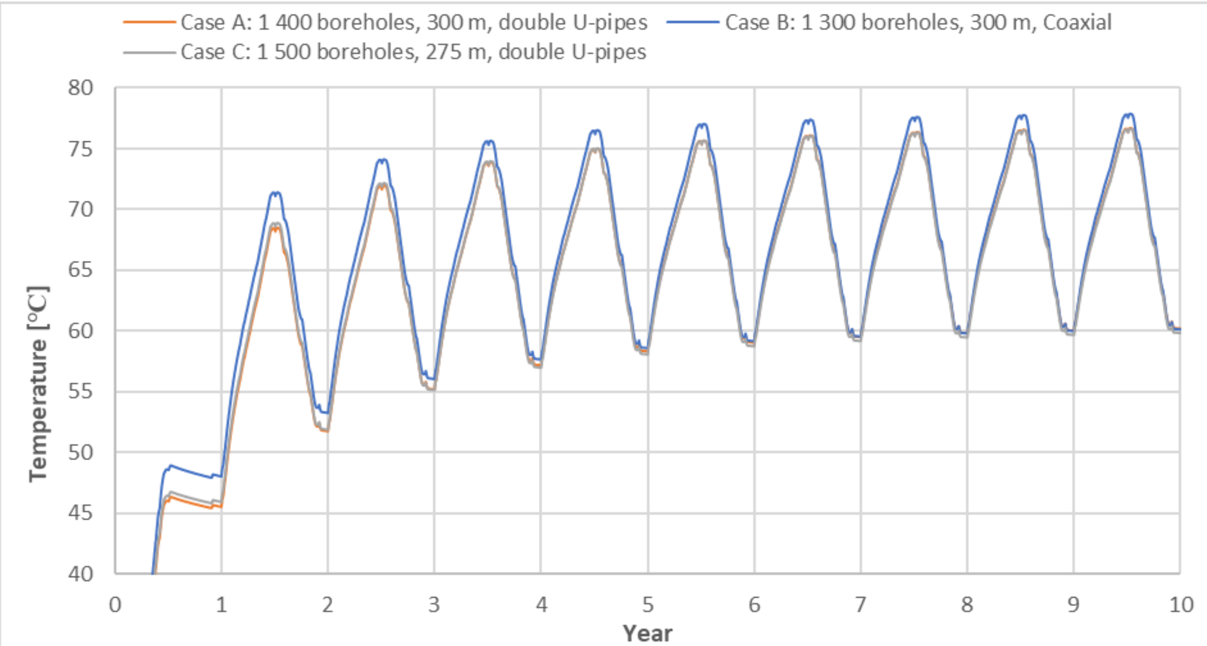


Figure 41 Average storage temperature for Case A, Case B and Case C.

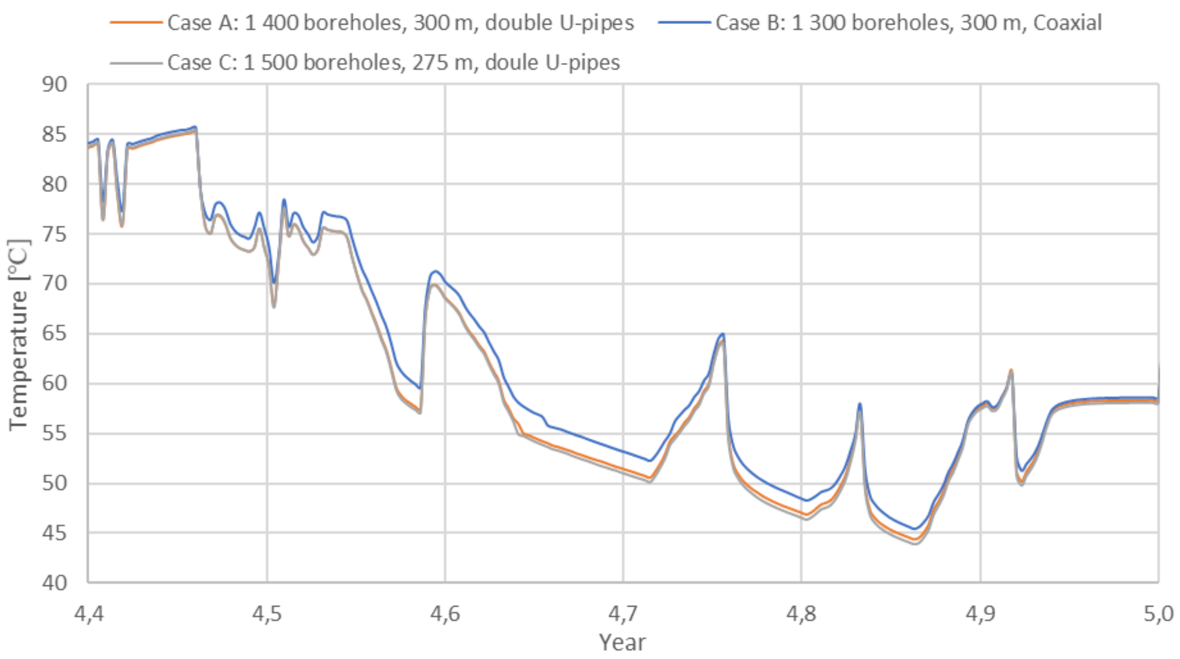


Figure 42 Outlet temperature from the BTES during discharge for Case A, Case B and Case C in the fifth simulation year.

In Figure 43 the energy consumption of the three cases A, B and C is shown, both for the circulation pumps in the BTES and the compressors in the heat pumps. The difference in energy consumption for the compressors is not significant. It can though be seen that the energy consumption for the circulation pumps in the BTES is lower for Case B compared to Case A and Case C, with the highest energy consumption for Case A.

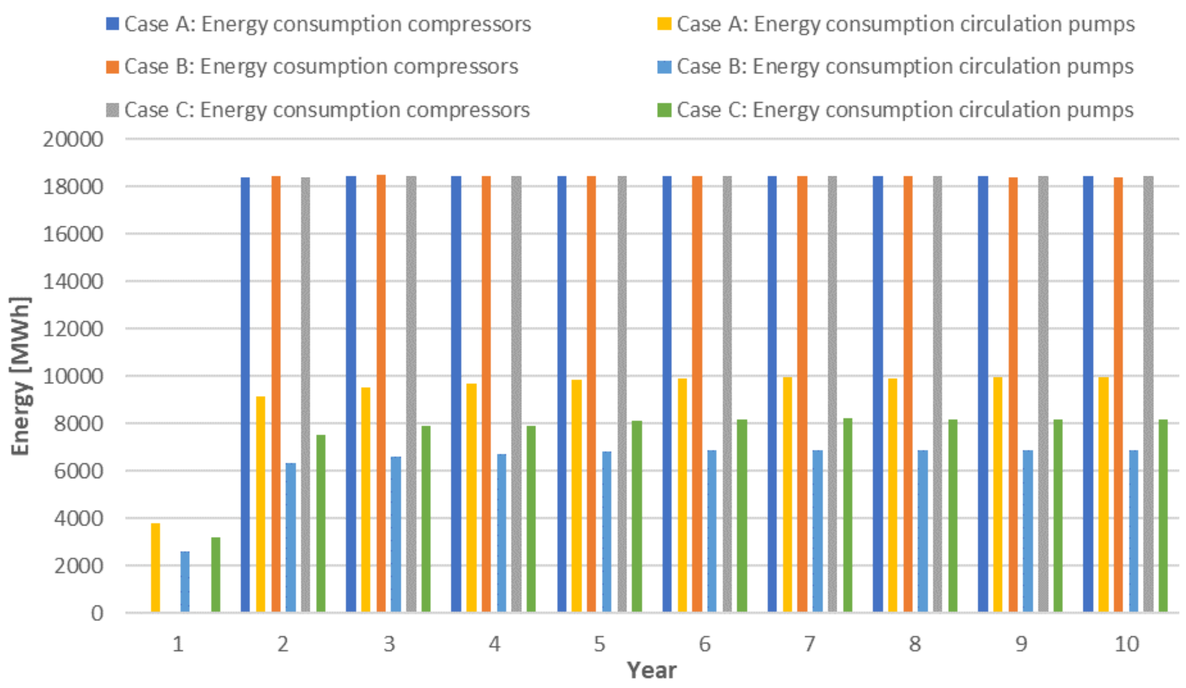


Figure 43 Energy consumption for the circulation pumps and the heat pump compressors for Case A, Case B and Case C.

In Figure 44 the seasonal performance factor for the heat pumps can be seen for the three cases A, B and C, excluding energy consumption for the circulation pumps. It can be seen that the SPF is higher for Case

B, especially in year 2 which is the first year when the BTES is discharged. It can also be seen that the SPF is slightly higher for Case A compared to Case C. Overall, there is though no significant difference.

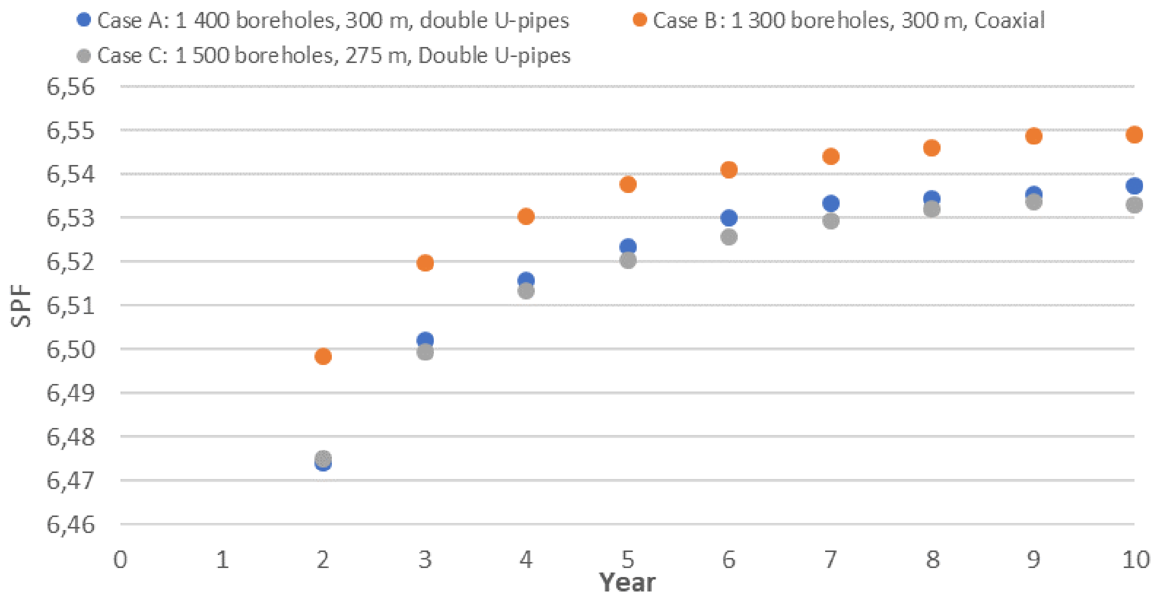


Figure 44 Seasonal performance factor (SPF) for Case A, Case B and Case C.

In Figure 45 the pressure drop in the collectors as a function of time for Case A, B and C can be observed. The borehole depth and the number of boreholes connected in series (three) are the same for Case A and B, and thereby also the total collector length in each loop of boreholes connected in series. However, the size of the BTES in case B is smaller than in Case A, 1 300 boreholes compared to 1 400 boreholes, which results in higher flowrate per borehole loop for Case B and thereby a higher pressure drop. However, the pressure drop is calculated for coaxial collectors in Case B and for double U-pipes in Case A, which results in a lower pressure drop in Case B. Overall, the pressure drop is lower for Case B than for Case A, as can be observed in Figure 45. Both in Case A and Case C the pressure drop is calculated for double U-pipes. Since the total number of borehole loops is lower, and hence the volumetric flowrate is higher per loop, for Case A compared to Case C the total pressure drop is higher for Case A. Furthermore, the borehole depth is lower for Case C, 275 m compared to 300 m. The pressure drop for the geometry in Case C is closer to the one for Case B. The pressure drop can though be considered high in all three cases.

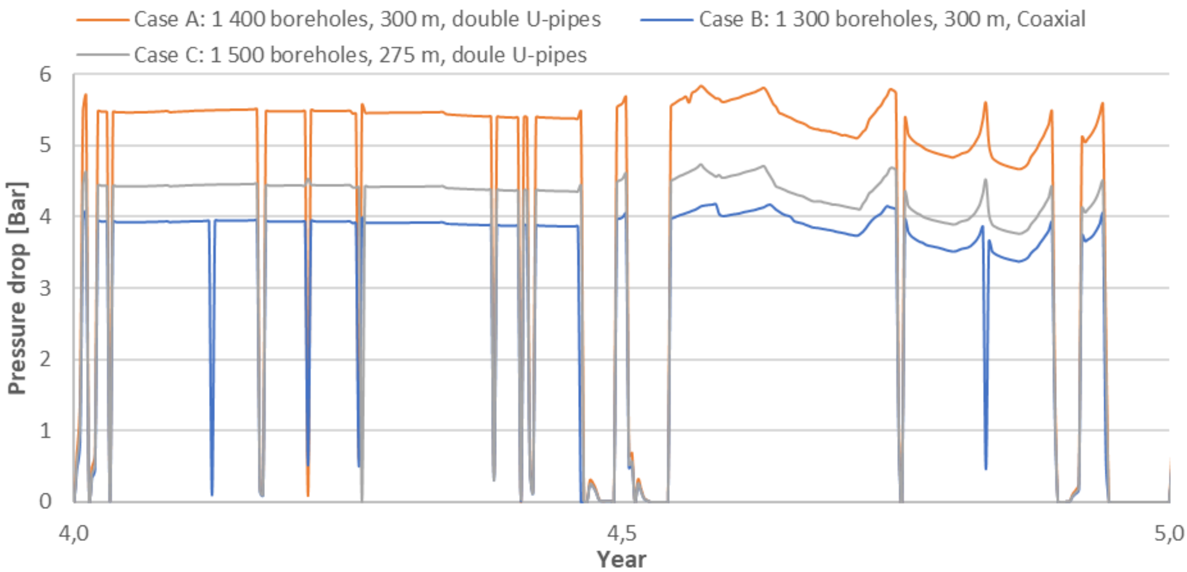


Figure 45 Pressure drop in the collectors for Case A (double U-pipes), Case B (coaxial) and Case C (double U-pipes).

7 Discussion

The model that has been used in this master thesis is based on several assumptions and simplifications of the reality, foremost as the reality is often too complex to model. The results should be considered as a guidance for the design and performance of the system, but it is worth underlying the limitations of the study and discuss how the model can affect the results.

The input data used in the model is based on daily averages. Therefore, no quick changes in the power demand has been simulated and how the system would react to sudden peaks in power demand has thereby not been investigated. It is though known that BTES systems can't handle large load variations. Furthermore, the Type22 control component that has been used in the TRNSYS model has nearly ideal behavior. It would be more likely to control the system with PID controllers which would result in time lags in the reaction of the system to changes in the power demand. One solution to this issue could be to add a hot water storage tank to the system to handle sudden changes in demand. The storage tank could also be used to store energy when discharge power would be too high due to sudden drops in demand.

During winter operation the delivered heating power from the heat pumps is controlled so that it does not exceed the power limit, with 50 MW as maximum peak power. A maximum of 50 MW has been set as limit since it is estimated that the district heating system can't handle higher power levels from the BTES. This constraint is controlled by varying the equivalent number of heat pumps running on full speed. As both the heating power, cooling power and the shaft power, are dependent on the equivalent number of heat pumps also the cooling power and compressor power are limited by the power limit. Hence, the discharge power from the BTES and the delivered heating power from the heat pumps does not vary markedly between the different storage sizes. Only when the storage volume gets too small for the system to be able to deliver the required discharge power during peak heat demand that the difference can be observed. In reality the discharge power could to some extent be increased for the power limit to be reached also for smaller sizes of the BTES. This would though result in a larger decrease of the storage temperature which is not desired. This is avoided in the model as the volumetric flow rate in the BTES is controlled to give a certain temperature drop over the evaporators (correlated to the number of heat pumps running).

When the heat pump model was developed a heat pump manufacturer were contacted. No standard heat pump is available on the market that suits the application for the required power levels. Heat pumps of this size would most probably be designed specifically for the project. Hence, simulations were required for the specific application performed by the heat pump manufacturer. Based on the resulting scenarios and relative outputs, curve fitting was applied to determine the coefficients of polynoms providing relationships between selected input variable (temperatures and mass flows in the brine loop and water loops) and the output variables Q1, Q2 and P. The resulting heat pump model works like a black box and does not provide a detailed description of the physical process within the heat pump. Furthermore, in the model obtained the COP of the heat pumps does not differ markedly between the different cases or the operational years. The resulting scenarios from the heat pump manufacturer though showed a somewhat stronger correlation between the COP and the discharge temperature from the BTES (input temperature to the evaporators) compared to the model. The operational cost for the heat pumps could thereby be expected to vary more between different cases and operational years than what was shown by the results. The model gives though a reasonable estimation of the required operational energy of the system.

Detailed knowledge of the behavior of these heat pumps would enable the construction of models to control both the number of heat pumps and the speed of the compressors. The construction of such complex heat pump model was though not possible within the time frame of this master thesis. Another limitation in the control of the heat pumps is that during some time windows in the beginning of the discharge period each year, the discharged temperature from the BTES is higher than the calculated temperature delivered to the CHP-plant on the warm side of the heat pump. At these periods the heat could be exchanged directly through the heat exchangers, and no heat pump would be required. This could be integrated in the model.

8 Conclusions

In this master thesis the potential for a HT-BTES combined with a CHP-plant (Gärstadverket) in Linköping has been investigated. A TRNSYS model has been used to simulate the system performance over time with different sizes of the BTES. It is for the intended application recommended to use heat pumps in the design of the BTES system to be able to assure the desired power and temperature levels. If vapor compression heat pumps would be used, which is used in the model in this project, it is recommended to use industrial heat pumps working with Ammonia as refrigerant, for example the ones delivered by Star Renewable Refrigeration. Ammonia is a natural refrigerant with zero ODP and GWP, very low explosive risk and is suitable in high temperature systems. The model used in this project is built as one 50 MW heat pump unit connected directly between the BTES and Gärstadverket. For future models it can though be considered to construct a model that consists of three heat pump units connected to the three separate parts of Gärstadverket (050, 061 and 062). In this configuration the water streams from the three different parts of Gärstadverket would not need to be mixed and the systems could be controlled separately.

The BTES was simulated with two different BHE: double U-pipes and coaxial. The geometries that showed most favorable result showed to be with 1 400 boreholes and 1 300 boreholes with double U-pipes and coaxial collectors respectively, both with borehole depth 300 m. When simulating with a lower borehole resistance, corresponding to coaxial collectors, the storage capacity of the system increased due to the improved heat exchange in the BHE and the size of the BTES could be decreased to 1 300 boreholes and 300 m borehole depth. Furthermore, a geometry with 1 500 boreholes and 275 m borehole depth with double U-pipes showed similar results regarding energy, power and system temperatures. The three systems showed a potential to store around 107 GWh/year and to extract around 93 GWh/year with the use of a GSHP. The resulting discharge temperature from the BTES ranges between 40-60°C, and up to 70°C in the initial discharge period in the tenth simulation year.

The pressure drop in the BHE is rather high for the simulated systems, highest for the case with 1 400 boreholes and double U-pipes and lowest for the case with 1 300 boreholes and coaxial collectors. It should be reminded that the simulations in this study has been made with three boreholes coupled in series. If the number of boreholes in series would be decreased the pressure drop would decrease markedly, both as the total collector length is decreased in each loop and that the number of loops would increase resulting in a lower volumetric flowrate in each loop. How this would affect the thermal performance should though be investigated further. Alternative hydraulic solutions could also be investigated, as well as the effect of a decreased volumetric flow rate, which in all simulations is set to vary between 0-1.1 m^3/s. Based on the results it is for the design of the HT-BTES recommended to consider using coaxial collectors, which both results in an improved heat exchange between the heat carrier fluid and the borehole wall and gives lower pressure drop in the pipes, and thereby decreased need for operational energy. It is however unknown if there are coaxial collectors available on the market that can work with the high temperatures in the BTES. This requires further investigation.

The three optimized designs are both cheaper in installation and operational cost than the reference case (1 500 boreholes with borehole depth 300 m and double U-pipes), except the case with 1 400 boreholes and 300 m borehole depth (double U-pipes). This case has lower installation cost, but higher operational cost. This is due to the high pressure drop in the system because of the higher volumetric flowrate per borehole loop compared to the reference case as the number of boreholes is lower, and thereby also the number of borehole loops. It should be emphasized that the cost for the required marginal energy for the total system not have been taken into consideration in this project. It could in reality be more feasible to build a larger BTES to increase the performance of the system and the storage temperature, and thereby decrease the need for marginal energy. A larger BTES also showed to give a higher thermal inertia and is thereby less sensitive to changes in power demand.

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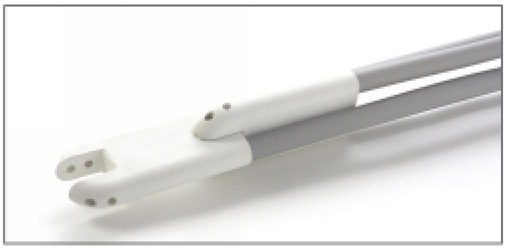
Appendix 1 – Product Sheet PE-Xa Double U-probe

1 GROUND-SOURCE PROBES & ACCESSORIES

RAUGEO PE-Xa Probe
Double-U-Probe
Made of cross-linked polyethylene (RAU-PE-Xa) according to DIN 16892/93, UV-stabilised, natural colour, with a grey protective layer of RAU-PE.
- High resistance against notches, grooves and point loads
- Curved probe tip, therefore no joint in the ground
- Additional protection of probe tip with GRP casing
- Factory tested with certification
- 10 year consequential loss warranty

Operating temperature: -40 °C to +95 °C
Delivery form: 1 Double-U-probe/pallet (4 coils)
Includes screws for connecting the probe feet of double-u probe



RAUGEO PE-Xa Probe 32 x 2,9
Probe tip diameter: 110 mm

Art.-No.		DeliveryLength [m]	d x s [mm]	Weight [kg]	Pipe Volume [l]
131693-050*		50	32 x 2,9	58	108
135873-060		60	32 x 2,9	70	129
135503-070		70	32 x 2,9	80	151
135513-080		80	32 x 2,9	91	173
135523-090		90	32 x 2,9	102	194
135533-100		100	32 x 2,9	114	216
135404-110		110	32 x 2,9	125	237
100741-120		120	32 x 2,9	136	259
135553-125		125	32 x 2,9	141	270
140138-130		130	32 x 2,9	147	280
135685-140		140	32 x 2,9	158	302
131703-150		150	32 x 2,9	169	323

* Produced to order

RAUGEO PE-Xa Probe 40 x 3,7
Probe tip diameter: 134 mm

Art.-No.		DeliveryLength [m]	d x s [mm]	Weight [kg]	Pipe Volume [l]
140143-050*		50	40 x 3,7	88	167
140153-060*		60	40 x 3,7	105	200
140163-070*		70	40 x 3,7	122	234
140173-080*		80	40 x 3,7	139	267
140183-090*		90	40 x 3,7	155	300
140193-100*		100	40 x 3,7	172	334
140203-110*		110	40 x 3,7	189	367
140223-125*		125	40 x 3,7	215	417
140753-130*		130	40 x 3,7	215	434
140233-150*		150	40 x 3,7	257	501
140243-175*		175	40 x 3,7	300	584
140253-200*		200	40 x 3,7	343	668
140263-225*		225	40 x 3,7	386	751
140273-250*		250	40 x 3,7	429	835

* Produced to order

Figure A1 Product sheet of RAUGEO PE-Xa Double U-probe, Source: (REHAU , 2014).