Integrating Ground Source Heat Pumps into Elderly Homes as Energy-efficient Heating and Cooling Solution

A Swedish Case Study

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Master thesis in Energy-efficient and Environmental Buildings
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Lund University

Lund University, with eight faculties and a number of research centres and specialized institutes, is the largest establishment for research and higher education in Scandinavia. The main part of the University is situated in the small city of Lund which has about 112 000 inhabitants. A number of departments for research and education are, however, located in Malmö. Lund University was founded in 1666 and has today a total staff of 6 000 employees and 47 000 students attending 280 degree programmes and 2 300 subject courses offered by 63 departments.

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The degree project is the final part of the master programme leading to a Master of Science (120 credits) in Energy-efficient and Environmental Buildings.

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Abstract

Energy efficiency and environmental questions have gained increased attention over the last decade. In combination with global warming and increasing cooling demands, the Swedish market for combined heating and cooling solutions such as ground source heat pumps expands. As it currently stands, the multi-family building market is highly dominated by district heating, while the market share of combined heating and cooling solutions is insignificant. Several factors are contributing to this market situation such as the widespread of district heating networks, the high installation cost of the borehole heat exchanger, and the site conditions.

This study aims to evaluate the integration of ground source heat pumps into Swedish elderly homes. The study is divided into the following four parts: (1) analysis of energy performance and thermal comfort conditions using a case study, (2) evaluation of the implementation of passive and active cooling measures to improve the indoor thermal comfort, (3) analysis of the ground source heat pumps combined heating and cooling contribution to reduced heating demand and achieving adequate indoor climate, (4) comparison of the ground source heat pump against alternative heating and cooling systems from a life-cycle costing perspective.

The results indicate that the integration of GSHPs in elderly homes significantly decreases the heating energy and highly contribute to the fulfilment of Swedish energy goals. The outlet fluid temperature of the boreholes were insufficient to provide the desired AHU cooling coil inlet temperature and additional room cooling units were required. The utilisation of free cooling was possible and reduced the cooling energy by 26 % and the capacity of the chiller by 29 % in comparison to utilising an external chiller only. Replacing boilers operating on electricity and oil with GSHPs was highly profitable while the replacement of less expensive heating sources such as district heating and boilers operating on pellets was highly dependent on electricity and pellets price. The economical outcome of the analysed cooling systems is dependent on whether the GSHP is used for combined heating and cooling solutions or for cooling only.

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Symbols, Definitions, and Units

Symbol	Definition	Unit
A	Area	[m^2]
A_{env}	Area of building envelope	[m^2]
A_{temp}	Area of heated floor	[m^2]
B_0	Annual savings or annual costs	[SEK]
B_1	Annual savings or annual costs after correction	[SEK]
$c_{p,\ air}$	Specific heat capacity of the air	[J/(kg·K)]
$c_{p,\ ground}$	Specific heat capacity of the ground	[J/(kg·K)]
E_c	Energy use for cooling	[kWh]
E_{DHW}	Energy use for domestic hot water	[kWh]
E_f	Energy use for facility equipment	[kWh]
E_{sh}	Energy use for space heating	[kWh]
g	Annual price increase	[%]
h	Heating degree hours	[Kh]
h_{pi}	Convection heat transfer coefficient of the U-pipe	[W/(m^2·K)]
i	Annual interest rate	[%]
k	Annual inflation	[%]
n	Number of years	[Year]
p	Pressure	[Pa]
P_c	Compressor electrical power	[W]
$P_{cooling}$	Cooling load	[W]
P_e	Evaporator thermal power	[W]
$P_{heating}$	Heating load	[W]
P_{hp}	Heat pump capacity	[W]
P_{max}	Maximal heating demand	[W]
P_{inf}	Infiltration heat loss rate	[W/K]
P_{tb}	Thermal bridge heat loss rate	[W/K]
P_{tran}	Transmission heat loss rate	[W/K]
q_{vent}	Supplied ventilation air flow	[m^3/s]
$q_{leakage}$	Infiltration leakage air flow at 50 Pa pressure difference	[m^3/s]
$q_{natural}$	Natural leakage air flow at atmospheric pressure	[(l/s)/m^2]
q_{vent}	Extract air flow of the ventilation system	[m^3/s]
U_{adj}	Adjusted mean thermal transmittance	[W/(m^2·K)]
U_{mean}	Mean thermal transmittance	[W/(m^2·K)]
η	Heat recovery efficiency	[%]
ρ_{air}	Air density	[kg/m^3]

Terminology and Abbreviations

EPW file: EnergyPlus Weather-file.

Overheating hours: Number of hours exceeding a predefined threshold for operative indoor temperature.

Specific energy use: Required energy for space heating, space cooling, production of domestic hot water, and facility electricity in (kWh/m^2)/year.

Boverkets Byggregler: The Swedish National Bord of Housing, Building and Planning building regulations.

Obligatorisk ventilationskontroll: Obligatory ventilation inspection that involves the inspection of the ventilation system and the functionality.

AHU	Air Handling Unit
BBR	Boverkets Byggregler
BHE	Borehole Heat Exchanger
CAV	Constant Air Volume
COP	Coefficient of Performance
DH	District Heating
EA	Extract Air
DHW	Domestic Hot Water
EER	Energy Efficiency Ratio
GHR	Global Horizontal Radiation
GSHP	Ground Source Heat Pump
LCC	Life-Cycle Costing
MFB	Multi-Family Building
NPV	Net Present Value
OH	Overheating Hours
OVK	Obligatorisk Ventilationskontroll
RH	Relative Humidity
SA	Supply Air
SEU	Specific Energy Use
SFB	Single-Family Building
SH	Space Heating
SHG	Sydsvenska Hälsogruppen
SHGC	Solar Heat Gain Coefficient
TRT	Thermal Response Test

1 Introduction

Energy efficiency and environmental questions have over the last decades gained increased attention, as climate change and global warming are becoming more apparent. Sweden´s determined policy regarding renewable energy and sustainability is continuously leading to new environmental goals targeting to decrease the country's environmental impact. Recently, the Swedish Energy Agency published a new goal targeting 100% renewable electricity by 2040, while their goal of increasing energy efficiency from 2005 to 2030 by 50% is still in progress (Swedish Energy Agency, 2020). These targets are ambitious and require increasing renewable energy use and more efficient utilisation within many sectors, especially in high energy use sectors.

The building sector is one of the sectors with the highest energy use as it approximately stands for one-third of Sweden's entire energy consumption (Swedish Energy Agency, 2015). Hence, offering substantial potential for improvements. Most of the building sector's energy (74%), is utilised for building operating purposes such as space heating, Domestic Hot Water (DHW), space cooling, ventilation, and lighting. Due to Sweden's heating-dominated climate, space heating and DHW use in households account for more than half of the energy use in the building sector (Swedish Energy Agency, 2015). Currently, cooling only represents a small share of the energy use, but is expected to increase as a result of global warming and strict thermal comfort requirements (Selvaraj, 2015).

With the increasing awareness of energy efficiency and sustainability in combination with increasing fossil fuel prices, heating systems have gradually shifted away from fossil fuels such as oil and natural gas. Instead, more energy-efficient heating systems such as heat pumps, boilers operating on biofuels and district heating distribution networks have taken a substantial share of the heating market over the past decades (Swedish Energy Agency, 2015). While state-of-the-art district heating systems significantly increase energy efficiency in comparison to pure fossil fuels, the availability of these plants is often limited to rural areas. District heating plants are generally built within a reasonable distance to large residential areas to minimize heat losses within the distribution system. As heating demands in rural villages are not large enough to justify the establishment of district heating plants, alternative energy-efficient heating systems are required.

The previously mentioned heat pump could serve as an energy-efficient alternative to district heating plants. One of the most efficient heat pump system is the Ground Source Heat Pump (GSHP). This system is considered to be renewable as thermal energy is extracted from a renewable ground-source. GSHPs are highly efficient at supplying both heating and cooling thus, they can contribute to the fulfilment of Sweden's sustainable goals . Sweden's history with geothermal energy sources stretches back over half a century (Andersson et al., 2020). During the oil crisis in the 1970s, many investigations were carried out to design an energy system independent of fossil fuels and oil . During the 1990s, geothermal energy technology accomplished significant development in Sweden and placed it at the front edge of research in geothermal energy systems (Andersson et al., 2020).

A frequently used way of extracting thermal energy stored in the ground is by a Borehole Heat Exchanger (BHE). This heat exchanger significantly increases the investment cost of the system in comparison to conventional air-source heat pumps. Because GSHPs are utilising a non-seasonal dependent source, they can provide higher capacities and are therefore more suitable for large-scale buildings as multi-family and industrial buildings. Since GSHPs only require electricity to operate, Sweden's low electricity prices and subsidies for renewable energies played an important role in increasing the market share of these systems. Currently, there are no subsidies on GSHPs but the labor costs deduction of 75 000 SEK (*Rot- Och Rutarbete - Privatpersoner*, 2021.).

At the moment, cooling systems in Sweden are primarily installed in buildings with strict thermal comfort requirements such as hospitals and offices and rarely in residential buildings. Nevertheless, the Earth's increasing average global temperature has contributed to increasing cooling demands in residential buildings in the USA, and Sweden could arguably follow this trend (Wang & Chen, 2014). This could potentially widen the market for cooling systems and for combined heating and cooling solutions. Cooling provided by GSHPs is often referred to as "free cooling" as it only requires electricity to power the pump circulating the heat transfer fluid in the BHE. Because GSHPs are highly energy-efficient at providing both heating and cooling, they could be utilised to achieve current and future energy goals, while also providing relatively adequate indoor climate during summer. Still, the market share of GSHPs within MFBs is relatively low. Currently, the Swedish market share for GSHPs in Single-Family Buildings (SFB) is around 20 %, while the market share for Multi-Family Buildings (MFB) is significantly lower (Sommerfeldt & Madani, 2018). The market share of GSHPs in SFBs is highly influenced by alternative heating systems such as air-source heat pumps with lower initial costs and district heating. The minimal share of GSHPs within MFBs could be connected to several factors; (1) most MFBs are located in urban areas and are already connected to district heating networks, (2) individual analysis of the free cooling capacities, (3) limited area for installing BHE, (4) high initial costs.

Previous studies from Sommerfeldt & Madani, (2018) and Alqaed et al., (2020) have shown that it is possible to implement GSHPs in MFBs but such implementation is climate, property, and ground condition dependent. Therefore, MFBs usually require individual analysis to dimension the GSHP system based on the local conditions. This thesis is analysing the implementation of a ground-source heat pumps in an elderly home in Southern Sweden, where the findings can be applied on buildings with similar characteristics, and ground conditions.

1.1 Goals and objectives

This thesis investigates the integration of GSHPs into an elderly home which is considered as a multi-family building in Sweden. A case study on an elderly home with strict thermal comfort requirements is performed. The study targets energy-efficiency, free cooling utilisation, and life-cycle costing of GSHPs in comparison to alternative heating and cooling systems such as boilers operating on different heating sources, district heating and air-source chiller. The goal of this thesis is to provide further knowledge about the implementation of GSHPs in buildings with strict thermal comfort requirements. The introduced type of MFBs are often found in rural areas in Sweden. Therefore, the findings of this thesis could motivate designers and decision makers to choose GSHPs over other heating and cooling systems on building with similar site conditions. To fulfil this goal, the thesis aims at addressing the following questions:

- How can the implementation of a ground source heat pump contribute to the fulfilment of the Swedish Energy Goals?
- How effectively can free cooling contribute to the enhancement of thermal comfort during summer?
- How does ground source heat pump's combined heating and cooling solution compared to other heating and cooling systems in terms of life-cycle costing?

1.2 Scope

The energy performance and thermal comfort condition of the elderly home was analysed by performing annual energy simulations. To improve thermal comfort conditions during summer, both passive and active measures were introduced by implementing external blinds, cooling the supply air and installing supplementary space cooling units. The building's recently installed heat pump and borehole heat exchanger exclusively operating in heating mode is analysed to determine the free cooling capacities. Further, the possibility of installing an additional heat pump utilising the existing borehole heat exchanger configuration is analysed. The economic aspects of the existing and additional simulated ground source heat pump's combined heating and cooling solutions are compared to alternative heating and cooling solutions.

1.2.1 A brief description of the case study

The analysed MFB is an elderly home owned by Sydsvenska Hälsogruppen (SHG). SHG is a family business focusing on residential care for elderly people. Residential care is provided in three different elderly homes spread over southern Sweden. This thesis focuses on the elderly home Nömmeberg, in Stensjön ($57°57'N, 14°82'E$). The elderly home consists of several building sections constructed between 1980 and 1990 as shown in Figure 1.

Figure 1: Building sections of Nömmeberg elderly home.

1.3 Assumptions and limitations

This study is limited to the integration of GSHPs in elderly homes and the evaluation of energy efficiency, thermal comfort, and Life-Cycle Costing (LCC) using energy simulations and spreadsheet calculations for the LCC. Performing energy simulations and life-cycle costings require numerous inputs. Several assumptions and limitations were made, the assumptions and limitations with the arguably highest impact on the results are described in the chapters below.

1.3.1 Case study

As the elderly home consists of many different sections, the scope of this study was limited to building sections D, E and F. These building sections are including various kind of rooms usually represented in elderly homes. The access to these sections was limited due to the ongoing pandemic situation and the retrieval of measured data was difficult. Therefore, the average Specific Energy Use (SEU) according to the energy declaration performed in 2019 was applied to sections D, E and F.

1.3.2 Building envelope

Construction drawings of the studied sections were not available and several assumptions regarding the building's envelope were made. Because the sections were built during different time periods with different regulations, their construction parts are unlikely to share the same materials and insulation thickness. Thus, making assumptions of the thermal transmittance for each different construction part would be inaccurate. Instead, the mean thermal transmittance (U_{mean}), of these sections is calculated based on the section's calculated transmission losses and building envelope area. Thermal bridges are not included.

1.3.3 Life-cycle costing

To assess the LCC of the GSHP implementation, the Net Present Value (NPV) for GSHPs providing heating and space cooling was calculated considering initial costs, operating costs and maintenance costs. Further, the NPV of alternative heating and cooling systems operating on different sources were calculated and compared to GSHP solutions. The material and operating costs of the GSHPs were acquired from the Swedish GSHP market leader. The installation costs of the included systems were assumed to be equal to 20 % of the corresponding material costs based on experience. The costs for the BHE were based on actual billing costs retrieved from the company Vatten & Borrteknik i Småland AB. The prices for different energy sources were partly obtained from statistical data and partly assumed when price fluctuations highly dependent on the location. The prices for the space cooling units were not considered due to the large discrepancy in costs from different manufacturers.

1.4 Overall approach

This study applies quantitative research methods to fulfil the goals of this thesis. The study was initiated with a comprehensive literature review of research papers, articles and other related sources about thermal comfort recommendations for elderly people and heat pump technology. Afterwards, a 3D-model was established of the analysed building sections using Autodesk Revit. The geometries and spaces of the 3D-model were validated using the software SimpleBim (*SimpleBim*, 2021) and the IDA-ICE add-on (*IDA-ICE Add-on for Simplebim 8.0 | Simplebim*, 2021). The energy simulations were performed using the software IDA-ICE with the borehole extensions (*Boreholes - Simulation Software | EQUA*, 2021). The life-cycle costings were carried out with spreadsheet calculations and the heat pump dimensioning program NIBEdim.

2 Literature review

The aim of this chapter is to introduce the reader to essential topics that will be mentioned throughout the thesis. The literature review provides the reader with introduction of the importance of thermal comfort for elderly people, heat pump technology, and the functionality and advantages of GSHPs.

2.1 Importance of thermal comfort for elderly people

Since elderly people generally have a reduced metabolic rate, reduced muscle mass, and body weight compared to the average person, higher effort is required to perform activities and maintaining body temperatures in relation to the average person. The national board of health and welfare has shown that there is a corelation between low indoor temperatures and cardiovascular and lung-related diseases. Also, a relationship between cold indoor temperatures and several physiological variables such as blood pressure and blood clots was observed. Therefore, elevated motility rates were observed during colder winter months. After extreme cold days, motility rates significantly increased for 50 % to 70 % for people aged 65 years and above, the cause of death being cardiovascular and lung related diseases. (Socialstyrelsen, 2005). Moreover, the national board of health and welfare specified that it is important to choose appropriate indoor temperature depending on the indoor activity. If the indoor climate is not adjusted according to the indoor activity, temperatures will not be within an adequate range and will negatively affect mental ability, mobility and strength (Socialstyrelsen, 2005). According to the Swedish public health authority guidelines for indoor climate, operative temperatures should not drop below 18 °C during winter, and should not exceed 26 °C during summer for longer periods (Folkhälsomyndigheten, 2019). These guidelines are outlined for the average person and should not be applied to people with special needs such as elderly. On account of elderly people having difficulties maintaining their body temperature, stricter indoor climate is required. During non-extreme weather conditions where exceptional low and high outdoor temperatures are not occurring, indoor temperatures of 22 °C should be maintained while 20 °C is the absolute minimum during winter (Socialstyrelsen, 2005). To fulfil these requirements, a stable source of thermal energy is required with a cooling system that operates during warm summers.

2.2 Heat pump technology

This section presents background of different types of heat pumps and their characteristics. Goldschmidt, (1984) describes a heat pump as “a device that extracts thermal energy from a low-temperature source (such as the outside air or ground) and transfers it to a higher-temperature sink (such as the heated indoors of a building)”. Heat pumps are versatile devices and serve for many different purposes such as space heating, space cooling and DHW (Rees, 2016). A heat pump produces thermal energy without a combustion process and absorbs energy from heat sources. The only required input electricity is needed for the mechanical work of the compressor, to transfer energy from the source to the sink. Therefore, it is considered to be a highly efficient and sustainable if the used electricity is obtained from renewable sources (NIBE, 2018). Around 96% of all heat pumps installed during 2013 were installed in SFBs (Åberg et al., 2020).

There are many heat pump variations and energy distribution systems. According to Sarbu & Sebarchievici, (2016), heat pumps can be differentiated by their function, source of thermal energy, and refrigeration cycle as shown in Table 1.

Table 1: Examples of different heat pump functionalities, energy sources, and refrigeration cycles.

Function	Energy source	Refrigeration cycle
Space heating	Ground	Air-to-water
Cooling	Water	Air-to-air
Domestic hot water	Outdoor air	Water-to-water
Ventilation	Exhaust air	Brine-to-water

Examples are provided with no particular order.

To provide heating and cooling, heat pumps can extract thermal energy from various sources with available energy and storage capacities (Goldschmidt, 1984). Energy sources listed in Table 1 are able to provide space heating and all but exhaust air can provide cooling. The main difference between these sources is their availability, heat storage capacities and temperature fluctuations throughout the year.

Most of the installed heat pumps are utilising outdoor air as the energy source and are providing heating through a convective or hydronic radiator system. The heating demand of the average single-family building in Sweden can mostly be supplied by air-source heat pumps. While air-source heat pumps are dominating the single-family market, their market share within multi-family buildings is significantly lower. The heat capacity of air is usually not sufficient to cover the heating demand in a feasible way. Larger systems would be needed, requiring high air flows, and potentially causing noise problems. Also, heating dominated climates are often combined with low outdoor air temperatures while heating demands are high. This reduces the heat extraction efficiency as the temperature difference of the energy source and heating sink increases. To supply building with larger heating demands such as MFBs, larger heat pump capacities in combination with higher capacity sources are required. These higher capacity sources are ground sources utilising the seasonal stored energy, providing stable source temperatures in comparison to seasonal dependent outdoor air (Perko et al., 2011). Ground temperatures are dependent on the location and bedrock depth and persistent ground temperatures of 4 °C to 12 °C can be reached in Sweden (NIBE, 2014). Further advantage of ground sources is the possibility to store thermal energy over seasons, as solar radiation and heat injected to the ground during cooling operations can be utilised during winter for heating operations. In comparison to the outdoor air, source temperatures are generally higher during the heating season and lower during the cooling season increasing the efficiency for both operation purposes (Sarbu & Sebarchievici, 2016). Additionally, obtaining energy from less fluctuating sources reduces the auxiliary energy demand during peak heating or cooling conditions. The advantages of ground-sources come with additional costs for drilling and the installation of the Borehole Heat Exchanger (BHE) which is significantly increasing the initial costs in comparison to air-source heat pumps (Sarbu & Sebarchievici, 2016).

The efficiency of a heat pump depends on the temperature difference between the source of energy and the desired heat sink temperature. Therefore, the efficiency of a heat pump increases if the temperature difference between the evaporator and condenser decreases (Warfvinge & Dahblom, 2010). Energy sources with high latent heat are preferable for heating purposes whereas low latent heat sources are preferable for cooling purposes. The efficiency of heat pumps under specific conditions is demonstrated by the Coefficient of Performance (COP). The COP specifies the ratio of obtained power to the input power required to operate the system. The COP enables effortless efficiency comparisons between other heat pumps under different conditions, or other heating systems (Warfvinge & Dahblom, 2010). Because the COP is season dependent, there are two ways of determining it: (1) by calculating it for specific outdoor conditions and heating setpoints, (2) calculating the average annual efficiency, considering seasonal variations through the year (Energimyndigheten, 2010). The annual average efficiency is also known as the Seasonal Coefficient of Performance (SCOP) or as the Seasonal Performance Factor (SPF) (Sarbu & Sebarchievici, 2016). According to Warfvinge & Dahblom, (2016) and Goldschmidt, (1984), the COP for heating operations is calculated according to equation (1), which is dependent on the heat supplied by the system ($P_{heating}$) which is equivalent to the thermal energy supplied by the evaporator (P_e) and the power required to operate the compressor (P_c).

$$COP = \frac{P_{heating}}{P_c} = \frac{P_e + P_c}{P_c} \tag{1}$$

Since the power supplied to the system also serves for heating, the COP will always be larger than one. Typical values for the SCOP for heat pumps ranges between three to five and therefore, three to five units of energy are gained for each energy unit supplied to the system.

The efficiency for cooling operations is specified as the Energy Efficiency Ratio (EER) and is calculated according to equation (2), which is dependent on the cooling supplied by the system ($Q_{cooling}$) and the power required to run the system (P_c) (Sarbu & Sebarchievici, 2016).

$$EER = \frac{P_{cooling}}{P_c} \tag{2}$$

The COP is dependent on the source temperature and the desired heating temperature. Because the average ground temperature is generally higher than the average outdoor air temperature during the heating season in heating dominated climates, higher COPs can be obtained in GSHPs. The correlation of ground temperatures and desired heat sink temperatures for a GSHP is visualised in Figure 2.

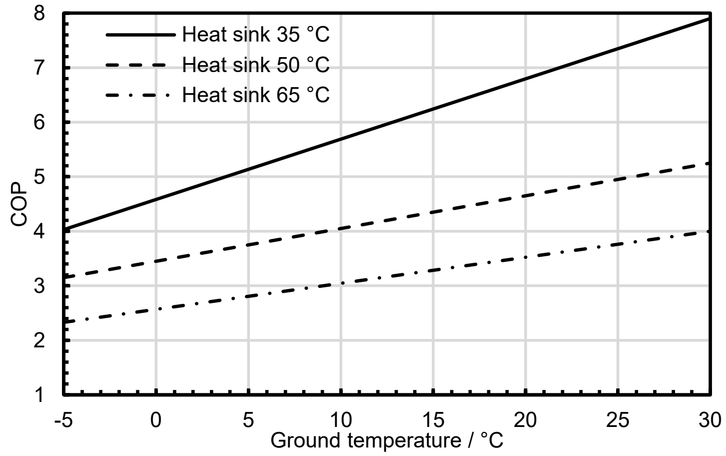

Figure 2: Coefficient of performance for different ground and heat sink temperatures, adapted from Hirvonen et al. (2018).

To extract energy from a given source, a refrigeration cycle is required. This cycle transfers energy from a low-temperature source to a high-temperature sink. Each heat pump features some kind of refrigeration cycle. This cycle can consist of different heat carriers and energy sources. Illustrated in Figure 3, heat is extracted from a ground-source via a water-to-water system. To perform a refrigeration cycle, the following components are required within a closed circuit: a compressor, condenser, evaporator, expansion valve, and refrigerant (Warfvinge & Dahblom, 2010). The refrigeration cycle of a water-to-water heat pump can be explained in the following five steps: (1) A fluid refrigerant absorbs thermal energy from a source and enters the evaporator. The refrigerant functions as the thermal energy transporting fluid within the heat pump, and has special properties allowing it to evaporate at low pressure and temperatures and condense at high pressure and temperatures. (2) The refrigerant gains enthalpy from the source and changes its aggregate state from liquid to gas. (3) The compressor pressurizes the gas to further increase its enthalpy and temperature. (4) Hot refrigerant gas then enters the condenser and transfers heat to a high-temperature sink such as a space heating system, by condensing and reducing its enthalpy. (5) Lastly, the refrigerant passes an expansion valve and returns to its initial stage (Warfvinge & Dahblom, 2010). Through this refrigeration cycle, heat can be supplied for space heating and domestic hot water. When reversing this cycle, heat is instead extracted from the inside and rejected to the ground, enabling the heat pump to be used for cooling.

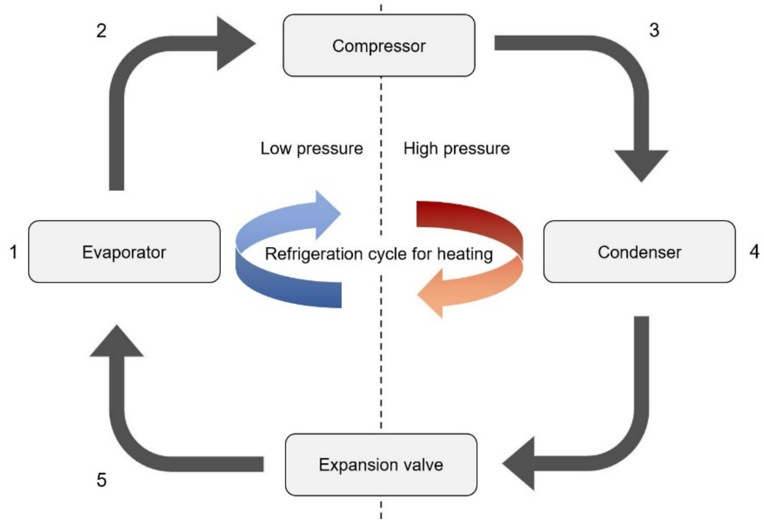

Figure 3: Refrigeration cycle for heat pumps operating in heating mode.

2.3 Ground source heat pumps

2.3.1 Background

Heat pumps extracting energy from ground-sources are referred to as GSHPs. To transfer thermal energy from a ground-source to a heat pump, some sort of heat exchanger is required. This heat exchanger could be buried below the ground surface, on a seabed, or put in vertical boreholes. The heat is transferred by circulating an anti-freeze solution of water and ethanol (Gehlin et al., 2016). Horizontal heat exchangers buried below the ground surface are more economical than vertical BHE but require more land area. Therefore, these are rarely used in urban areas, where land areas are limited. Vertical BHE installations require reduced land area and are therefore suitable for a wider range of buildings. A simplified model of a GSHP with a vertical BHE operating in heating and cooling model is presented in Figure 4. To supply the heating demands of MFBs in heating-dominated climates, several BHEs are connected in parallel. In Sweden, conventional borehole depths range between 200 m and 300 m, but according to Gehlin et al., (2016), borehole depths have increased over the years. The reasons for that are increasing pump efficiencies which are reducing the operation costs of circulation pumps, and the trends of higher capacity systems, with higher Sizing Factors (SF). Because boreholes directly interfere with the ground, steel casings are required within the first six meters of boreholes to protect the soil and groundwater from pollutions (Swedish Geological Survey, 2016).

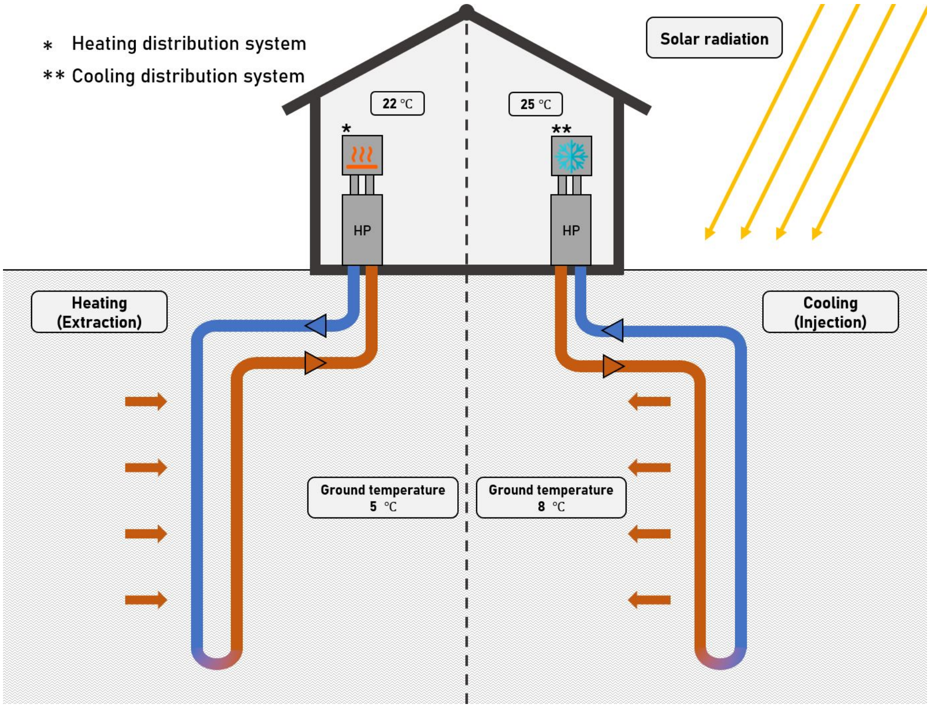

Figure 4: Illustration of a ground source heat pump operating in heating and cooling mode.

Currently, GSHPs are not well represented within MFBs but there are many factors pointing towards higher market shares in the near future. One of these factors is the possibility to provide space cooling, prompting better indoor climates during summer. As seen in Figure 5, GSHPs are able to provide free cooling by bypassing the heat pump and circulating the heat transferring fluid through the BHE. By doing this, the fluid cools down and injects thermal energy to the ground. The cold circulating fluid is then transferred to the buildings cooling system by a heat exchanger. As the ground heats up with time, the free cooling capacities might not be enough for the entire cooling season, requiring active cooling solutions. Reversable GSHPs are capable of reversing the refrigeration cycle and are able to provide active cooling at high EERs (NIBE, 2014). The combination of free and active cooling could arguably be very useful in buildings with high thermal comfort standards such as hospitals or strict offices. But also, residential buildings could implement the combined heating and cooling solution to increase thermal comfort.

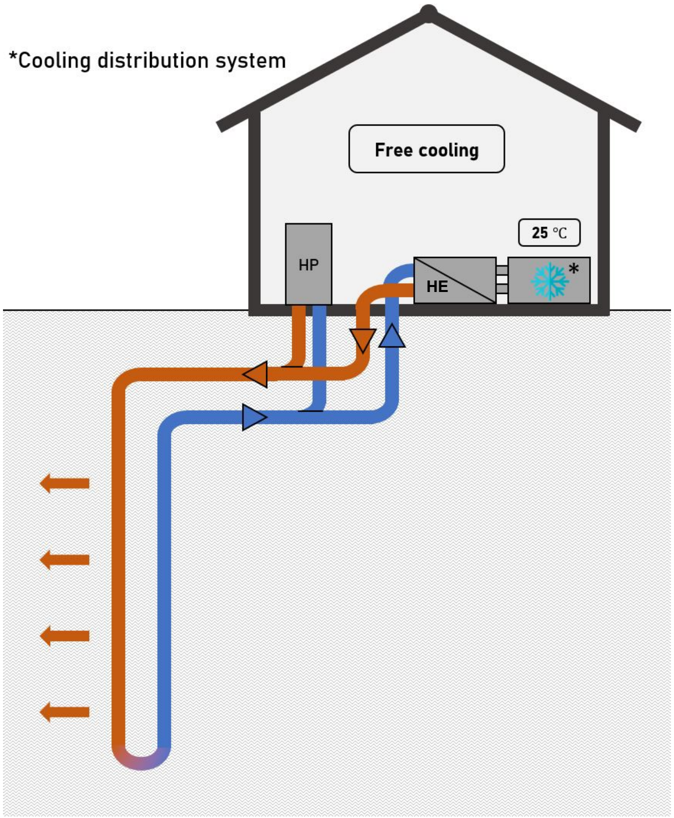

Figure 5: Ground source heat pump with free cooling utilisation.

Costs associated with the BHE installation are accountable for large shares of the total GSHP installation costs. Therefore, these heat exchangers should be dimensioned carefully. There are several factors affecting the BHE costs. For example, there is a correlation between increasing borehole depth and costs as drilling deeper causes higher drill wearing and increases the risk for complications (Gehlin et al., 2016). Additionally, deeper boreholes are causing higher pressure drops, requiring larger circulation pumps and additional operating costs. Because the first BHE meters require casing to protect the groundwater from pollutions, increasing the number of boreholes instead of drilling deeper could also lead to increased costs. All these factors are highlighting the importance of appropriate dimensioned BHE configurations and the balance between the initial costs and future savings generated.

2.3.2 System sizing

There are several steps involved in dimensioning a GSHP. The first step is to estimate the building's total and peak heating demand. This is achieved by either hand calculations or preferably through energy simulations that consider detailed evaluation. Once the heating demands are obtained, the next step is to determine the GSHP's capacity. This is accomplished by choosing a Sizing Factor (SF) which is defined as the ratio between the GSHPs maximum capacity (P_{hp}) to the buildings peak heating demand (P_{max}) according to equation (3).

$$SF = \frac{P_{hp}}{P_{max}} \quad (3)$$

The SF influences crucial factors such as the SCOP, BHE size and installation costs. Typical sizing factors range between 0,5 and 0,7 meaning that 50 % to 70 % of the peak heating load is covered by the GSHP. By covering 70 % of the annual peak heating loads, roughly 95 % of the total annual heating demand could be covered by the GSHP. Since peak heating loads usually occur during very short time periods seen over the entire heating season, they are covered by auxiliary heating system instead (Warfvinge & Dahblom, 2010). By doing this, the GSHP capacity could be significantly reduced, which decreases the initial costs of the entire system. Once the heating demand and SF are determined, the next step is to dimension the BHE required to extract the thermal energy (Sarbu & Sebarchievici, 2016).

The required total borehole depth depends on several factors, such as the amount of thermal energy to be extracted, the available land area, geothermal properties of the ground and the layout and configuration of the BHE. An overall chart that illustrates the factors influencing the required borehole depth is shown in Figure 6.

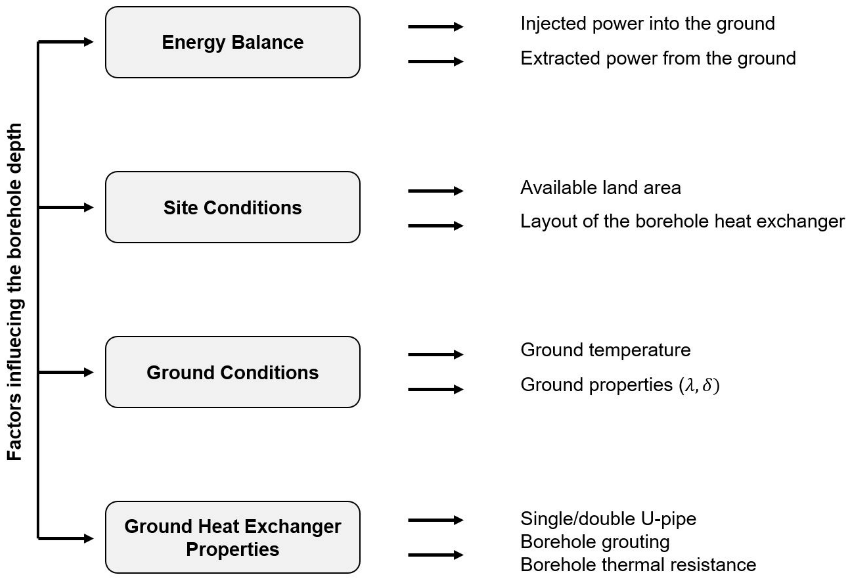

Figure 6: Factors influencing the required borehole depth.

A careful consideration to the energy balance of the ground must be given. The energy balance consists of the thermal energy extracted from the ground during winter and the energy injected from solar radiation and cooling operations during summer. If more energy is extracted than injected over years, the ground temperature will decrease, causing borehole depletion and reducing the SCOP of the system (Zhao et al., 2018).

To decrease borehole depletion, the layout of the BHE should consider the number of boreholes, their depth and spacing between them. If the spacing between boreholes is insufficient, they will negatively interfere with each other. In places with limited usable land areas such as urban areas, spacing between the boreholes is a limiting parameter, causing negative interference. To compensate for area limitations, fewer but deeper boreholes could be drilled. As seen in Figure 7, the average borehole depth is continuously increasing as GSHP gains popularity in urban area (Mazzotti et al., 2018).

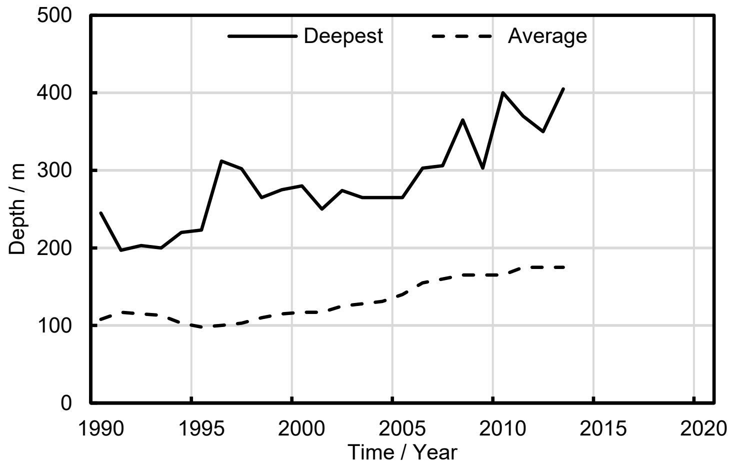

Figure 7: Increasing borehole depth over the last decades, adapted from Mazzotti et al. (2018).

Ground properties such as the ground temperature, specific heat capacity and thermal conductivity are influencing the required total borehole depth. Because the COP of GSHPs increases with increased source temperatures, high ground temperatures are preferable for heating purposes. But also increased specific heat capacity and thermal conductivity of the ground could benefit the GSHP system (Mazzotti et al., 2018).

2.3.3 Borehole heat exchanger thermal resistance

Additional factor influencing the total borehole depth is the configuration of the BHE. The BHE configuration influences the thermal resistance and heat transfer rate from the ground to the circulating fluid in the U-pipes and the other way around. The BHE thermal resistance depends on the borehole dimensions, number of U-pipes and their properties, the mixture of the circulating fluid and the borehole grouting (Kurevija et al., 2017). A detailed review of existing methods for calculating boreholes thermal resistance is available in (Javed & Spitler, 2016) and (Javed & Spitler, 2017). Multipole method is recommended for calculating thermal resistance of grouted boreholes (Bennet et al., 1987). Simplified formulas of multipole expansions are available for single and double U-tube configurations in (Claesson & Javed, 2018; Claesson & Javed, 2019; Claesson & Javed, 2020). Since groundwater levels in Sweden generally are high, boreholes in Sweden are often filled with groundwater (Gehlin et al., 2016). The advantage of groundwater filled boreholes in comparison to grouted are decreased thermal resistances within the borehole configuration, increasing the efficiency and SCOP of the system (Kurevija et al., 2017). According to Javed & Spitler (2016), the thermal resistance of groundwater filled boreholes (R_b) is calculated according to equation (4) and is dependent on the convective thermal resistance at the borehole wall (R_{BHWc}), the outer thermal resistance of the U-pipe (R_{poc}), the conductive thermal resistance of the U-pipe (R_{pc}) and the inner thermal resistance of the U-pipe (R_{pic}). Each resistance and temperature measurement point for a groundwater filled single U-pipe BHE are shown in Figure 8.

$$R_b = R_{BHWc} + \frac{(R_{poc} + R_{pc} + R_{pic})}{2} \tag{4}$$

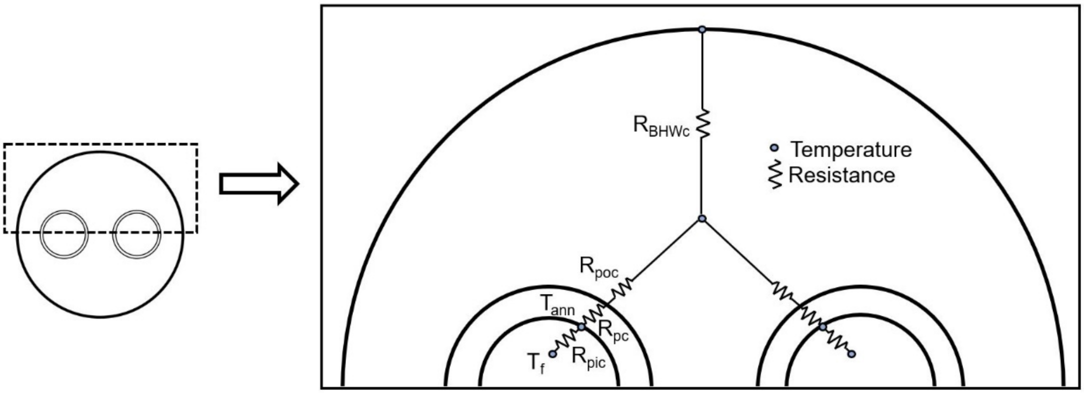

Figure 8: Borehole resistances and temperatures for a groundwater filled single U-pipe, adapted from Spitler et al. (2016).

3 Methodology

The methodology chapter presents the quantitative research approach of the study. Additional information about the case study, the simulation cases, the approach to estimate the building's mean thermal transmittance and further details about the LCC are provided.

3.1 Case Study

Sections D, E and F of the previously introduced elderly home is providing residential care for 42 care recipients and accumulate a wide range of different room types for both caretakers and staff. The different rooms are individual rooms for caretakers, large living rooms, large dining room, kitchen, spa, expeditions, and offices. Floor plans for the sections are provided in Appendix A and technical characteristics of the sections are provided in Table 2.

Table 2: Technical characteristics of the studied sections.

Heated floor area / m^2	1 458
Measured building envelope area / m^2	3 442
Window type	Double-pane*
Window U-value / $W/(m^2{\cdot}K)$	2,5*
Solar Heat Gain Coefficient (SHGC)	0,76*
Measured window area / m^2	138

**Estimated*

According to the energy declaration performed in 2019 and shown in Appendix B, the Specific Energy Use (SEU) of the entire elderly home was 164 kWh/m^2, which corresponds to energy class D. The achieved energy class is dependent on the building's energy performance in relation to modern requirements. All classes and the modern elderly home requirements are shown in Table 3.

Table 3: Energy classes according to energy declaration.

Energy class	Percentage of modern requirements*
A	$\leq$ 50
B	> 50 - $\leq$ 75
C	> 75 - $\leq$ 100
D	> 100 - $\leq$ 135
E	> 135 - $\leq$ 180
F	> 180 - $\leq$ 235
G	< 235

**Modern specific energy use requirements for elderly homes, 126 kWh/m^2 and year.*

The energy performance is adequate for buildings constructed during the 1990s and 2000s, but as higher requirements are set on the thermal performance of building envelopes, ventilation systems and heating plants, it is difficult to compare the building with modern buildings.

3.1.1 Description of energy and ventilation system

Sections D, E and F are sharing the same heating plant. Like many buildings constructed during the 1980s and 1990s, heating is provided by boilers operating on fuels, in this case pellets, distributed by a hydronic radiator system. To avoid boiler short-cycling, three buffer tanks for space heating and DHW are present with a total capacity of 1,25 m^3. In 2020, the heating plant of sections D, E and F underwent renovation, and the boilers were substituted by a non-reversible 40 kW GSHP as described in Appendix C. The boilers are now serving as auxiliary heating. As it currently stands, no cooling distribution system exists and the free cooling possibilities are not utilised. Therefore, the heat pump is only operating in heating mode. Four vertical boreholes were drilled to supply the heat pump with low-grade energy from the ground. The layout of the boreholes and the configuration of the BHE are visualised in Figure 9.

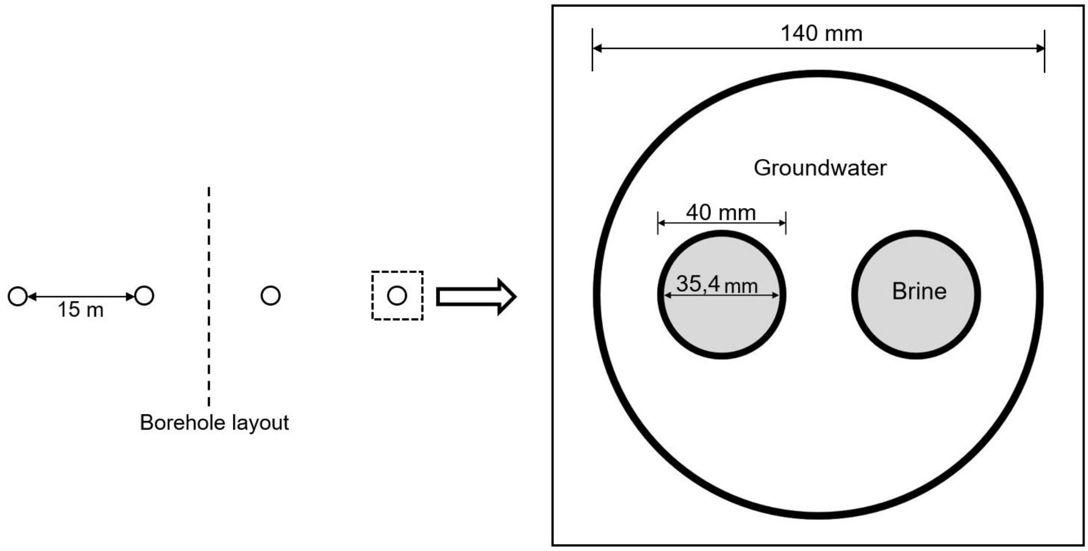

Figure 9: Configuration and layout of the borehole heat exchanger.

Further technical characteristics of the borehole heat exchanger and circulating fluid are provided in Table 4.

Table 4: Technical characteristics of the borehole heat exchanger.

Borehole	
Layout	1x4
Spacing	15 m
Total effective depth	920 m
Heat exchanger	
Configuration	Single U-pipe
Inner pipe radius	35,4 mm
Pipe thickness	2,3 mm
Pipe thermal conductivity	0,40 W/(m·K)
Spacing of shanks	Uncontrolled
Filling	Groundwater
Filling thermal conductivity	0,6 W/(m · K)
Filling specific heat capacity	4180 J/(kg · K)
Circulation pump efficiency	80 %*
Circulating fluid	
Type	Ethanol (20 %)
Freezing point	-15 ℃
Average borehole liquid mass flow	1 kg/s*
Thermal conductivity	0,4 W/(m·K)
Specific heat capacity	4100 J/(kg·K)

**Estimated*

Sections D, E and F are supplied by two Constant Air Volume (CAV) ventilation systems, LA1 and LA2. The Specific Fan Power (SFP) and the supply and extract air volumes for each room are found in Appendix D. The second floor of section D is supplied by LA1 and the remaining spaces are supplied by LA2. Both AHUs do not have cooling coils but are equipped with enthalpy wheel heat exchangers only operating in heating mode.

3.1.2 Site and climate analysis

The elderly home is located on a hill next to cliffs leading to a lake on the north and west side. Apart from several trees in the cliffs, there are barely any obstacles providing shading besides the adjacent buildings, which exposes the sections to solar radiation. To analyse the climate conditions of the case study, several climate-dependent parameters such as the outdoor dry-bulb temperature, the global horizontal radiation and the relative humidity were acquired from the weather file of Jönköping airport. The ground conditions from the site were obtained from geological maps from the Geological Survey of Sweden (SGU) (*Geological Survey of Sweden*, 2021) and the undisturbed ground temperature of Jönköping was obtained from (Xing, 2014).

3.2 Determination of building envelope thermal transmittance

Because construction drawings were unavailable, the mean thermal transmittance of the studied sections was estimated from the total transmission losses. The transmission losses are acquired from the building's heating balance according to equation (5). To calculate the transmission losses, all other factors included in the heating balance are required. The heating balance includes the heat losses from the building which includes transmission losses (P_t), ventilation losses (P_v), infiltration losses (P_{inf}), and the heat gains include space heating (P_{sh}), solar gains (P_s) and internal gains (P_i).

$$P_t + P_v + P_{inf} = P_{sh} + P_s + P_i \tag{5}$$

The following section describes the derivation of the building heat losses and heat gains.

3.2.1 Heat loss rate

Now that the heat gains of the heat balance in equation (5) are specified, the heat losses are required to obtain the section's transmission losses. The heat losses consists of transmission losses, ventilation losses and infiltration losses. Further details about these losses and the approach to estimate these are provided below. The losses are calculated as specific losses and require multiplied with the location's heating degree hours.

Infiltration loss (Q_{inf}), is the heat lost through leaks in the building's envelope. Leaks are normally located in the building's construction and around openings such as windows and doors. The infiltration losses are estimated according to equation (6) and depends on the density of the air (ρ_{air}), the specific heat capacity of the air ($c_{p,air}$) and the infiltration leakage air flow ($q_{infiltration}$).

$$P_{inf} = \rho_{air} \cdot c_{p,air} \cdot q_{infiltration} \tag{6}$$

In many cases, the leakage airflow ($q_{leakage}$), is stated at 50 Pa pressure differences between the indoor and outdoor. According to equation (7), the leakage air flow is converted to the infiltration air flow at normal pressure conditions. The leakage air flow is assumed to be equal to 0,6 l/s per m^2 envelope area (Boverket, 2020).

$$q_{infiltration} = 0{,}05 \cdot q_{leakage} \tag{7}$$

Ventilation losses (Q_v), are the heat lost through the supply air. During the exhaustion process of the building's extract air, the sensible heat of the extract air is not completely recovered. The specific ventilations loss is calculated according to equation (8), dependent on the density of the air exhaust air (ρ_{air}), the specific heat capacity of the exhaust air ($c_{p,air}$), the extract air flow (q_{vent}) and the efficiency of the heat recovery (η).

$$P_v = \rho \cdot c_p \cdot q_{vent} \cdot (1-\eta) \tag{8}$$

Transmission losses (Q_t), are the heat lost through the building's envelope. The loss occurs through the building's construction parts such as external walls, foundation, roof and thermal bridges. The specific transmission loss is calculated according to equation (9), dependent on the thermal transmittance of the construction parts (U_i), the construction part's corresponding surface area (A_i) and the thermal bridges ($\mathrm{Q_{tb}}$).

$$P_t = Q_{tb} + \sum_i U_i \cdot A_i \tag{9}$$

3.2.2 Heat gains

The heat gains of a building consists of space heating, solar gains and internal gains from equipment and occupancy. The heating plants of buildings are generally dimensioned to heat the ambient air to 17 °C while solar gains and internal gains are contributing to achieve indoor temperatures of 21 °C to 22 °C. Therefore, the solar and internal gains of the heating balance can be set to zero when determining the sites heating degree hours (h) for 17 °C. The heating degree hours were calculated from hourly weather data measured at Jönköping airport for the same year as the energy declaration was performed.

The space heating demand of the studied sections is derived from the elderly home's average SEU according to the energy declaration. The SEU includes the energy use for heating (E_h), energy use for cooling (E_c) and the facility energy (E_f) according to equation (10).

$$E_{tot} = E_h + E_c + E_f \tag{10}$$

The total energy use for the studied section's is derived from equation (13) and is dependent on the elderly home's SEU and the studied sections heated floor area (A_{temp}).

$$\mathrm{E} = SEU \cdot A_{temp} \tag{11}$$

The studied section's cooling energy use is set to zero since no cooling is provided. The studied section's facility energy use is estimated according to the elderly home's average facility energy use according to the energy declaration, which is equivalent to 10,9 kWh/m^2. The studied section's facility energy use is calculated according to equation (12).

$$E_f = 10{,}9 \cdot A_{temp} \tag{12}$$

The studied section's heating energy in equation (10) is divided into the energy use for space heating (E_{sh}) and the energy use for DHW (E_{DHW}) according to equation (13).

$$E_h = E_{sh} + E_{DHW} \tag{13}$$

The energy use for DHW is estimated according to the elderly home's average DHW use according to the energy declaration, which is equivalent to 34,1 kWh/m^2. The studied section's DHW energy use is calculated according to equation (14).

$$E_{DHW} = 34{,}1 \cdot A_{temp} \tag{14}$$

Now that the energy for heating and DHW is calculated, the studied sections space heating demand can be obtainable as expressed in equation (13).

3.2.3 Thermal transmittance

Now that the sections transmission losses are specified, the section mean envelope thermal transmittance (U_{mean}) can be specified according to equation (15), dependent on the space heating energy (Q_{sh}), the specific ventilation and infiltration loss, the heating degree hours (h), and the sections envelope area (A_{env}).

$$U_{mean} = \frac{Q_{sh} - (Q_v + Q_{inf}) \cdot \mathrm{h}}{A_{env} \cdot \mathrm{h}} \tag{15}$$

The mean envelope thermal transmittance requires adjustment to consider the heat gains and losses from glazed construction parts with significant higher thermal transmittances such as windows. This is accomplished by calculating the adjusted mean envelope thermal transmittance (U_{adj}) according to equation (16), which is considering the sections window area ($A_{windows}$) and their thermal transmittance ($U_{windows}$).

$$U_{adj} = \frac{U_{mean} \cdot A_{tot} - U_{windows} \cdot A_{windows}}{A_{tot}} \tag{16}$$

3.3 Simulation cases

This study features several simulation cases to analyse the energy performance and indoor climate of the base case and for the integration of GSHPs. The 3D-model developed in Revit and exported to IDA-ICE which is used for all simulation cases is shown in Figure 10. Further details about the cases and simulation inputs are provided in the sections below.

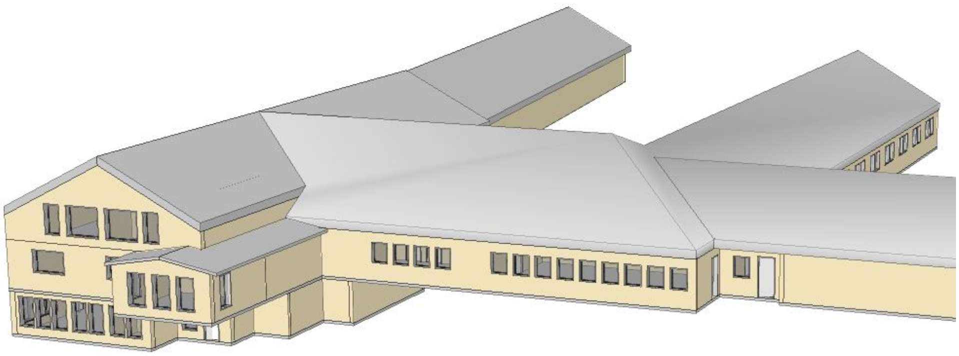

Figure 10: Visualisation of the energy model in the energy simulation software IDA-ICE.

The different simulation cases and the workflow behind the simulations are presented in Figure 11 and consists of the following processes; the energy performance and indoor climate conditions of the base case were evaluated, three different cooling measures were simulated to achieve the desired indoor climate, the possibility of covering the heating and cooling demands to achieve the improved indoor climate with a GSHP is evaluated, finally the GSHP's combined heating and cooling ability and energy performance is compared to the base case and alternative heating and cooling systems from an LCC perspective.

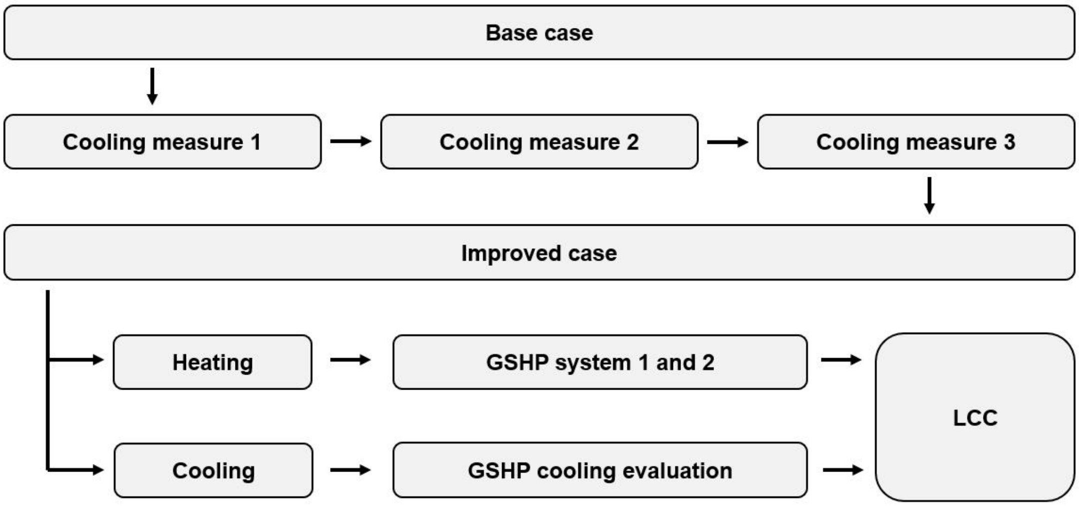

Figure 11: The workflow between the simulated cases.

3.3.1 Base case simulation

The base case is intended to simulate the studied sections before the GSHP installation. This case is created to analyse the thermal comfort of the sections and the energy performance of the old boilers fuelled by pellets. Inputs for the heating system, ventilation system, internal gains and schedules of the base case are provided in Table 5. The occupancy schedule for working spaces such as offices, expeditions and kitchens were set to 08:00-17:00 for every day of the week. The number of occupants was set between one and ten depending on the floor area of the rooms. For care recipient rooms a normal living schedule was applied and the occupancy of each care recipient room was set to one. Additional internal gains of computers in offices, staff rooms and expeditions were set to 100 W per unit. The additional heat gains from the kitchen were considered to be equal to 20 W/m^2. Internal gains from lighting were neglected as lighting is provided by LEDs.

Table 5: Base case inputs for the heating system, ventilation system and the internal gains.

Heating system	
Heating plant	Boiler
Fuel type	Pellets
Heat distribution system	Hydronic radiators (55 °C / 45 °C)
Heating setpoint / °C	22
Buffer tank size / m^3	1,25
Mechanical ventilation	
Type	CAV
Heat recovery efficiency / %	80
Air flows	According to Appendix D
Supply air temperatures	Winter 19 °C, summer 17 °C

The configuration of the heating plant modelled in IDA-ICE including the boiler, buffer tank and heating setpoints of the DHW and AHU are shown in Figure 12.

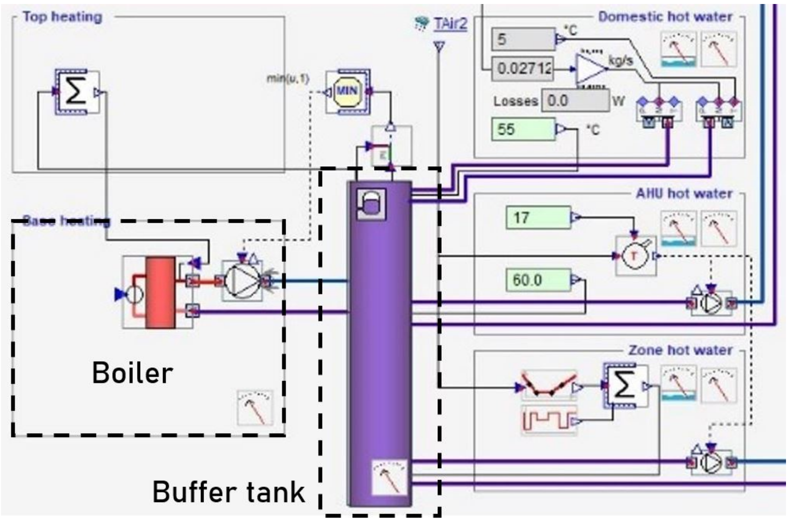

Figure 12: Base case heating plant configuration in IDA-ICE software.

The configuration of the AHU modelled in IDA-ICE including the enthalpy wheel heat recovery and supply air heating coil is shown in Figure 13. The supply air temperature is configured according to a graph dependent on the outdoor and indoor air temperature.

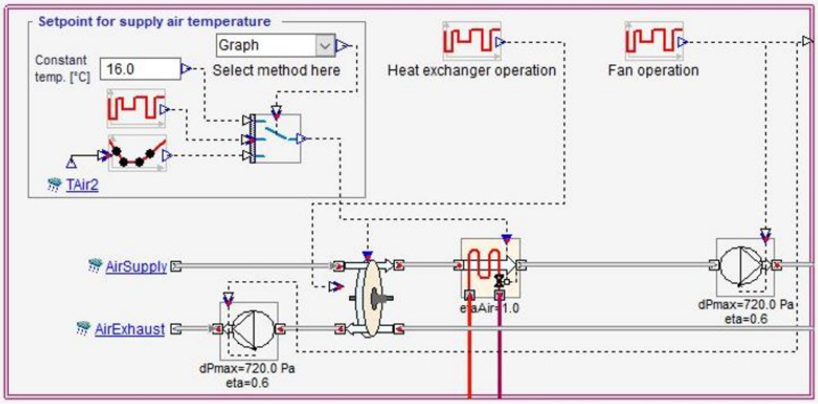

Figure 13: Base case air handling unit configuration in IDA-ICE.

3.3.2 Cooling measures

Since the base case of the studied sections was not provided with space cooling, several simulations evaluating the implementation of passive and active cooling measures were made. These cooling measures aims at providing indoor climates suitable for elderly people. Therefore, a maximum operative temperature of 25 ℃ was targeted during summer, to guarantee temperatures below the maximum temperature recommendations of 26 ℃ according to the guidelines of the Swedish public health authority (Folkhälsomyndigheten, 2014).

3.3.2.1 Cooling measure 1 – Addition of external shading

Generally, the implementation of passive cooling measures are introduced before active cooling measures. The advantage of passive cooling measures is the possibility of reducing the building's cooling demand and peak cooling loads without causing additional operating costs. The passive cooling measure introduced as measure 1 is the implementation of external shading devices. The windows of each base case room exposed to Overheating Hours (OHs) were equipped with external blinds with a Solar Heat Gains Coefficient (SHGC) of 0,14. Considering the importance of maintaining the mental health of the elderly home's care recipients which depends on daylight access, the external blinds were configured to minimize periods of completely drawn blinds. This was accomplished by configuring the external blinds to be dependent on the Global Horizontal Radiation (GHR). The external blinds were completely drawn for solar radiation levels exceeding 85 % of the annual maximum level. For the remaining time, the external blinds were completely open.

3.3.2.2 Cooling measure 2 – Addition of cooling coil in the AHU

After reducing the cooling demands and peak cooling loads by passive measures, the implementation of the first active measure was introduced. The first active measure introduced as measure 2 is the implementation of a cooling coil in the existing CAV-system. This is accomplished by replacing the existing AHU with a new one including a cooling coil to cool the supply air. To avoid additional costs and draft, the section's existing ventilation duct system and diffusers are kept, maintaining the current supply and extract air flows according to Appendix D. The new cooling coil was dimensioned to provide supply air temperatures of 17 ℃ during the cooling season. The cooling capacities are dependent of the supply air flows to each room.

The required AHU replacement offers the opportunity to reduce the AHU's fan operating energy by decreasing the Specific Fan Power (SFP). To calculate the potential operating energy reduction, several assumptions of the different components and the respective pressure drops were made according to Nilsson, (2003), presented in Table 6. The fans of the existing AHU were assumed to operate at reduced efficiency due to clogging in the components and duct system. The design inlet temperature of the cooling coil was set to 7 ℃.

Table 6: Air handling unit components and corresponding pressure drops.

Component	Old AHU	New AHU
Supply air filter / Pa	170	150
Heat recovery / Pa	190	170
Silencer / Pa	40	40
Heating coil / Pa	40	40
Cooling coil / Pa	-	60
Humidifier / Pa	-	110
Fan efficiency / %	50	75

The pressure drops of the studied sections duct system was estimated from the supply air fan power (P) according to the OVK and equation (17), which is dependent on the supply air flow (q), the system pressure drops (p) and the fan efficiency (η).

$$P = \frac{q \cdot \Delta p}{\eta} \tag{17}$$

3.3.2.3 Cooling measure 3 – Addition of passive beams

After the implementation of measure 1 and 2, rooms still exceeding the operative temperature threshold of 25 ℃ for more than 100 hours were equipped with additional room cooling units. Passive beams were favoured over active beams as the space cooling unit to minimize draft and additional installation costs caused by the modifications on the supply air diffusers. The design inlet temperature of the passive beams was specified to 14 ℃. The cooling capacity of the passive beams was calculated to reach the desired indoor climate.

After the implementation of cooling measure 1, 2 and 3, all rooms of the studied sections should achieve the desired indoor climate. This case is referred to as the "Improved case" in Figure 11.

3.3.3 Integration of ground source heat pumps

After the heating demand of the base case and the cooling demand of the improved case were determined, the impact of the recently installed GSHP is analysed. This was divided into two parts, the heating part and the cooling part.

The heating part consists of two analysed GSHP systems. The first system, system 1, was analysing the energy performance of the 40 kW heat pump and the installed borehole heat exchanger utilising the existing boiler as auxiliary heating to cover the peak heating loads.

The second system, system 2, consists of the 40 kW heat pump in combination with an additional 16 kW heat pump utilising the same borehole heat exchanger configuration. The energy performance of both GSHP systems was compared to the base case, both short term and long-term fluid temperatures are performed to analyse the feasibility of system 2. Further details about the heating plant configuration used for system 1 and 2 consisting of the BHE, the heat pump, buffer tank and the boiler used during peak demands are shown in Figure 14. The borehole resistances of both systems were assumed to 0,06 K/(W/m) based on the model of (Spitler et al., 2016).

The cooling simulations consist of the free cooling evaluation to determine if the free cooling capacities of the ground are enough to cover the entire cooling season or if additional cooling systems are required. To analyse the free cooling capacities, the fluid temperatures of the boreholes are simulated and controlled whether or not they exceed the required inlet temperature of the passive beams.

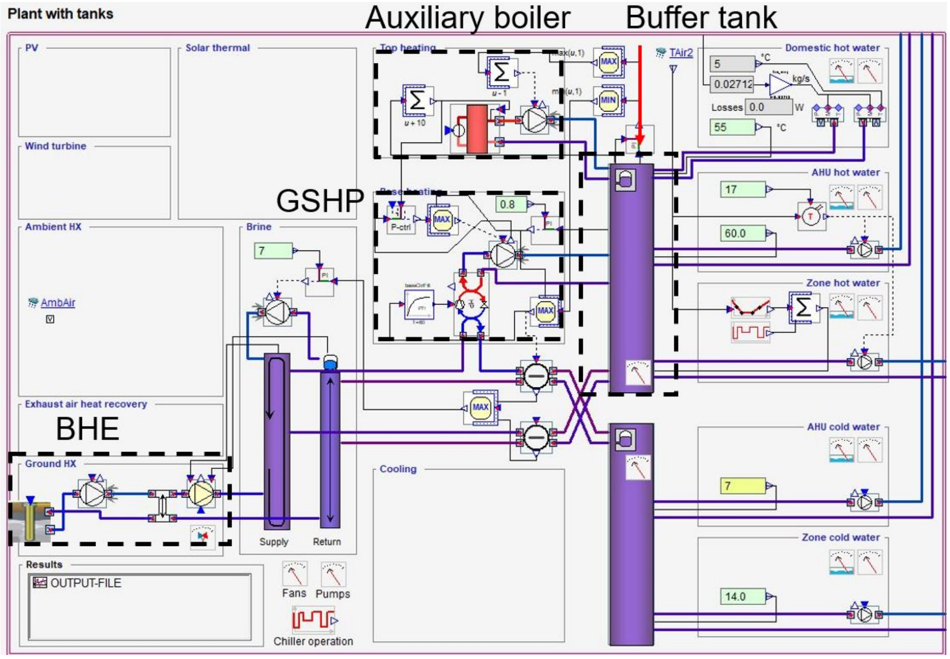

Figure 14: Improved case heating plant configuration in IDA-ICE software.

3.4 Life cycle costing

To analyse the installation and operation costs of the installed GSHP, the different heating and cooling solutions proposed in the previous section were compared to alternative systems. The life cycle costs for heating operations of the recently installed 40 kW and the proposed 56 kW GSHP system were compared to the life cycle costs of district heating and the base case boiler operating on different fuels. To accomplish this, the potential savings or losses generated by replacing the boilers operating on different heating sources and district heating of the GSHP systems were calculated. When replacing the boiler and district heating with the

GSHP system, the replaced system is assumed to serve as auxiliary heating. The life cycle costs of the GSHP cooling solutions consisting of free and active cooling were compared to cooling provided by air-source chillers. To compare the LCC of the cooling solutions, the NPV of the different systems was calculated.

To calculate the savings or losses generated by replacing the boilers operating on different sources and district heating, the NPV for each successive year is calculated according to equation (18), dependent on the initial costs of the GSHP installation and the annual savings and maintenance costs.

$$\mathrm{NPV} = -\mathrm{Initial\ costs} + \mathrm{savings} - \mathrm{maintenance} \tag{18}$$

The savings and maintenance costs for each year (P), were calculated according to equation (19), dependent on the annual savings and maintenance costs (B_1), the annual price increase of the heating sources and maintenance (g), the interest rate (i), and the time span (n).

$$P = B_1 \left[\frac{1 - (1 + g)^N \cdot 1 - (1 - \mathrm{i})^{-N}}{i - g} \right] \tag{19}$$

The annual maintenance costs at year one (B_1), were calculated according to equation (20), dependent on the maintenance costs at year zero (B_0) and the annual price increase (g).

$$B_1 = B_0 \cdot (1 + g) \tag{20}$$

The decreasing buying power over time is considered by calculating the real interest rate (i_r), according to equation (21), dependent on the nominal interest rate (i_n) and the inflation rate (k).

$$(1 + i_n) = (1 + i_r)(1 + k) \tag{21}$$

The costs of the different boiler heating sources and district heating were based on statistical data or assumptions if recent price fluctuations were high or were highly dependent on the local distributors. The fixed electricity price was calculated based on hourly statistical data and additional costs for grid fees, taxes, and green energy certificates were added. The pellets price was obtained from the Swedish Pellets Association (Pellets förbundet, 2018), the price for district heating was obtained from the Swedish statistics centre (Statistikcentralen, 2019), the price for oil was estimated to 12 750 SEK/m^3 with a boiler efficiency of 80 %. Further details about the initial costs, maintenance costs, heating source prices and economical inputs are provided in Table 7. The prices of the different heating sources were assumed to increase equally over the considered timespan. To consider the long lifespan of the borehole heat exchanger a calculation time of 40 years was considered for the heating LCC. Therefore, the GSHPs are replaced after 20 years. The replacement costs of the boilers serving as auxiliary heating were not considered. Because the operating costs of GSHPs are highly influenced by the electricity price, a further sensitivity analysis was carried out, considering an electricity price of 2,0 SEK/kWh.

Table 7: LCC calculation inputs.

Economical		
Nominal interest rate	2%	
Inflation rate	1%	
Timespan heating	40 years	
Timespan cooling	20 years	
Initial costs		
Boreholes	178 000 SEK	
GSHP System 1	209 000 SEK	
GSHP System 2	326 000 SEK	
Air cooled chiller	168 000 SEK	
Circulation pump	30 000 SEK	
Annual maintenance		
Boiler	2 500 SEK	
GSHP System 1	1 500 SEK	
GSHP System 2	2 000 SEK	
Air cooled chiller	1000 SEK	
Annual price increase	1%	
Heating prices		
Electricity	1,45	SEK/kWh
Pellets	0,67	SEK/kWh
Oil	1,6	SEK/kWh
DH	0,8	SEK/kWh

4 Results

This chapter presents the results of the site and climate analysis, the energy performance of the introduced cases, the indoor climate evaluation of the base and the impact of the cooling measures, the results from the GSHP integration and life cycle costings.

4.1 Site and climate analysis

The climate conditions of Jönköping from 2020 are presented in Figure 15. The average outdoor temperature is indicating that the location is heating dominated. The lowest temperature was reached on the 27th of February and the highest dry-bulb temperature was reached on the 25th of June.

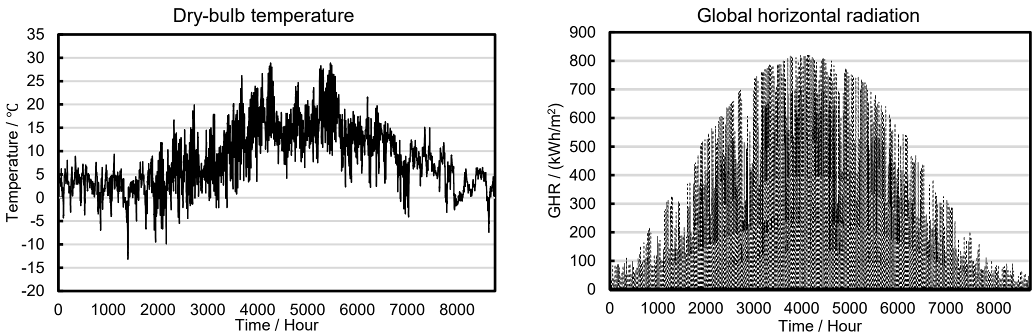

Figure 15: Dry-bulb temperature and global horizontal radiation of Jönköping.

Further details about the climate conditions are provided in Table 8.

Table 8: Minimum, average, and maximum values.

Parameter	Min	Average	Max
Dry-bulb temperature / °C	-13,2	8,2	28,9
Global horizontal radiation / (W/m^2)	-	119	824
Relative humidity / %	20,0	78,4	100

The approximate ground conditions of the site obtained from Geological Survey of Sweden (SGU) are presented with further geothermal properties and the thermal resistance of the groundwater filled borehole in Table 9.

Table 9: Ground conditions.

Geothermal properties	**Value**
Type	Granite
Density / (kg/m^3)	2 700
Specific heat capacity / (J/kg·K)	7 900
Thermal conductivity / (W/(m·K))	3,0
Undisturbed ground temperature / ℃	8,9
Borehole thermal resistance / (K/(W/m)	0,06

4.2 Base case

In this section, the results of the heat losses and heat gains calculations and the mean thermal transmittance of the studied section's envelope area are presented. Moreover, the energy performance of the base case and the indoor climate conditions are analysed.

4.2.1 Energy performance

The studied section's SEU according to the energy declaration and the heated floor area is provided in Figure 16. Space heating is accountable for the largest share followed by DHW.

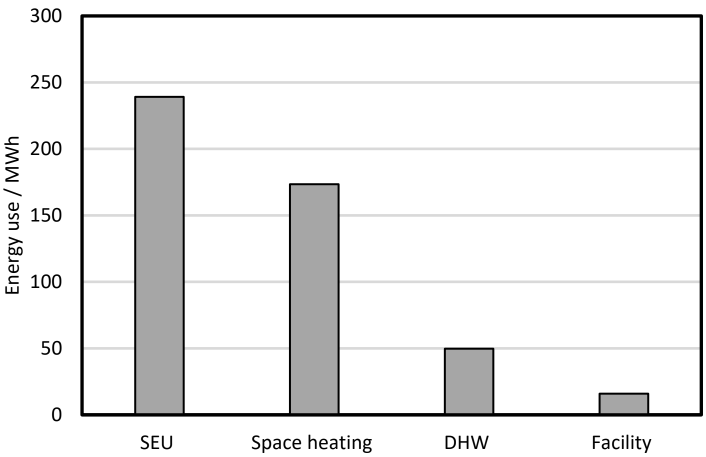

Figure 16: Base case energy use based on energy declaration.

The heating degree hours required to calculate the section's heat losses were calculated to 86 700 Kh according to the weather data measured during the same year as the energy declaration was carried out. The specific and total heat losses required to determine the transmission losses are presented in Table 10. The transmission losses were calculated to 125 000 kWh and were accountable for 71,9 % of the total heat losses.

Table 10: Base case calculated losses.

Heat losses	Specific losses / (W/K)	Total losses / (kWh/year)	Share / %
Ventilation	349	35 700	20,6
Infiltration	127	11 000	7,5
Transmission	1 222	125 000	71,9

The mean thermal transmittance of the section's envelope was calculated to be 0,44 W/(m^2·K) according to the transmission losses and the section's envelope area of 3 442 m^2. To consider the special thermal properties and thermal transmittance of the windows, the adjusted thermal transmittance of the opaque building envelope surfaces was calculated to be 0,357 W/(m^2·K)

4.2.2 Energy simulations

The annual heating demand and the peak heating loads of the base case were determined by annual energy simulations and the previously calculated mean thermal transmittance. The simulated annual heating demand was to 224 000 kWh and the peak heating loads of 67 kW. The heating balance of the base case consisting of the simulated heat gains and heat losses is illustrated in Figure 17.

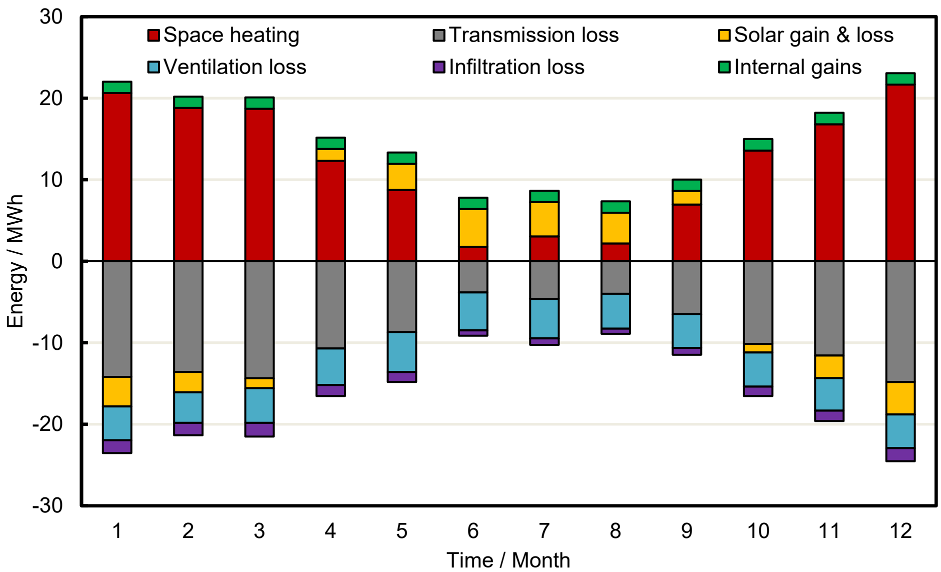

Figure 17: Base case energy balance.

The simulated heating profile of the base case shows variations of the annual heating demand is illustrated in Figure 18. The highest heating loads were occurring during winter as the demand for space heating increased. The heating demand during summer consisted of DHW and AHU heating for conditioning the ventilation air to the required supply air temperature.

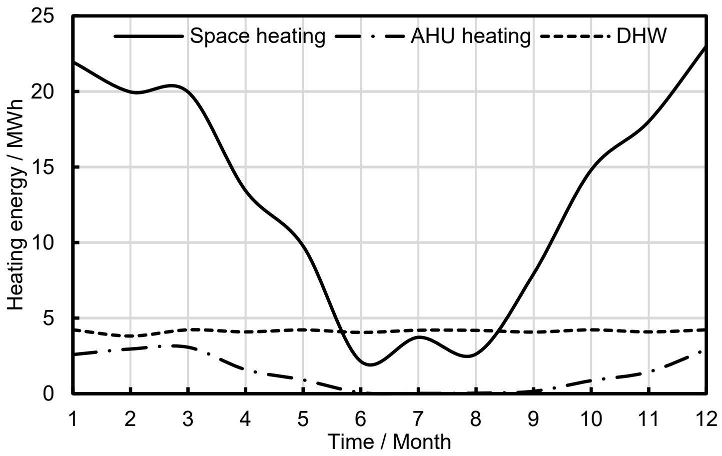

Figure 18: Base case heating profile.

4.2.3 Transition from calculated to simulated results

The manual calculations of heat losses and heat gains were compared to the simulated results to verify the transition from the calculated heat gains and losses to the simulated ones and the accuracy of the mean thermal transmittance approach, the manually calculated and simulated values were compared in Figure 19. The highest difference was observed in the ventilation losses (27 %) and the transmission losses (8 %). The differences in the ventilation losses are related to distinctions of the assumed heat recovery efficiency and the actual simulated efficiency based on the supply and extract air flows. The differences in the transmission losses were caused by simplified heating degree methods and simulations based on hourly detailed heat balance.

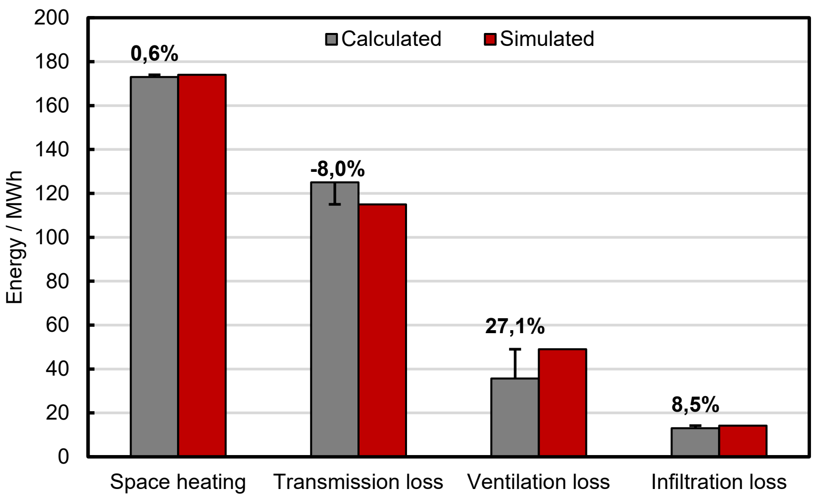

Figure 19: Comparison of hand-calculated and simulated results.

4.2.4 Thermal comfort

The energy simulations of the base case were also used to evaluate the indoor climate. Different kind of rooms located at the corner of the studied sections were chosen to illustrate the impact of the different cooling measures. The chosen rooms of the 1st and 3rd floors are shown in Figure 20 and the rooms of the 2nd floor are shown in Figure 20. The rest of the evaluated rooms and further details are provided in the Appendix E. The 1st and 3rd floors are only featuring two important rooms, the staff room, and a large office.

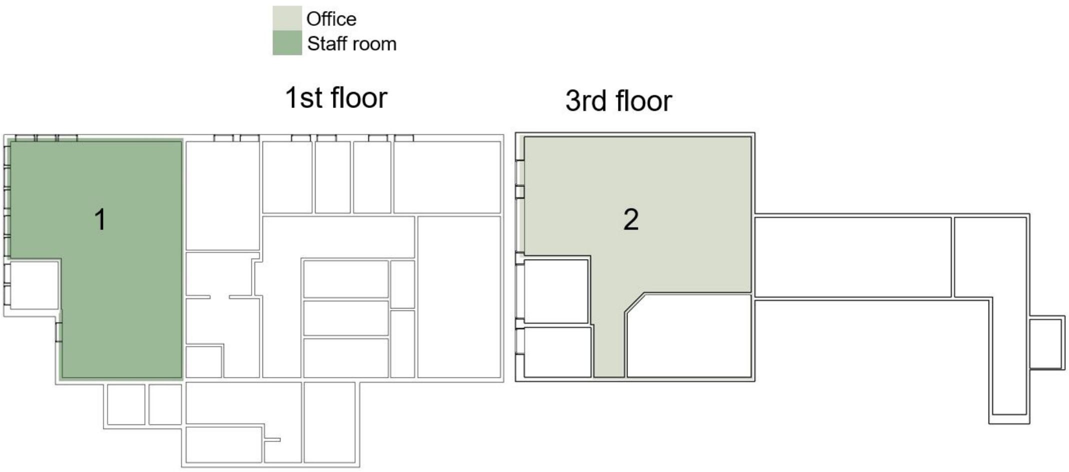

Figure 20: Selected rooms for the thermal comfort analysis of the 1st and 3rd floor.

The 2nd floor is including various types of rooms and spaces for elderly people. Most of these rooms are care recipient rooms facing different orientations, located at the corners of the floor plan as seen in Figure 21.

Figure 21: Selected rooms for the thermal comfort analysis of the 2nd floor.

The results of the base case indoor climate evaluation and the impact of the different cooling measures is illustrated in Figure 22. Rooms oriented towards the south and west were exposed to the highest OHs. Because shading provided from adjacent buildings and obstacles were minimal, these orientations were directly exposed to direct solar radiation. The rooms oriented towards the north were exposed to fewer OHs and the OHs for rooms towards the west were highly dependent on the window-to-wall ratio.

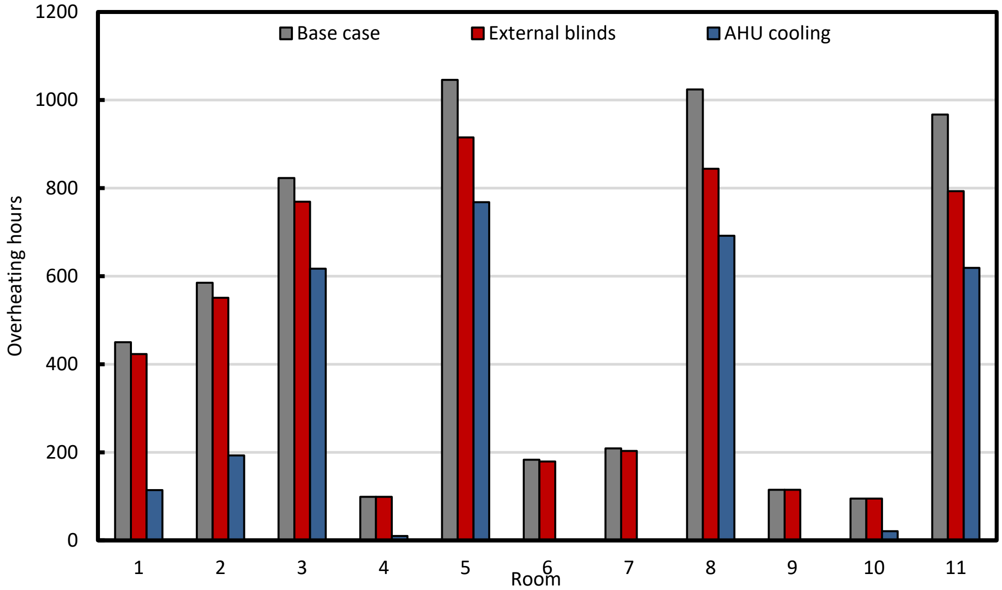

Figure 22: Impact of cooling measures on crucial rooms.

All rooms exposed to OHs were equipped with external blinds configured to be completely drawn at GHR above 700 W/m2. These shading devices were proven to be the most efficient for rooms oriented towards the south as OHs were decreased by up to 22 %. The external blinds were proven to be the least efficient for rooms oriented towards the north.

The implementation of airborne cooling reduced the OHs for each analysed room. The largest improvements were seen in rooms with high supply air flows such as the office and staff room. The airborne cooling was sufficient to achieve the desired indoor climate for the care recipient rooms towards the north and the kitchen and dining room towards the west. The annual cooling demand for the AHU cooling was simulated to 7 300 kWh and the peak cooling loads to 23,2 kW.

To determine the potential energy savings of the required AHU replacement, the pressure drops in the duct system were calculated to 230 Pa. By replacing the AHU, the SFP could potentially decrease by 21,5 %, yielding annual electricity savings of 6 100 kWh. The decreasing electricity use was caused by the higher efficiency of the fans and reduced pressure drops in the new components. Further characteristics of the existing and the existing and new system are provided in Table 11.

Table 11: Characteristics of the existing and new air handling unit.

Variable	Existing system	New system
Pressure drop ducting / Pa	230	230
Pressure drop SA / Pa	550	800
Pressure drop EA / Pa	590	550
SFP / (kWh/(m^3/s))	2,39	1,9
Annual fan electricity (kWh/year)	29 800	23 700
Cooling coil size / kW	n.a.	23,2
Heating coil size / kW	n.a.	16,4

All rooms still exposed to OHs larger than 100 h after the implementation of AHU cooling were equipped with passive beams. Because these rooms would achieve the desired indoor climate of zero OHs, the results are not shown in Figure 22. The annual cooling demand including room cooing units was simulated to 9 800 kWh and the peak cooling loads to 32,7 kW. The required passive beam capacity for each room is provided in the Appendix E.

The section’s annual cooling profile for the airborne cooling and the additional room units is visualised in Figure 23. The two larger peaks were occurring during the outdoor dry-bulb temperature peaks observed in Figure 15.

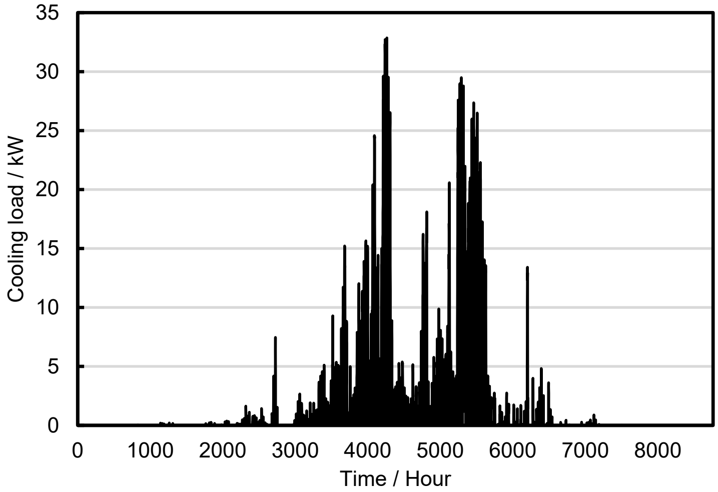

Figure 23: Improved case cooling profile.

4.3 Integration of ground source heat pumps

4.3.1 Heating demand coverage

To analyse the energy performance of the recently installed 40 kW GSHP and the proposed additional 16 kW heat pump utilising the existing borehole heat exchanger configuration, the systems heating coverage were simulated. The sizing factor of system 1 with the individual GSHP was calculated to 60 % and the heating demand of the base case was reduced by 68 %. The sizing factor of system 2 with the additional GSHP was calculated to 84 % and covered 72 % of the base case heating demand. The heating demand coverage ratio of the GSHP and auxiliary heating is provided for both systems in Figure 24.

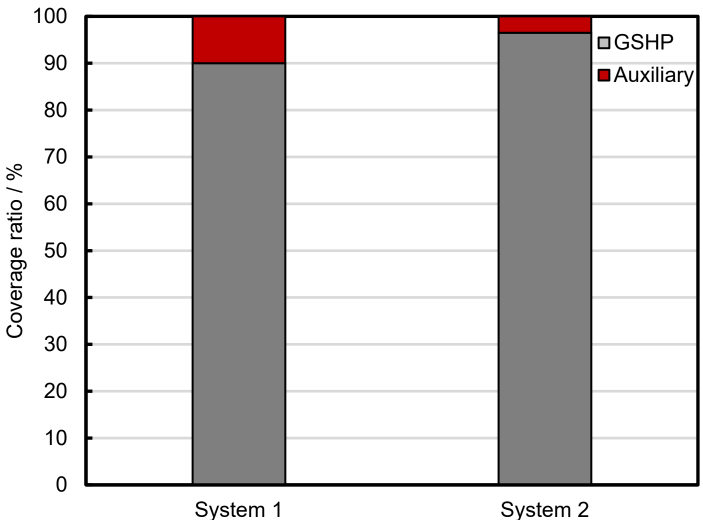

Figure 24: Ratio of heat coverage by the GSHPs auxiliary system.

The energy use for both GSHP systems is lower in comparison to the energy use of the base case due to the high SCOP of the GSHP. The energy use for the base case and both GSHP systems are shown in Figure 25.

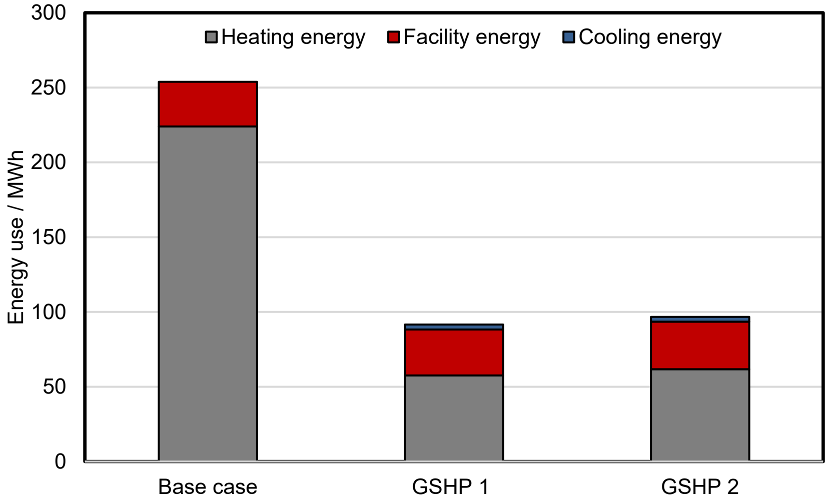

Figure 25: Energy use for the simulated cases.

Since different energy sources have a different environmental impact the primary energy use was calculated according (Boverket, 2020). The energy use is weighted based on the energy source. According to Boverket, electricity has a weighting factor of 1,8 and pellets under the biofuel category of 0,6. The primary energy use is presented in Figure 26.

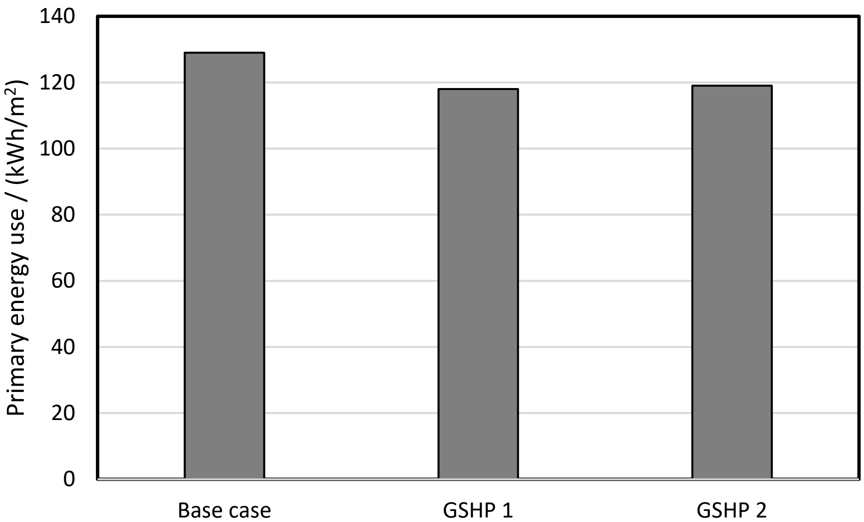

Figure 26: Primary energy use for the simulated cases.

The integration of GSHP system 1 and 2 reduced the primary energy use of the studied sections despite the significantly higher weighting factor of electricity in comparison to pellets. The reason for that is the high SCOP of GSHPs. The primary energy use of both systems is lower than the requirement for modern buildings, attaining energy class C according to Table 3.

4.3.2 Borehole fluid temperatures in system 1

To analyse the free cooling potential of the existing borehole configuration and the short-term borehole depletion, the inlet and outlet fluid temperatures of the different systems were simulated. The on-site measured inlet and outlet temperatures of the boreholes providing the individual heat pump of system 1 are presented in Figure 27.

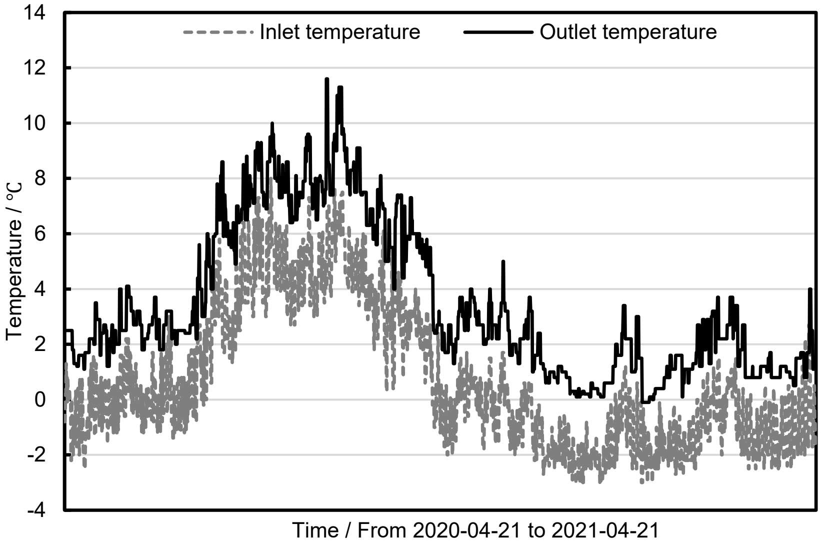

Figure 27: Measured fluid temperatures of GSHP system 1.

The free cooling capacities of the existing borehole configuration and the short-term inlet and outlet temperature fluctuations were analysed by simulating the fluid temperatures of system 1 with the individual GSHP for one year as seen in. Because free cooling is being utilised, the inlet fluid temperatures increased after passing through the cooling distribution system. Additionally, the simulated inlet temperatures are lower in comparison to the measured temperatures during the heating season and larger temperature differences between the inlet and outlet were observed. These differences are related to the distinctions of the assumed and actual conditions of the ground properties and the borehole mass flow. The outlet fluid temperatures are peaking at 12 °C and are therefore sufficient to supply the passive beams with the design inlet temperature of 14 °C. However, the outlet fluid temperatures are insufficient to supply the AHU cooling coil with the design inlet temperature of 7 °C for the entire cooling season.

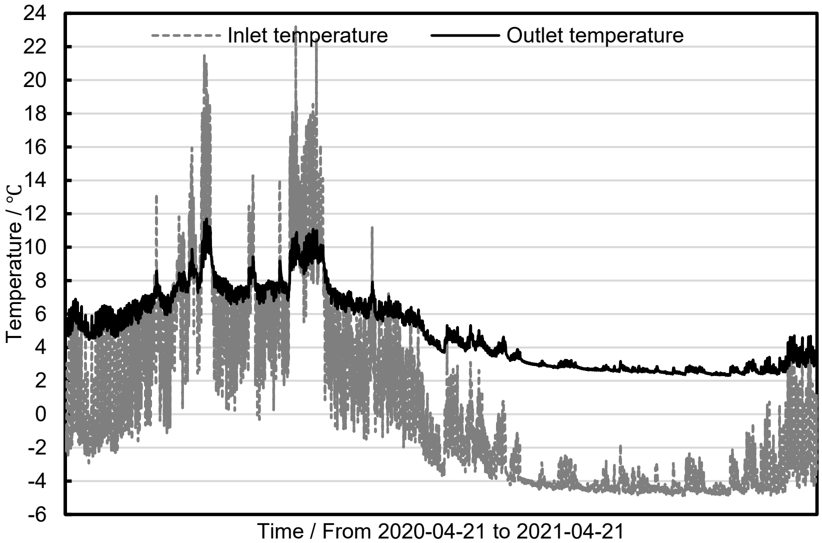

Figure 28: Simulated fluid temperatures of heating system 1 utilising free cooling.

The long-term borehole depletion of system 1 was analysed by simulating the fluid temperatures over five years as seen in Figure 29. The inlet and outlet fluid temperatures are decreasing over time. The largest decrease was observed in the inlet temperature during the heating season, reaching -5,5 °C at the end of the fifth year. Additionally, it was observed that the largest decrease of the inlet temperature was seen between simulation year one and two. After the second year, the lowest inlet fluid temperature stabilized.

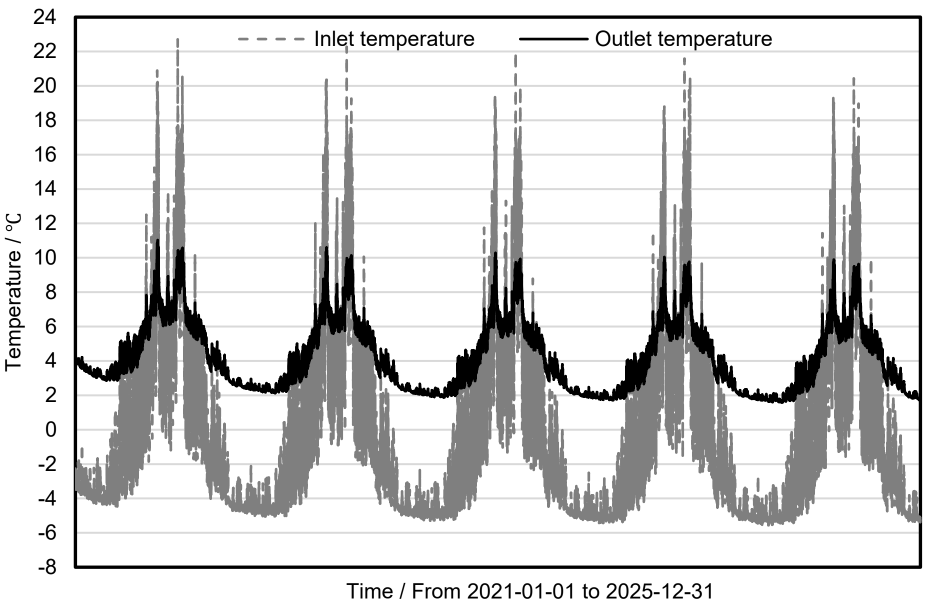

Figure 29: Simulated fluid temperatures of heating system 1 over 5 years.

4.3.3 Borehole fluid temperatures in system 2

To analyse the impact of the additional GSHP of system 2 utilising the existing borehole configuration, the inlet and outlet fluid temperatures were simulated for one year and are shown in Figure 30. As a result of the increased heat extraction rate caused by the extended Sizing Factor (SF), the lowest inlet fluid temperature further decreased to -7 ℃.

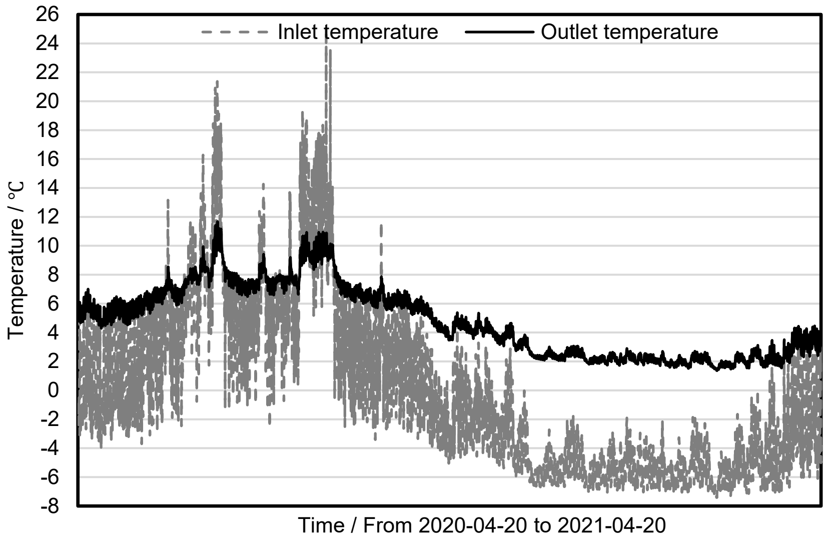

Figure 30: Simulated fluid temperatures of heating system 2 utilising free cooling.

The long-term borehole depletion of system 2 was analysed by simulating the fluid temperatures over five years as seen in Figure 31. The fluid temperature of system 2 are following the same trend as seen in system 1. The lowest inlet temperature is reached at the end of the fifth year at -8,0 ℃.

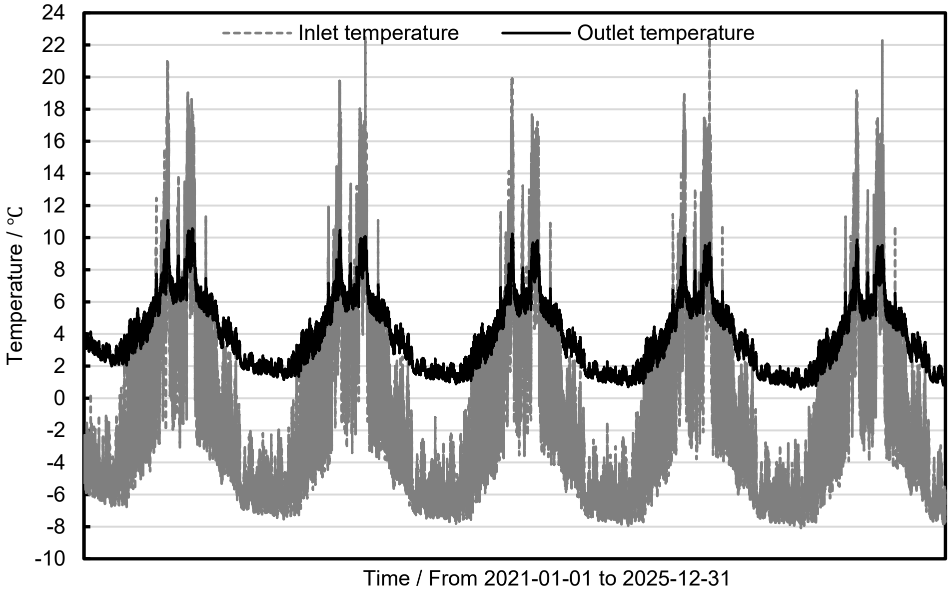

Figure 31: Simulated fluid temperatures of heating system 2 over 5 years.

4.4 Life cycle costing

4.4.1 System 1 with individual ground source heat pump

To analyse the initial and operating costs of system 1, the ratio of the initial and operating costs were calculated and presented in Figure 32. During the heat pump's lifetime of 20 years, the largest costs were associated with the compressor operation costs and the heat pump installation of 76,8 % and 7,1 % respectively.

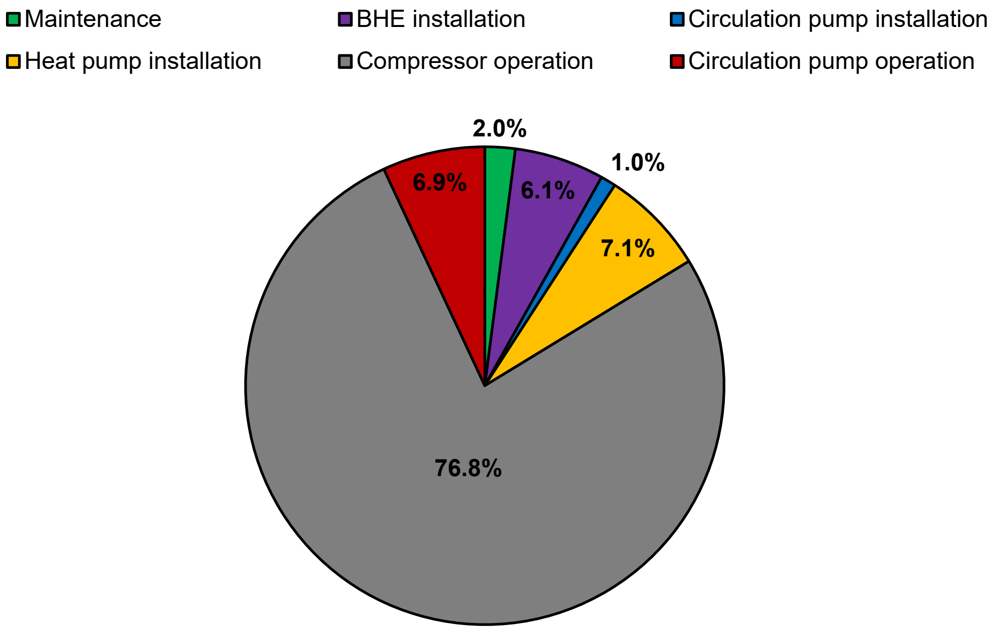

Figure 32: Ratio of the life-cycle costs of system 1 with individual GSHP.

The life cycle costs of system 1 and system 2 were evaluated to determine if the savings generated from replacing alternative heating solutions such as district heating and boilers operating on different fuels are sufficient to cover GSHPs initial costs. The evaluation of system 1 with the individual heat pump is shown in Figure 33. The highest potential savings were generated by replacing the boilers operating on oil and electricity. The pay-back time of these heating sources were below three years as indicated by the crossing of the zero-line. Replacing boilers operating on oil and electricity generated potential savings exceeding five million SEK during the considered time span. However, replacing district heating and boilers operating on less expensive sources such as pellets increased the pay-back time to five and seven years respectively. After 20 years, the heat pump required replacement, causing a sharp drop in all curves.

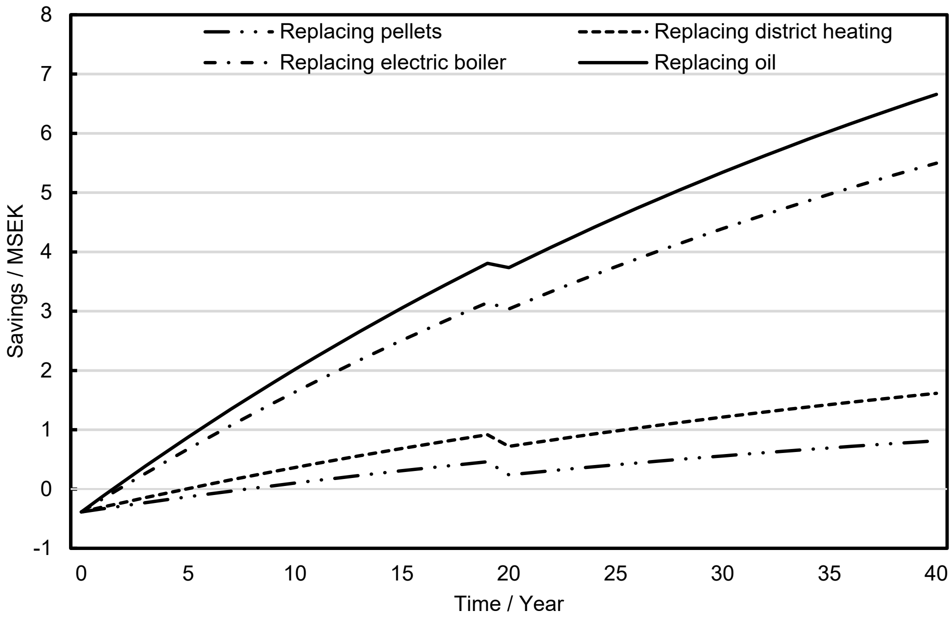

Figure 33: Savings by replacing various heating sources with GSHP system 1 with individual GSHP.

Since the GSHP operating costs are highly dependent on the electricity price, the impact of increasing electricity price was studied and the resulted savings of system 1 are presented in Figure 34. The electricity price increase generated higher savings after replacing electric boilers. However, the savings from replacing other heating sources drastically decreased and no savings were obtained by replacing pellets with the GSHP system 1 over the studied life-cycle.

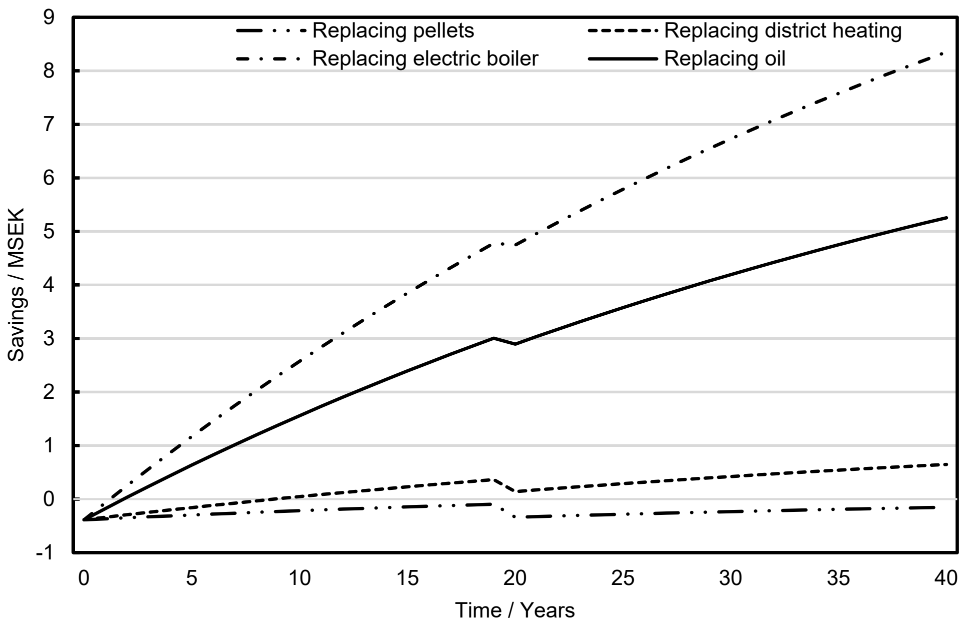

Figure 34: Impact of increasing electricity price on the savings of system 1 with individual GSHP.

4.4.2 System 2 with additional ground source heat pump

To compare the initial and operating costs of system 1 and system 2, the ratio of the initial and operating costs was also calculated for system 2 according to Figure 35. Since system 2 is considering an additional heat pump, the installation costs of system 2 increased by 37 % in comparison to system 1. The compressor operation and the heat pump installation costs remained the highest.

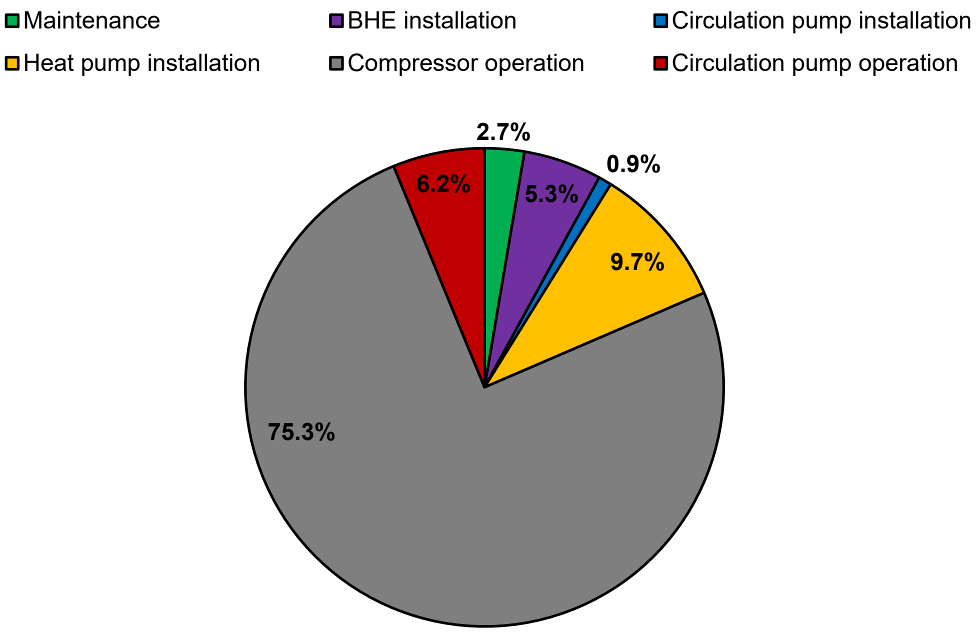

Figure 35: Ratio of the life-cycle costs of system 2 with additional GSHP.

To compare the potential life-time savings of both systems, the potential life-time savings compared to alternative heating sources were also calculated for system 2 and the results are presented in Figure 36. The additional heating savings generated by system 2 were not sufficient to cover the installation costs of the additional heat pump. The pay-back time and the total time span savings of replacing boilers operating on oil and electricity remained approximately the same. However, replacing the less expensive heating sources such as pellets and district heating increased the pay-back time by several years and decreased the total savings of the considered time span.

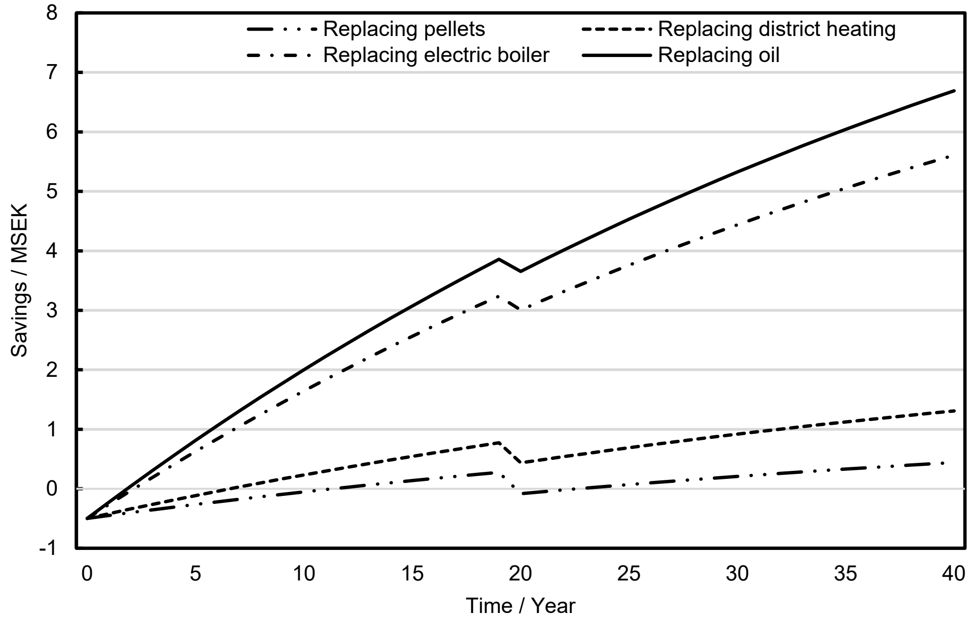

Figure 36: Savings by replacing various heating sources with GSHP system 2 with additional GSHP.

The impact of changing the electricity price to 2,0 SEK/kWh for system 2 is shown in Figure 37. The results are following the same trend as seen in system 1, but the negative impact of the increased electricity price is larger in system 2. Replacing district heating resulted in minimal savings, while the replacement of pellets caused losses of approximately 700 000 SEK over the considered 40 year time span.

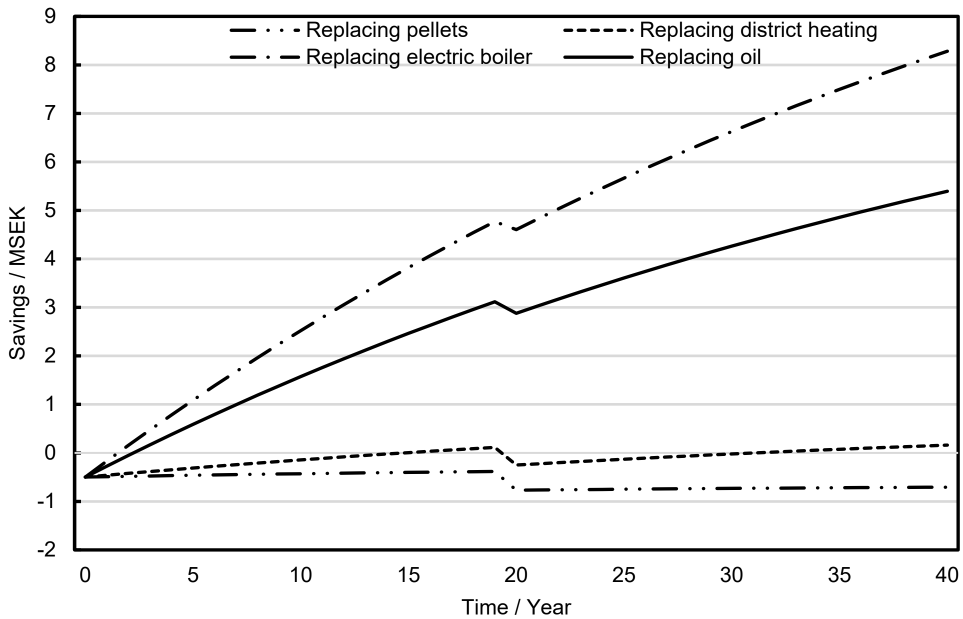

Figure 37: Impact of increasing electricity price on the savings of system 1 with additional GSHP.

4.4.3 Cooling life-cycle costs

The life-cycle costs of GSHP related cooling solutions in comparison to air cooled chiller were evaluated. The GSHP cooling solutions consist of free cooling only requiring boreholes and a circulation pump and active cooling requiring boreholes and a reversible GSHP. Because the outcome of the LCC is dependent on if the GSHP installation is intended as a combined heating or cooling solution or exclusively cooling, both scenarios were analysed for the studied sections. The NPV considering the GSHP cooling operation exclusively is presented in Figure 38. The NPV of free cooling only demanding the borehole installation and circulation pump is significantly lower than the active cooling solution demanding a reversible GSHP. The NPV of the air cooled chiller was the most economical solution for lower cooling demands as the initial costs are fairly low in comparison to GSHP cooling solutions. However, the lower Energy Efficiency Ratio (EER) of the air cooled chiller and higher maintenance costs significantly increased the NPV for higher cooling demands.

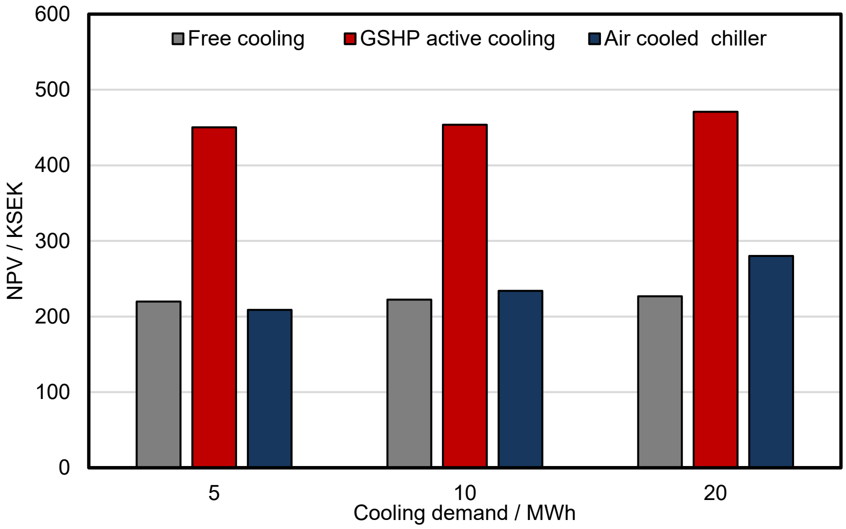

Figure 38: Life-cycle costs considering the ground source heat pump installation for cooling only operations.

If the GSHP is installed for heating purposes and the free and active cooling possibilities of the boreholes and a reversible GSHP is considered as an addition to the heating operations, the NPV of the GSHP cooling solutions significantly increases as seen in Figure 39. In this case, only the operational costs for the GSHP cooling solutions are relevant while the installation costs of the air cooled chiller are considered as this is exclusively installed for cooling purposes.

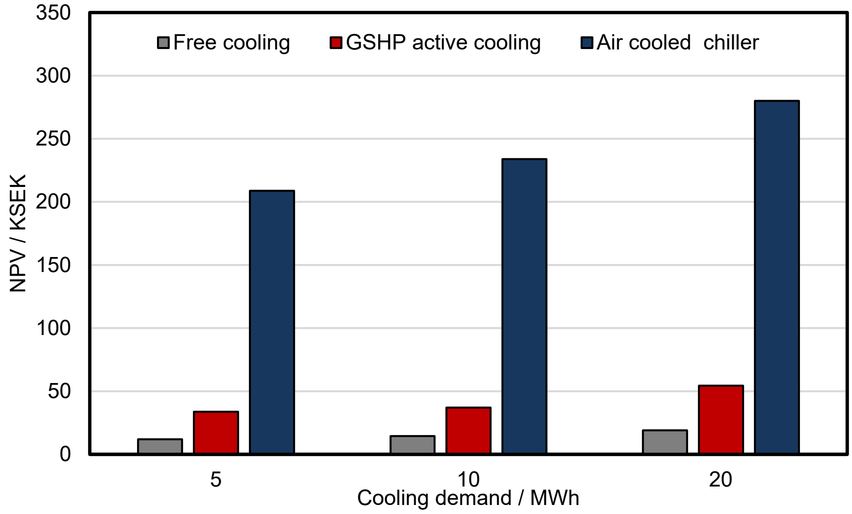

Figure 39: Life-cycle costs considering the ground source heat pump installation for both heating and cooling operations

5 Discussion

Integrating GSHPs in an elderly home increases thermal comfort and provides efficient heating and cooling was investigated using energy simulations. The results of the implementation indicate that the combined heating and cooling solution could highly contribute to the fulfilment of Swedish energy goals by significantly reducing the building's energy required for heating and simultaneously improving thermal comfort. The feasibility of the GSHP system is dependent on the type of replaced reference heating source and electricity price. A discussion is provided below to address the most important findings in the study.

5.1 Derivation of the buildings mean thermal transmittance

The average thermal transmittance of the studied sections was derived from the SEU of the entire elderly home. Therefore, distinctions between the actual and estimated mean U-value of the specific sections are likely. Instead of calculating the mean U-value based on the average transmission losses, the U-value of each building part could have been estimated. The discrepancies caused by estimating the U-value for each construction parts are arguably higher than the discrepancy caused by the approach introduced in this study.

5.2 Discrepancy between simplified and simulated results

The comparison of the heat losses and gains from the simplified heating degree hours calculations and simulations based on hourly detailed heat balance introduced distinctions of up to 27,1 % as shown in Figure 19. The differences in the ventilation losses are related to discrepancies between the assumed heat recovery efficiency and the actual simulated efficiency based on the supply and extract air flows. The discrepancy between the transmission losses and ventilation losses cancelled each other and reduced the difference in space heating demand to only 0,6 %.

The presented measured and simulated inlet and outlet temperatures of system 1 in Figure 27 and Figure 28 were inconsistent with each other. These inlet and outlet temperatures are dependent on many different factors such as the extracted energy, the on-site ground conditions and the mass flow rate in the boreholes making it difficult to determine the specific cause. The differences between the actual and assumed mass flow rates and ground conditions are arguably the factors having the largest impact on the observed inconsistency. But other factors such as on-site measurement uncertainties and that the logging of the measured data might not represent the reality could cause the observed inconsistency. In addition, the algorithms used in the simulation program to predict the borehole fluid temperatures play a substantial role. Due to the observed inconsistencies, conclusions drawn from the simulated fluid temperatures might be inaccurate.

5.3 Impact of the analysed cooling measures

The external blinds impact on the OHs for rooms oriented towards the north and east were negligible. The reason for that is the path of the sun and the shading devices high solar radiation thresholds set to increase the daylight access and mental health of the care recipients. The installation of shading devices for these rooms is still recommended to avoid glare problems. More economic solutions such as internal blinds are preferable. The impact of the AHU cooling was insignificant as it was unable to provide the targeted indoor climate for most rooms. The integration of additional room cooling units was required to reach the desired indoor climate. Passive beams were preferred over active beams to reduce drafts in the occupancy zones which is potentially caused by the high air velocities and low supply temperatures in active beams. However, the placement of the passive beams should be regarded with care to satisfy both architectural and technical requirements.

5.4 Heating coverage by ground source heat pumps

The integration of two GSHP systems did not decrease the primary energy use of the studied building sections as the weighting factor for electricity is high. Still, the GSHP systems reduced the energy use of the building sections and can help with the fulfilment of the Swedish energy efficiency goals. The system with the individual GSHP managed to cover 90 % of the sections heating demand as seen in Figure 24. Since system 1 is already covering the majority of the studied section's heating demand, the additional heat coverage ratio gained by system 2 was insignificant. Although the additional heat coverage ratio was minimal, the additional savings from replacing boilers operating on oil and electricity were sufficient to cover the installation costs of the additional GSHP. However, the additional savings from replacing less expensive heating sources such as district heating and boilers operating on pellets were insufficient to cover the installation costs of the additional GSHP. Therefore, the installation of the additional GSHP for the studied sections is not recommended considering the life-cycle costs. Since the auxiliary heating of the studied sections is supplied by pellets which are considered renewable, the installation of the additional GSHP is also not motivated from an environmental perspective. For these reasons, the installation of the additional GSHP for the studied section is not recommended. Additionally, the decreased fluid temperatures caused by the additional GSHP are increasing the temperature difference between the evaporator and condenser which in turn decreases the overall heating efficiency of the system 2.

Besides contributing to energy efficiency goals, GSHPs could also contribute to the fulfilment of Swedish and European net-zero carbon emission goals. Since GSHPs are extracting energy from a renewable source without the demand of combustion processes, zero-emissions occur during the operation. It is important to also consider the energy efficiency and environmental impact of the auxiliary heating plant. Preferably, the auxiliary heating plant shall consist of environmentally friendly sources such as pellets or those that come from modern district heating plants. The integration of GSHPs is associated with increased electricity use for compressor and circulation pump operation. This highlights the importance of utilising electricity generated by renewable sources. In Sweden, large shares of the electricity are generated with renewable sources such as hydropower, compromising the increased electricity use (Royal Swedish Academy of Engineering Sciences, 2016). To ensure that the provided electricity is generated from renewable sources, electricity labelled as sustainable should be chosen.

5.5 Cooling coverage by ground source heat pumps and other solutions

The simulated outlet temperatures of both systems were sufficient to provide the design inlet temperature of the passive beams and free cooling utilisations was possible. Although, the outlet temperatures of the boreholes were insufficient to provide the design inlet temperature of the air handling unit cooling coil, causing the demand for an additional chiller. The possibility of utilising free cooling from the boreholes supplying the passive beams reduced the cooling demand of the required chiller by 38 % and the total capacity by 29 %. There are several alternative solutions for covering the remaining cooling loads of the cooling coil. Two solutions are introduced and further discussed. The first solution is to install an air-cooled chiller with a cooling capacity to exclusively cover the cooling demand of the AHU. The second solution is to install a chiller connected to the boreholes providing active cooling at a higher Energy Efficiency Ratio (EER). The chiller utilising the boreholes has the advantage of providing cooling at higher EER due to the lower temperatures supplied to the chiller in comparison with air-cooled chillers. The disadvantage of the chiller providing active cooling utilising the boreholes is that heat is injected into the ground, which will increase the fluid temperatures and negatively impact the free cooling potential. Therefore, the capacity of the chiller utilising the boreholes for active cooling has to cover the peak loads of the AHU cooling coil and the passive beams to ensure that the required cooling is provided over the entire year. This would increase the initial costs of the chiller. The configuration of the chiller utilising the boreholes for active cooling requires further analysis to ensure if the required design inlet temperatures still would be met. For that reason, the first option of installing an air cooled chiller exclusively covering the AHU cooling demand is recommended.

5.6 Integration of ground source heat pumps into conventional MFBs

This study investigated the integration of GSHPs in elderly homes which are considered as Multi-Family Buildings (MFB). Currently, cooling is provided infrequently in conventional MFBs as the life-cycle costs of the required cooling system are high. However, if global warming continues and ambient temperatures increase, new building regulations could be developed. These regulations could enhance the application of combined heating and cooling solutions. The combination of residential areas and working spaces in elderly homes contribute to unique occupancy schedules and internal loads in comparison to conventional MFBs. Because the share of elderly homes is limited within MFBs, the discussion of whether or not the results of this study are applicable in conventional MFBs was initiated. The possibility of decreasing the energy use in conventional MFBs by the integration of GSHPs is undoubted. However, the free cooling utilisation is expected to be different. The internal gains of conventional MFBs are arguably higher with increased equipment loads and activity levels. Since the average person requires less consideration of drafts in comparison to elderly people, active beams can be used instead of passive beams. The higher inlet temperatures of active beams that increase free cooling utilisation could compensate for the increased internal gains in conventional MFBs.

5.7 Current market share and future transitions

Since many MFBs are located in urban areas and are connected to district heating networks, a brief discussion about district heating is provided in this section. State-of-the-art district heating plants are often Combined Heat and Power plants (CHP), are operating on renewable energy sources and are providing low-temperature heating with low distribution losses. These plants are efficient at providing heat utilising renewable sources. However, most of the district heating plant networks in Sweden are 3rd generation networks. Many of these 3rd generation district heating networks generate heat and electricity by burning waste material and biofuels with relatively high distribution losses (Werner, 2017). While heat and electricity generated from burning waste materials are practical to avoid wasted energy, the utilisation of biomass in CHPs is often debated. The debate if biomass should be considered as renewable or not is frequently initiated. The use of biomass has several advantages and was important during the transition from fossil fuels to greener energy sources. But the use of biomass is also associated with several disadvantages such as deforestation, the loss of biodiversity and the controversial discussion of carbon neutrality (Montanaro, 2020). However, the potential advantages of utilising biomass instead of fossil fuels are larger than the disadvantages. But it is important to ensure that the harvested biomass is continuously restored to decrease the advantages. The possibility of installing GSHPs in rural areas offering the required area for installing boreholes is high. Since most MFBs are located in urban areas with limited area for borehole installations, the demand for alternative solutions expands. One solution to this is the combination of large-scale GSHPs and district heating. This combination is seen in the municipality of Lund in Sweden. This increases the energy efficiency of the district heating plants although the large distribution losses of district heating networks persist. To reduce these heat losses and further increase the efficiency of GSHP in combination with district heating, lower distribution temperatures are favourable. These Low Temperature District Heating systems (LTDH) increase the efficiency of large-scale GSHP systems, while reducing the distribution losses and could serve as the energy system of the future (Schmidt et al., 2017).

5.8 Future work

This study paves the way towards further possible future studies. The building sections included in the analysis in this study were limited to sections D, E and F of the elderly home. The remaining sections could also be analysed in terms of thermal comfort and energy efficiency. Moreover, the scope of this study did not cover the life-cycle assessment. This research topic becomes increasingly important and future studies can analyse the integration of GSHPs into MFBs under this topic.

6 Conclusions and recommendations

The goal of this study was to analyse the implementation of ground source heat pumps in a Swedish elderly home. The study addressed Sweden’s energy efficiency goals, the indoor climate conditions and life-cycle costing of different heating and cooling systems. By using the simplified approach to calculate the mean U-value of the building sections and using the energy simulation tool IDA-ICE, the energy performance and the thermal comfort of the studied sections were analysed. Both passive and active measures were introduced to enhance thermal comfort. Two different ground source heat pump systems were compared to enhance the section's energy performance. The savings of replacing several alternative heating sources with ground source heat pumps and the life-cycle costing of different cooling solutions were calculated. The main conclusions of this study and the recommendations for the case study building are presented below.

- The integration of shading devices was only effective at reducing overheating hours for rooms oriented towards the south and west. For rooms oriented towards the north and east, less expensive internal shading devices are recommended. Further cooling solutions consisting of airborne cooling and passive beams were required to reach the desired indoor climate. The impact of airborne cooling was low for rooms oriented towards the south.

- The free cooling capacities of the boreholes were sufficient to provide the inlet temperature of the passive beams over the entire cooling season. Because the free cooling is insufficient to provide the required inlet temperature of the air handling unit cooling coil an additional chiller is required. Since the free cooling covered the cooling demand of the passive beams, the cooling demand of the required chiller was reduced by 26 % and the capacity by 29 %. Since cooling loads are relatively low, the installation of an air cooled chiller is recommended.

- The integration of ground source heat pumps highly contributed to the Swedish energy efficiency goals as the energy use of the studied building sections could be significantly reduced. The higher sizing factor of system 2 with the additional ground source heat pump is contributing to increased heating coverage ratios but the additional savings were insufficient to cover the additional installation costs. The installation of system 2 is not recommended for the studied sections considering the low costs and environmental impact of the auxiliary heating system. Despite the high weighting factor for electricity, ground source heat pumps were still able to reduce the primary energy use due to the high season coefficient of performance.

- The outcome of the comparison of the life-cycle costs between ground source heat pump related cooling solutions and air cooled chillers is dependent on whether the ground source heat pump installation is intended for mainly heating or cooling exclusively. The ground source heat pump installations intended for mainly heating are outperforming the air cooled chiller as operational costs are lower.

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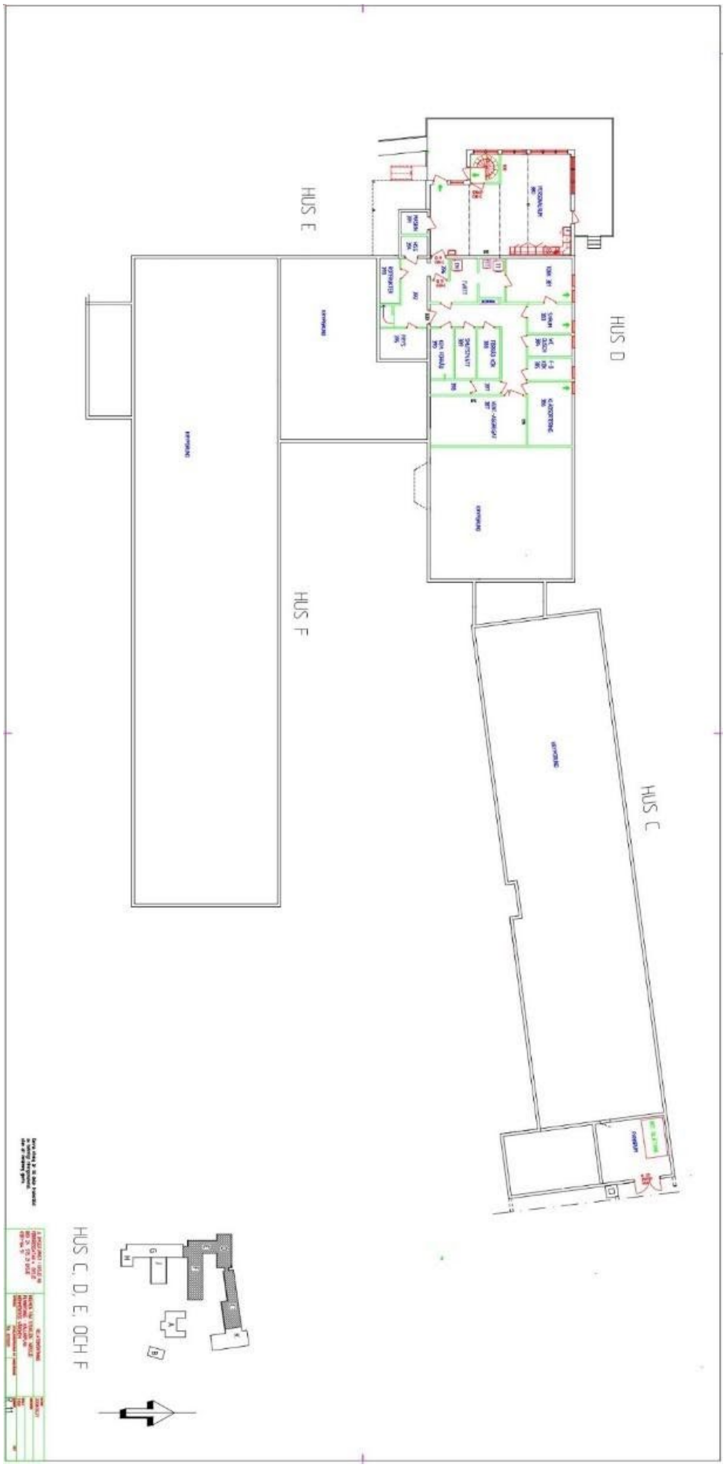
Appendices

Appendix A – Plan drawings of the studied sections




HUS E
HUS D
HUS F
HUS C
HUS C, D, E, OCH F

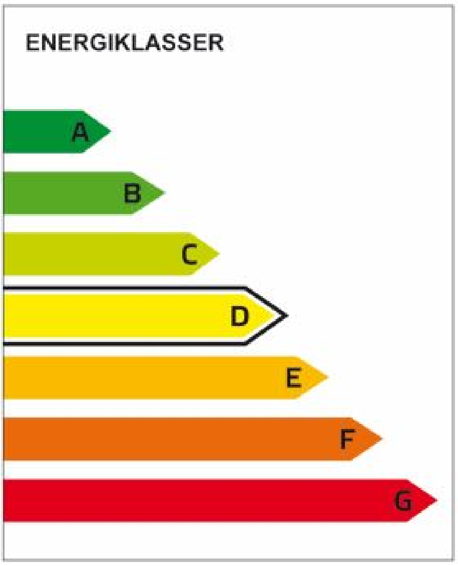
Appendix B – Energy declaration of the studied sections

sammanfattning av

ENERGIDEKLARATION

Nömmeberg Vårdhem 1, 571 73 Stensjön
Nässjö kommun

Nybyggnadsår: 1980
Energideklarations-ID: 1028933



DENNA BYGGNADS ENERGIKLASS

Energiprestanda, primärenergital:
170 kWh/m² och år

Krav vid uppförande av ny byggnad, primärenergital:
Energiklass C, 126 kWh/m² och år

Specifik energianvändning (tidigare energiprestanda):
164 kWh/m² och år

Uppvärmningssystem:
Flis/pellets/briketter och eldningsolja

Radonmätning:
Inte utförd

Ventilationskontroll (OVK):
Utförd

Åtgärdsförslag:
Har lämnats

Energideklarationen är utförd av:
Roger Johansson, RJ Energiteknik AB, 2019-12-19

Energideklarationen är giltig till:
2029-12-19

Energideklarationen i sin helhet finns hos byggnadens ägare.

För mer information:
www.boverket.se

Sammanfattningen är upprättad enligt Boverkets föreskrifter och allmänna råd (2007:4) om energideklaration för byggnader.

Energianvändning

Mätperiod
Vilken 12-månadersperiod avser energiuppgifterna?
(ange första månaden i formatet ÅÅMM)

1801 - 1812

Beräknad energianvändning
Beräknad energianvändning vid normalt brukande och ett normalår anges för byggnader där det inte går att få fram uppgifter om den uppmätta energianvändningen.

☐

Hur mycket energi har använts för värme och varmvatten angiven mätperiod?
Värdena ska vara korrigerade för normalt bruk. (BFS 2016:12)
Angivna värden ska inte vara normalårskorrigerade.

	Energi för uppvärmning	Energi för tappvarmvatten	
Fjärrvärme (1)			kWh
Eldningsolja (2)	121652	38798	kWh
Naturgas, stadsgas (3)			kWh
Ved (4)			kWh
Flis/pellets/briketter (5)	223990	71436	kWh
Övrigt biobränsle (6)			kWh
El (vattenburen) (7)			kWh
El (direktverkande) (8)			kWh
El (luftburen) (9)			kWh
Markvärmepump (el) (10)	45816		kWh
Värmepump-frånluft (el) (11)			kWh
Värmepump-luft/luft (el) (12)			kWh
Värmepump-luft/vatten (el) (13)			kWh
Tappvarmvatten (el) (14)		14033	kWh

Omvandlingsfaktorer för bränslen i tabellen nedan gäller om inte annat uppmätts:

Eldningsolja	10 000 kWh/m³
Naturgas	11 000 kWh/1 000 m³ (effektivt värmevärde)
Stadsgas	5 880 kWh/1 000 m³
Pellets	4 500-5 000 kWh/ton, beroende av träslag och fukthalt

Källa: Energimyndigheten
För övriga biobränsle varierar värmevärdet beroende av sammansättning och fukthalt. Det är expertens ansvar att omräkna bränslets vikt eller volym till energi på ett korrekt sätt.

Övrig el som ingår i energiprestanda

Fjärrkyla (15)		kWh
El för komfortkyla (16)	2000	kWh
Fastighetsel[1] (17)	39821	kWh

Övrig energi (ingår inte i energiprestanda)

Hushållsel[2] (18)		kWh
Verksamhetsel[3] (19)		kWh

Energi för uppvärmning, tappvarmvatten, komfortkyla och fastighetsel

Summa 1 - 17[4] 557546 kWh

Finns solvärme? (●) Ja ( ) Nej — Ange solfångararea 84 m² — Beräknad energiproduktion 16566 kWh/år

Ort (Energi-Index)

Nässjö

Finns solcellsystem? ( ) Ja (●) Nej — Ange solcellsarea m² — Beräknad elproduktion kWh/år

Byggnadens energianvändning[5]
(Normalårskorrigerat värde (Energi-index))

596466 kWh/år

Byggnadens primärenergianvändning[6]

618328 kWh/år

Energiprestanda (primärenergital)	Referensvärde 1 (enligt nybyggnadskrav)	Referensvärde 2 (liknande byggnader)	Referensvärde 3 (nybyggnadskrav för denna byggnad)
170 kWh/m² ,år	126 kWh/m² ,år	214 kWh/m² ,år	kWh/m² ,år

[1] Den el som ingår i fastighetsenergin.
[2] Den el som ingår i hushållsenergin.
[3] Den el som ingår i verksamhetsenergin.
[4] Den energimängd som levereras till byggnaden vid normalt brukande.
[5] Enligt definition i Boverkets byggregler (2011:6) - föreskrifter och allmänna råd.
[6] Underlag för energiprestanda.

Appendix C – Ground source heat pump of system 1

Appendix D – Obligatory ventilation control

Luftflöde
Driftstider/Märkeffekt

	Referensnummer: 22376	Systemnummer: LA2		L6
L1 Fastighetsbeteckning: NÄSSJÖ-HOLMEN 1:61	Byggnadsnamn:	Byggnadsnr:		Sidnr.:
Aggregatbenämning: LA2	Ritning:	Flödesenhet: m^3/h ☐ l/s ☑		Datum: 2018-12-19

Driftstider: 24h/dag	Märkeffekter: Se E2

L2

	Rum. nr.	Benämning	Projekterad Tilluft	Uppmätt Tilluft	% av proj Tilluft	Mät-metod	Projekterad Frånluft	Uppmätt Frånluft	% avproj Frånluft	Mät-metod	Anm.
1		**Hus F Markplan**									
2	1	Rum 1	10				10	13			Golvinblåsning
3	1	Rum 2	10				10	7			Golvinblåsning
4	1	Rum 3	10				10	11			Golvinblåsning
5	1	Rum 4	10				10	8			Golvinblåsning
6	1	Rum 5	10				10	13			Golvinblåsning
7	1	Rum 6	10				10	ej åtk			Golvinblåsning
8	1	Rum 7	10				10	11			Golvinblåsning
9	1	Rum 8	10				10	9			Golvinblåsning
10	1	Rum 9	10				10	9			Golvinblåsning
11	1	Rum 10	10				10	8			Golvinblåsning
12	1	Rum 11	10				10	8			Golvinblåsning
13	1	Rum 12	10				10	8			Golvinblåsning
14	1	Rum 13	10				10	12			Golvinblåsning
15	1	Rum 14	10				10	9			Golvinblåsning
16											
17											
18											
19											
20											

Anm.
1 Rum uppräknade från Hus E moturs. Projekt. värden skall vara 13/15 (som i övriga rum)

Mätteknitker
RJ/SP

Namnteckning

Mätmetod: A=Kanal, B=Frånluft, C=Tilluft

1 = A1, Punktvis hast.mätn.m prandtlrör
2 = A2, Fasta flödesmätdon
3 = A3, Punktvis hastmätn m varmtrådsanemometer
4 = A4, Spårgasmätning
5 = B1, Punktvis mätn m varmtrådsanemo rekt galler
6 = B21, Tryckfallsmätning med sond
7 = B22, Tryckfallsmätning med fast mätuttag
8 = B3, Mätn. m stosförsedd anemometer
9 = C1, Mätning av referenstryck
10 = C21, Mätning m stos, direkt metod
11 = C22, Mätning m stos, indirekt metod
12 = Övrigt

FunkiS v 1.1

Luftflöde Driftstider/Märkeffekt	Referensnummer 22376	Systemnummer LA2	L5
L1 Fastighetsbeteckning NÄSSJÖ-HOLMEN 1:61	Byggnadsnamn	Byggnadsnr	Sidnr.
Aggregatbenämning LA2	Ritning	Flödesenhet m³/h ☐ l/s ☑	Datum 2018-12-19

Driftstider	Märkeffekter
24h/Dag	Se E2

L2

	Rum. nr.	Benämning	Projekterad Tilluft	Uppmätt Tilluft	% av proj Tilluft	Mät-metod	Projekterad Frånluft	Uppmätt Frånluft	% avproj Frånluft	Mät-metod	Anm.
1		**Hus E, Markplan**									
2		Matsal	300	350							
3		Frd		20							
4		Kontor		20				50			
5											
6											
7		**Hus F Markplan**									
8		Städ (öd saknas)					20	3			dammtäppt
9		Kontor					?	17			
10		Wc					25	30			öd saknas
11		Godiskiosk					20	17			öd saknas
12	1	Sköljrum	35	30			70	66			öd saknas
13		Fotvård	50	-			200	80			ombyggt
14		TV-rum	100								golvinblåsning
15		Samtalsrum	?	10							öd saknas
16											
17											
18											
19											
20											

Anm.	
1	luktar avlopp troligen pga undertyck. Montera öd (överluftsdon)

Mätteknker

RJ/SP

Namnteckning

Mätmetod: A=Kanal, B=Frånluft, C=Tilluft

1 = A1, Punktvis hast.mätn.m prandtlrör
2 = A2, Fasta flödesmätdon
3 = A3, Punktvis hastmätn m varmtrådsanemometer
4 = A4, Spårgasmätning
5 = B1, Punktvis mätn m varmtrådsanemo rekt galler
6 = B21, Tryckfallsmätning med sond
7 = B22, Tryckfallsmätning med fast mätuttag
8 = B3, Mätn. m stosförsedd anemometer
9 = C1, Mätning av referenstryck
10 = C21, Mätning m stos, direkt metod
11 = C22, Mätning m stos, indirekt metod
12 = Övrigt

FunkiS v 1.1

Luftflöde
Driftstider/Märkeffekt

Referensnummer: 22376	Systemnummer: LA2	L4	
L1 Fastighetsbeteckning: NÄSSJÖ-HOLMEN 1:61	Byggnadsnamn:	Byggnadsnr:	Sidnr.:
Aggregatbenämning: LA2	Ritning:	Flödesenhet: m³/h ☐ l/s ☑	Datum: 2018-12-19

Driftstider	Märkeffekter
24h/Dag	Se E2

L2

		Benämning	Projekterad Tilluft	Uppmätt Tilluft	% av proj Tilluft	Mät-metod	Projekterad Frånluft	Uppmätt Frånluft	% avproj Frånluft	Mät-metod	Anm.
1		**Hus D källare**									
2		Personalrum	145	177			150	170			
3		Tvätt					120	80+67			+50
4	1	Tork FF					?	200			
5		Smutstvätt						20			
6		Mangelrum/syrum						25			
7		Omkl						?			låst
8		FRD						?			låst
9		Kem frd						10			
10		Frd kök						11			
11		Trapp		14							
12		**Markplan**									
13		Wc vid kök					25	30			
14		Korridor vid kök	40+20	100							
15		Minikontor köksperс		saknas							
16		Frd kök					25	20			
17		Kök	200	150							
18		FF12-FF13					400	180			
19		FF Ugn					?	150			
20		Städ i korr					10	11			

Anm.

1 FF startar när inställt börvärde överskrids på termostat vid torktummlare.

Mättekniker

RJ/SP

Namnteckning

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10 = C21, Mätning m stos, direkt metod
11 = C22, Mätning m stos, indirekt metod
12 = Övrigt

FunkiS v 1.1

Luftflöde
Driftstider/Märkeffekt

	Referensnummer 22376	Systemnummer LA2		L3
L1	Fastighetsbeteckning NÄSSJÖ-HOLMEN 1:61	Byggnadsnamn	Byggnadsnr	Sidnr.
	Aggregatbenämning LA2	Ritning	Flödesenhet m³/h ☐ l/s ☑	Datum 2018-12-19

Driftstider	Märkeffekter
24h/dag	Se E2

L2		Benämning	Projekterad Tilluft	Uppmätt Tilluft	% av proj Tilluft	Mät-metod	Projekterad Frånluft	Uppmätt Frånluft	% avproj Frånluft	Mät-metod	Anm.
1		Rum 301	13	13			15	8			
2		Rum 302	13	8			15	17			
3		Rum 303	13	12			15	6			
4		Rum 304	13	13			15	16			
5		Rum 305	13	8			15	0			fd dammtäppt
6		Rum 306	13	11			15	10			
7		Rum 307	13	*			15	11			*golvinblåsning
8		Rum 308	13	*			15	18			*golvinblåsning
9		Rum 309	13	*			15	22			*golvinblåsning
10		Uppehållsrum 329	80	68							
11		Pentry 328					85	70			
12		Korridor 321									
13		Exp		10				46			
14	1	Beh 326	40	35			50	50			Drag
15	1	Exp 325	40	40			50	50			Drag
16		Wc 324					10	10			
17											
18											
19											
20											

Anm.

1 Rekommenderar forc/grudflöde i rum 326 och 325, personal klagar på drag (oftast 2 p/r)

Mätttekniker

RJ/SP

Mätmetod: A=Kanal, B=Frånluft, C=Tilluft

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11 = C22, Mätning m stos, indirekt metod
12 = Övrigt

Namnteckning

FunkiS v 1.1

Appendix E – Additional cooling measure results

Section	Room nr.	Room type	Orientation	Max. temp.	OH
D					
	301	Care recipient	North/West	33,3 °C	823
	302	Care recipient	North	29,9 °C	228
	303	Care recipient	North	29,8 °C	182
	304	Care recipient	North	29,7 °C	176
	305	Care recipient	North	29,7 °C	171
	306	Care recipient	North	29,6 °C	151
	307	Care recipient	North	29,7 °C	120
	308	Care recipient	North	29,6 °C	113
	309	Care recipient	North	29,5 °C	99
	322	Expedition	South	34,1 °C	1311
	325	Expedition	South	34,0 °C	1046
	329	Living room	Central	28,1 °C	66
	380	Staff room	West	33,1 °C	450
	399	Office	West	33,0 °C	585
E					
	343	Dining room	West	32,5 °C	209
	344	Kitchen	West	31,4 °C	183
F					
	201	Care recipient	South	33,1 °C	1024
	202	Care recipient	South	33,1 °C	1058
	203	Care recipient	South	33,1 °C	1060
	204	Care recipient	South	33,1 °C	1060
	205	Care recipient	South	33,1 °C	1060
	206	Care recipient	South	33,1 °C	1061
	207	Care recipient	South	33,1 °C	1060
	208	Care recipient	South	33,1 °C	1060
	209	Care recipient	South	33,1 °C	1058
	210	Care recipient	South	33,0 °C	967
	211	Care recipient	North	29,4 °C	95
	212	Care recipient	North	29,6 °C	109
	213	Care recipient	North	29,6 °C	109
	214	Care recipient	North	29,5 °C	104
	215	Living room	North	30,0 °C	115

Measure 1 – Addition of external shading

Section	Room nr.	Room type	Orientation	Max. temp.		OH	Improvement	
D								
	301	Care recipient	North/West	33,2	℃	769	6,6	%
	302	Care recipient	North	29,9	℃	224	17,5	%
	303	Care recipient	North	29,8	℃	182	-	%
	304	Care recipient	North	29,7	℃	176	-	%
	305	Care recipient	North	29,7	℃	170	-	%
	306	Care recipient	North	29,6	℃	151	-	%
	307	Care recipient	North	29,7	℃	120	-	%
	308	Care recipient	North	29,6	℃	113	-	%
	309	Care recipient	North	29,5	℃	99	-	%
	322	Expedition	South	33,6	℃	1145	12,7	%
	325	Expedition	South	33,7	℃	315	12,5	%
	329	Living room	Central	28,1	℃	65	-	%
	380	Staff room	West	33,1	℃	423	6,0	%
	399	Office	West	32,9	℃	551	5,8	%
E								
	343	Dining room	West	32,4	℃	203	2,8	%
	344	Kitchen	West	31,4	℃	179	-	%
F								
	201	Care recipient	South	32,9	℃	844	17,6	%
	202	Care recipient	South	32,9	℃	861	18,6	%
	203	Care recipient	South	32,9	℃	863	18,6	%
	204	Care recipient	South	32,9	℃	863	18,6	%
	205	Care recipient	South	32,9	℃	863	18,6	%
	206	Care recipient	South	32,9	℃	863	18,7	%
	207	Care recipient	South	32,9	℃	863	18,6	%
	208	Care recipient	South	32,9	℃	863	18,6	%
	209	Care recipient	South	32,9	℃	861	18,6	%
	210	Care recipient	South	32,8	℃	793	18,0	%
	211	Care recipient	North	29,4	℃	95	0,0	%
	212	Care recipient	North	29,6	℃	109	0,0	%
	213	Care recipient	North	29,5	℃	109	0,0	%
	214	Care recipient	North	25,5	℃	104	0,0	%
	215	Living room	North	30,0	℃	115	0,0	%

Measure 2 – Implementation of airborne cooling

Section	Room nr.	Room type	Orientation	Max. temp.		OH	Improvement	
D								
	301	Care recipient	North/West	31,5	°C	617	19,8	%
	302	Care recipient	North	26,5	°C	10	95,5	%
	303	Care recipient	North	26,8	°C	18	90,1	%
	304	Care recipient	North	26,8	°C	18	89,8	%
	305	Care recipient	North	26,8	°C	19	88,8	%
	306	Care recipient	North	26,7	°C	16	89,4	%
	307	Care recipient	North	26,8	°C	14	88,3	%
	308	Care recipient	North	26,7	°C	11	90,3	%
	309	Care recipient	North	26,7	°C	10	89,9	%
	322	Expedition	South	32,5	°C	1053	8,1	%
	325	Expedition	South	29,3	°C	287	8,9	%
	329	Living room	Central	22,7	°C	0	100	%
	380	Staff room	West	27,9	°C	114	73	%
	399	Office	West	29,2	°C	193	65	%
E								
	343	Dining room	West	25,6	°C	0	100	%
	344	Kitchen	West	24,0	°C	0	100	%
F								
	201	Care recipient	South	31,2	°C	692	18,0	%
	202	Care recipient	South	31,5	°C	788	8,5	%
	203	Care recipient	South	31,6	°C	797	7,6	%
	204	Care recipient	South	31,6	°C	798	7,5	%
	205	Care recipient	South	31,6	°C	798	7,5	%
	206	Care recipient	South	31,6	°C	798	7,5	%
	207	Care recipient	South	31,6	°C	798	7,5	%
	208	Care recipient	South	31,6	°C	797	7,6	%
	209	Care recipient	South	31,6	°C	786	8,7	%
	210	Care recipient	South	31,1	°C	619	21,9	%
	211	Care recipient	North	27,2	°C	21	77,9	%
	212	Care recipient	North	27,3	°C	22	79,8	%
	213	Care recipient	North	27,5	°C	22	79,8	%
	214	Care recipient	North	27,2	°C	17	83,7	%
	215	Living room	North	23,2	°C	0	100	%

Measure 3 – Implementation of passive beams

Section	Room nr.	Room type	Orientation	Max. temp.		OH	Cooling unit	
D								
	301	Care recipient	North/West	25,0	°C	0	700	W
	302	Care recipient	North	24,2	°C	0	-	
	303	Care recipient	North	23,9	°C	0	-	
	304	Care recipient	North	23,8	°C	0	-	
	305	Care recipient	North	23,8	°C	0	-	
	306	Care recipient	North	23,8	°C	0	-	
	307	Care recipient	North	23,8	°C	0	-	
	308	Care recipient	North	23,8	°C	0	-	
	309	Care recipient	North	23,9	°C	0	-	
	322	Expedition	South	25,1	°C	0	900	W
	325	Expedition	South	25,1	°C	0	1 200	W
	329	Living room	Central	21,8	°C	0	-	
	380	Staff room	West	25,2	°C	0	1 400	W
	399	Office	West	25,1	°C	0	1 200	W
E					°C		-	
	343	Dining room	West	24,7	°C	0		
	344	Kitchen	West	22,7	°C	0	-	
F					°C		-	
	201	Care recipient	South	25,1	°C	0	600	W
	202	Care recipient	South	25,1	°C	0	600	W
	203	Care recipient	South	25,1	°C	0	600	W
	204	Care recipient	South	25,1	°C	0	600	W
	205	Care recipient	South	25,1	°C	0	600	W
	206	Care recipient	South	25,1	°C	0	600	W
	207	Care recipient	South	25,1	°C	0	600	W
	208	Care recipient	South	25,1	°C	0	600	W
	209	Care recipient	South	25,1	°C	0	600	W
	210	Care recipient	South	25,1	°C	0	600	W
	211	Care recipient	North	24,0	°C	0	-	
	212	Care recipient	North	24,0	°C	0	-	
	213	Care recipient	North	24,0	°C	0	-	
	214	Care recipient	North	23,7	°C	0	-	
	215	Living room	North	21,7	°C	0	-	

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