A Comparative Analysis of Ground Source and Air Source Heat Pumps for Heating of an Office Building in Bergen

Master's Thesis in Energy

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Abstract

More efficient and sustainable heating systems could reduce greenhouse gas emissions and contribute to meeting the 1.5 or 2-degree goal set by the Paris Agreement. Two heating systems are suggested as alternatives for an example office building in Bergen, Norway. One uses ground heat as its low-temperature heat source, while the other uses air. The building's annual heat demand is calculated to be 728,484 kWh, and the cooling demand to 82,525 kWh.

The design of a ground source heat pump system is often based on empirical rules, which may not be the most effective approach due to the complexity and diversity of factors involved.

The ground heat exchanger is the ground system consisting of several boreholes in which heat is collected. It is estimated that the ground heat exchanger must consist of 36 boreholes of 210 meters depth per borehole. The spacing between these boreholes was found to be 26 meters.

The most cost-effective system for the building in Bergen was the ground source heat pump, which, if correctly sized, is also the most sustainable solution. Other factors to consider when choosing a heat pump system, such as location, electricity prices, and environmental impacts, were also discussed.

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List of Abbreviations

ASHP	Air Source Heat Pump
CFD	Computational Fluid Dynamics
CFL	Courant-Friedrichs-Lewy
COP	Coefficient of Performance
C	Cooling
D	Daytime
GSHP	Ground Source Heat Pump
GTIP	Ground Thermal Imbalance Performance
HP	Heat Pump
N	Nighttime
SPT	Simple Payback time

Nomenclature

α	Thermal diffusivity	m^2s^{-1}
Δ	Difference	
$\dot{Q}$	Heat rate	Wm^{-2}
$\dot{V}$	Ventilation rate	$m^3h^{-1}m^{-2}$

ϵ	Efficiency	
ϵ	Emissivity	
∇	Gradient	
$\overline{q}$	Average heat flux	Wm^{-2}
∂	Change	
ρ	Density	kgm^{-3}
Σ	Sum	
σ	Stefan-Boltzmann constant	
A	Area	m^2
C_p	Specific heat capacity	$J°C^{-1}kg^{-1}$
C_v	Volumetric heat capacity	$Jm^{-3}°C^{-1}$
d	Thickness of the tube wall	m
F	View factor	
h	Heat transfer coefficient	$Wm^{-2}°C^{-1}$
k	Thermal Conductivity	$Wm^{-1}°C^{-1}$
L	Length	m
m	Mass	kg
N	Grid dimensions	
n	Number og gridpoints	
Q	Heat	W
q	Heat	W
T	Temperature	°C
t	Time	s
U	Conductance rate for heat transfer	$Wm^{-1}°C^{-1}$
V	Volume	m^3
W	Work	J
x	Distance in x direction	m

y	Distance in y direction	m
z	Distance in z direction	m

Subscript

air	Air
C	Low-temperature
$Carnot$	Carnot cycle
cf	Collector fluid
$coil$	Ventilation coil
des	Design point
$dist$	Distance between wells
ex	Extracted
f	Fluid
g	Ground
geo	Geothermal
H	High-temperature
hp	Heat pump
ID	Indoor
in	Into the system
$internal$	Internal loads
max	Maximum
OD	Outdoor
rad	Radiatior
$recup$	Recuperator
s	Solid
$surr$	Surrounding

tot	Total
undertemp	Under-temperature
vent	Ventilation

Contents

Chapter 1

Introduction and motivation

1.1 Motivation

The need for sustainable and energy-efficient solutions to reduce greenhouse gas emissions has become increasingly urgent as the threat of global warming and climate change continues to escalate. The world is facing a major international energy crisis. However, the oil crisis in the 1970s accelerated the transition towards more efficient and sustainable systems, and a similar acceleration may occur in response to the current energy crisis [1].

To limit global warming and meet the 1.5 or 2-degree goal set in the Paris Agreement, the world's energy systems need to become less dependent on fossil fuels [2]. Fossil fuels meet more than 60% of global heating demand, with natural gas dominating these statistics [3]. However, the gas shortage could accelerate the growth of less carbon-intensive heating systems. Heat pumps, efficient installations for space heating and cooling, could play a significant role in addressing this challenge, producing 3 or 5 times as much energy as is added to the system [4].

More efficient energy systems are particularly important in countries with high shares of fossil fuels in their electricity mix. Even in countries with near-zero fossil fuels in their electricity mix, such as Norway [5], where the carbon footprint from electricity production is low, more efficient systems are welcome due to limited energy supply and high electricity prices.

The global market for heat pumps has experienced steady growth in recent years, with a remarkable surge in sales in 2022. The sales of heat pumps increased by 11% globally, with an increase of 40% in European sales compared to the previous year. Heat pumps have become increasingly popular in all areas of the world, owing to their ability to provide both heating and cooling throughout the years. Currently, heat pumps account for about 10% of the global heating needs in buildings. To meet the existing climate and energy pledges, this figure should rise to 20% by 2030. However, if the current installation rate persists, this target is reachable [6].

The current heat pump market is dominated by small-scale air-source heat pumps, primarily installed in residential houses with relatively low heat demands [6]. Ground source heat pumps, on the other hand, use the ground as both heat source and sink, owing to the stable temperature in the ground being higher than the outside air temperature in winter and lower in summer. This way, by utilizing the thermal properties of the subsurface, one can obtain a sustainable and efficient way to both heat and cool buildings [7]. However, when the heating demand is high, an incorrectly sized system can lead to a decrease in the ground temperature, resulting in low efficiency or unnecessarily high costs. Ground source heat pumps used mainly for cooling are less sensitive to under- or over-sizing, but if substantially undersized, efficiency would decrease [8, 9].

The conventional approach for sizing ground heat exchangers is to relate the total borehole length to building heat loads using rules of thumb. However, this approach may not be the best, as the design of ground source heat pumps can vary significantly with several factors. On the other hand, finding a value that covers a wide range of applications is nearly impossible [10]. The design of a ground source heat exchanger depends on factors like climate, thus building heating or cooling demand, the ground's thermal properties, and the overall system's efficiency [11]. Furthermore, the total amount of energy extracted from the ground must be considered to prevent the ground temperature from decreasing to a level that could significantly reduce the heat pump's efficiency over time [12].

Rules of thumb vary, even if they are initially made to meet the heat load in the same area. Horizontal loops are a series of underground pipes circulating a refrigerant buried in horizontal trenches a few meters under the surface. In the UK, horizontal loop heat pumps are said to typically require an area that is 2 to 4 times larger than the floor space being heated to extract a sufficient amount of heat, another source state that 15 W/m pipe is a good estimate [8, 13–15].

Vertical loops are vertical U-pipes with a circulating fluid absorbing the ground heat from drilled boreholes. In the United States, it is estimated that the heat extraction per meter borehole ranges from 30 to 111 W/m [16–

19]. These numbers show that using the rules of thumb is not an accurate way of sizing the energy wells. Finding a generalized way to size the heat exchanger is challenging because the sizing depends on factors like climate and ground properties, among others.

In this thesis, an example office building located in Bergen, Norway, was investigated. Two different heat pump systems were suggested: one using outside air as the low-temperature heat source, the other ground heat. It is often stated that using a ground source heat pump system would be the most cost-effective solution in the long due to its high efficiency, even though the investment costs related to the boreholes are much higher than for the air source heat pump. To confirm or disprove this statement, a cost comparison was conducted. Lastly, other essential thinking points on what system to choose were discussed.

1.2 Objective

This thesis specifically seeks to:

- Define a building and its heating and cooling needs
- Define two heat pump systems to be compared, one using outside air as the low-temperature heat source, the other using ground heat
- Compare the two systems economically
- Discuss other beneficial factors when choosing a heating system.

Chapter 2

Theory

2.1 Heat transfer

Heat is the amount of energy transferred from one point to another due to the temperature difference between the two points. Energy is transferred in the direction of temperature decrease. This transfer can occur in three ways; by conduction, convection, or radiation [20, 21].

2.1.1 Conduction

Conduction is heat transferred either from one part of a body with high temperature to another with lower temperature or from one body at higher temperature to an adjacent body, which physically touches the first, that has lower temperature. The process takes place at molecular level by transfer of energy and momentum from molecules with higher energy to molecules with lower energy.

At macroscopic level, the heat flux is proportional to the temperature gradient. Fourier's law describes conduction,

$$\frac{dq}{dA} = -k(\frac{\partial T}{\partial x} + \frac{\partial T}{\partial y} + \frac{\partial T}{\partial z}) = -k\nabla T, \tag{2.1}$$

where the left side of the equation represents the heat flux, k is the thermal conductivity, and ∇T is the temperature gradient in three-dimensional space with directions x, y, and z. The gradient is negative due to the direction of the heat flow, which is from high temperature to low temperature [22].

2.1.2 Convection

Convection is heat transfer between a solid and a fluid in motion. The convective flux is proportional to the temperature difference between the solid and the fluid and follows Newton's law of cooling

$$\frac{q}{A} = h(T_s - T_f), \tag{2.2}$$

where the left side of the equation is the heat flow rate, q, per surface area, A. On the right side, T_s is the temperature of the solid, T_f is the temperature of the fluid, and h is the heat transfer coefficient. Eq. (2.2) is similar to Fourier's law, Eq. (2.1), for conduction through a solid material, but unlike k, the heat transfer coefficient, h, is not merely a property of the material. The heat transfer coefficient is found from the flow pattern and thermal properties of the fluid [23].

There are two primary forms of convection: forced and natural convection. Forced convection occurs when an external device, such as a fan or a pump, causes fluid to flow. On the other hand, natural convection occurs when buoyancy forces created by differences in temperature and density cause fluid motion [22].

2.1.3 Radiation

Thermal radiation is the energy transfer through space by electromagnetic waves [23]. All solids, liquids, and some gases of temperature above 0K emit thermal radiation. Radiation does not require material to transfer heat, as with conduction. Thermal radiation is most effective in vacuum [22].

Calculations of thermal radiation on macroscopic level are based on Stefan-Boltzmann's law. The radiation emitted by a black body, an ideal radiator, is proportional to the body's temperature to the fourth power

$$\frac{q}{A} = \sigma T^4, \tag{2.3}$$

where the left side represents heat emitted per unit area and *sigma* is the Stefan-Boltzmann constant.

However, surfaces rarely perform as black bodies, and Stefan-Boltzmann's law needs to be modified to account for this. For actual surfaces, the law reads

$$\frac{q}{A} = \epsilon\sigma T^4 \tag{2.4}$$

where ϵ is the emissivity of the surface, the emissivity is a value between 0 and 1, where a black body has $\epsilon = 1$. In the case where two black bodies, 1 and 2, exchange heat, net heat exchange is expressed as

$$q = \sigma A_1(T_1^4 - T_2^4). \tag{2.5}$$

Here, T_1 is the temperature of body 1, and T_2 is the temperature of body 2. Emissivity, in this case, is 1, as they are both blackbodies. However, if only a fraction of the radiation that is emitted by body one is intercepted by body 2, the law will look like this:

$$q = \sigma A_1 F_{12}(T_1^4 - T_2^4), \tag{2.6}$$

where F_{12} is the view factor. Its value is the fraction of energy from body 1 that is intercepted by body 2 [22].

2.2 Heat Pumps

Naturally, heat flows from the point of higher temperature to a point of lower temperature. By using a heat pump, this process can be reversed. Heat can be retrieved from a low-temperature source and transported to a point of higher temperature by adding some work to the system, as sketched in Fig. (2.1) [20].

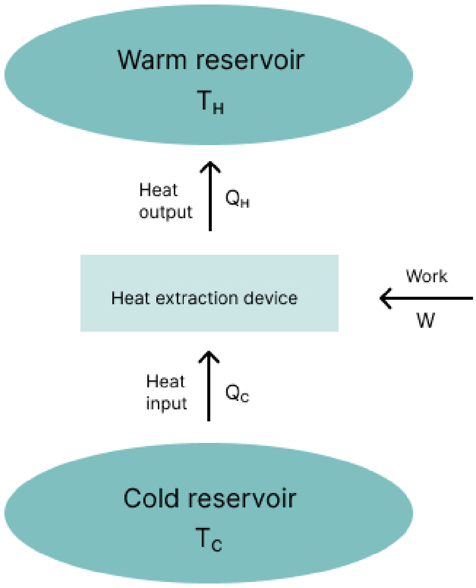


Figure 2.1: Heat pump principle [24].

2.2.1 Classification of heat pumps

Heat pumps can be categorized based on several characteristics, and one of the most significant features is the type of thermodynamic cycle used. Two primary cycles are used: vapor compression and absorption heat pumps.

The conventional vapor compression heat pump operates on an inverse cycle, which comprises four main components: the evaporator, compressor, condenser, and expansion valve. These systems typically involve a compressor powered by an energy input source, such as an electric motor or an engine. The working principle of a vapor compression heat pump will be elaborated further in Section (2.2.2).

Unlike the vapor compression heat pump, absorption heat pumps do not have compressors. Instead, they use a mixture of refrigerants with different vapor pressures. The mixture typically comprises a high-pressure refrigerant here called the absorbent, and a low-pressure refrigerant, here simply called the refrigerant. The refrigerant is absorbed into the absorbent and moved to another location where heat input releases it from the absorbent. Once the refrigerant is released, it evaporates and then recombines with the absorbent, creating a low-pressure mixture. The refrigerant and the absorbent are then circulated back to their starting points to repeat the process. In this thesis, vapor compression heat pumps are the only ones in focus.

The refrigerant is important to consider when designing a heat pump system. The refrigerant should fulfill several requirements. It should be environmentally friendly, not flammable, chemically stable, not toxic, have a low freezing point, and be compatible with the heat pump's lubricating oils and other components. The refrigerant should also have a relatively low boiling point so that it evaporates at low temperatures [25].

Further, we can differentiate according to the heat source. In this thesis, the chosen heat source will be the only characterization explored in depth. The most commonly used low-temperature heat sources are outside air, ground heat, and surface water.

2.2.2 Heat pump technology

Heat pumps are thermal installations that are based on the reverse Carnot cycle. This cycle requires some energy input, often in the form of electricity [26]. The four main components of a heat pump are shown in Fig. (2.2). In addition to these components, the heat pump has a circulating refrigerant carrying the thermal energy through the cycle. Heat pumps are one of the most efficient ways to heat buildings in relatively mild climates [27].

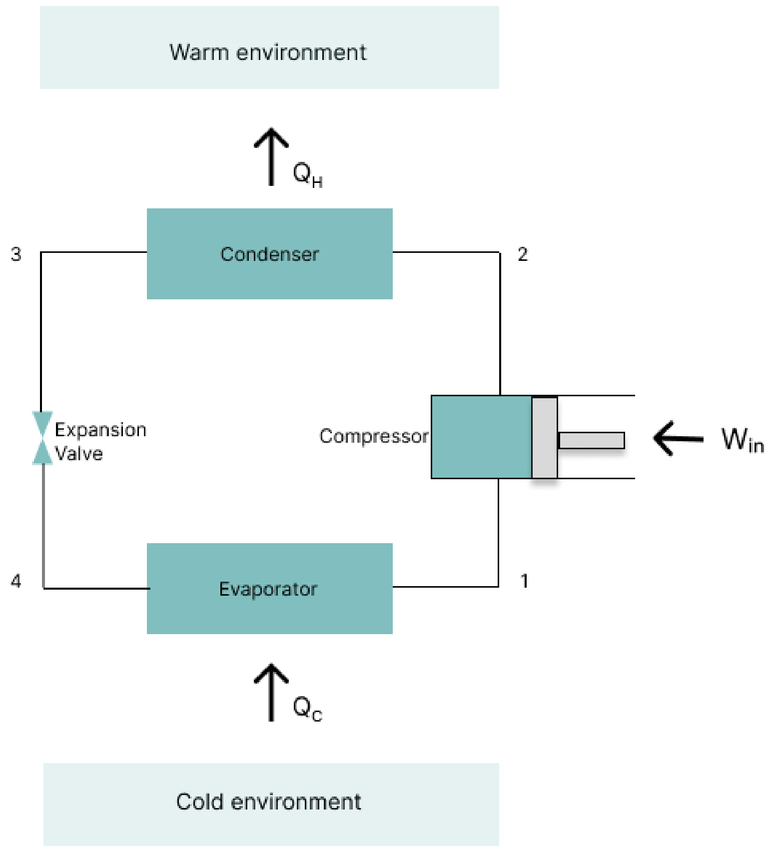


Figure 2.2: The components of a heat pump [20].

Evaporator

In the evaporator, the refrigerant absorbs heat from a low-temperature source, causing it to reach its boiling point and evaporate. The refrigerant should have a low boiling point for the medium to evaporate easily at low temperatures.

Compressor

From the evaporator, the vapor proceeds to the compressor. Here the refrigerant is compressed, and thus there is a pressure and temperature increase. The compressor requires addition of energy, usually electricity.

Condenser

In the condenser, heat will be transferred from the vapor with increased pressure and temperature to the space that is to be heated, most often the inside of a building. The heat loss makes the vapor condense, moving into the expansion valve.

Expansion valve

The warm fluid flows through an expansion valve where pressure and temperature are reduced to initial values. The cycle starts over [4].

2.2.3 Heat sources

The only two to be described in depth are outside air and ground heat, as they are the sources of the two heat pump systems to be compared in this thesis.

Outside air

Outside air as a heat source is widely used, especially in small-scale systems. One of the most significant disadvantages is that in winter, when the heat demand is high, the outside air is at a lower temperature, and the heat pump is thus less effective than in warmer seasons when the heat demand is lower. Therefore, air-source heat pumps are best suited in coastal climates or other climates where temperatures are relatively mild.

Air-to-air heat pumps are commonly used in houses, small apartment buildings, and other applications where the space to be heated is of limited area [28]. Air-to-air heat pumps transfer heat directly or through ventilation ducts to the inside air [29]. A ducted transfer could be advantageous, as the distribution of heat is more even than when using direct transfer. On the other hand, ductwork installation is associated with higher costs unless ducts are already installed. Ductless installation requires no construction work except connecting the outside and inside modules [30].

Air-to-air heat pumps offer a range of advantages. Some of the most noteworthy are listed below.

- Most of the heat pump models on the market can be used for cooling in summer by reversing the cycle
- Installation requires a minimum of modification and construction and is easily installed in almost any residence
- Investment costs are relatively low
- The heat source is always available, regardless of geographic location
- No local greenhouse gas emissions

There are also disadvantages:

- Another heating system may be needed for peak load in areas where the temperatures are especially low
- At outside temperatures below approximately 2-5°C, it is necessary to defrost the machine to remove ice on the outdoor condenser. This is optimal for heat absorption, but the heat pump's efficiency is reduced while defrosting
- Damp and salty air, typically in coastal areas, could lead to corrosion. This might reduce the lifetime of the heat pump
- Noise pollution [28]

Air-to-water heat pumps are commonly used in larger residential buildings with several housing units and office buildings. Waterborne heat ensures even heat distribution throughout the entire building through underfloor heating, radiators, or a combination of the two. Air-to-water heat pumps have higher investment costs than air-to-air, thus longer pay-back time [31].

Some significant advantages of air-to-water heat pumps are:

- Can be used for cooling in summer
- Heat source is outside air, an abundant and widely available resource. This eliminates the need for a limited and more expensive heat source, such as a ground source reservoir
- Reasonably low investment cost
- Supplies facilities of waterborne heat, which provides even distribution of heat and a better indoor climate than heat from combustion of fuel

- No local greenhouse gas emissions

The disadvantages of air-to-water heat pumps are the same as those for air-to-air heat pumps mentioned above [31].

Ground heat

Ground source heat pumps can recover low-temperature geothermal energy almost everywhere. They can be used for cooling in summer. Thus, geothermal heat pumps can heat buildings in winter and cool them in summer [32]. Efficient geothermal systems are attainable for residential, apartment, and office buildings [33]. They can be used locally, for a single or a few buildings, or district heating [34].

A ground source heating system contains three main subsystems, the ground connection system, the heat pump system, and the heat distribution system. Ground source heat pumps have two main categories: ground-water systems and ground-coupled systems. Ground-coupled heat pump systems are the most common of the two [26], which will be discussed further in this thesis. In further discussion, ground-coupled heat pumps are called ground-source heat pumps.

Heat can be extracted from both horizontal and vertical collector loops. In this thesis, vertical loops are the most relevant. Most are closed systems, where heat is transferred from the ground to water mixed with an antifreeze agent, which is circulated through the energy well. Usually, a heat exchanger connects the ground system to the heat pump system, where the heat is further transferred to a refrigerant which circulates in the heat pump system [24].

Compared to outside air, the temperature in the ground is much more stable, and borehole temperature remains near-constant throughout the year. The temperature in the ground is assumed to hold approximately the same temperature as the annual average temperature [35].

Ground-source heat pumps are often known to have a longer lifetime than air-source heat pumps, partly due to less or no temperature fluctuations. Air source heat pumps, especially in colder climates, operate at higher loads, which causes more strain on the condenser. Heat pumps also perform better at higher temperatures, requiring less energy to move heat from the low-temperature source to the space that is to be heated. Consequently, ground-source heat pumps often have lower annual operating costs than air-source heat pumps [36].

Ground-source water-to-water heat pumps have many advantages:

- Suitable for residential buildings with high heat and hot water demand
- Waterborne heating assures even heat distribution
- Can be used for cooling in summer with small adjustments
- No visible parts of the heat pumps system outdoors
- Lifetime is longer than for other types of heat pumps
- No or little noise, both inside and outside
- No local greenhouse gas emissions
- Low maintenance and operating costs
- In Norway, and several other countries, installation of ground source heat pumps qualify for subsidies; in Norway, subsidies for a more rapid energy transition are distributed by Enova [37].

Ground source heat pumps have few disadvantages. They have high investment and installation costs, in addition, one would need an appropriate piece of land. The higher investment cost is primarily due to the cost of the boreholes. The heat pumps have a very long lifetime, and if a new heat pump is to be installed after the lifetime of the first, the same boreholes can be used again. Then the only investment costs are a new heat pump and collector tube, given that the wells are sustainably designed.

2.2.4 Heat pump efficiency

Coefficient of Performance - COP

The coefficient of performance (COP) is a measure of the efficiency of a heat pump and is defined as the ratio of heat output to work input

$$COP = \frac{Q_H}{W_{net,in}}, \tag{2.7}$$

where Q_H is the heat output from the heat pump, and $W_{net,in}$ is the work input. A higher COP means a more efficient heat pump, as more heat is produced from a given amount of energy [24].

The equation above can also be expressed as

$$COP = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - \frac{Q_L}{Q_H}}, \tag{2.8}$$

where Q_L is the amount of heat absorbed from the low-temperature heat source [20].

Carnot efficiency

The efficiency of any heat pump is restricted by an upper theoretical limit called the Carnot limit, which is the efficiency of the Carnot cycle. The Carnot efficiency can be used to see how efficient a heat pump system is compared to what is theoretically possible. The Carnot limit is decided based on the temperatures in the heat source and the sink [24].

A Carnot heat pump operates as a reversible heat pump by following the same cycle as the Carnot cycle but reversing the directions of heat and work. The reversed Carnot cycle is shown in Fig. (2.3) and consists of the following four processes:

- 4-1 Isothermal heat absorption in the evaporator
- 1-2 Isentropic compression in compressor
- 2-3 Isothermal heat rejection in the condenser
- 3-4 Isentropic expansion in turbine

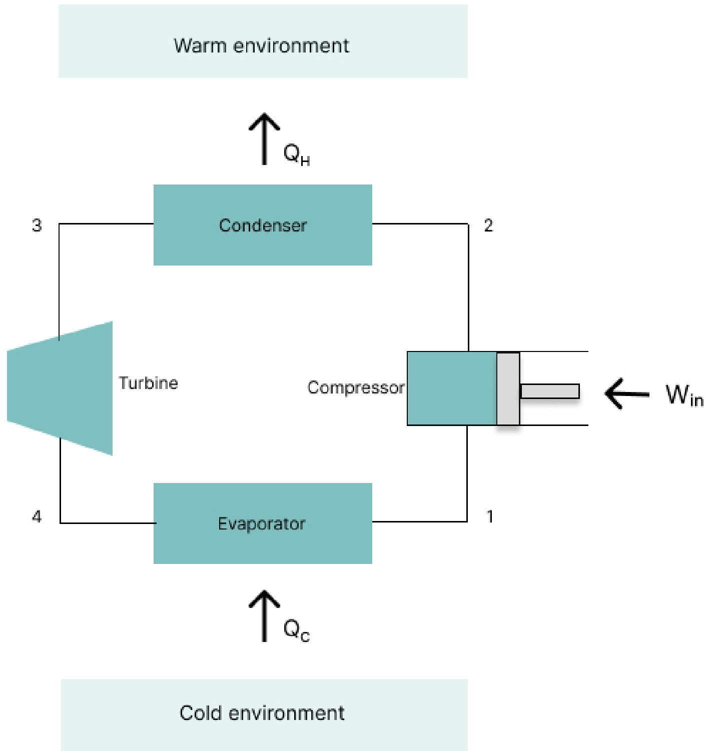


Figure 2.3: Reversed Carnot cycle [20].

This cycle is theoretically the most efficient refrigeration cycle operating in a specific span of temperatures. Therefore, looking at it as an ideal cycle for heat pumps would make sense. However, it is not suitable for heat pumps and is very unrealistic. The isentropic processes in the compressor and expansion valve (processes 1-2 and 3-4) cannot be practically approximated due to the second law of thermodynamics. This law dictates that entropy cannot increase, limiting these processes' practical feasibility. Due to losses in the system, entropy is not conserved, and thus, the cycle is irreversible. In further detail, the compressor must handle a vapor-liquid, not saturated gas, and the turbine must expand a refrigerant with a very high moisture content. To fix the problems with the isentropic processes, the isothermal processes would be challenging. In other words, a compromise must be found. The best use of the reversed Carnot cycle is as a reference point used to compare actual cycles [20].

For reversible refrigeration cycles, such as the Carnot heat pump, the heat terms from Eq. 2.8 can be written using absolute temperatures:

$$COP_{carnot} = \frac{T_H}{T_H - T_L} = \frac{1}{1 - \frac{T_L}{T_H}}, \tag{2.9}$$

where T_H is the temperature of the room of higher temperature that is to be heated, and T_L is the temperature in the low-temperature heat source. Real heat pump systems will always have COP lower than the Carnot heat pump efficiency [20].

Using Eq. (2.9), it can be seen why ground-source heat pumps usually have higher COP than air-source heat pumps. It is beneficial if the difference between the source and sink temperatures is as small as possible. The temperature difference is relatively large when using outside air as the low-temperature heat source in the winter. If ground source heat is used, the temperature is higher, even in the coldest periods of the year.

The ideal vapor-compression refrigeration cycle

Both issues mentioned above can be solved by vaporizing the refrigerant entirely and replacing the turbine with an expansion valve. The result of these changes is the ideal vapor-compression refrigeration cycle. The cycle consists of the four processes listed below; see also Fig. (2.2).

- 4-1 Isobaric heat absorption in an evaporator
- 1-2 Isentropic compression in a compressor
- 2-3 Isobaric heat rejection in a condenser
- 3-4 Isentropic expansion in an expansion valve

In this cycle, the refrigerant enters the compressor in state 1 as saturated vapor and is compressed isentropically to condenser pressure, resulting in a temperature increase. The refrigerant enters the condenser as superheated vapor in state 2 and leaves the condenser as a saturated liquid in state 3. Then the pressure is reduced to evaporator pressure by an expansion valve, and the temperature decreases due to the pressure reduction. Unlike the reversed Carnot cycle, the ideal vapor-compression refrigeration cycle is not reversible, as it involves an irreversible process in the expansion valve [20].

Actual vapor-compression refrigeration cycles

There are several differences between an actual vapor-compression refrigeration cycle and an ideal one, primarily due to irreversibilities in different forms and components. Two main reasons for irreversibilities are pressure loss due to friction and heat transfer between the refrigerant and the surroundings.

In the ideal cycle, the refrigerant is in the form of saturated vapor when leaving the evaporator, and this is impossible to control precisely and, therefore, impossible in practice. To approximate the actual cycle to the ideal, the refrigerant is slightly superheated when reaching the compressor, ensuring that the refrigerant is completely vaporized when entering the compressor. The line connecting the evaporator and the compressor (see Fig. (2.2)) is usually long. As a result, we will have pressure and heat loss between the evaporator and the compressor.

The inefficiencies resulting from superheating, pressure loss, and heat loss necessitate an increased workload on the compressor, leading to a higher energy input than the ideal cycle. Consequently, the efficiency of the heat pump decreases.

The superheating, pressure loss, and heat loss result in an increased workload on the condenser, thus, higher energy input than for the ideal cycle, lowering the heat pump's efficiency. There would be some friction and, therefore, the pressure loss in the condenser, and the refrigerant is also subcooled before reaching the expansion valve to ensure that it is fully condensed when reaching the expansion valve. The condenser and the expansion valve are usually close; therefore, there is little pressure loss in the connecting line between the two [20].

2.3 Geothermal energy

Geothermal energy is a renewable energy source with increasing interest. Unlike wind, solar, and other variable energy sources, geothermal energy is a reliable and continuous energy source [38]. Averaged over Earth's surface, energy flows outwards at a rate of $\overline{q}_{geo} = 87$ mW/m^2. Integrated over the Earth's surface, this adds up to approximately 44 TW [24].

This thermal energy is of different origins, where 60% is from radioactive decay in the crust, and the majority of the remaining energy is from the cooling of the planet's interior. In other words, a little less than 40% of the geothermal heat originates from the Earth's formation many billion years ago [21]. The heat is transported from the core to the surface through convection, conduction, and thermal radiation [39]. Conduction, convection, and radiation are described in sections (2.1.1), (2.1.2), and (2.1.3), respectively.

The heat from the degradation of radioactive isotopes and heat from Earth's formation is not easily accessible. To access these reservoirs, energy wells of several kilometers length are needed, unless in a very volcanic area, such as Iceland. Using this energy source for electricity production or district heating could be financially practical in these areas due to low drilling depths. On the other hand, this would be an economically ineffective way to extract energy here in Norway. However, there are other ways to use the subsurface heat, for example through the use of heat pumps.

2.3.1 Classification of geothermal energy

Geothermal sources are often classified as either shallow or deep. Geological Survey of Norway classifies shallow, or ground source, geothermal energy as energy stored less than 300 meters below the surface. Deep geothermal energy is defined as energy stored from 300-400 meters below the surface to the core [40]. These numbers are approximate, as these limits are not definite and change based on geographical location.

2.3.2 Deep geothermal energy

Deep geothermal energy is primarily used for electricity production or heating purposes. The heat is extracted using wells that are usually more than a kilometer deep, and water is used to transport the heat. Unlike shallow resources, the temperature in deep geothermal sources is much higher, and thus no heat pump is required for heating applications. Both open and closed systems are used to harness deep geothermal energy, where open systems use water from the ground directly as the working fluid and closed systems use a circulating fluid to absorb heat [41].

2.3.3 Shallow geothermal energy

To utilize the ground heat, wells must be drilled, and a system must be connected to these wells. The use of open and closed systems is explained in the following.

Open loop systems

Open loop system uses groundwater directly as the working fluid. Water is extracted from one or more boreholes and passed through a heat pump system before being pumped down one or more boreholes. The discharged water has undergone a temperature change, which can be either positive or negative, depending on whether it is used for cooling or heating, respectively [35]. A typical open loop system is shown in Fig. (2.4), where the heated groundwater is extracted from the borehole on the left side, and colder water is rejected to the borehole on the right.

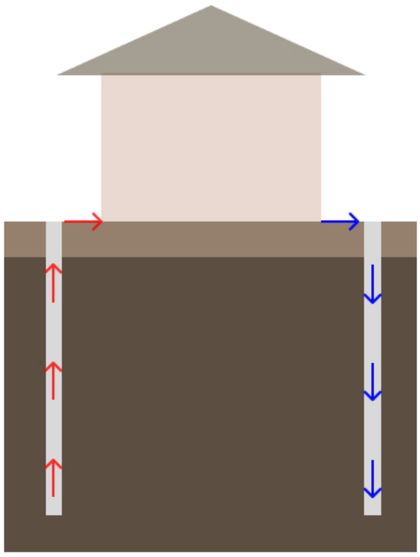


Figure 2.4: Open loop system for heating, where the extracted warmer water is shown using red arrows and colder water using blue arrows.

Closed loop systems

In closed-loop systems, the heat is indirectly retrieved using collector tubes from horizontal or vertical wells. The two are shown in figures (2.5) and (2.6), respectively.

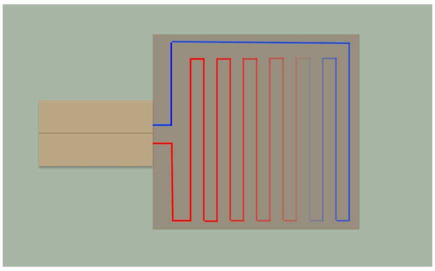


Figure 2.5: Closed horizontal loop system for heating, red pipes representing higher temperature, blue pipes colder.

In the sketch of a closed horizontal loop in Fig. (2.5), the temperature of the water is gradually increasing through the length of the loop. The red color denotes a higher temperature, and the blue color shows the parts of the pipe where the temperature is lower.

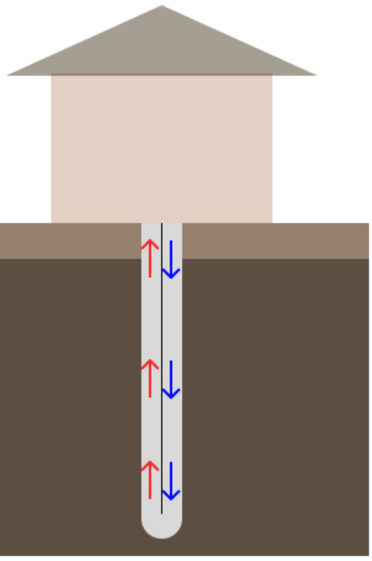


Figure 2.6: Closed vertical loop system for heating, red arrows depict water heated by the ground, and blue arrows represent colder water being pumped back into the well.

The red arrows in Fig. (2.6) represent water with higher temperatures extracted and chilled water returned to the energy well. The vertical, closed loops are the ones that are used for the calculations in this thesis.

There are advantages and disadvantages for both horizontal and vertical loops. For ground source heat pumps, the higher costs compared to systems using other heat sources are mainly due to drilling costs. For horizontal systems, drilling is usually unnecessary, and relatively low costs are the main advantage of horizontal systems. Disadvantages of these systems are that at such short distances below ground, temperature is more weather dependent, in addition, more surface area is needed to extract the required amount of heat. Vertical wells are more expensive, but in return, the ground temperature is more stable through the year, and as a result, we get a more stable and efficient system [11].

2.3.4 The ground as heat storage

The subsurface works as a thermogeological accumulator and is warmed mainly by solar energy [24]. During summer, when heat is in surplus, heat can be deliberately injected into the ground. Examples of surplus heat that could be discarded into the ground are

- Solar radiation harvested by solar panels or collectors
- Heat from cooling of the building(s) in question
- Waste heat from industrial processes
- Surplus heat from combined heat and power installations [11].

Specific heat capacity, C_p

A substance's ability to store heat is decided by the specific heat capacity of the substance, C_p. The heat loss in a medium is given by

$$Heat\,loss = mC_p\Delta T, \tag{2.10}$$

where m is mass of the medium, C_p is the specific heat capacity and ΔT is the difference between final and initial temperature [11].

Volumetric heat capacity

In this thesis, using volumetric heat capacity instead of specific is beneficial. The two heat capacities are easily related by the density of the substance by

$$C_v = \rho C_p, \tag{2.11}$$

where C_v is the volumetric heat capacity, and ρ the density of the material in question. The use of volumetric heat capacity is more beneficial in the means of energy wells than mass because the volume of the energy wells is easily calculated using the well walls' area and the boreholes' depth [11].

2.3.5 Energy wells

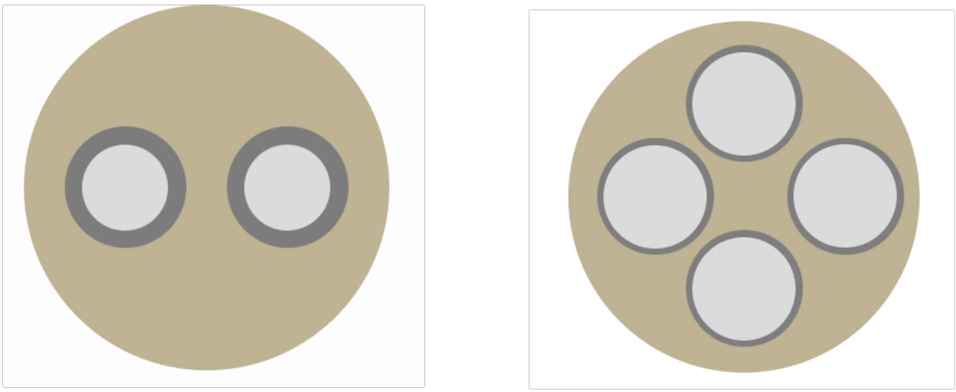
Energy wells are closed vertical loops that make utilization of ground heat possible. When a sufficient depth in the ground is reached, the temperature remains reasonably constant throughout the year [42]. The sizing of the energy wells is an important part of planning the geothermal heat pump system. If they are sized to be too large, the installation of the ground source heat pump system is unnecessarily expensive. If it is undersized, one might have a decrease in power output due to the depletion of the reservoirs [43]. Such a reservoir depletion would decrease the system's efficiency [44]. Sustainable wells are described further in section (2.3.6).

Collector tubes

Various collector tubes are used for closed-loop ground source heat pump systems. A single U-tube is the most common for heating systems, while a double U-tube is often used where the cooling demand is higher. Both alternatives are shown in Fig. (2.7). Double U-tubes usually have thinner walls, thus a more extensive area for heat transfer than a single U-tube. Double U-tubes are often used for cooling because these properties reduce the thermal resistance in the tube, defined as

$$Thermal\, resistance = \frac{d}{kA} \tag{2.12}$$

where d is the thickness of the tube wall, k is the thermal conductivity and A is the wall area that the heat transfers through [11]. Double U-tubes reduce thermal resistance by increasing the area of heat transfer and decreasing the tube thickness. Due to the heating demand being dominant in Norway, single U-tubes are the most commonly used collector.



(a) (b)

Figure 2.7: A sketch of (a) single and (b) double U-tube.

In each well, there is a U-tube collector containing a circulating solution of water mixed with an anti-freeze agent. The space between the tube and the well is generally filled with a fluid, usually groundwater or brine that is added after drilling. The purpose of filling this void, which is depicted as the darker area in Fig. (2.7, is to optimize the heat transfer from the ground to the collector fluid. Compared to air, water has higher conductivity. Looking at Eq. (2.1) that shows Fourier law, we see that the higher conductivity of water would enhance the heat transfer.

Collector fluid

The collector pipes have a circulating fluid, making the borehole work like a heat exchanger. Heat is transferred from the ground through the pipe and moved to the condenser by the fluid. This fluid is usually a mixture of water and ethanol or glycol to ensure that the fluid does not freeze in the colder months of the year [45].

Potential

The theoretical potential for energy extraction from the energy well(s) can be calculated using the volumetric heat capacity of the ground.

$$Energy\ extracted = VC_v, \tag{2.13}$$

where V is the ground volume used for wells, and C_v is the volumetric heat capacity of the ground formation. Eq. (2.13) gives the amount of energy that can be extracted from the ground volume per one degree of temperature decrease in the ground. The equation is simplified, as a variety of factors will influence the amount of heat one can extract from the energy wells. The most common factor is groundwater flow, transferring heat into the wells in winter, when heat is extracted, and out of the wells when the cooling demand is higher in summer.

The sizing of energy wells and borehole parks are today, as mentioned in the introduction, dominated by different rules of thumb when it comes to power output, energy output, and area needed for energy wells. This gives an unnecessary number of ground source heat pump systems that are wrongly sized, either over- or undersized. Several factors should be considered to ensure that the energy wells are appropriately sized.

To ensure that the subsurface heat reservoir meets the required heat load and is not oversized, both the maximum power load and the annual energy load should be considered. Power load is what the energy wells are exposed to in the colder months, at design outdoor temperature. Therefore, when the heat pump works at full capacity, this power load is usually short-term and is the deciding factor for the total length of the boreholes. Annual energy load decides the volume of the borehole park. This includes the depth and number of wells, distance between wells, and formation of the boreholes. This volume has to be sufficiently large so that the temperature decrease in winter is not too high, affecting the efficiency of the heat pump system [46].

The heat that is extracted from the ground is given by

$$Q_{ex} = L(U\Delta T), \tag{2.14}$$

where Q_{ex} is the amount of heat extracted from the ground, L is the total length of the boreholes, U is the conductance rate for heat transfer from the ground to the circulating collector fluid, and ΔT is the temperature difference between the ground and the collector fluid [16].

The geological formation at the drilling site is essential. Different kinds of rock have different conductivity, thus some rock species are more suitable for energy transportation than others. This will have an effect if groundwater is present at the side, as heat is more easily transported through water than the rock [12]. It is beneficial if the distance down to solid rock is as small as possible from an economic point of view. The casing is needed when drilling through the overburden, which makes drilling costs higher.

2.3.6 Thermal balance and sustainable wells

During the summer, if cooling demand is higher than heating demand, the wells can also be used for space cooling. However, over the year, in Norway, the heating demand is generally higher, and the amount of extracted energy would be larger than the amount returned. This unbalanced situation can lead to a decrease in temperature over time [47].

Indicator of thermal imbalance

For a borehole or borehole field to be sustainably sized, the energy output through the year has to be compensated by an equal amount of energy input. In the top few hundred meters underground, where the vertical collector tubes are, radiation from the sun is stored as heat [48]. Therefore the solar radiation in the heat pump's geographical area must be considered.

The desired scenario for an energy well system is that the temperature is stable during the entire lifetime of the system. Therefore, heat out of the energy well must equal heat into the well. A way to investigate the balance in a system of boreholes is to define an indicator called ground thermal imbalance performance (GTIP). This is defined as:

$$GTIP = (1 - \frac{\Sigma Q_{borehole,heat}}{\Sigma Q_{borehole,cooling}}) * 100\%. \tag{2.15}$$

where $\Sigma Q_{borehole,heat}$ is the sum of heat out of the borehole, while $\Sigma Q_{borehole,cooling}$ is the total amount of heat going into the ground [49].

GTIP should be as low as possible, preferably 0%. If a borehole system has a high value of GTIP, one could install systems to add heat. An example could be solar collectors, where heat from the sun could be stored and used to balance the wells. In appliances where cooling demand is higher than heating demand, dry coolers could be used, dumping heat to the outside air rather than the well.

Temperature in the well

There are several possible causes for an imbalance in the energy wells. The most important ones, in this case, are insufficient distance between boreholes or low groundwater flow. If the wells are placed too closely together, they will affect each other. The result would be that the heat pump system extracts more energy than is restored. This could be problematic over the borehole field's lifetime, as the ground might have decreasing annual mean temperature [11]. If the groundwater flow is insufficient, heat in the wells would be restored at a slower rate, which would have the same consequences for the temperature in the ground over time.

Low restoring heat rate will decrease the efficiency of the heat pump. In the Norwegian climate, heating demand is much higher than cooling demand. Therefore, it is common for the ground temperature coupled with ground source heat pumps to decrease over time due to wrongly sized borehole fields. In addition to lower COP, consequences could be freezing temperatures in the ground, which would further lower groundwater flow and worsen the existing problem.

Summarized below are three reasons why decreasing ground temperature is undesirable [11].:

- Economic reasons connected to lower efficiency of the heat pump system. The main reason for choosing a ground source heat pump system over an air source system is mainly gone, and the additional investment cost of a ground source system is unnecessary.
- The mean temperature of the collector fluid should remain above zero degrees Celsius to prevent permafrost.
- Minimum temperature in the collector fluid must never sink below its freezing temperature (-15°C if 30% ethanol is used [50])

2.4 Heat transport within the ground

To get a better picture of the temperature development within the ground after having extracted heat for some time, one can use techniques that are explained in the following section.

2.4.1 Discretization

Discretization is when a closed-form mathematical expression, like in this case a partial differential expression, that has an infinite continuum of values through its domain is approximated by an expression that prescribes values at a finite number of discrete points in the domain, called grid points. Discretization aims to end with a system of algebraic equations that can be solved numerically in discrete points. Another name for this is the method of finite differences [51]. An example of a grid and naming of the discrete grid points are shown in Fig. 2.8.

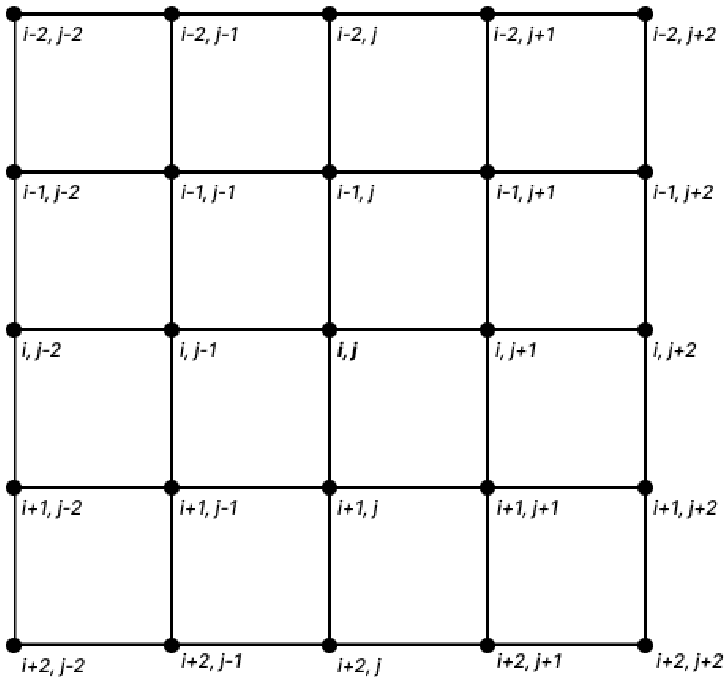


Figure 2.8: Example of the grid and naming system of discrete grid points.

2.4.2 Explicit and implicit approaches

To solve a problem, a technique has to be chosen. The methods fall into one of two approaches: implicit or explicit.

An explicit approach is when each difference equation only has one unknown and can be solved explicitly for this variable. In other words, in this case, the explicit method only requires the current state of the flow to calculate the state at the next time step. This approach is easier to implement and is more computationally efficient for problems with shorter time scales. The downside is that this method could easily become unstable for problems with more extended time scales or when the time step is large.

An implicit method, on the other hand, requires several unknowns that have to be computed simultaneously. The numerical solution at the next time step is computed from a system of equations, including the current and the next time step. This makes the method more inefficient than the explicit method, as all the grid points have a set of equations that must be solved rather than one. However, the solutions are more stable and accurate for problems with longer time scales or with larger time steps [51].

2.4.3 Equations used for estimating the ground temperature development

The one-dimensional heat conduction equation is given by

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}, \tag{2.16}$$

where the left side of the equation represents the temperature change over time and the second partial derivative of temperature in the x direction. α is the thermal diffusivity, which is given as

$$\alpha = \frac{k}{\rho c_p} \tag{2.17}$$

where k is the thermal conductivity of the ground, ρ is the density of the ground, and c_p is the specific heat capacity at constant pressure.

Eq. (2.16) can be represented as an explicit model. Choosing to represent $\frac{\partial T}{\partial t}$ as a forward difference, and $\frac{\partial^2 T}{\partial x^2}$ as a central difference, one obtains

$$\frac{T_i^{n+1} - T_i^n}{\Delta t} = \frac{\alpha(T_{i+1}^n - 2T_i^n + T_{i-1}^n)}{(\Delta x)^2} \tag{2.18}$$

By rearranging the equation, we get:

$$T_i^{n+1} = T_i^n + \alpha \frac{\Delta t}{(\Delta x)^2}(T_{i+1}^n - 2T_i^n + T_{i-1}^n). \tag{2.19}$$

To get a picture of the temperature in the ground, Eq. (2.19) can be used, but it is more beneficial to have a 2D or 3D picture. A two-dimensional explicit finite-difference model that can be used to have a better look at the temperature development from conduction can be represented as [51].

$$T_{i,j}^{n+1} = T_{i,j}^{n} + \Delta t \alpha \frac{T_{i+1,j}^{n} - 2T_{i,j}^{n+} + T_{i-1,j}^{n}}{\Delta x^2} + \Delta t \alpha \frac{T_{i,j+1}^{n} - 2T_{i,j}^{n} + T_{i,j-1}^{n}}{\Delta y^2}. \quad (2.20)$$

2.4.4 The Courant-Friedrichs-Lewy stability criterion

A stability criterion is necessary for the stability of numerical methods used to solve partial differential equations. The Courant-Friedrichs-Lewy (CFL) stability criterion relates the time step size to the spatial grid size. Specifically, for the two-dimensional model given as Eq. (2.20), the CFL condition can be expressed as:

$$\Delta t \leq \frac{(\Delta x)^2}{4\alpha}. \quad (2.21)$$

The condition ensures that the numerical method remains stable and that the error does not grow with each time step [51].

2.5 Economy

Simple payback time can be used to compare the costs of different heat pump systems. Simple payback time is the time it takes to earn back the investment associated with a heat pump system. In the case of heat pumps, the investment is both the cost of the heat pump itself and the maintenance and operating costs. For ground source heat pumps, the cost of the boreholes has to be included as investment costs as well.

The payback is, in most cases, decided based on income. For heat pumps or other energy-producing installments, the payback is the costs saved by investing. For a heat pump, this is the cost of the energy that is retrieved from the low-temperature energy source [52].

The formula used to calculate simple payback time (SPT) is given as

$$SPT = \frac{Total\ investment\ cost}{Cost\ of\ saved\ energy}. \quad (2.22)$$

Chapter 3

Methods

3.1 Temperature data

The main objective of this thesis is to find the best heat pump solution from an economic point of view. To calculate the operating costs of a heat pump, one has to know the required electrical input and, thus, the annual heat demand of the building. We need a history of temperature data from the chosen area because heat demand depends on the outside temperature. It will also be used to calculate the evaporation temperature directly for the air-source system and by using the annual average temperature for the ground-source system.

Temperature data for Bergen was retrieved from Norsk Klimaservicesenter [53]. The location was set to Bergen, Florida, and the time resolution to hours. Data were retrieved for 2022, starting on 01.01.22 at 00.00 local time and ending on 01.01.23 at 00.00. This gives 8760 data points, one for each hour of 2022. The temperature data is presented chronologically through 2022 in Fig. (3.1).

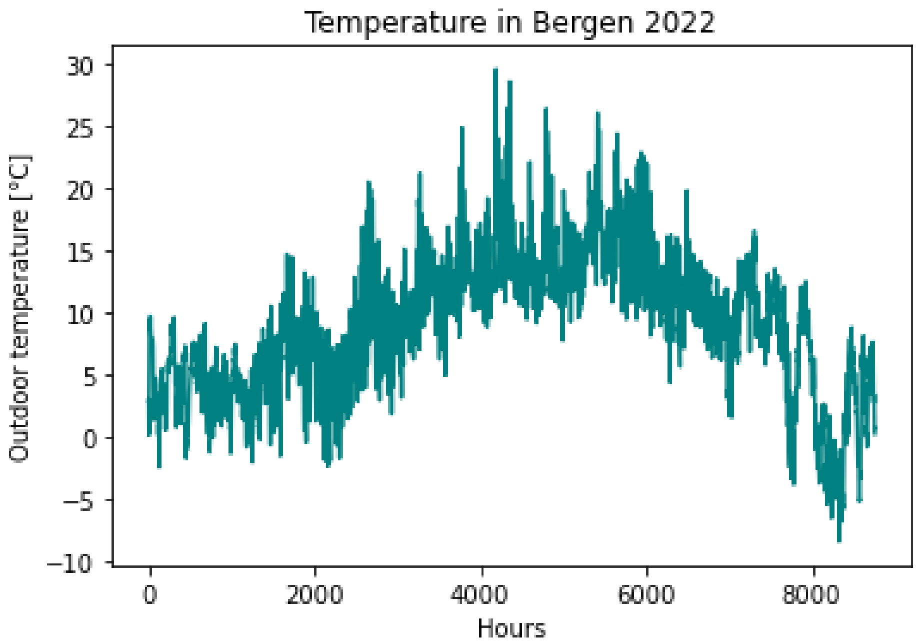


Figure 3.1: The temperature in Bergen every hour of 2022, in chronological order.

3.2 Example building

The building to be analyzed in this thesis is an office building located in Bergen, Norway. It is an example building where the values used for calculations are assumptions, not known values as they would be for an existing building.

The total area of the building is 17000 m^2, population density is 20 m^2 per person, and ventilation rate is 12 m^3/h per unit area. The outdoor design temperature is set to be -12°C, which is assumed to be the lowest probable temperature in Bergen, and the indoor design temperature to 22°C. The average outside temperature is calculated to be 8.9°C, using the temperature data. These properties are summarized in Table (3.1).

3.2.1 Building heat demand - Design point

The design point heat demand is found at full ventilation without including internal heat sources. Transmission and infiltration losses at the design point, $Q_{loss,des}$, are assumed to be 325 kW, following Norwegian building codes [54].

Table 3.1: Properties of the example building

Property	Value	Unit
Area	17000	m^2
Population density	20	m^2 pr person
Ventilation rate	12	m^3/h per m^2
Outdoor design temperature	-12	°C
Indoor design temperature	22	°C
Average outside temperature	8.9	°C

Air from the ventilation system is supplied at an under-temperature assumed to be 3°C below the design indoor temperature, making the colder air mix efficiently due to differing density. Radiator design heat rate takes this temperature difference and $Q_{loss,des}$ into account. The heat needed to raise the temperature of the ventilation air, $\dot{Q}_{undertemp,des}$, is

$$\dot{Q}_{undertemp,des} = \dot{V} A c_{p,air} \rho_{air} \Delta T, \tag{3.1}$$

where $\dot{V}$ is the ventilation rate per area, A is the area of the building, $c_{p,air}$ is the specific heat capacity of air at constant pressure, ρ_{air} is the density of air and ΔT is the under-temperature at which the ventilation air is supplied, here 3°C.

The total radiator heat demand at design point, $Q_{rad,des}$ is given as

$$Q_{rad,des} = Q_{loss,des} + Q_{undertemp,des} \tag{3.2}$$

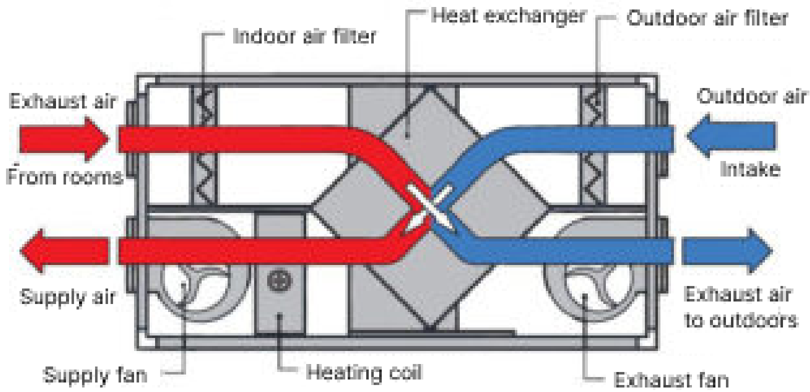


Figure 3.2: Sketch of the different components in a recuperator [55].

Eq. (3.1) is also used in the calculation of heat demand for the outside air coming into the ventilation system, $Q_{vent,coil,des}$. A recuperator, similar to the one shown in Fig. (3.2), is used to recycle excess heat in the ventilation system. A heating coil, like the one shown in the sketch, is used to heat the air. The heat supplied by the heating coil must raise the temperature of the air by

$$\Delta T = T_{ID,des} - \Delta T_{coil,room} - T_{OD,recup,des}, \tag{3.3}$$

where $T_{ID,des}$ is the indoor design temperature, and $\Delta T_{coil,room}$ is the temperature difference between the air leaving the heating coil and the design indoor temperature. This temperature difference is found using the assumption that the heat is delivered at an under-temperature of 3°C, warming of approximately 1°C due to friction in the transfer from the inlet to the heating coil, and a temperature increase of 1°C due to heat added from fan work in the ventilation system. This gives a $\Delta T_{coil,room}$ of 5°C. $T_{OD,recup,des}$ is given by

$$T_{OD,recup,des} = T_{OD,des} + \epsilon_{recup}(T_{ID,des} - T_{OD,des}). \tag{3.4}$$

$T_{OD,des}$ is the outdoor design temperature, and ϵ_{recup} is the efficiency of the recuperator. Now the total design heat demand of the building can be found from:

$$\dot{Q}_{tot,des} = \dot{Q}_{rad,des} + \dot{Q}_{vent,coil,des} \tag{3.5}$$

The values used in the design point heat calculations are assumed values, except for the properties of air. They are presented in Table (3.2).

Table 3.2: Values for calculation of design point heat demand

$c_{p,air}$ [56]	1.006	kJ/kgK
ρ_{air} [57]	1.204	kg/m^3
$\Delta T_{coil,room}$	5	°C
ϵ_{recup}	0.8	

3.2.2 Building heat demand - Off design

There are very few days, if any, where the outside temperature reaches as low as the outdoor design temperature of -12°C. Therefore, transmission and infiltration loss, Q_{loss}, at different outside temperatures has to be known. The plot assumes that the loss at the outdoor design temperature of -12°C is 325 kW and 0 kW at the indoor design temperature of 22°C. These numbers are used to make the linear plot shown in Fig. (3.3). Note that from the figure, it is also assumed that the losses become negative at temperatures higher than the indoor design temperature, meaning that the temperature inside the building will rise above 22°C.

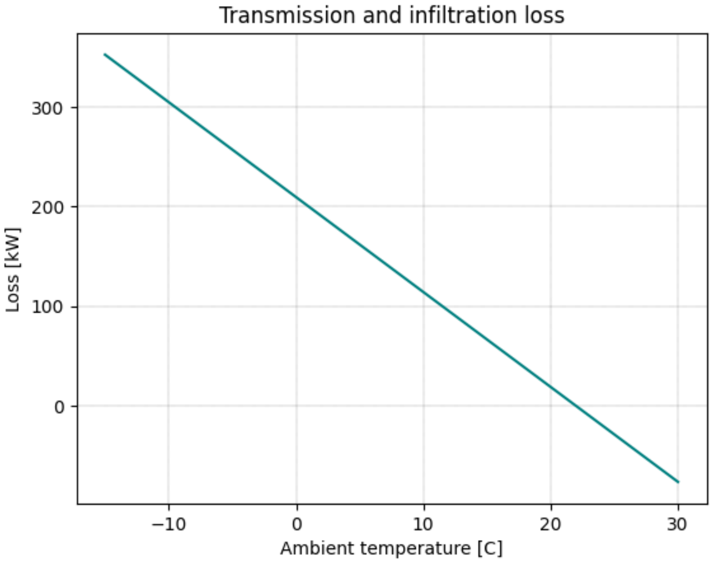


Figure 3.3: Transmission and infiltration loss as a linear function of the outside temperature.

Other load factors have to be considered as well, as they are considerable contributions to the heat budget; these are here called internal loads, $Q_{internal}$. When the building is in use, several internal loads must be included in the calculation. They are listed in Table (3.3).

Table 3.3: Internal loads

Internal load	Heat	Unit
People	80	W/person
Computers	100	W/person
Light	10	W/m^2
Miscellaneous	1	W/m^2

Heat loads from people, equipment, and light are the major heat sources in a regular office building. A miscellaneous load will also be included to account for heat that is not in one of the three categories and is assumed to be constant.

To make a more accurate calculation of the heat demand of the building, these internal loads are distributed into four different categories: weekdays daytime, weekdays nighttime, as well as weekend daytime, and nighttime.

Daytime is defined as from 07.00 to 19.00 and nighttime from 19.00 to 07.00. Weekdays are naturally defined as Monday to Friday, and weekends as Saturday and Sunday. The internal load distribution for the four categories is specified in Table (3.4). Day- and nighttime is denoted by D and N, respectively. In addition to the internal load distribution, ventilation intensity is included. It is denoted as I_{vent} and represents the relative rate of ventilation compared to the ventilation rate at its maximum capacity.

Table 3.4: Data for distribution of internal heat loads

Internal source	Workday D	Workday N	Weekend D	Weekend N
People	80%	2%	5%	0%
Computers	80%	2%	5%	0%
Light	100%	10%	15%	5%
Miscellaneous	100%	100%	100%	100%
Ventilation intensity	80%	5%	10%	0%

The data from Table (3.4) is used to calculate the total heat demand using equations (3.6)-(3.9).

The amount of heat required to raise the temperature of the air supplied from the ventilation system to room temperature, $Q_{undertemp}$, is given by

$$Q_{undertemp} = I_{vent} \times Q_{undertemp,des}, \tag{3.6}$$

where $Q_{undertemp,des}$ is found using Eq. (3.1), and ventilation intensity is read from Table (3.4). $Q_{undertemp}$ is then used to calculate the net radiator heat demand $Q_{net,\,rad}$

$$Q_{net,\,rad} = Q_{loss} + Q_{undertemp} - Q_{internal}, \tag{3.7}$$

where Q_{loss} represents the transmission and infiltration losses, which are read from Fig. (3.3) for the outside temperature at the time in question, and $Q_{internal}$ is the sum of all the internal loads at the same specific time.

The net ventilation heating coil demand is the heat required to heat the ventilation air to the 19°C at which it is delivered. This is the air that is supplied at an under-temperature of 3°C. To find this value at a given point, the design point net ventilation heating coil demand is multiplied by the ventilation intensity at the specific time.

$$Q_{net,\, vent,\, coil} = I_{vent} \times Q_{vent,\, coil\, ,des}. \tag{3.8}$$

In the end, to find the total heat demand for the hour in question, net radiator demand and net ventilation heating coil demand are added together.

$$Q_{total} = Q_{net,\, rad} + Q_{net,\, vent,\, coil}. \tag{3.9}$$

3.3 Evaporation temperature

3.3.1 Air as low-temperature heat source

The temperature data for Bergen is used to find the evaporation temperature for a heat pump system using the outside air as the heat source. The air flows directly over the evaporator, delivering heat. There is a temperature loss, or a reduction of the quality of the thermal energy. The amount of energy is the same, but in the heat transfer, there will inevitably be a loss of temperature or energy quality through any heat exchange. The air temperature will be reduced by 3°C. The temperature of the air is assumed to be reduced by another 3°C when heat is absorbed in the evaporator. This gives an evaporation temperature of 6°C below the outside air temperature.

3.3.2 Ground heat as low-temperature heat source

As stated in section (2.3.5), the ground temperature is assumed to equal the average outside temperature throughout the year. It is assumed that the mean temperature in the collector fluid is approximately 6°C below the ground temperature when the heat pump is working at full load. The collector fluid is assumed to have a temperature increase of 3°C through the entire length of the collector tube. Therefore, the temperature of the fluid out of the evaporator is estimated to be 7.5 °C below the round temperature, while the temperature of the fluid going into the evaporator has a temperature of 4.5°C below ground temperature.

The heat exchange in the evaporator is assumed to cause a temperature difference of 3°C. The final evaporation temperature is, therefore, 10.5°C below the ground temperature.

In summary, evaporation temperature in the ground source heating system is assumed to be more or less constant due to the stable temperature in the ground. When using outdoor air as a low-temperature heat source, the evaporation temperature depends on the outside temperature and is calculated based on the temperature data collected for Bergen.

3.4 Heat pump systems

For the comparison of the two low-temperature heat sources, two heat pump systems have been chosen. ABK-Qviller, one of the well-known retailers of heat pump systems in Norway, has provided system propositions and efficiency data. These are used in calculating operational costs and heat extraction from the ground.

The heat pump size was chosen based on the wanted heat pump performance. Heat pump performance was set to be 95% of the total annual energy demand. The desired heat pump capacity was then found to be 215 kW. This number was sent to ABK-Qviller, and two systems were proposed as the best options for comparison.

3.4.1 Air-source heat pump system

For the air-to-water heat pump, a machine from FrigoPlus has been chosen. The heat pump has a heating capacity of 208.4 kW. The electric power consumption of the compressor is at heating capacity 64.3 kW. Efficiency in this selected working point is 3.03. Evaporation and condensation temperature are set to -3.4°C and 59°C, respectively. The refrigerant used is propane, R290. The inlet/outlet temperature of the heating medium is 45°C/55°C out of the condenser/radiator system. The information about the air-to-water heat pump system at maximum capacity is summarized in Table (3.5)

Table 3.5: Information on FrigoPlus.

	Value	Unit
Heating capacity	208.4	kW
Cooling capacity	144.1	kW
Electrical input	64.3	kW
COP heating	3.03	-
COP cooling	2.24	-
Evap./Cond. temperature	-3.4/59	°C
Inlet/outlet temperature	45/55	°C

3.4.2 Ground-source heat pump system

The heat pump system used in the analysis of the ground-source system is from Enrad, and it consists of two modules HP700. The system provides a nominal heating capacity of 183.4 kW and electrical input of 43.8 kWh, resulting in a coefficient of performance equal to 3.42. The maximum heating capacity is 236.5 kW, and efficiency is reduced to 2.93 when operating at full capacity.

The refrigerant and the radiator distribution system's inlet/outlet temperatures are the same as for the proposed air-source system.

The heat is retrieved from boreholes. The borehole system consists of 36 wells with a depth of 210 meters, and the wells are placed in a square formation, and the distance between each well is 26 meters. In addition to the ground source heating system, some of the peak load has to come from another source, like district heating or electricity. Key properties of the system are summarized in Table (3.6).

Table 3.6: Information on Enrad HP700.

Number of modules	2	-
MAximum heating capacity	236.5	kW
Electrical input (max)	80.8	kW
COP (max)	2.93	-
Nominal heating capacity	183.9	kW
Electrical input (nom)	53.8	kW
COP (nom)	3.42	-
Maximum cooling capacity	167.9	kW
Nominal cooling capacity	129.5	kW
Inlet/outlet temperature	45/55	°C

3.5 Energy wells

To ensure sufficient heat extraction from the ground and simultaneously prevent the temperature in the ground from becoming too low, it is essential to consider both energy and power extraction when sizing the energy wells. Assuming the temperature in the collector to have an average temperature of 6 degrees below the average temperature in the ground and ground properties as stated in Table (3.7), one can determine the required size of the ground heat exchanger.

Table 3.7: Properties of the ground.

Property	Symbol	Value	Unit
Temperature of the ground	T_g	8	°C
Temperature in collector	T_{cf}	2	°C
Density of the ground	ρ	2500	kg/m^3
Specific heat capacity	c_p	900	J/°Ckg
Volumetric heat capacity	c_v	2.2	MJ/m^3K
Thermal conductivity	κ_g	2.5	W/m°C
Thermal diffusivity	α	$1.1 \cdot 10^6$	m^2/s
Total conductivity, ground to collector [11]	U	3.58	W/m°C
Geothermal heat flux	q_{geo}	0.06	W

3.5.1 Sizing based on energy extraction

As explained in Section (2.3.5), the volume of the ground surrounding the energy wells must be sufficient to sustain an acceptably high temperature in the ground. The volumetric heat capacity is used to quantify the amount of heat stored in the ground, in addition to the annual energy extraction.

Ideally, the ground temperature should not decrease at all, as the heat pump's efficiency will decrease with decreasing evaporation temperature. This, however, is not realistic in this case, as the heating demand is significantly higher than the cooling demand. Therefore, it is here chosen to tolerate a reduction in the temperature of $1^{\circ C}$ every ten years. The required thermal capacity is then given as follows:

$$Thermal\ capacity = Annual\ energy\ extraction \times 10 years \tag{3.10}$$

The volume corresponding to the assumptions made will then be given as

$$Volume = \frac{Thermal\ capacity}{Volumetric\ heat\ capacity} \tag{3.11}$$

The ground volume needed to supply enough energy is then used to find the surface area required for the energy well system. This is done by assuming a depth of 250 meters, 210 meters for each well, and 40 meters below.

3.5.2 Sizing based on power extraction

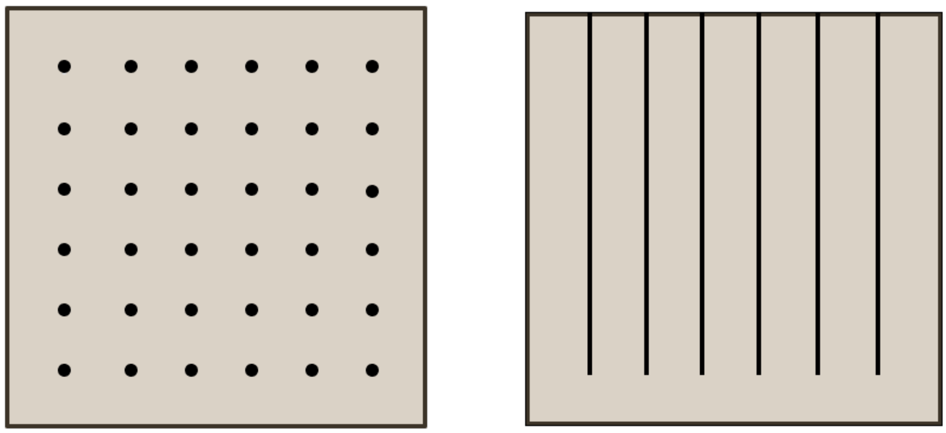
The total borehole length is important to ensure that the power extracted from the energy wells is sufficient to meet peak load. A simplified and approximate method of calculating the total borehole length is using Eq. (2.14), in the rewritten form:

$$L = \frac{Q_{ex,max}}{U(T_g - T_{cf})} \tag{3.12}$$

3.6 Temperature development in the ground

A Fortran program was written to understand better the temperature development underground and the impact of cooling on the ground. The ground heat exchanger is a three-dimensional system. Still, due to the complexity of the system and the restricted time frame of this thesis, it was chosen to use two 2D representations for this analysis. One is the ground heat exchanger seen from above, now called the horizontal plane, and the other is seen from the side, called the vertical plane. Sketches of the two representations are shown in Fig. (3.4).

To be able to compare the temperature development, four different systems were made. Two systems represent the horizontal plane and two the vertical plane. For each 2D representation, calculations were made for two scenarios; one that solely accounted for heat extracted from the ground and one where the cooling demand is subtracted from the total annual extracted energy. This highlights the effect that re-injecting heat to the ground has on ground temperature and, consequently, the overall efficiency of the heat pump system. Ground properties used in the calculations are displayed in Table (3.7).



(a) (b)

Figure 3.4: Sketches of (a) the system seen from above, and (b) the side.

3.6.1 The horizontal plane

Values used to build the program for the horizontal plane are presented in Table 3.8.

Table 3.8: Dimensions, horizontal numerical model.

	Symbol	Value	Unit
Distance between grid points	Δx, Δy	1	m
Distance between each well	n_{dist}	26	grid points
Surrounding ground	n_{surr}	18	grid points
Time step	Δt	3600	s
Lifetime	-	25	years

Creating the initial grid for the horizontal plane

The initial grid was established by setting the dimensions to be N×N, where N represents the number of grid points in each direction. N is calculated using Eq. (3.13). As seen from Table (3.8), the distance between two grid points, in both x and y direction, Δx and Δy, was set to be equal to 1 m. From this N becomes:

$$N = n_{surr} \times 2 + n_{dist} \times (6 + 1) + 1, \tag{3.13}$$

where n_{surr} is the width of the frame that represents the surrounding ground that is outside the calculated heat exchanger size, described in Section (3.5.1), this additional surrounding ground is there to represent the heat transfer from the surroundings and is the part of the grid that is outside the green frame in Fig. (3.5). n_{dist} is the number of grid points representing the distance between wells, which is multiplied by the number of wells plus one so that there are n_{dist} grid points on both sides of the outermost wells. Values for both n_{surr} and n_{dist} are read from Table (3.8.

When the grid dimensions were calculated, the initial temperature in all grid points was equal to the assumed ground temperature. The annual average temperature for 2022, and thus the approximate temperature of the ground, was calculated to be 8.9°C. To further ensure that the ground source heat exchanger is of sufficient size, it was chosen to use 8°C as the ground temperature.

To represent the 36 boreholes, the initial temperature in 36 of the grid points was changed to be 6°C below ground temperature, as described in Section (3.3.2). The outermost wells were placed at a distance of $n_{surr} + n_{dist}$ grid points from the grid boundary. Further, there were n_{dist} grid points between the wells in the x and y directions. The initial grid is shown in Fig. (3.5), where the wells are represented as green dots. In the figure, each grid line represents two grid points.

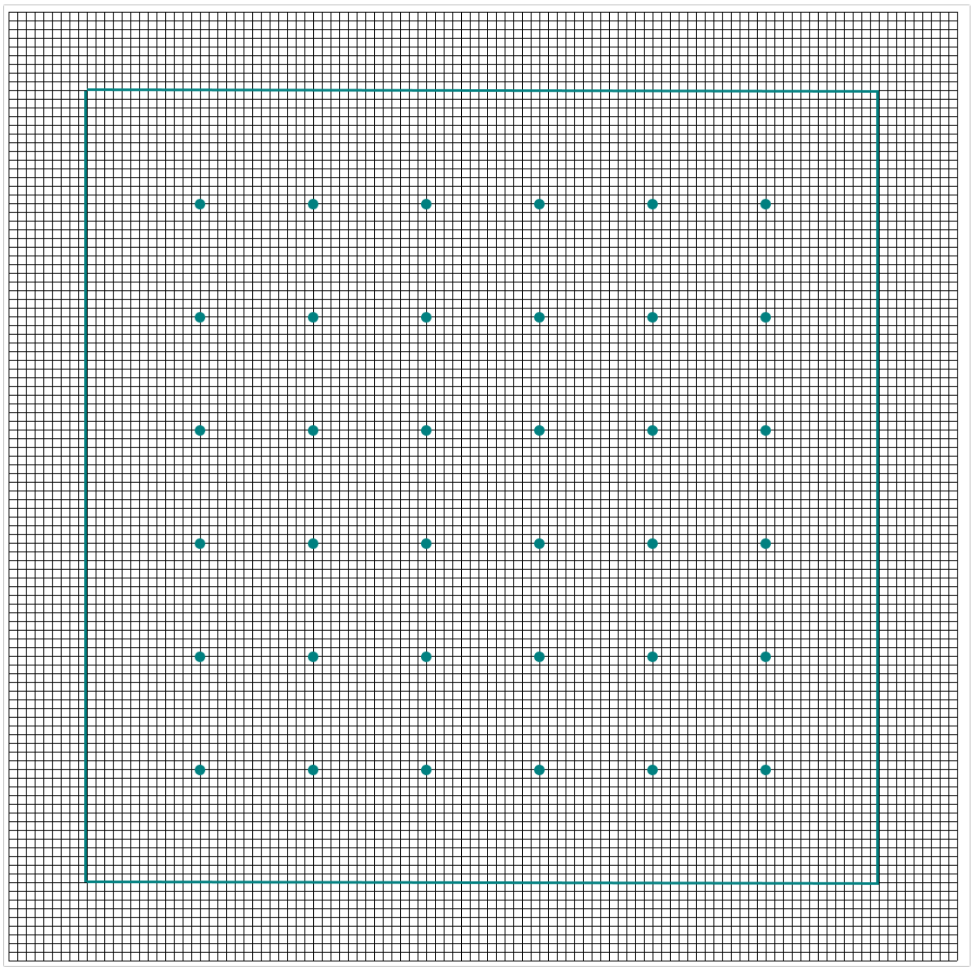


Figure 3.5: Initial grid for calculations in the horizontal plane. Each grid line represents two grid points.

Calculations in the horizontal plane when cooling to the ground is not included

As stated in Section (3.3.2), the temperature in the collectors is assumed to be 6°C below the annual average temperature at full load. Therefore hereafter, full load equivalent hours is the term used to describe how many hours the system would have to work at full load to account for the total heat extracted from the ground over a year. The seasonal temperature fluctuations are neglected as the calculations are done over 25 years. Full load equivalent hours are found by dividing total extracted energy over a year, $E_{ex,tot}$, by the maximum heating capacity of the heat pump, $Q_{hp,max}$:

$$Full\ load\ equivalent\ hours = \frac{E_{ex,tot}}{Q_{hp,max}}. \tag{3.14}$$

In the horizontal plane, no other heat fluxes than the one described by Eq. (2.20) were included in the calculation. The equation was used for calculations by looping over all grid points. The temperature difference in each point was saved in a separate grid until all calculations within the same time step were done. This is because the calculations depend on the adjacent grid points, and adding the temperature difference while still in the same time step would thus give incorrect results. After iterating through all grid points, the temperature difference in each point was added to the main temperature grid.

The time step was set to 3600 s. The calculations were looped over the full load equivalent hours and further over the 25 years that is assumed to be the heat pump's lifetime. After every time step, the temperature difference in the grid points that are the boreholes are set to be zero. This is to keep the temperature of the collector fluid constant at 2°C. To get valid values around the boundary, the values at the grid boundary were set to equal the temperature in an adjacent grid point within the boundary. The code that was used is presented in Appendix A.

Calculations in the horizontal plane when cooling is included

When cooling of the building is included, calculations are performed using the exact same method as before, but changing the number of full load equivalent hours. Again, due to the long time frame of the calculations, fluctuations are neglected. Therefore, the net energy extracted from the ground is used to calculate full load equivalent hours.

$$Full\ load\ equivalent\ hours = \frac{E_{ex,net}}{Q_{hp,max}}, \tag{3.15}$$

where $E_{ex,net}$ is given as

$$E_{ex,net} = E_{ex,tot} - E_{re,tot} \tag{3.16}$$

where $E_{re,tot}$ is the total annual cooling demand and corresponding energy input.

3.6.2 The vertical plane

Values used to build the program for the vertical plane are presented in Table 3.9.

Table 3.9: Dimensions, vertical numerical model.

	Symbol	Value	Unit
Distance between grid points	Δx, Δy	1	m
Distance between each well	n_{dist}	26	grid points
Surrounding ground, sides	n_{side}	30	grid points
surrounding ground, below	n_{below}	39	grid points
Time step	Δt	3600	s
Life time	-	25	years

Creating the initial grid for the vertical plane

The vertical system was built similarly to the horizontal system, but rather than assigning the collector temperature to 36 single grid points, all grid points in vertical columns were set to 2°C. Therefore, the system does not take all 36 wells into account but only one of the sides, equating to six wells.

The initial grid was made by setting dimensions N1×N2, where N1 and N2 are the numbers of grid points in the y and x directions, respectively. They are calculated using Eq. (3.17) and (3.18). As for the horizontal plane, and as can be read from Table (3.9), the distance between two grid points, Δx and Δy, is 1 meter. Thus, N1 becomes

$$N1 = n_{depth} + n_{below} + 1, \tag{3.17}$$

where n_{depth} is the number of grid points corresponding to the depth of the wells and n_{below} is the number of grid points below the wells. Finally, we compute N2 from

$$N2 = n_{side} \times 2 + n_{dist} \times (6 + 1) + 1, \tag{3.18}$$

where n_{side} is the number of grid points that make up the additional ground on the sides of the heat exchanger, and n_{dist} is the number of grid points between each well. Table (3.9) presents all values used to make the initial system.

Further, the temperature in all grid points is initially set to be equal to 8°C. Then, to represent the entire length of the well, the temperature of 6 columns is changed to be 2°C. The temperatures are changed from the top of the grid, representing the surface, to a length corresponding to 210 meters, which is the depth of the well (see Table (3.9)).

The initial grid in the vertical plane is shown in Fig. (3.6). The lighter green lines show the boundaries of the calculated size of the heat exchanger, and the darker green lines represent the energy wells with a temperature of 2°C.

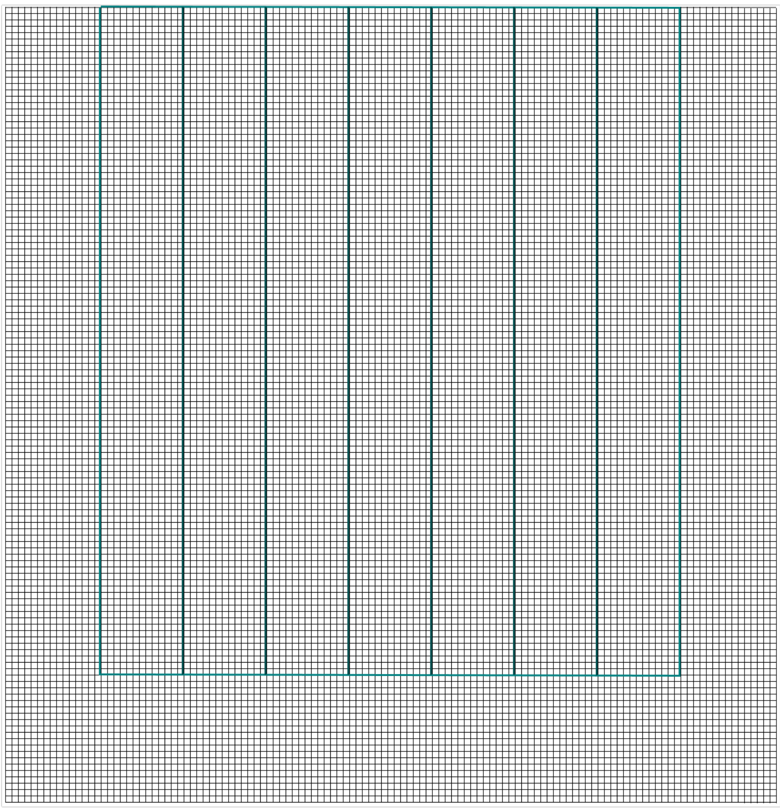


Figure 3.6: Initial grid for calculations in the vertical plane. Each grid line represents two grid points.

Calculations in the vertical plane

Calculations in the vertical plane were made similarly to those in the horizontal plane. In addition to the heat transfer described by Eq. (2.20), a heat flux was added to account for the geothermal heat from below the wells. Eq. (3.19) was used to calculate the temperature added from the heat transfer from below by adding the temperature difference to the temperature in the bottom grid points.

$$\Delta T_{bottom} = \frac{q\Delta t}{\rho c_v \Delta y} \tag{3.19}$$

Solar radiation also contributes to energy being replenished to the ground. Another simplification done, as a consequence of neglecting the seasonal fluctuations, was setting the surface temperature to be constant. Over the year, the average surface temperature would be approximately constant and equal to the annual average temperature.

Looping over the full load equivalent hours, both when including cooling and not, was done in the same manner as described for the horizontal system. The code used in the calculations for the vertical representation is presented in Appendix B.

The Courant-Friedrich-Levy stability criterion from section (2.4.4) is checked to ensure the error does not grow over the chosen time frame.

3.7 Economy

If the comparison of the costs of the air source and the ground source heat pump systems are to be reasonably accurate, the costs of both investment, operation, and maintenance should be included in the calculation.

3.7.1 Overview

All the costs associated with a ground source heat pump system can be allocated between a series of categories:

- **Heat pump unit:** Includes the investment cost of the heat pump and required equipment.
- **Ground heat exchanger:** Includes the cost of drilling boreholes, the collector tubes, casings, and other equipment needed for the heat exchanger.

- **Distribution system:** Includes the cost of the distribution piping, ducts, and necessary pumps, valves, controls, etc.
- **Installation costs:** Includes the costs of labor and equipment associated with installing the system.
- **Maintenance and operating costs:** This includes the cost of routine maintenance and energy costs associated with operating the system.

The costs associated with air-source heat pumps are allocated into these categories:

- **Heat pump unit:** Includes the investment cost of the heat pump and required equipment.
- **Distribution system:** Includes the cost of the distribution piping, ducts, and necessary pumps, valves, controls, etc.
- **Installation costs:** Includes the costs of labor and equipment associated with installing the system.
- **Maintenance and operating costs:** This includes the cost of routine maintenance and energy costs associated with operating the system.

Some categories include equal costs for both air-source and ground-source heat pumps and will thus not influence any choices of which heat pump system is most feasible economically. The distribution system is assumed to be identical for the two systems, and installation costs are also assumed to be irrelevant to this thesis and are therefore not considered.

3.7.2 Cost ground-source heat pump system

The total cost of the ground-source heat pump system is divided into two main categories. Investment costs include the cost of the heat pump and the energy wells and running costs, which consist of annual maintenance and electricity costs.

Investment costs

In addition to efficiency data, ABK-qviller has provided a budget price of 1,200,000 NOK for the heat pump described in section (3.4.2). A price estimate for another project using ground heat through vertical boreholes is used to estimate the cost of the energy wells. The prices used to calculate the cost of the energy wells are shown in Table (3.10)

Table 3.10: Energy well cost budget. All prices are in NOK

Component	Price per unit [NOK]	Unit
Access and mobilization	45,000	Park
Drilling	250	Meters borehole
Iron casing	5100	Well
Casing shoe	1750	Well
TurboCollector 210 m	26,500	Well
Container for residue	11,000	Park
Lockable lids	2500	Well

Running costs

The running costs are annual maintenance cost and annual operating cost. Annual maintenance cost for the ground source heat pump system is assumed to be 2% of the price of the heat pump itself. Using the budget price from ABK-qviller, this would be an annual cost of 24,000 NOK.

Electricity prices fluctuate and have increased significantly in the last few years. Assuming they will stay elevated from earlier times but stabilize, this thesis assumes an average of 2 NOK/kWh. Using Eq. (2.7) and rewriting it as shown in Eq. (3.20), the annual energy cost can be found using the calculated heat demand.

$$W = \frac{Q}{COP}, \tag{3.20}$$

3.7.3 Cost Air-source heat pump system

As for ground source heat pump systems, the total cost of the air source heat pump system is divided into the categories of investment costs, which are the cost of the heat pump, and annual costs, which consists of yearly maintenance costs and electricity costs.

Investment costs

The budget price received from ABK-qviller for the heat pump described in Section (3.4.1) is 1,250,000 NOK. Somewhat higher but still similar to the ground source heat pump. Investment costs for the system using air as a low-temperature heat source are low compared to the ground source, as the energy wells are unnecessary.

Annual costs

The maintenance costs for the air-source heat pump system are assumed to be higher than for the ground-source heat pump, and this is due to the fluctuating evaporation temperatures, making it work harder. Here it is assumed that the annual maintenance costs are 3% of the investment cost, which equals 37,500 NOK.

The energy input to the compressor was found the same way as the ground source heat pump, using Eq. (3.20). For the air source heat pump, the other components that require energy are not considered as they were for the system using ground heat. Table (3.11) shows the data used to calculate the total power required. As described in the last section, the electricity price was assumed to be 2 NOK/kWh.

Table 3.11: Electrical power consumption Air-source heat pump.

Component	Power [W]	When
HP in off-mode	100	Off-mode
Crankcase heater on	800	Off-mode
Thermostat off	200	Off-mode
Total fan power	4430	On-mode

3.7.4 Cost comparison

To make the economic comparison of the systems, the total cost over the lifetime of each system and simple payback time, as explained in Section (2.5), will be used. The total cost is a good measure, but the simple payback time of the systems could give another perspective. Eq. (2.22) is used to calculate simple payback time.

Chapter 4

Results and discussion

4.1 Heat demand

4.1.1 Building heat demand at design point

At design point, meaning outside temperature -12°C, ventilation at full load, and no internal loads, the heat demand was calculated as described in Section (3.2.1). The results are presented in Table (4.1). The radiator system has to account for both the transmission and infiltration losses and to heat the ventilation air that is assumed to be delivered at an under-temperature of 3°C. The ventilation heating coil heats the air in the ventilation system to 19°C. According to the calculations, the total heat demand at the design point is 655 kW.

Table 4.1: Building heat demand at design point.

	Notation	**Heat demand [kW]**
Transmission and infiltration loss	$Q_{loss,des}$	325
Under-temperature	$Q_{undertemp,des}$	205
Ventilation heating coil	$Q_{vent,coil,des}$	125
Total building heat demand	$Q_{tot,des}$	**655**

The total heat demand at design point is never reached, as the outside temperature in Bergen very rarely drops to -12°C. The temperature curve in Fig. (3.1) shows that it never happened in 2022. Even if it occurred, there would be internal heat sources in the building if it happened in the daytime on a weekday. Alternatively, the ventilation rate would be very low, sometimes zero, if it happened outside of the office times.

4.1.2 Building heat demand - Off design

As the heat demand at design point is never reached, it is more interesting to look at the actual heat demand. Fig. (4.1) shows the actual heat demand over the entire year, calculated as described in Section (3.2.2). Here, the outside temperature calculates the transmission and infiltration in the given hour, and internal loads are considered. The results are categorized after the time of day and week and sorted from highest to lowest heat demand.

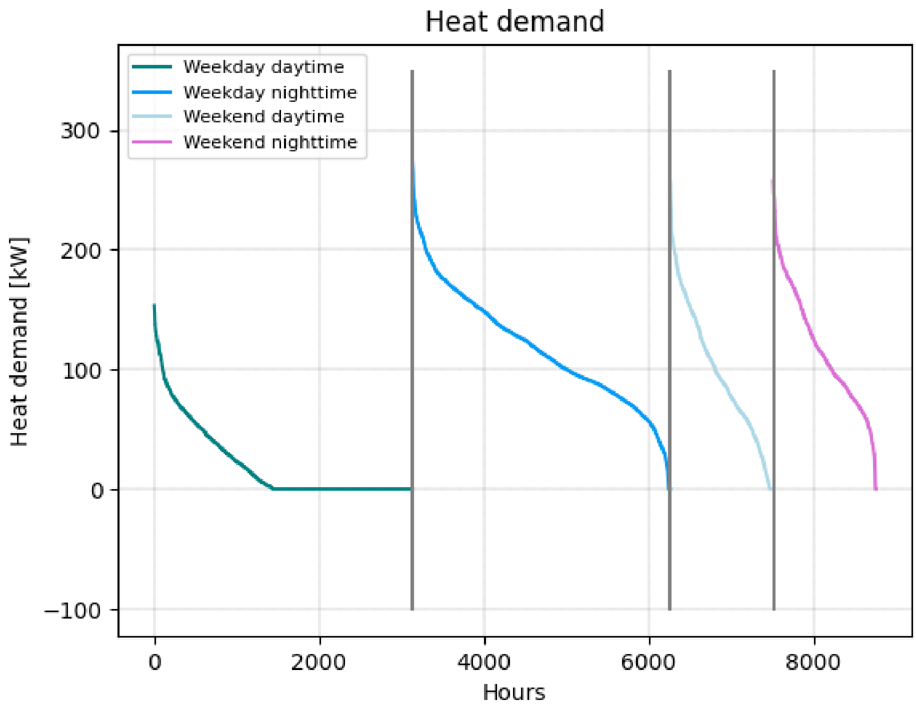


Figure 4.1: Total heat demand, distributed over the week and time of day. Internal loads are included.

The significance of the internal loads is clearly seen in Fig. (4.1). The heat demand is generally lower in the daytime on weekdays, when the heat load from people and computers has their highest contributions. The temperature being generally higher in the daytime could contribute as well. Still, looking at the curves for weekend days and nights, the difference between the two does not indicate that the slightly higher temperature would have a significant effect. Outside ordinary office times, the heat demand is substantially higher when internal heat loads are small.

The heat pump performance, being 95% of the annual energy demand, is calculated to be 692 GWh. This results in a desired peak heating and cooling capacity of 215 kW and 158 kW. According to the information retrieved from ABK-qviller, the highest heating and cooling power for the ground source heat pump system is 236.5 kW and 167.9 kW, respectively. This is in the desired range. The air source system's highest heating and cooling capacity is 208.4 kW and 144.1 kW, slightly under the requested capacity but still within a reasonable range. This information is summarized in Table (4.2), where heating and cooling capacity for ground source and air source heat pump systems are denoted as GSHP and ASHP, respectively.

Table 4.2: Heat pump performance, highest heating and cooling capacity for the ground source heat pump.

Heat pump performance	692060	kWh
Maximum heating demand	215	kW
Maximum cooling demand	158	kW
Maximum heating capacity GSHP	236.5	kW
Maximum cooling capacity GSHP	167.9	kW
Maximum heating capacity ASHP	208.4	kW
Maximum cooling capacity ASHP	144.1	kW

One issue that must be discussed is the depletion of the ground heat reservoir. Decreasing temperature in the ground leads to a decrease in evaporation temperature, hence lower efficiency for the ground source heat pump. Ideally, one would like the temperature in the ground to be constant, and this would require that the amount of heat extracted is also replenished. Without any artificial means to restore the heat, this is impossible in Norway, as the heating demand is much higher than the cooling demand. The cooling demand is shown in Fig. (4.2). The annual heating and cooling demand and the heat extracted from and injected into the ground are summarized in Table (4.3).

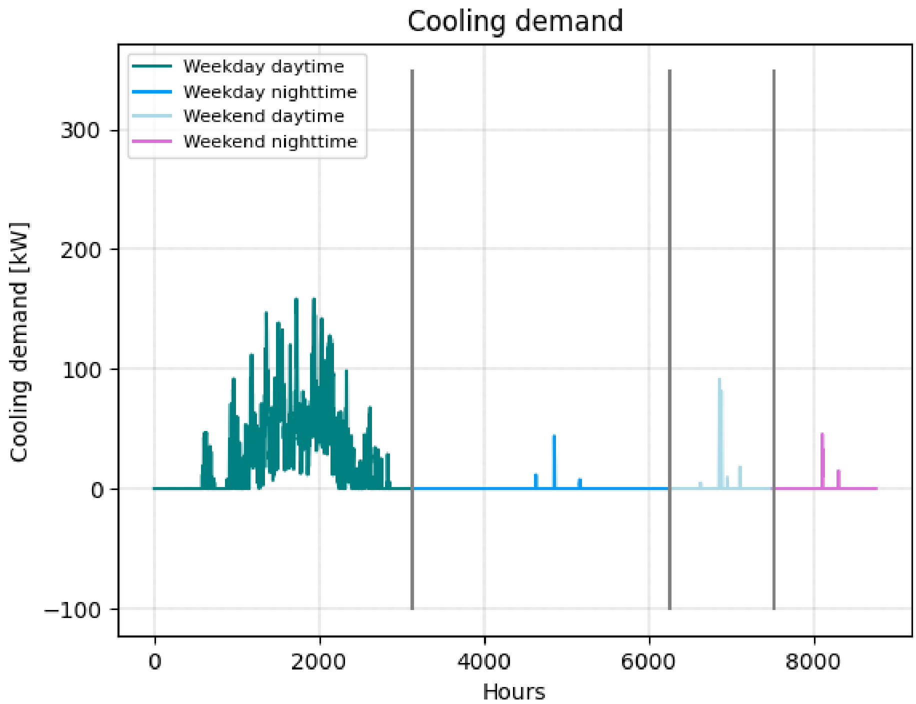


Figure 4.2: Cooling demand distributed over the week and time of the day.

When cooling demand is included, the temperature in the ground would not be constant, but it would slow the depletion of the reservoir, which is desirable. This would also mean that the heat pump's efficiency is higher than it would be if cooling of the building were done in any other way. If the temperature in the ground were constant, a way of pumping heat into the ground would have to be installed. This could, for instance, be a solar collector.

Table 4.3: Heating demand, cooling demand, heat extracted from and injected to the ground.

Total annual heating demand	728,500	kWh
Total annual cooling demand	82,500	kWh
Energy extracted from the ground	507,200	kWh
Energy injected to the ground	119,700	kWh

4.2 Sizing of the energy wells

As explained in sections (2.3.5) and (3.5), sizing the energy wells should be done considering both the annual energy extraction and the peak heat load. More precisely, both the ground volume and the total borehole length must be calculated.

4.2.1 Sizing based on energy extraction

How much energy that is extracted from the ground decides the required volume of subsurface heat storage. The calculations are done both with and without, including cooling.

No cooling

The results of the calculations, using equations (3.10) and (3.11), and values from Table (3.7), are presented in Table (4.4). The total required volume of the reservoir is calculated to be $8.3 \cdot 10^6$ m^3. If the volume is divided by 250, that is, the depth of the wells, 210 meters, plus an additional 40 meters under the wells, the surface area needed is $3.32{\cdot}10^4$ m^2. Assuming a square formation, this corresponds to 180 × 180 meters.

Table 4.4: Sizing of the energy wells based on energy extraction.

Annual energy extraction	1.826×10^{12}	J
Thermal capacity	1.826×10^{13}	J
Volume	8.3×10^{6}	m^3
Ground area	3.32×10^{4}	m^2

Cooling included

The only difference between the calculations with and without cooling is the net amount of energy extracted from the ground. When cooling is included, the annual cooling demand, and required input for cooling, are subtracted from the amount of energy extracted to satisfy the heating demand. The results from the calculations are presented in Table (4.5).

The total required volume of the reservoir is calculated to be 1.530×10^{12} J. Thus, the area is calculated to be 2.8×10^4 m^2. The square formation will then have sides of 166 meters, which is somewhat smaller than when cooling is not considered. It must be noted, that in cities, where the population is steadily increasing, using as little of the surface as possible is preferable.

Table 4.5: Sizing of the energy wells based on energy extraction, cooling included.

Annual energy extraction	1.530×10^{12}	J
Thermal capacity	1.530×10^{13}	J
Volume	7.0×10^{6}	m^3
Ground area	2.8×10^{4}	m^2

4.2.2 Sizing based on power extraction

Using Eq. (3.12), the total borehole length required to meet peak load is calculated to be 7263 meters. Using boreholes with a depth of 210 meters, 35 energy wells are needed. This results in a heat extraction rate of approximately 22 W per meter borehole. This number is in the lower aspects of the extraction rates presented in the paragraph on rules of thumb in the introduction of this thesis. This would make sense, as the annual average temperature in Norway, and thus the temperature in the ground, is relatively low compared to other locations. A summary of the results is presented in Table (4.6).

Table 4.6: Sizing of the energy wells based on power extraction.

Maximum extraction of heat	156	kW
Spacing between boreholes	10	m
Heat per meter borehole	22	W/m
Calculated number of boreholes	35	-

The required number of energy wells is calculated to be 35. In this calculation, it is assumed that the temperature in the ground remains constant. As this is not the case, another energy well is added to ensure the peak heat demand is met. The ground heat exchanger will consist of 36 wells in a 6 $\times$ 6 square formation. To ensure the ground's temperature level remains reasonable over time, it is helpful to look at the temperature development in the subsurface. This is done in the following.

4.3 Temperature development in the ground

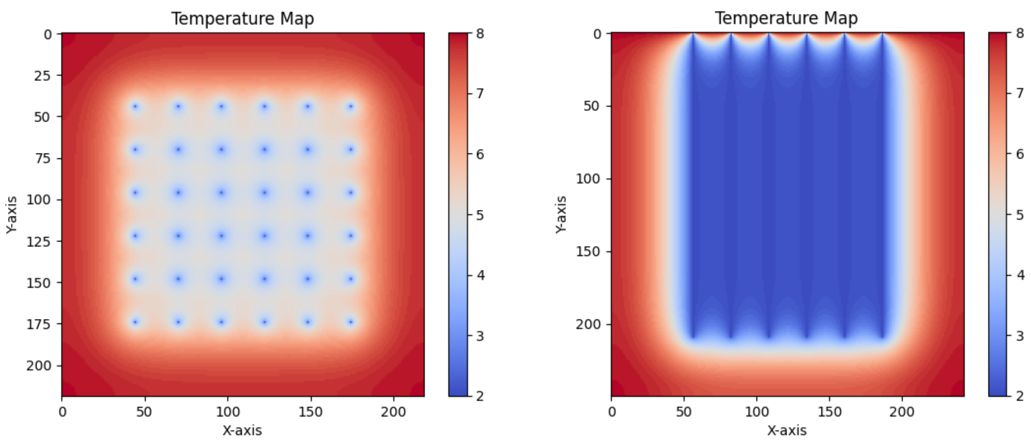
The average temperature was set to decrease 1°C over the entire volume of the ground every ten years, as described in Section (3.5.1). Consequently, the temperature of the ground close to the wells has decreased more, and the ground further from the wells less. This is expected, as the heat transfer between the ground and the wells is higher than the heat transfer within the ground.

A numerical analysis of the temperature development in the ground was performed as described in Section (3.6). The iterations were done for peak operation equivalent hours, including cooling and without subtracting the cooling demand. They are presented in Table (4.7).

Table 4.7: Full load equivalent hours.

no cooling	2926
cooling	1638

Temperature maps of the ground, both for the horizontal and vertical systems, as well as with and without cooling, are shown in Figures (4.3) and (4.4). Comparing the two systems, the effect of using a heat pump system used for cooling and heating is apparent.



(a) (b)

Figure 4.3: Temperature maps for the ground when cooling is not implemented seen (a) horizontally and (b) vertically.

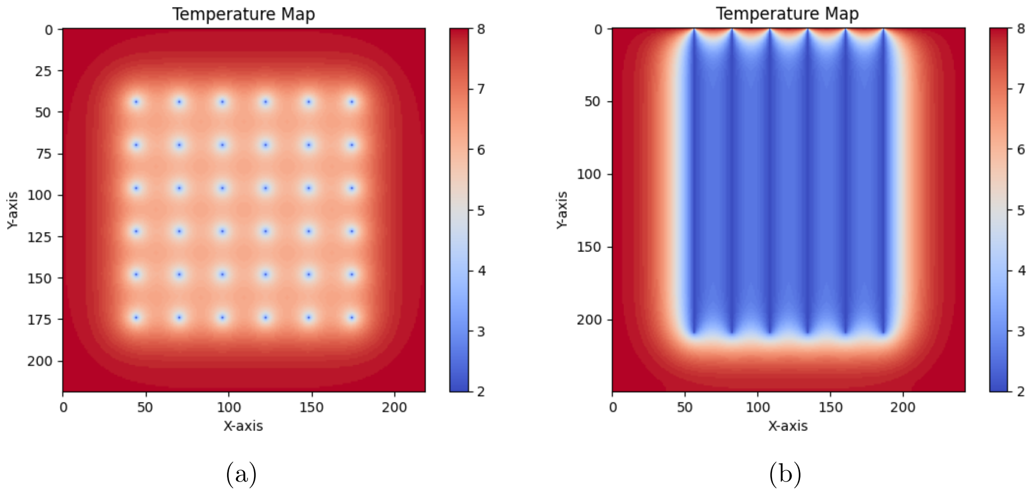


Figure 4.4: Temperature maps for the ground when cooling is implemented seen (a) horizontally and (b) vertically.

Figures (4.3) and (4.4) show significant differences in temperature for two systems that should be the same. As mentioned in Section (3.6), each system was made into two 2D grids/systems. This will not give correct numbers, and the actual temperatures between boreholes can be assumed to be somewhere between the ones found for each heat pump system.

These deviations from the actual situation are due to how the programs are built. The temperature in a grid point is found based on the adjacent grid points. The program for the horizontal view has only 36 grid points representing the energy wells, while the vertical system has entire columns.

One would also have to consider that by iterating over the peak load equivalent hours, the heat inflow from the surrounding ground is incorrect. The natural heat inflow will be higher than shown in the four temperature maps due to the remaining hours not being put into the program.

The calculations of temperature distribution in the ground have been done, despite the incorrect assumptions and resulting values. This is, as mentioned, to give a better view of the importance of cooling. It also shows that second-guessing all the rules of thumb is necessary, as small differences in heat demand, and thus heat extraction and injections have significant consequences. Obtaining precise results for the evolution of ground temperature was not the primary objective of the thesis and was, therefore, not of the highest priority.

4.4 Economic assessment

The economic comparison of ground and air source systems was performed as described in Section (3.7).

4.4.1 Cost ground-source heat pump system

The investment cost of the boreholes is calculated in Table (4.8), using the computed number of wells and total borehole length, as well as the prices from the price offer used as reference. The borehole cost and the other costs of the ground source heat pump system are presented in Table (4.9). Investment costs consist of the cost of the heat pump, as well as the cost of the boreholes. Annual maintenance costs and operation costs are running costs.

Table 4.8: Energy wells cost calculation. All prices are in NOK.

Component	Number of units	price per unit	total price
Access and mobilization	1	45,000	45,000
Drilling	7560	250	1,738,800
Iron casing	216	850	183,600
Casing shoe	36	1750	63,000
Turbo Collector 210m	36	26,500	954,000
Container for residue	1	11,000	11,000
Lockable lids	36	2500	90,000
Total			**3,085,400**

Table 4.9: Investment cost ground source heat pump.

Component	Cost [NOK]
Heat Pump	1,200,000
Borehole	3,085,400
Annual maintenance cost	24,000
Annual operating cost, no cooling	440,660
Annual operating cost, cooling included	514,660
Total	**4,184,720**

4.4.2 Cost air-source heat pump system

The air source heat pump system has much lower investment costs than the ground source system, and the only investment cost is the heat pump. On the other hand, from section (3.4.1), we see that the COP is lower than for the ground source system. Therefore, operating costs are higher. Table (4.10) shows the annual operations cost calculation. The yearly maintenance costs are also expected to be somewhat higher, 3% instead of 2% for the ground source system. The machine is run at a higher load for more extended periods of time due to the fluctuations in evaporation temperature.

These higher annual maintenance and operation costs must also be considered before discussing the most economically favorable solution. The cost of 15000 kWh for defrosting the machine is also added to the operation cost, which also adds to the higher operating cost of the air source heat pump system compared to the ground source heat pump system.

Table 4.10: Electrical power consumption air-source heat pump.

Component	**Power [W]**	**Hours**	**Energy [kWh]**
HP in off-mode	100	2897	289.7
Crankcase heater on	800	2897	2317.6
Thermostat off	200	2897	579.4
Total fan power	4430	5863	25,973.1
Defrosting			15,000
Heat pump			256,315
Cooling			36,731
Total energy, cooling			**337,206**
Total energy, no cooling			**300,475**

Table 4.11: Investment cost air source heat pump

Component	**Cost [NOK]**
Heat pump	1,250,000
Annual maintenance cost	37,500
Operating cost, cooling	674,463
Operating cost, no cooling	601,000

4.4.3 Economic assessment of the systems

When choosing a heat pump system, it is essential to consider all costs. Table (4.12) summarizes all costs associated with the two systems. The savings for each system is also calculated, and saved energy is the energy retrieved from the two low-temperature heat sources. Simple payback time is also presented.

Table 4.12: Investment, annual operational, and lifetime cost for four different systems. Costs are presented as NOK and payback time as years.

	GSHP C	**GSHP**	**ASHP C**	**ASHP**
Investment	4,285,400	4,285,400	1,250,000	1,250,000
Annual operation	538,660	464,660	711,963	638,500??
Lifetime	17,751,900	15,901,900	19,049,075	17,212,500
energy cost savings	1,014,485	1,014,485	942,518	942,518
simple payback time	17.5	15.7	20.2	18.3

The total costs of the systems through the lifetime of 25 years are presented in the linear plot shown in Fig. (4.5). The system that has the lowest costs is the ground source heat pump system. The cost difference between the two systems is approximately 1.5 million NOK, meaning the simple payback time for an air source heat pump system is about 1.5 years longer than for the ground source system.

The example building requires cooling independently of which heat pump system is chosen due to the significant cooling demand of the building. If the heat pumps are not used to cool the building, another cooling system must be implemented. Therefore, it would not be economically beneficial to use another cooling solution. Due to the minor cost differences in cooling for the two systems, further discussion will not differentiate between cooling or no cooling, only which low-temperature heat source or sink is used. In addition to the potential external cooling system's added costs, the ground source heat pump maintains its high efficiency by using the energy wells for cooling and heating purposes.

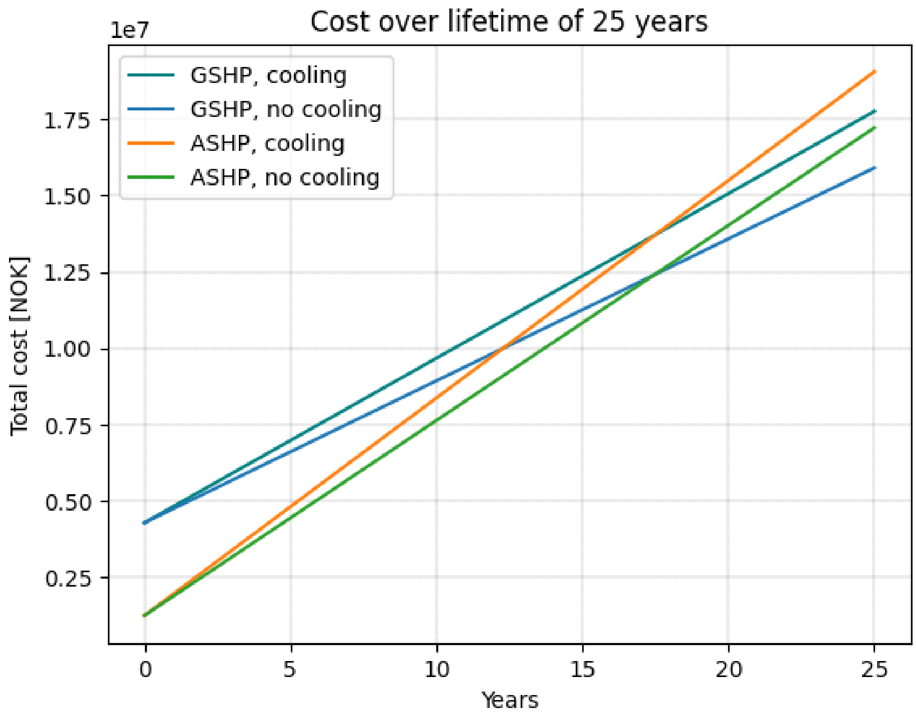


Figure 4.5: Cost of the four different systems, ground source, and air source, both with and without cooling.

4.5 Factors to consider when choosing a heat pump

As discussed in section (2.2.3), there are advantages and disadvantages for both ground-source and air-source heat pump systems. The main disadvantages of the air source system are that the efficiency decreases significantly at lower temperatures when the heating demand is at its highest and that at very low temperatures, the heat pump is shut off. For the particular heat pump discussed in this thesis, this temperature is -12°C. This is not an issue in Bergen, as the temperatures rarely reach this temperature, even in the midst of winter, but this could be a challenge in colder locations.

Other factors include investment costs against running costs, fluctuating electricity prices, and differences in available space. If the building in question is located in a densely populated area, a ground source heating system could be problematic, as the surface area needed for the energy wells is relatively large.

Location

To give an example of the significance that climate has when choosing a heating system, Karasjok, a small town in northern Norway, is chosen. Hourly temperature data was retrieved from Norsk Klimaservicesenter and sorted from low to high temperatures. The temperature curve for Karasjok is presented in Fig. (4.6). The sorted temperature curve shows how many hours during a year the temperature has been at a given level or below. Karasjok has an annual average temperature of 0°C. As can be seen from the temperature curve, the temperature was below 0°C for 4058 hours and below -12°C for 1339 [53]. Hours below -12°C have the highest heat demands, but the air source heat pump is very ineffective at these temperatures.

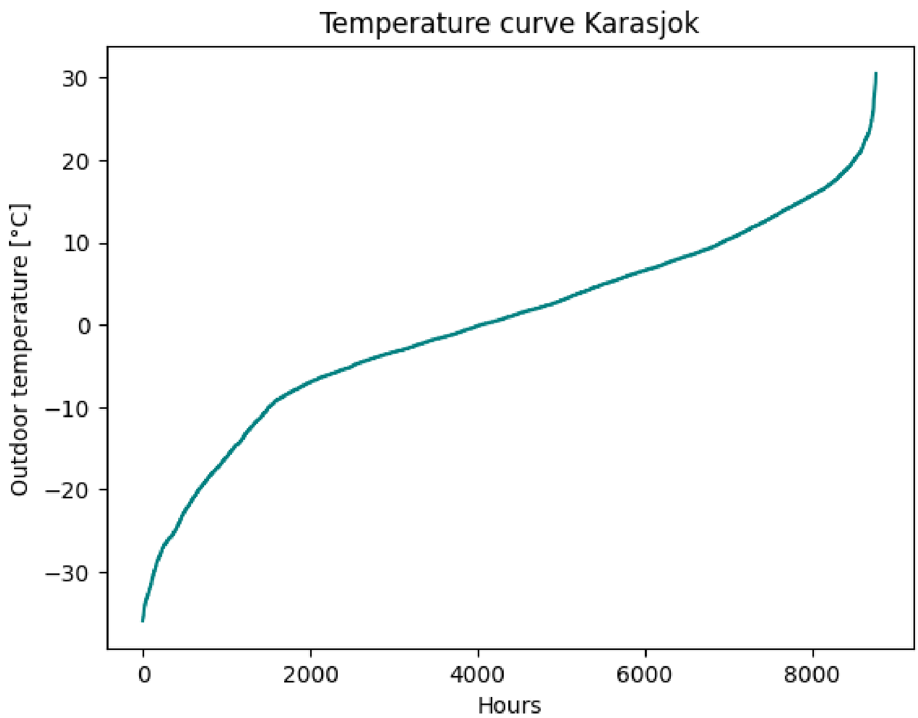


Figure 4.6: Temperature curve Karasjok sorted from low temperature to higher temperature.

Assumptions and data used to estimate the cost of an air source and a ground source heating system in Karasjok are summarized in Table (4.13). A bigger heat pump than the one used for the building in Bergen would be needed, and therefore the budget for the heat pump is raised to 2 million NOK. Similarly, one would need a more extensive ground heat exchanger, and the budget is increased to 5 million NOK. These are only rough estimates made based on the higher heating demand. Annual maintenance costs are found using the same method as for Bergen. And the cost of alternative heating for when the air source heat pump uses the energy demands for the hours in question.

Table 4.13: Data and assumptions Karasjok.

Lowest working temperature ASHP	-12	°C
Total annual energy demand	1,445,361	kWh
Cost heat pump	2,000,000	NOK
Cost boreholes	5,000,000	NOK
Annual maintenance cost GSHP	40,000	NOK
Annual maintenance cost ASHP	60,000	NOK
Operating cost GSHP	730,814	NOK
Operating cost ASHP	616,706	NOK
Cost alternative heating ASHP	948,258	NOK

The cost estimates for an air source heat pump system and a ground source system for the same example building used before are presented in Fig. (4.7). It is seen from the figure that the ground source heat pump system is the best choice if looking at the economy alone. The ground source heat pump system has a lower lifetime cost of more than 15 million NOK.

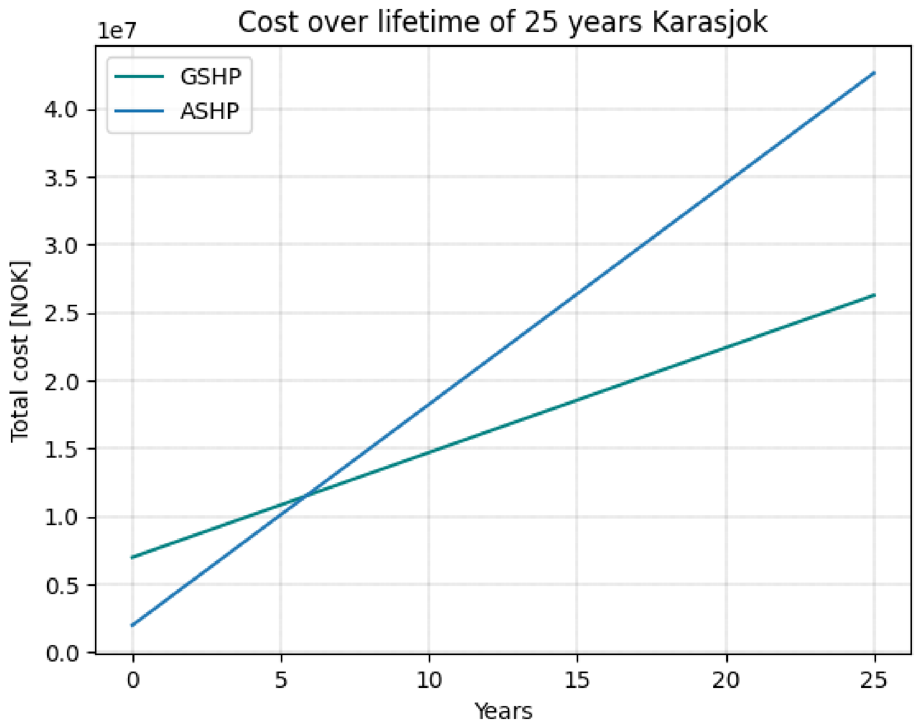


Figure 4.7: Cost over the lifetime for two systems located in Karasjok.

A ground source heat pump system is more impractical when the temperatures are this low. The temperature in the ground is just above the freezing point. Thus, the amount of heat stored in the ground is more limited. The annual incoming solar radiation is also lower than in Bergen, meaning that the amount of heat extracted from the ground is higher, and the amount of heat restored is much lower. Consequently, the ground volume, and thus area, would have to be bigger, the number of wells higher, and the spacing between the wells longer. It would be more challenging to design a sustainable well. However, it would still be economically beneficial if it was possible to do it reasonably and practically based on the available area at the chosen location.

Electricity prices

Comparing the investment and maintenance and operating costs from Table (4.12), the ground source heat pump's investment costs are almost four times the air source system's investment. On the other hand, the air source heat pump system's lifetime costs are higher. The critical difference is in the operating costs. The high electricity prices in the last period of time resulted in a significant increase in operating costs.

Statistics from 2010 to 2020 show that the average electricity prices for Western Norway in this period were less than 0.5 NOK/kWh, and in only a few months the electricity prices exceeded this. Still, they never were higher than 0.65 NOK/kWh [58]. This is more than four times less than the electricity price of 2 NOK/kWh used when calculating the costs of the heat pump system. If using 0.5 NOK/kWh when doing the cost calculations from Section (4.4), the lifetime cost development would be as shown in Fig. (4.8).

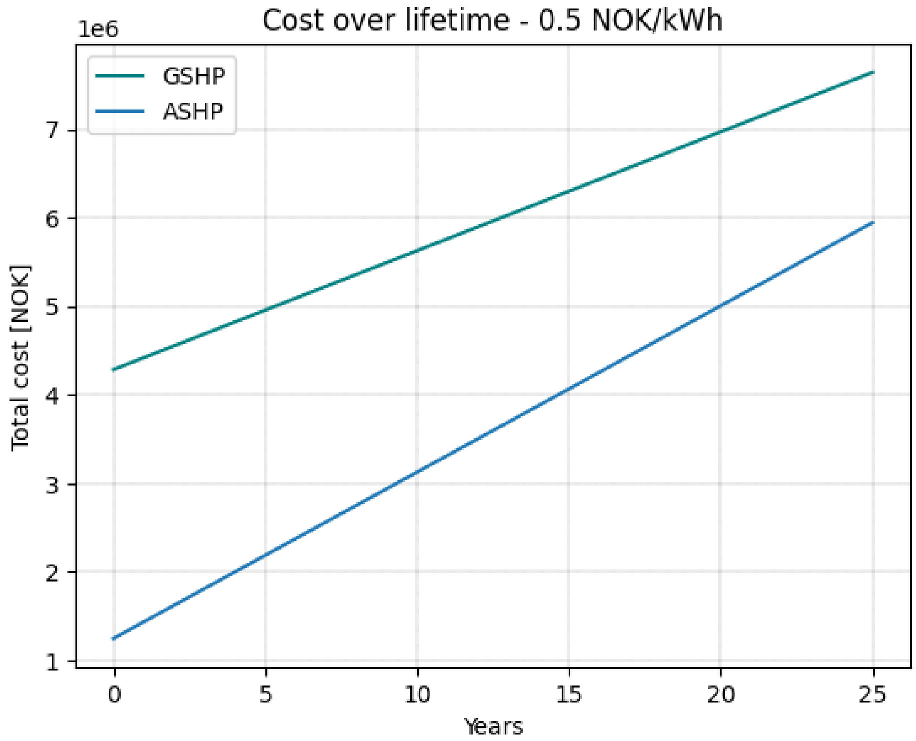


Figure 4.8: Total cost of the ground source and air source heat pump system, given electricity price of 0.5 NOK/kWh.

From these results, the air source heat pump system would be an obvious choice economy-wise. This shows that electricity prices are essential when choosing a system. The high electricity prices in recent months might stabilize and decrease in the future, but there is no indication that the prices will go back to the level of 0.5 NOK/kWh in the next few years.

When looking at energy prices, one could also argue that the higher fossil fuel prices mentioned in the introduction would make it even more economically beneficial to choose more efficient systems, like heat pumps, compared to heaters using natural gas or oil as fuel.

Environmental impact and efficiency

Heat pumps are not emitting greenhouse gases while running, as they are purely electrical installations [59]. However, the electricity used to run them is not necessarily carbon-free. As a matter of fact, in most countries, a significant share of the produced energy is from fossil fuels. Even in Europe, one of the most developed continents of the world, the percentage of combustible fuels for electricity generation was 41.9% in 2021 [60]. When the share of carbon-intensive fossil fuels is high, high efficiency is even more critical.

Due to both population growth and economic growth in several parts of the world, the world has increased energy demand, especially electricity [61]. Even though electricity production from renewable sources is growing significantly, the increase in the share of renewable sources in global electricity production is not. More cost-effective production of electricity, using coal and natural gas, is still installed due to well-known technology and availability to meet the increasing demand. By choosing more efficient technology, one could slow the increase in electricity and thus delay the installation of new power plants.

4.5.1 Summary

All of these factors must be considered when choosing a heating system. What is more important in the specific case? If the economy is the deciding factor, the choice could be straightforward. On the other hand, there would still be insecurities connected to electricity prices, as the time scope is large. If energy savings is the most important for environmental purposes, the ground source heat pump would be the best choice. Location is an essential factor when considering both economy and practicality. The ground source heat pump has more even running conditions, which could influence the heat pump's lifetime. This could also have economic consequences when lifetime costs are the most critical factor.

Chapter 5

Conclusion

Two different heat pump systems, for an example office building of 1700 m^2 located in Bergen, Norway, were compared against each other. Heat demand was calculated using temperature data from the entire year of 2022. One of the heat pumps assessed used outside air as the low-temperature source, and the other ground heat. ABK-qviller provided data for both heat pump systems. An economic assessment was done, and other essential factors to consider were discussed.

The total building heating and cooling demand for a year was calculated to be 728,484 kWh and 82,525 kWh. The received COPs for the two systems were used to compute the required energy input, which was found to be 220,330 kWh and 256,314 kWh for the ground source and air source, respectively, and to estimate how much heat would be extracted from the ground. The total heat extracted during one year was 507,242 kWh, and the total heat re-injected was 119,735 kWh.

Sizing energy wells is important for designing a ground source heat pump, as incorrect dimensions could have severe consequences for the heat pump's operation. Both the amount of energy extracted from the ground and ensuring that the amount of heat from the wells meets peak load are crucial. To meet these requirements, the calculated surface area for the ground heat exchanger was 180 $\times$ 180 meters, and the total borehole length was 7263 meters.

The temperature in the ground was roughly mapped using a Fortran program that shows the temperature development after 25 years to give a picture of the importance of considering cooling in the design of a ground source heating system.

The investment costs for the ground source and air source heat pumps were 4,285,400 NOK and 1,250,000 NOK, respectively. Running annual maintenance and operating costs were 538,660 NOK and 711,963 NOK. The total costs for the two systems after 25 years of operation were found to be 17,751,900 NOK and 19,049,075 NOK, meaning that, with the assumptions made in this thesis, the ground source heat pump is the most beneficial choice economically.

Other factors worth considering were discussed, including climate and location, environmental considerations, and energy prices. Using a more efficient heating system, like ground source heat pumps, could help in the ongoing energy transition that is necessary to fight the effects of global warming.

Chapter 6

Future work

In the future, the demand for efficient, sustainable solutions for heating will continue to increase. For more energy and cost-effective systems, it could be helpful to consider improving some of the following points.

- Advanced heat exchanger design: The heat exchanger design is critical to the performance of a GSHP system. Future research could focus on developing more efficient and cost-effective heat exchangers that can improve heat transfer efficiency.
- Thermal storage technologies: Thermal storage can help improve the efficiency of ground source heat pumps by allowing excess energy to be stored ad used later when needed. Focusing on developing more cost-effective solutions and more energy-effective solutions would make it more beneficial to implement sustainable solutions rather than heating systems fueled by fossil energy sources.
- More accurate models for sizing: Accurate modeling of the ground heat exchanger is critical for well-functioning ground source heat pump systems.
- Hybrid ground source heat pump systems that combine different renewable sources like solar power and wind power would reduce greenhouse gas emissions in regions with high shares of fossil fuels in their electricity mix.
- Focus on solutions allowing the use of waste heat from unconventional sources would reduce the costs that come with drilling boreholes.

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Appendix A

Fortran code - Horizontal temperature development

```
module variables
        implicit none
        integer :: i, j, N, time, years, avstand,
           wells, grid
        integer :: dx_grid, rundt, gridrundt
        real :: T_i, T_b, dx, dy, k, rho, c_v, a, dt,
            dtdx, dtdy
        real :: sizexy, lowest
        real, dimension (500,500) :: T, delta
    end module variables

    program small_park
        use variables
        implicit none
        call readfile
        call fill_array
        call temperature_change
        call loop_year
        call loop_life
    end program small_park

    subroutine readfile
        use variables
        open(10, file='input_master', status = 'old')
        read(10,*) T_i, T_b, k, rho, c_v
        dx = 1
        dy = dx
        avstand = 26
        wells = 6
```

```
		rundt = 18
		dx_grid = 1
		grid = avstand/dx_grid
		sizexy = avstand*(wells+1)+(2*rundt)
		N = (sizexy/dx) + 1
		gridrundt = rundt/dx_grid
		dt = 3600
		a=k/(rho*c_v)
		print*, dx
	end subroutine readfile

	subroutine fill_array
		use variables
		do i = 1, N
			do j = 1, N
				T(i,j)=T_i
			end do
		end do
		do i = (gridrundt)+grid+1, N-2-gridrundt, grid
			do j= (gridrundt)+grid+1, N-2-gridrundt, grid
				T(i,j) = T_b
			end do
		end do
	end subroutine fill_array

subroutine temperature_change
		use variables
		do i = 2, N-1
			do j=2, N-1
				dtdx=(T(i+1,j)-2*T(i,j)+T(i-1,j))/dx**2
				dtdy=(T(i,j+1)-2*T(i,j)+T(i,j-1))/dy**2
				delta(i,j)=a*dt*(dtdx+dtdy)
			end do
		end do
		do i= (gridrundt)+grid+1, N-2-gridrundt, grid
			do j=(gridrundt)+grid+1, N-2-gridrundt, grid
				delta(i,j)=0
			end do
		end do
		do i = 2, N-1
			do j= 2, N-1
				T(i,j)=T(i,j)+delta(i,j)
			end do
		end do
```

```

        T(:,1) = T(:,2)
        T(:,N) = T(:,N-1)
        T(1,:)=T(2,:)
        T(N,:)=T(N-1,:)
    end subroutine temperature_change

    subroutine loop_year
        use variables
        do time= 1,2926
            call temperature_change
        end do
    end subroutine loop_year

subroutine loop_life
        use variables
        do years = 1, 25
            call loop_year
        end do
        do i = (gridrundt)+grid+1, N-2-gridrundt, grid
            do j= (gridrundt)+grid+1, N-2-gridrundt, grid
                T(i,j) = T_b
            end do
        end do
        lowest=8
        do i =1, N
            do j = 1,N
                if (T(i,j) /= 2) then
                    if (T(i,j).lt.lowest) then
                        lowest = T(i,j)
                    end if
                end if
            end do
        end do
        open(60, file='smallpark_life.txt')
        do i =1, N
            write(60, '(248(f5.1))') (T(i,j), j=1, N)
        end do
        print*, lowest
    end subroutine loop_life
```

Appendix B

Fortran code - Vertical temperature development

```
module variables
        implicit none
        integer :: i, j, N1, N2, time, years, avstand, wells
        integer :: grid, dx_grid, rundt, gridrundt, depth, under
        real :: T_i, T_b, dx, dy, k, rho, c_v, a, dt, dtdx, dtdy
        real :: sizex, sizey, dT_bottom, q, deltaT
        real, dimension (500,500) :: T, delta
end module variables

program park_vertical
        use variables
        implicit none
        call readfile
        call fill_array
        call temperature_change_active
        call loop_year
        call loop_life
end program park_vertical

subroutine readfile
        use variables
        open(10, file='input_master', status = 'old')
        read(10,*) T_i, T_b, k, rho, c_v
        dx = 1
        dy = dx
        avstand = 26
```

```
                depth = 210
                wells = 6
                rundt = 30
                under = 39
                dx_grid = 1
                grid = avstand/dx_grid
                gridrundt = rundt/dx_grid
                sizex = avstand*(wells+1)+(2*rundt)
                sizey = depth + under
                N1 = (sizey/dy) + 1
                N2 = (sizex/dx)+1
                dt = 3600
                a=k/(rho*c_v)
        end subroutine readfile

        subroutine fill_array
                use variables
                T = T_i
                do i = 1, depth/dx_grid
                        do j =(gridrundt)+grid+1,N2-2-gridrundt,grid
                                T(i,j) = T_b
                        end do
                end do
        end subroutine

        subroutine temperature_change_active
                use variables
                q = 0.06
                dT_bottom = (q*dt)/(rho*c_v*dy)
                do i = 2, N1-1
                        do j=2, N2-1
                                dtdx=(T(i+1,j)-2*T(i,j)+T(i-1,j))/dx**2
                                dtdy=(T(i,j+1)-2*T(i,j)+T(i,j-1))/dy**2
                                delta(i,j)=a*dt*(dtdx+dtdy)
                        end do
                end do
                delta(1,:) = 0 !constant temperature at surface
                do i=1, depth/dx_grid
                        do j =(gridrundt)+grid+1,N2-2-gridrundt,grid
                                delta(i,j) = 0
                        end do
                end do
                do i =2, N1-1
                        do j = 2, N2-1
```

```
                        T(i,j) = T(i,j) +delta(i,j)
                end do
        end do
        !boundary conditions
        T(1,:) = T_i
        T(N1,:)=T(N1-1,:) + dT_bottom
        T(:,1) = T(:,2)
        T(:,N2) = T(:,N2-1)
        do j = gridrundt+grid+1,N2-2-gridrundt,grid
                i = 1
                T(i,j) = T_b
        end do
end subroutine temperature_change_active

subroutine loop_year
        use variables
        do time = 1, 2926
                call temperature_change_active
        end do
end subroutine loop_year

subroutine loop_life
        use variables
        do years = 1, 25
                call loop_year
        end do
        open(60, file='verticals_life.txt')
        do i = 1, N1
                write(60, '(243(f5.1))') (T(i,j), j
                   =1,N2)
        end do
end subroutine loop_life
```