KU LEUVEN

FACULTEIT INGENIEURSWETENSCHAPPEN

Design and Assessment of Low-carbon Residential District Concepts with (Collective) Seasonal Thermal Energy Storage

Robin Haesen
Louis Hermans

Thesis voorgedragen tot het behalen van de graad van Master of Science in de ingenieurswetenschappen: werktuigkunde

Promotoren:
Prof. dr. ir. L. Helsen
Prof. ir. W. Boydens

Assessoren:
Dr. ir. S.Carli
Ir. S.Kushwaha

Begeleiders:
Ir. A. Uytterhoeven
Ir. W. Peere

Academiejaar 2020 – 2021

© Copyright KU Leuven

Without written permission of the thesis supervisors and the authors it is forbidden to reproduce or adapt in any form or by any means any part of this publication. Requests for obtaining the right to reproduce or utilize parts of this publication should be addressed to Faculteit Ingenieurswetenschappen, Kasteelpark Arenberg 1 bus 2200, B-3001 Heverlee, +32-16-321350.

A written permission of the thesis supervisors is also required to use the methods, products, schematics and programmes described in this work for industrial or commercial use, and for submitting this publication in scientific contests.

Zonder voorafgaande schriftelijke toestemming van zowel de promotoren als de auteurs is overnemen, kopiëren, gebruiken of realiseren van deze uitgave of gedeelten ervan verboden. Voor aanvragen tot of informatie i.v.m. het overnemen en/of gebruik en/of realisatie van gedeelten uit deze publicatie, wend u tot Faculteit Ingenieurswetenschappen, Kasteelpark Arenberg 1 bus 2200, B-3001 Heverlee, +32-16-321350.

Voorafgaande schriftelijke toestemming van de promotoren is eveneens vereist voor het aanwenden van de in deze masterproef beschreven (originele) methoden, producten, schakelingen en programma's voor industrieel of commercieel nut en voor de inzending van deze publicatie ter deelname aan wetenschappelijke prijzen of wedstrijden.

Preface

More or less a year ago, we got to know each other rather accidentally. We actually were drafted to work together at a project for a shared course in our curriculum. Immediately, our shared interest in energy surfaced and a productive collaboration started to grow. Lifting each other to a higher level through shared motivation and interests, that first project was brought to a good end. As a result of this positive experience and growing connection, we decided to also collaborate for our master's thesis. As we both share the same passion for energy, it was not difficult to find a subject that really interested us both. We got lucky and got assigned our most preferred subject. This was the start of an even more productive collaboration and the start of a friendship. Week in week out, passionate by our work, we worked together and had great times. The result of this long adventure is all combined in this text. We hope you will enjoy reading it as much as we enjoyed doing the research and writing the text.

We would also like to take this opportunity to thank our promoters prof. Helsen and prof. Boydens for their support throughout the year. Moreover, we would like to say thank you to our mentors Anke and Wouter, for their weekly advice and tips. This really propelled us forward. Next, our sincere gratitude also goes to Arjan Goemé who came up with this interesting subject. Moreover, we want to thank all people that helped us to collect all the necessary data. Lastly, we would also like to thank the jury for reading this text.

Robin Haesen
Louis Hermans

Contents

Abstract

Achieving carbon neutrality by 2050 is a major challenge for all European countries, including Belgium. It requires a transition away from fossil fuels and towards renewable energy sources in all energy-intensive sectors. Heating of residential buildings is one of these sectors, currently dominated by use of fossil fuels. A potential alternative is proposed by renewable solar energy. However, a mismatch exists between solar energy, which is largely available in the summer and the heating demand, which is mainly situated in the winter. This mismatch can be overcome by applying a form of seasonal thermal energy storage as part of a district heating system. In this thesis, different concepts are designed for such a district heating system. In total, 13 concepts are found, each with an a and b version. The concepts are divided based on the type of seasonal storage that is applied, resulting in 6 concepts with seasonal tank storage, 4 concepts with low-temperature borefield storage and 3 concepts with high-temperature borefield storage. Furthermore, two methods are applied to assess the different concepts: a first, simplified method and a second, detailed method. The concepts are assessed both on their costs and CO_2 emissions. The results of both methods are compared and significant differences are observed. This leads to the conclusion that applying simplifications in the calculations creates a distorted picture. A detailed assessment of the systems under consideration in this thesis is therefore required. The detailed assessment shows that two concepts with low-temperature borefield storage are superior to all other concepts, based on their considerably lower costs. These concepts are compared to two benchmark cases, one in which each dwelling has an individual gas boiler and one in which each dwelling has an individual heat pump and solar PV. The results show that both concepts of a district heating system are cost competitive with the benchmark cases, while achieving lower primary energy use and CO_2 emissions.

Samenvatting

CO_2-neutraliteit bereiken tegen 2050 is een grote uitdaging voor alle Europese landen, ook voor België. Het vereist een overgang van fossiele brandstoffen naar hernieuwbare energiebronnen in alle energie-intensieve sectoren. De verwarming van residentiële gebouwen is één van deze sectoren, die momenteel gedomineerd worden door het gebruik van fossiele brandstoffen. Een potentieel alternatief wordt voorgesteld door hernieuwbare zonne-energie. Er bestaat echter een mismatch tussen zonne-energie, die grotendeels in de zomer beschikbaar is, en de warmtevraag, die vooral in de winter is gesitueerd. Deze mismatch kan overwonnen worden door een vorm van thermische seizoensopslag toe te passen als onderdeel van een stadsverwarmingssysteem. In deze thesis worden verschillende concepten ontworpen voor een dergelijk stadsverwarmingssysteem. In totaal zijn er 13 concepten gevonden, elk met een a- en b-versie. De concepten zijn opgedeeld op basis van het type seizoensopslag dat wordt toegepast, wat resulteert in 6 concepten met tankopslag, 4 concepten met lage-temperatuur-boorveld-opslag en 3 concepten met hoge-temperatuur-boorveld-opslag. Verder worden er twee methodes toegepast om de verschillende concepten te beoordelen: een eerste, vereenvoudigde methode en een tweede, gedetailleerde methode. De concepten worden beoordeeld op zowel hun kosten als CO_2-uitstoot. De resultaten van beide methodes worden vergeleken en significante verschillen worden geobserveerd. Dit leidt tot de conclusie dat het toepassen van vereenvoudigingen in de berekeningen een vertekend beeld geeft. Een gedetailleerde beoordeling van de systemen die in deze thesis worden besproken, is daarom vereist. De gedetailleerde beoordeling toont aan dat twee concepten met lage-temperatuur-boorveld-opslag superieur zijn aan de andere concepten, op basis van hun aanzienlijk lagere kosten. Deze concepten worden vergeleken met twee benchmarks, één waarin elke woning een individuele gasketel heeft en één waarin elke woning een individuele warmtepomp en zonnepanelen heeft. De resultaten tonen aan dat beide concepten van een stadsverwarmingssysteem qua kosten competitief zijn met de benchmarks, terwijl ze een lager primair energiegebruik en lagere CO_2-emissies behalen.

List of Figures

List of Tables

List of Abbreviations and Symbols

Abbreviations

ATES	Aquifer Thermal Energy Storage
BTES	Borehole Thermal Energy Storage
DDA	Detailed Dynamic Assessment
DHW	Domestic hot water
HP	Heat pump
HTC	Heat transfer coefficient
MB HP	Micro booster heat pump
NPV	Net Present Value
PEX	Cross-linked polyethylene
PTES	Pit Thermal Energy Storage
PUR	Polyurethaan
SDA	Simplified Dynamic Assessment
STES	Seasonal Thermal Energy Storage
TTES	Tank Thermal Energy Storage
UTES	Underground Thermal Energy Storage

Symbols

c	Specific investment cost of tank storage [€/m^3]
c_p	Specific heat capacity of a 50/50 water-glycol mixture [J/kgK]
$c_{p,w}$	Specific heat capacity of water [J/kgK]
G	Solar irradiance [W/m^2]
g_c	Gas consumption [kWh]
h_s	Dimensionless heat loss factor [-]
k_s	Thermal conductivity of the ground [W/mK]
L	Total length of the borefield [m]
r	Interest rate [%]
R	Market interest rate [%]
R_i	Inflation rate [%]
R_b^*	Equivalent borehole resistance [mK/W]
T_{amb}	Ambient temperature [K]
T_b	Borehole wall temperature [K]
T_f	Average fluid temperature between in and outlet of the borefield [K]
$T_{f,col}$	Average fluid temperature in the solar collector [K]
T_H	Temperature of the hot volume in the stratified tank [°C]
T_L	Temperature of the cold volume in the stratified tank [°C]
V	Tank volume [m^3]
ρ	Density of a 50/50 water-glycol mixture [kg/m^3]
ρ_w	Density of water [kg/m^3]

Introduction

The effects of climate change are visible across the globe. Engineers and scientists are faced with the challenge of mitigating climate change, which involves significantly reducing greenhouse gas emissions. All member states of the European Union aim to be climate-neutral by 2050, exhibiting net-zero greenhouse gas emissions [19]. With this objective in mind, a radical reduction of CO_2 emissions is necessary in all energy-intensive sectors. Among these is the building sector, which is responsible for approximately 40% of the final energy use and 36% of the CO_2 emissions in the EU [20]. In residential buildings, or households, more than three quarters of the final energy is used for space heating and water heating [24]. The domain of residential space heating and water heating, supplemented with space cooling, is the main focus of this text.

Today, most of the energy demand of buildings is met by the use of fossil fuels, resulting in significant CO_2 emissions. To reach carbon neutrality by 2050, many countries aim to transition towards renewable energy sources within the heating of buildings. One way to achieve this transition is by electrifying the heating of buildings with heat pumps. These heat pumps can be powered by electricity from renewable sources, such as solar energy. However, solar energy is mainly available in the summer, whereas the majority of heating demand is situated in the winter. This seasonal mismatch between the availability of solar energy in the summer and the heating demand of buildings in the winter limits the direct use of renewable electricity. Moreover, the most well-known form of electrical energy storage, i.e. battery storage, does currently not provide a long-term storage solution to overcome the seasonal mismatch. Electrification alone of residential heating will therefore not suffice to reach the strict goals of CO_2 emission reductions.

Hence, another way to incorporate solar energy in the heating of buildings is by introducing a district heating system with Seasonal Thermal Energy Storage (STES). In this system, solar energy is captured by solar thermal collectors and transferred to a long-term thermal energy storage, which is connected to multiple houses via a district heating network. In that way, the solar energy available in the summer can be collectively stored and subsequently used in the next heating season. Designing concepts of a district heating system with STES is a challenging task with many design options. It is the objective of this thesis, along with an assessment of the environmental and economic aspects of these concepts.

The text is divided into three main parts, based on three central questions that are asked in this thesis. These parts are respectively: Concept Design, Concept Assessment and Benchmarking.

- **Concept Design** How does the selection and interconnection of different components lead to different concepts of a district heating system with STES?
- **Concept Assessment** How do different concepts of a district heating system with STES compare to each other on an environmental (CO_2 emissions) and economic (Net Present Value) basis?
- **Benchmarking** How do interesting concepts of a district heating system with STES compare to two benchmark cases with individual heating systems?

At the end of each part, the reader will be able to answer the central question corresponding to the part. To come to this point, the parts are divided into different chapters, which helps to build the reasoning in each part of the thesis. The reasoning starts in the first chapter of the first part, which handles the design of concepts.

Part I

Concept Design

Chapter 1

General System Overview and Requirements

Before starting the design process of a district heating system, it is important to understand the lay-out of such a system and the components that are used. This chapter gives an overview of the global district heating system, as well as the requirements for the system. Chapter 2 dives deeper into the specific components that are part of the system. Note that the term 'district heating system' is used to refer to the overall system, which is divided into different components, as discussed in Section 1.1. Subsequently, Section 1.2 mentions the requirements for the system, specified in the Belgian climate.

1.1 System Overview

In district heating, heat is collectively provided to a densely built-up district, meaning that the heating demand is highly concentrated. This thesis considers a residential district of 50 low-energy dwellings. These low-energy dwellings are buildings with an energy-efficient design that require less energy than if built according to conventional building criteria [6]. In Section 1.2, the low-energy character of the houses is illustrated by their annual space heating demand of 30 $\frac{kWh}{m^2}$. Furthermore, the dwellings in the district are designed as stand-alone houses with a total floor area of 150 m^2, corresponding to more or less the average for this type of dwellings in Belgium [35].

A district heating system provides the residential district with heat for space heating via a network of insulated pipes. This underground network of pipes is referred to as the '**district heating network**'. It uses water as a heat carrier to distribute heat to the dwellings in the district. Along with space heating, the district heating network can potentially also provide heat for domestic hot water. However, this is not necessarily the case, as is demonstrated by the different concepts that are developed in Chapter 3. The district heating network is further discussed in Section 2.3.

In addition to space and water heating, dwellings require **cooling** as well. Therefore, this thesis also takes into account the cooling demand of low-energy dwellings, corresponding to 10 $\frac{kWh}{m^2}$ per year (see Section 1.2). Cooling in the global system can be provided either by a separate system in each house or by the same network that is used for heating. In the second option, the existing pipework is used and cold water provides cooling to the dwellings.

Furthermore, the overall district heating system also includes a form of heat generation. A number of different options exist for heat generation in district heating systems, e.g. fossil fuels, biomass or solar energy. In this thesis, the focus is on solar energy. It is integrated in the overall system by use of **solar thermal collectors**, which form another component of the district heating system. For the location of the solar collectors, there is a choice between a central solar collector field on the one hand and the placement of solar collectors on the dwellings' rooftops on the other hand. As this thesis considers both options, the availability of space for a central solar collector field in the district is taken as a starting point. The solar thermal collectors are examined in detail in Section 2.2.

Thus far, the district heating system already includes the district heating network, the component of cooling and the solar thermal collectors. Furthermore, the system also includes a **Seasonal Thermal Energy Storage (STES)**. This component provides the possibility to store energy, captured by the solar collectors, at a seasonal timescale. The STES is connected to the solar collectors on the one hand and via the district heating network, to the dwellings on the other hand. A number of different technologies exist for Seasonal Thermal Energy Storage, which are discussed in Section 2.1.2.

In addition to the previously mentioned components, the district heating system can be extended with **supplementary heating systems**. These systems can help to provide the dwellings with the required heat, in addition to the heat captured by the solar collectors. Section 2.4 mentions the supplementary heating systems that are considered in this thesis.

The last component that is included in the district heating system is the **heat emission system**. Each dwelling is equipped with such a system, providing heat inside the rooms of the dwelling. In Section 2.5, the heat emission system is discussed in detail.

In conclusion, the overall district heating system consists of the following components: a Seasonal Thermal Energy Storage (STES), solar thermal collectors, a district heating network, potential supplementary heating systems, the heat emission systems and the component of cooling. This last component can either coincide with the district heating network or correspond to separate cooling systems in the dwellings. Chapter 2 examines these different components in detail.

An important remark is that pumps and heat exchangers in the district heating system are not considered in the above mentioned system components. Although they will be used in the system, they are not examined in detail in this thesis. Therefore, in the remaining of this text, pumps and heat exchangers are not taken into account in the calculations. This means that if heat exchangers are used, it is assumed that heat is transferred with 100 percent efficiency. Regarding the pumps, it is assumed that no pumping power is required.

As the lay-out of the overall district heating system is now determined, the next step is to take a look at the requirements that this system needs to meet. This is discussed in the next section.

1.2 System Requirements

As mentioned in Section 1.1, this thesis considers both the heating and cooling demand of 50 low-energy dwellings with a floor area of 150 m^2. The heating demand includes both the space heating demand and the domestic hot water demand. The district heating system has to be designed in such a way that all demands are met.

The demand profiles for heating and cooling are obtained from the company *boydens engineering*. Corresponding to the low-energy character of the dwellings, the annual space heating demand is 30 $\frac{kWh}{m^2}$ and the annual cooling demand is 10 $\frac{kWh}{m^2}$. The demand of domestic hot water corresponds to a daily demand of 6.27 kWh per dwelling. This demand remains constant throughout the year. The space heating and cooling demand on the other hand, show an evolution throughout the year, as seen in Figure 1.1 for a single dwelling of 150 m^2. This figure illustrates the peak space heating demand of 2.9 kW and the peak cooling demand of 1.9 kW for a single dwelling. The peak demand of domestic hot water corresponds to 0.3 kW for a single dwelling. The annual demands and peak demands for the entire district of 50 dwellings are shown in Table 1.1.

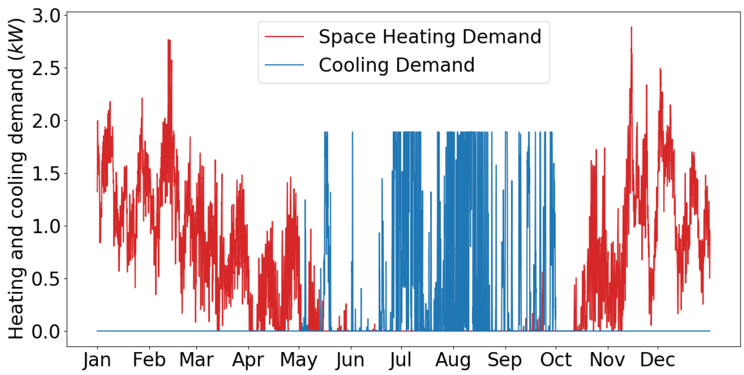


Figure 1.1: Evolution of the space heating and cooling demand of a single dwelling.

	Annual demand	Peak demand
Space heating	225 MWh	145 kW
Cooling	75 MWh	95 kW
Hot water	115 MWh	15 kW

Table 1.1: Annual and peak demand of the district of 50 dwellings.

Sizing of the components in the district heating system is based on these demand profiles. Along with the demand profiles, the solar irradiance influences the sizing of the system components as well, more in particular of the solar collectors. The profile for the solar irradiance over the year is illustrated in Figure 1.2. It corresponds to a typical Belgian year with a total irradiation of 990 $\frac{kWh}{m^2}$ over the year. Furthermore, Figure 1.3 shows the evolution of the ambient temperature for a typical Belgian year.

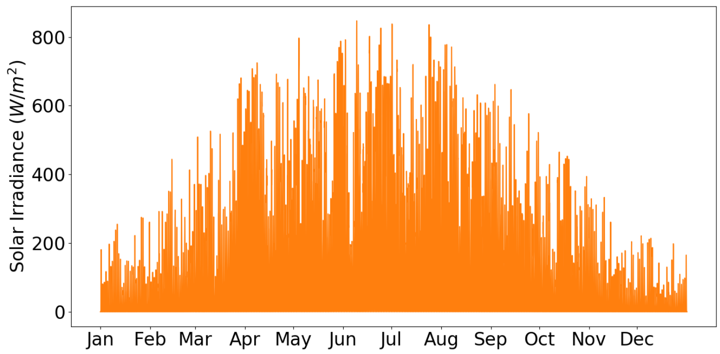


Figure 1.2: Evolution of the solar irradiance for a typical Belgian year.

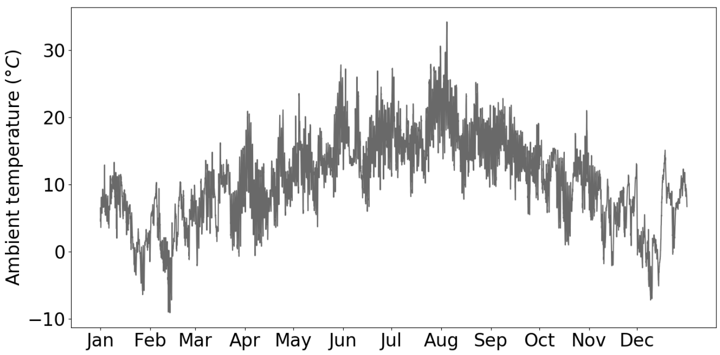


Figure 1.3: Evolution of the ambient temperature for a typical Belgian year.

As the lay-out of the district heating system, as well as the system requirements are now known, the components of this system are studied in detail in Chapter 2.

Chapter 2

Literature Study of Possible System Components

The district heating system consists of multiple components that are interconnected. This chapter provides a detailed study of these components from literature. Each component is discussed in a separate section, i.e. the *Seasonal Thermal Energy Storage*, the *Solar Thermal Collectors*, the *District Heating Network*, the *Supplementary Heating Systems* and the *Heat Emission System*.

2.1 Seasonal Thermal Energy Storage

This section first gives a general overview of Thermal Energy Storage (TES). Subsequently, the different technologies for Seasonal Thermal Energy Storage (STES) are studied.

2.1.1 Thermal Energy Storage

Solar thermal energy offers great potential for substituting fossil fuels in residential heating applications. According to Pinel et al., the annual solar irradiation incident on the roof of a dwelling exceeds its annual energy use [54]. However, the main problem in exploiting this solar energy for space heating and domestic hot water preparation is its intermittency. Indeed, fluctuations in solar radiation occur on a daily, weekly and seasonally basis with the majority of energy being available at midday and during summer. On the other hand, energy use is higher during winter due to a higher space heating demand, especially at morning and night. This leads to a mismatch between energy supply and demand at different time scales [54, 11]. The mismatch at seasonal scale is illustrated in Chapter 1 by comparing the evolution of the space heating demand in Figure 1.1 to the evolution of the solar irradiance in Figure 1.2.

Thermal Energy Storage (TES) provides a solution to overcome this mismatch by storing excess energy from the renewable source and making the energy available at a later stage. At daily or weekly scale, short-term storage, e.g. in the form of water tanks, can compensate for the diurnal offset relatively easy with limited heat loss. At seasonal scale, thermal energy storage becomes more complex and more expensive. In this case, Seasonal Thermal Energy Storage (STES) is applied to store energy captured by the solar thermal collectors in the summer and discharge the energy in the winter to meet the higher heating demand [54, 11].

Before looking into different technologies for STES, it is worth noting that TES systems can generally be classified into three types based on their storage mechanism: chemical, latent and sensible storage. Chemical and latent storage are currently in research phase and, although promising, they are economically not suitable for thermal energy storage at seasonal scale [54, 30].

Therefore, this text focuses on the type of TES that is well demonstrated, clearly understood and widely applied, i.e. sensible storage. In sensible storage, heat is stored as internal energy by increasing the temperature of a solid or liquid storage medium. A good storage medium must fulfill a number of requirements:

- Firstly, the medium should have a high thermal capacity (i.e. density multiplied by specific heat). Together with a relatively large temperature increase of the storage medium, a satisfactory energy density can be obtained in that case [54].
- Furthermore, the capacity of the medium to absorb and release heat at a sufficient rate is important, since it allows fast charging and discharging of the storage. The rate of heat transfer is determined by either the thermal diffusivity for a solid medium or the convective heat transfer rate for a liquid medium [54].
- Lastly, an important incentive to select a storage medium is its cost and typically used storage mediums include low-cost options such as water, soil and rock [54, 27].

In sensible heat storage, difficulties can occur while charging and discharging the storage. On the one hand, charging the storage increases the storage medium's temperature and heat transfer to the storage becomes problematic when it is almost fully charged. On the other hand, discharging the storage decreases the medium's temperature and it becomes difficult to obtain the low-quality heat from the storage when it is almost discharged. Therefore, so-called stratification of the storage can be applied to improve heat transfer to and from the system [54]. A stratified storage system, e.g. a storage tank, consists of a hot and a cold zone, separated by a thermocline region that prevents mixing of the two zones. "Stratification [...] allows continuous possibilities to transfer heat to the cooler regions of an almost fully charged store while it also results in a proper energy quality being available from the warmer regions of an almost discharged store" [54]. The principle of stratification is illustrated in Figure 2.1, which respectively shows a well-stratified, a moderately stratified and a fully mixed storage tank. It shows that a smaller thermocline region corresponds to a higher degree of stratification.

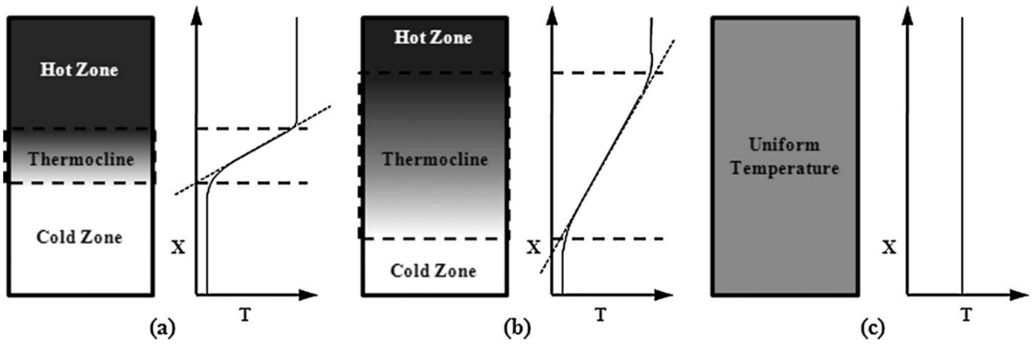


Figure 2.1: Different levels of stratification in a storage tank: (a) well-stratified, (b) moderately stratified, (c) fully mixed [54].

The main drawback of sensible heat storage is that it is often characterised by a considerable amount of self-discharge through heat loss. The temperature difference between the storage and the surroundings, as well as the generally large volume and heat exchange area of the sensible storage, lead to significant heat losses. Self-discharge is especially problematic for the long storage periods observed in seasonal thermal energy storage. Therefore, measures are often in place to reduce heat loss to the surroundings. These measures include insulating the storage, burying the storage system and designing the storage with a low surface to volume ratio [54].

As the basic principles of sensible heat storage are now known, the next step is to take a look at the different STES technologies that are available. This is done in the next section.

2.1.2 STES Technologies

Seasonal thermal energy storages are mainly built underground because of their large volumes and proximity to residential areas. Moreover, fully or partially burying the storage systems adds to the insulation. In some cases, the soil itself is the storage medium [30]. STES systems are therefore often referred to as Underground Thermal Energy Storage (UTES) systems [30, 11, 51].

Literature distinguishes four main technologies for large-scale UTES that have been applied to date:

- Tank Thermal Energy Storage (TTES)
- Pit Thermal Energy Storage (PTES)
- Borehole Thermal Energy Storage (BTES)

- Aquifer Thermal Energy Storage (ATES)

They rely on the principles of sensible heat storage and are illustrated in Figure 2.2. Determining the most suitable STES technology for a project always follows from a techno-economic analysis, taking into account the specific boundary conditions. These conditions are obtained from the overall system in which the STES is integrated [51]. This is later illustrated by the concepts in Chapter 3. What follows here is an overview of the four STES technologies.

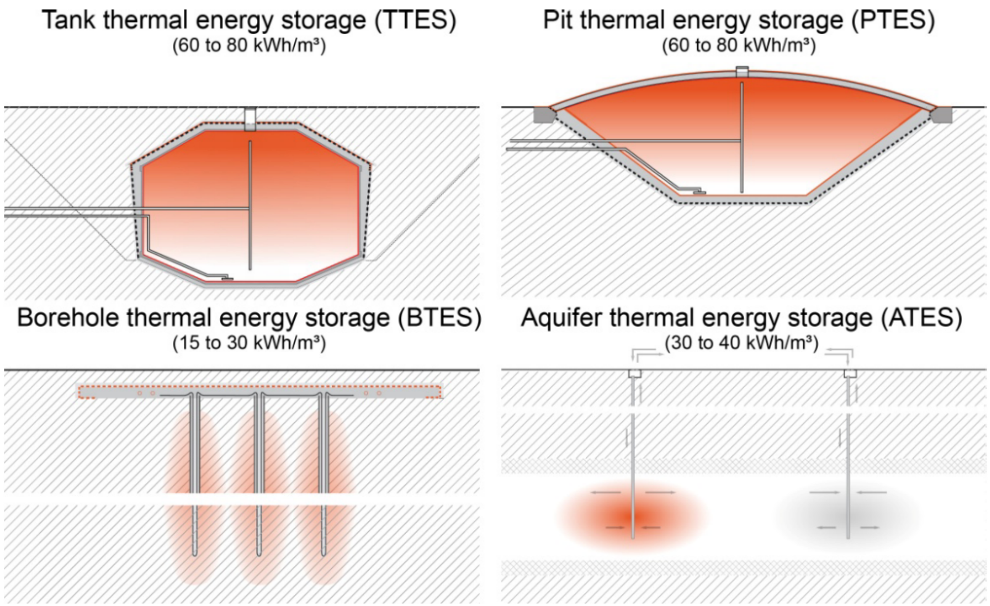


Figure 2.2: Different STES technologies [30].

Tank Thermal Energy Storage

Tank Thermal Energy Storages (TTESs) are mostly built with a reinforced concrete structure, utilizing either in-situ concrete or prefabricated concrete elements. Alternatively, stainless steel or fiber reinforced plastics can be used. Concrete tanks are state of the art for seasonal, underground TTES, whereas steel tanks are generally used as above ground buffer tanks [30, 51]. On the inside surface of a concrete tank, an additional liner made of stainless steel or polymer is normally mounted to obtain a watertight construction. The outside surface of the tank is equipped with insulation material to reduce heat loss [30].

In tank storage, water is commonly used as the storage medium. The water can be heated up to 98°C, in case that stainless steel is applied as liner material [30]. Even higher temperatures can be achieved if the tank is under internal pressure but are not relevant in district heating applications. With a sufficiently large temperature increase of the water, TTES provides relatively high heat storage capacities between

60 and 80 $\frac{kWh}{m^3}$ [30, 51].
Other main advantages of tanks are: a wide variety of potential building locations since no strict geological requirements are needed; a large freedom of design allowing easy optimization of the surface to volume ratio and thereby the heat loss; a high level of stratification, improving the energy quality of the storage [28, 11, 30].
On the other hand, the main drawbacks of tanks include high construction costs and potential leakage problems [28, 11].

Pit Thermal Energy Storage

Pit Thermal Energy Storages (PTESs) are built without a static structure. An excavated pit is used as an artificial pool filled with the storage medium. The walls of the pit are tilted under a certain angle and are coated with a polymeric liner material that provides water tightness [30, 51]. A polymeric liner allows a maximal temperature of only 80 to 90°C, compared to 98°C for a stainless steel liner. The pit storage is closed by a heat-insulated lid and in some cases, the walls are insulated as well. The construction of the lid depends on the storage medium inside the pit. A first possibility is to use a mixture of gravel and water (with gravel fractions up to 70%) as the storage medium, in which case the lid can easily be carried by the gravel filling. The second option is to only use water as the storage medium. This leads to a more complex construction of the lid: it can either be floating on the water or it can be built as a self-supporting roof structure [30].

Using water as the storage medium in PTES is the most interesting option from a thermodynamic point of view, since it offers the highest heat storage capacity (60 to 80 $\frac{kWh}{m^3}$). Adding gravel to the storage reduces the storage capacity to approximately 30 to 50 $\frac{kWh}{m^3}$, depending on the gravel fraction. This means the required volume for a gravel-water pit storage is 1.3 to two times larger than for a water-filled pit storage to store the same amount of energy [30]. However, a gravel-water pit storage provides the advantage of using the space over the storage, e.g. for a parking lot [30, 51].

Similarly to tank storage, pit storage can be applied in a large number of locations that provide stable ground conditions. Furthermore, design of pits, as well as for tanks, has to take into account potential leakage problems. Differences with tank storage lie in the more reasonable construction costs and reduced flexibility in the design of pit storage [28, 11]. The latter is due to restrictions in the slope angle of the walls, which depends on the friction coefficient of the soil [48].

Borehole Thermal Energy Storage

In Borehole Thermal Energy Storage (BTES), heat is stored in the underground geology [51]. To transfer heat to and from the underground, U-pipes are inserted into vertical boreholes, which are arranged in a borefield. As a result, a large heat exchanger is formed, typically with water as a heat transfer fluid. While charging the storage, the flow direction of the water is from the center of the borefield to the boundaries. This induces high temperatures in the center and lower ones at the storage boundaries and therefore creates a horizontal temperature stratification. While discharging the storage, the flow direction of the water is reversed [30, 51]. Furthermore, heat insulation of the storage is only applied at the top surface, since the side walls and bottom are inaccessible. As a result, the storage volume is not exactly demarcated and heat losses to the surrounding ground are significant [51].

Compared to tank and pit storage, BTES is more restricted in the number of potential locations. Firstly, drillable ground must be present on-site with no or very limited groundwater flow. Secondly, the underground geology must provide a high thermal capacity and high thermal conductivity. Nevertheless, the thermal capacity of soil or rock is significantly smaller compared to water, resulting in a heat storage capacity of borefields between 15 and 30 $\frac{kWh}{m^3}$. Storage sizes of BTES are therefore three to five times larger than TTES to obtain the same amount of stored heat [51]. Moreover, thermal conductivity of the ground is rather limited as well, leading to a lower charging and discharging power compared to tank storage. To overcome this problem, a buffer storage is often integrated into the system [30].
Borefield storage does however provide advantages over the previous storage technologies in its lower construction costs and modular design, which allows the borefield to be easily expanded with additional boreholes [51, 11].

Two different usages of BTES can be distinguished: high-temperature borefield and low-temperature borefield. In the first option, the ground can be heated up to 98°C with the sole purpose of using BTES for heating applications. In the second option, the ground temperature is kept below 25°C and BTES is applied in a system that provides both heating and cooling [30]. Both possibilities will be dealt with separately in the following chapters. Note that for a high-temperature borefield, a yearly imbalance between the heat extraction and heat injection is necessary to increase the temperature of the soil over the years.

Aquifer Thermal Energy Storage

In Aquifer Thermal Energy Storage (ATES), heat is stored in a naturally occurring, water-filled ground layer [30]. The geological composition of the aquifer layer is a combination of permeable sand, gravel, sandstone or limestone on the one hand and groundwater on the other hand [51]. The storage medium is therefore a combination of water and sand or rock, which leads to a heat storage capacity of approximately 30 to 40 $\frac{kWh}{m^3}$. Heat is transferred to and from the aquifer storage via two wells that are drilled into the aquifer layer and which allow extraction and injection of groundwater. Charging of the storage is achieved by extracting cold groundwater from the cold well and injecting the groundwater into the warm well after heating by a heat source. Vice versa, discharging of the storage is achieved by extracting warm groundwater from the warm well and injecting the groundwater into the cold well after cooling by a heat sink [30, 51].

Aquifers are restricted in the number of potential locations due to the limited occurrence of aquifers and the strong requirements to use aquifers for heat storage [30, 51, 11]. These requirements include a high porosity and high hydraulic conductivity of the aquifer layer. Furthermore, the aquifer layer must be enclosed in two impermeable layers and natural groundwater flow must be negligible. Hence, an extensive hydro-geological investigation is conducted at the beginning of the project to verify these requirements and thereby the possibility of ATES [51].
Another significant drawback of aquifers is that the storage volume cannot be thermally insulated, resulting in relatively high thermal losses [51, 11]. The temperatures inside the storage will therefore mostly be restricted, which in turn opens up the opportunity to use aquifer storage for both heating and cooling applications [51].
The main advantage of ATES systems is that they can be realised with low construction costs, if the building site fulfills the required geological conditions [11].

This concludes the literature study on the four possible STES technologies, i.e. TTES, PTES, BTES and ATES. In the next section, a literature study is performed on the next system component: the solar thermal collectors.

2.2 Solar Thermal Collectors

This section gives a short overview of the different solar collector technologies that exist. Only a high-level description is provided.

A solar collector is a device that absorbs the thermal energy of the sun and transfers this energy to a working fluid. This working fluid then carries the thermal energy to places where it can be used for specific purposes. The working fluid can be water, a water-glycol mixture or other fluids such as air or molten salts.
Furthermore, solar collectors can be divided into two main groups, i.e. non-tracking solar collectors and tracking solar collectors [59]. Non-tracking collectors have a fixed position and do not move. Therefore, it is important to orientate these collectors appropriately [60]. Tracking solar collectors on the other hand make sure that they are always perpendicular to the solar irradiation by following the movement of the sun. In this thesis, only non-tracking solar collectors are considered. For these non-tracking collectors, two main types of collectors exist, i.e. flat plate solar collectors and vacuum tube collectors. Although there is actually a third type of non-tracking collector, a compound parabolic collector, this type is not considered [59]. In the next two sections, the structure and working principle of both flat plate collectors and vacuum tube collectors are discussed. In Section 2.2.3, the efficiency of these collectors is discussed.

2.2.1 Flat Plate Solar Collectors

A flat plate solar collector, shown in Figure 2.3, consists of four main components: an absorber plate, tubes for the working fluid, a glass cover and an insulation layer. The tubes for the working fluid are usually fixed or embedded in the absorber plate. The material of the absorber plate usually combines a high absorption coefficient with a small emissivity coefficient [49]. When the sun shines, the absorber plate absorbs the thermal energy of the solar irradiation and consequently its temperature rises. The absorber plate then transfers the heat to the working fluid flowing through the tubes. At the back of the absorber plate and at the sides of the collector, insulation material is provided to prevent heat loss to the environment. The glass cover is placed a couple of centimeters in front of the absorber plate, with an air layer separating the glass cover and the absorber. This cover limits the convection and irradiation losses to the environment. Other materials than glass can be used for this cover as well if they have a high transmissivity of short-wave radiation, a low transmissivity of long-wave radiation and a good UV-resistance [60, 49].

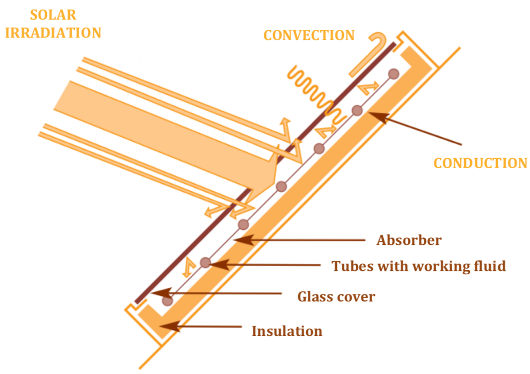


Figure 2.3: A flat plate solar collector [49].

2.2.2 Vacuum Tube Solar Collectors

The main component of a vacuum tube collector is a glass enclosure in which a heat pipe is located. This glass enclosure is under vacuum pressure, making it an ideal insulator. As a result of this good insulation, vacuum tube collectors often have a higher efficiency, especially at higher temperatures [49]. The heat pipe inside this vacuum can be seen as a closed tube which has a working fluid such as water, ethanol or methanol inside [59]. This working fluid evaporates under influence of the sun. As a result of buoyancy, the vapour then rises to the top of the tube. At the top, there is a heat exchanger section where the vapour transfers its heat to a secondary closed circuit and as a result the vapour condenses again [49]. After condensation, the liquid working fluid flows back to the bottom of the heat pipe and the cycle can restart. A vacuum tube collector consists of several of these glass enclosures next to each other. At the top, a second working fluid such as water or a water-glycol mixture collects the heat coming from the condensing working fluids. Subsequently, this second working fluid can transfer the heat to for example a hot water storage tank. Figure 2.4 schematically shows this working principle of a heat pipe in vacuum tube collectors. A complete vacuum tube collector is shown in Figure 2.5

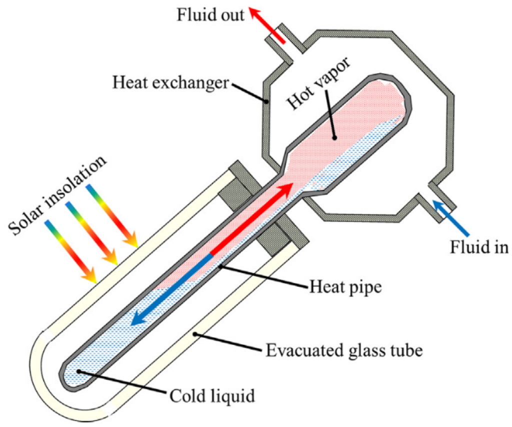


Figure 2.4: Working principle of a heat pipe inside vacuum tube solar collectors [59].

Figure 2.5: A complete vacuum tube collector [69].

2.2.3 Efficiency of Solar Collectors

To determine the amount of thermal energy a solar collector can absorb, it is necessary to know its efficiency. This efficiency is defined as follows:

$$\eta_{collector} = \frac{\dot{Q}_{absorbed}}{G} \tag{2.1}$$

Where $\dot{Q}_{absorbed}$ is the amount of useful thermal power absorbed per unit area in $\frac{W}{m^2}$ and G is the solar irradiance in $\frac{W}{m^2}$. To calculate $Q_{absorbed}$, it is important to know the area of the solar collector. Three definitions of this area exist: the gross collector area, the absorber area and the aperture area. The gross collector area is the result of length times width of the external dimensions. It is thus the total area that is covered by the collector. The absorber area is simply the total absorber area of the collector. Finally, for the aperture area the company Viessmann gives the following definition [66]:

> *The aperture area is the biggest projecting surface through which insolation can enter.*
> *In the case of flat-plate collectors, the aperture area is the visible part of the glass panel, in other words that area inside the collector frame through which light can enter the appliance.*
> *In vacuum tube collectors - both with flat as with circular absorbers without reflector areas - the aperture area is defined as the total of the longitudinal cross-sections of all glass tubes. Since the tubes contain smaller areas at the top and bottom with no absorber panels, the aperture area of these collectors is slightly larger than the absorber area. For tube collectors with back-mounted reflectors, the projections of these mirror areas are defined as the aperture area.*

Since multiple definitions of area exist, different efficiencies are defined as well. In general and independent of the area definition, the following equation is used to calculate collector efficiency [66]:

$$\eta_{collector} = \eta_0 - k_1 \frac{T_{f,col} - T_{amb}}{G} - k_2 \frac{(T_{f,col} - T_{amb})^2}{G} \tag{2.2}$$

where n_0 is the optical efficiency, k_1 is a heat loss correction value in $\frac{W}{m^2K}$, k_2 is a heat loss correction value in $\frac{W}{m^2K^2}$, $T_{f,col}$ is the average fluid temperature in the collector in K and T_{amb} is the ambient air temperature in K. The optical efficiency and both correction values are determined according to the procedure that is described in the European Standard EN 12975. Hence, these values are part of the technical specifications of a solar collector and are provided by the manufacturers. Usually the manufacturers of solar collectors report the values of η_0, k_1 and k_2 for all definitions of collector area. Typical values of these parameters are shown in Table 2.1. Note that the heat loss correction factors of flat plate collectors are larger than those of vacuum tube collectors. This suggests that the efficiency of flat plate collectors decreases more than the efficiency of vacuum tube collectors in case of increasing temperatures. This is clearly visible in Figure 2.6, which shows the efficiency as a function of the temperature difference.

Collector Type	Optical efficiency η_0	Heat loss correction value k_1	Heat loss correction value k_2
Flat plate collector	0.8	4	0.1
Vacuum tube collector	0.8	1.5	0.005

Table 2.1: Typical values for η_0, k_1 and k_2.

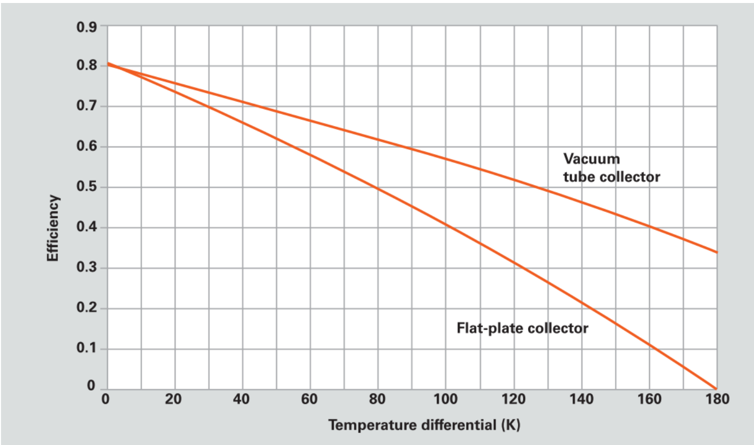


Figure 2.6: Efficiency curve of a flat plate collector and a vacuum tube collector [66].

The average temperature of the working fluid $T_{f,col}$ in Equation 4.17 is defined as follows:

$$T_{f,col} = \frac{T_{out} + T_{in}}{2} \tag{2.3}$$

Here, T_{out} is the temperature of the working fluid when it exits the solar collector, while T_{in} is the temperature of the working fluid when it enters the solar collector.

2.3 District Heating Network

This section continues with another system component: the district heating network. It briefly discusses the aspects of a district heating network that are relevant for this thesis.

A district heating network connects the dwellings in the district via a network of insulated pipes. In these pipes, water is circulated, acting as a heat carrier. The district heating network allows to collectively provide the dwellings with heat for space heating and potentially for domestic hot water as well. The heat can originate from a variety of heat sources, with solar thermal energy being the source that is considered in this thesis (see Section 1.1).

District heating has been used since the end of the 19$^{\text{th}}$ century and has evolved considerably over the years. Lund et al. describes the fourth generation of district heating as the necessary development for district heating to play an important role in future energy systems [40]. One of the aspects of fourth generation district heating is the transition towards lower distribution temperatures. This trend was also identified in the previous three generations. Low distribution temperatures decrease grid losses and are better suited for integrating renewables in future energy systems. Three different temperature levels are proposed for supplying heat in district heating, i.e. 55°C, 45°C and 35°C [41]. The return temperature is set at 25°C in all cases. These three temperature levels are used for the design of concepts in Chapter 3. Furthermore, a fourth temperature level is added to the options, i.e. district heating with an average supply temperature of 15°C. In contrast to the aforementioned temperature levels, a supply temperature of 15°C does not allow direct heating in the dwellings. Indeed, direct heating with an underfloor heating system requires supply temperatures of at least 35°C (see Section 2.5).

The district heating network that is used in this thesis is a two-pipe network with one supply line and one return line. The supply line delivers heat to the dwellings, where heat is extracted from the network to obtain the required comfort. The return line returns cooled water back to the heat source. Inevitably, the distribution of heat is accompanied by a certain heat loss from the network to the surroundings.
During summer, the two-pipe district heating network can potentially function as a cooling network, since no space heating is required then. This is only the case if the network does not provide domestic hot water, since hot water is required throughout the entire year. Another condition to use the network for cooling is the presence of a heat sink in the system, e.g. a low-temperature borefield (see Section 2.1.2). A heat sink allows cooling of the network by transferring heat from the network to this low-temperature heat sink. Cooling in the district heating system concepts is further handled in Sections 3.1.4 and 3.2.2.

2.4 Supplementary Heating Systems

This section briefly covers the different supplementary heating systems that are used in this thesis. In general, many different heating systems exist, e.g. gas boilers, electric heaters, heat pumps, etc. In this thesis, only heat pumps are considered as supplementary heating systems. A heat pump is a device that transfers heat from a low-grade heat source to a high-grade heat sink and it uses electrical power to do this. In other words, a heat pump is said to upgrade heat to a higher temperature. The main purpose of heat pumps is usually heating, but they can also be used for cooling. Based on the mediums between which heat is transferred, different types of heat pumps exist [44]. In this thesis three types are considered: a water-to-water heat pump, an air-to-water heat pump and an air-to-air heat pump. The working principle of these different types is the same and is based on refrigeration cycles [44].

An example of a water-to-water heat pump that is used in this thesis is a micro booster heat pump. This type of heat pump is often used in combination with a district heating network at lower temperatures [39]. This heat pump serves the purpose of lifting the temperature to the required temperature level of 55°C for domestic hot water [39]. To limit the necessary power levels and to cope with irregular domestic hot water demand, a small water storage tank is always coupled with this type of heat pump [42].

To express the efficiency of a heat pump, two measures are mainly used. The first one is the Coefficient of Performance or COP. This measure is defined as follows:

$$COP = \frac{Q_H}{W} \tag{2.4}$$

Q_H is the heat transferred to the medium at high temperature and is expressed in $Watt$. W is the electrical power, expressed in $Watt$. The COP is the measure of efficiency if a heat pump is used for heating. The second measure of efficiency is the Refrigeration Coefficient of Performance or COP_R. This measure corresponds to the cooling mode of a heat pump. It is defined as follows:

$$COP_R = \frac{Q_L}{W} \tag{2.5}$$

In this equation Q_L is the heat extracted from the low-temperature medium and W is still the electrical power. Hence, for a specific heating or cooling demand, respectively the COP and COP_R determine the corresponding electricity use of the heat pump.

2.5 Heat Emission System

This last section considers the heat emission system component in the district heating system. There are different types of heat emission systems to heat buildings. In this thesis, only underfloor heating is considered. In an underfloor heating system, pipes are embedded in a concrete layer under the floor. Hot water is sent through these pipes, which results in heating of the room above. In contrast with other widespread emission systems such as radiators or convectors, underfloor heating systems allow low supply temperatures of around 35°C [72]. This makes it very interesting to use low-grade heat sources in combination with a heat pump. After all, a heat pump generally runs more efficiently if the temperature difference between heat source and sink is smaller. Another advantage of underfloor heating is its uniform temperature distribution which provides a comfortable indoor thermal environment [7]. Yet another characteristic of these underfloor heating systems is their large thermal inertia. This can be advantageous, as it can allow for a shift in time between the production of heat and the actual heating of the room [62]. However, this large thermal inertia can also be disadvantageous, as the system has difficulties to adapt to fast changing conditions. Therefore, a good control of the system is necessary.

Chapter 3

Design of Concepts

In the previous chapter, the components of the district heating system were studied. These components are now used in the design process of concepts for the district heating system. The development of the concepts is based on a system matrix, i.e. a representation of the whole system based on its components. Following from this system matrix, a total of 13824 concepts can be theoretically formed. By applying a set of specific 'rules of thumb', the number of concepts is drastically reduced to 26 concepts. These concepts are interpreted as 13 practical concepts with versions a and b.
First, in Section 3.1, the system matrix is explained, followed by the rules of thumb in Section 3.2. Finally, an elaboration of the final concepts is provided in Section 3.3.

3.1 System Matrix

The system matrix is derived from dividing the entire system into its components, which form the building blocks of the district heating system. The matrix with its system components is shown in Figure 3.1.

The rows of the system matrix mention the system components and their variables, with a choice to be made on each row. An example of a system component is the *Seasonal Thermal Energy Storage (STES)*, for which three variables are listed with their possibilities. For all three variables, i.e. *Technology*, *Location* and *Supplementary STES*, a possibility must be selected. The same goes for the variables of other system components in the matrix.

A single concept is developed from the system matrix by completing the choices for all variables of the system components. On each row, exactly one option for the variable is chosen before continuing to the next row. By following this procedure from top to bottom for all rows of the system matrix, a single concept is formed. Clearly, different choices for the system components' variables lead to different concepts. In total, 13824 concepts can be formed this way. By applying some 'rules of thumb', this number can be drastically reduced. This will be explained in Section 3.2. First, all the different components of the system matrix are elaborated in the next sections.

System Components		Possibilities			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Figure 3.1: System matrix.

3.1.1 Seasonal Thermal Energy Storage

The first system component is the *Seasonal Thermal Energy Storage (STES)*, for which three choices must be made. Firstly, there is the choice of technology between the following STES options: *Tank*, *Borefield (< 25° C)* and *Borefield (> 25° C)*. These technologies are explained in Chapter 2. Other STES options that were mentioned in that chapter are not considered in the further assessment, i.e. pit storage and aquifer storage, based on the following reasoning:

- Pit storage is conceptually similar to tank storage in case water is used as a storage medium. Therefore, if water is used as storage medium, both storage types are interchangeable in the overall district heating system. The different construction method of pit storage would however lead to different heat losses compared to tank storage, depending on the level of insulation that is applied. As this design variable is case-dependent, the distinction between tank and pit storage is not well-defined, because they can have better or worse insulation. In what follows, only tank storage with a well-defined degree of insulation is considered.

- Aquifer storage is not considered due to its limited choice of locations. This results from a limited occurrence of aquifers and strong hydro-geological requirements, as was mentioned in Section 2.1.2.

The system matrix shows a distinction between a low-temperature borefield option, limited to 25°C and a high-temperature borefield option, without a theoretical temperature limit. There are two separate reasons to make this distinction: a first, non-technical reason is that Belgian laws are unclear whether the ground temperature in a borefield should be limited to 25°C, as is the case for aquifers [71]. Because of this unclarity, a distinction is made based on the same temperature. A second, technical reason is that a low-temperature borefield (< 25°C) is balanced over a year, meaning that the average ground temperature is more or less constant. This is achieved by balancing the energy that is injected in the borefield and the energy that is extracted, including the heat losses. On the other hand, a high-temperature borefield (> 25°C) only has a lower temperature limit of 0°C and its upper temperature limit is increased, which allows more energy to be stored in the borefield. In theory, there is no upper limit, but in practice this will be the case due to heat losses and specific permits. The ground temperature in a high-temperature borefield can increase over the years by adding more energy to the borefield than is extracted and lost. Therefore, a high-temperature borefield is said to be imbalanced.

The second variable for the Seasonal Thermal Energy Storage is the location of the STES. This variable offers a choice between a centralised storage and a decentralised storage. A centralised storage is located at a central position in the district, whereas a decentralised storage is located at each individual dwelling.

Lastly, there is the choice whether a supplementary STES is added to the system. In that case, the supplementary STES is a tank that acts as a centralised storage for hot water. The hot water is delivered to the dwellings by a separate heating network with a supply temperature of 55°C. This means that two heating networks are present in that case: one for space heating and one for domestic hot water. As will be explained in Section 3.2, this supplementary tank is only used in combination with a low-temperature borefield with centralised solar collectors for hot water production.

3.1.2 Solar Collectors Space Heating

The second component in the system matrix consists of the solar collectors for space heating. Both the location and the technology of the collectors have to be selected. Regarding the location, the solar collectors can be either centralised at a solar field or decentralised, whereby the collectors are placed on the rooftops of the individual houses. Regarding the technology, there is the choice between flat plate collectors and vacuum tube collectors (see Section 2.2).

3.1.3 Solar Collectors DHW

Similarly to the solar collectors for space heating, there are two variables for the solar collectors for domestic hot water. The location can be either centralised or decentralised, corresponding to a central solar field and collectors on the rooftops respectively. In case the solar collectors are decentralised, they are part of a local hot water production unit, that also includes a storage tank and domestic hot water (DHW) heat pump (see subsection 3.1.6 on Supplementary Heating Systems). Finally, the technology offers again the choice between flat plate collectors and vacuum tube collectors.

3.1.4 Cooling

The fourth system component in the matrix is cooling of the houses during the summer months. This can possibly be integrated in the system. In that case, cooling is provided to the dwellings via the district heating network. On the other hand, a separate cooling system can be added in each dwelling. In that case, no cooling is provided by the heating network. This separate system is an air-to-air heat pump, with an outdoor unit and indoor units in the rooms that require cooling.

3.1.5 District Heating Network

The fifth system component is the district heating network, which distributes heat at certain temperature levels. As mentioned before, the first three temperature levels (55, 45 and 35°C) constitute three options for the supply temperature of the district heating network that allow direct space heating (see Section 2.3). The last option on this row is a district heating network with an average supply temperature of 15°C, making it incapable of providing direct heating.

3.1.6 Supplementary Heating System

The sixth component in the system matrix consists of supplementary heating systems. Both for space heating and domestic hot water, a supplementary system can be added.

Regarding supplementary systems for space heating, there is a choice between a large central heat pump and a small decentral heat pump. The large central heat pump is a water-to-water heat pump, located centrally in the district, that provides supplementary heating for all dwellings. The small decentral heat pump is a water-to-water heat pump, located at each dwelling, that provides individual supplementary space heating. Besides, there is also the option of not adding a supplementary space heating system at all.

Regarding supplementary systems for domestic hot water, two different heat pumps are listed on this row: a micro booster heat pump and a domestic hot water (DHW) heat pump. The micro booster heat pump is a water-to-water heat pump, producing hot water in each individual dwelling by using water from the district heating network as source (see Section 2.4). It includes a small storage tank to locally store hot water that is produced with the micro booster heat pump. The domestic hot water (DHW) heat pump is an air-to-water heat pump that is part of a local hot water production unit. This local unit for DHW also includes a storage tank and decentralised solar collectors on the rooftop. Lastly on this row, there is also the option of not using any supplementary systems for domestic hot water.

3.1.7 Emission System

The seventh and last component in the system matrix is the heat emission system. On this row, the system matrix only provides one option, i.e. underfloor heating. Looking at low-energy buildings, underfloor heating provides a more energy-efficient solution compared to traditional radiators. As mentioned in Section 2.5, it is the only option considered in this thesis.

This concludes the explanation of the components of the system matrix. The next step in finding the relevant concepts is applying some rules of thumb to these components and their relationships.

3.2 Rules of Thumb

Considering all combinations in the system matrix, a total of 13824 concepts can be formed. However, not all combinations are useful and the number of concepts can be narrowed to 26 concepts by applying a total of ten rules of thumb. These rules of thumb result from both reasoning and basic calculations. They can be divided into two groups: on the one hand there are three 'excluding rules', which exclude options from the system matrix. On the other hand, there are seven 'incompatibility rules', which prohibit the combination of certain options (e.g. choice X for the technology of the STES and choice Y for the location of the solar collectors cannot be chosen together). In what follows, both types of rules are further elaborated.

3.2.1 Excluding Rules

The first set of rules are three excluding rules, which exclude some of the possibilities in the system matrix. Figure 3.2 shows the excluding rules in the system matrix. These rules are explained below.

System Components		**Possibilities**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Excluding Rule

Figure 3.2: System matrix with the excluding rules.

First Excluding Rule

"A decentralised location of the Seasonal Thermal Energy Storage is excluded."

The first excluding rule prescribes that a decentralised location of the STES is excluded and that the location of the STES should therefore always be centralised. There are several reasons for applying this rule of thumb:

- It is assumed that the space in dwellings is too limited to apply decentralised STES.
- A single centralised STES reduces heat losses in the storage by decreasing the area to volume ratio compared to multiple decentralised storages. Since an energy efficient solution is preferred, centralised STES is clearly the better option.

- A centralised STES benefits from economies of scale, since the specific investment cost (€/m^3) decreases with an increasing storage volume [32].

Second Excluding Rule

"A decentralised location of the solar collectors for space heating is excluded."

The second excluding rule excludes the option of a decentralised location of the solar collectors for space heating. Reasons to only consider a centralised location of the solar collectors for space heating are as follows: firstly, there are less heat losses during transport from the solar collectors to the STES, since the first excluding rule prescribes a centralised location of the STES as well. Secondly, there is more freedom to find the optimal orientation of the collectors in a central solar field to increase their energy yield, compared to a limited orientation freedom on rooftops. Thirdly, the installation and maintenance of a central solar collector field is easier compared to solar collectors on rooftops. It is important to note that this second excluding rule only holds under the condition that space is available to install a central solar field. Since this was stated as a starting point in Section 1.1, the condition is fulfilled and the second excluding rule is applied.

Third Excluding Rule

"The district heating network with a supply temperature of 45° C is excluded."

The third excluding rule considers the three temperature levels of the district heating network that can provide direct space heating. Since a temperature of around 55°C is required for domestic hot water, the network with a supply temperature of 55°C can directly provide domestic hot water in each dwelling via an instantaneous heat exchanger.

In the case of a district heating network with a supply temperature of 35 or 45°C, hot water cannot be provided directly and supplementary heating with a micro booster heat pump is required. A comparison between these two supply temperatures on a primary energy basis leads to the conclusion that the network at 45°C can be excluded as an option. Indeed, calculations show that the decrease in heat loss for a network at 35°C outweighs the surplus in electricity use of the micro booster heat pump, compared to a network at 45°C. These calculations are provided in Appendix A.1.

3.2.2 Incompatibility Rules

The second set of rules are seven incompatibility rules. These rules prescribe that certain options from different rows in the system matrix cannot be combined with each other.

First Incompatibility Rule

"A supplementary tank for DHW is incompatible with STES technologies at higher temperatures."

For the STES system component, the option of a supplementary tank on the third row is incompatible with the STES technologies *Tank* and *Borefield (>25°C)* on the first row. This rule of thumb follows from the supplementary tank's function as an energy storage for hot water. In case of a tank and high-temperature borefield STES, energy can be stored at higher temperatures (higher exergy), allowing the production of hot water. Therefore, it is unnecessary to add a supplementary tank for hot water to a system with tank or high-temperature borefield STES.

On the contrary, in a low-temperature borefield (<25°C), energy is stored at lower temperatures (lower exergy). Starting from low temperatures, the production of hot water would require a lot of electricity use for the heat pumps. Therefore, a supplementary tank for hot water storage can be added to the district heating system with a low-temperature borefield. This is only the case when the solar collectors for hot water are located centrally. In the other case when solar collectors are located decentralised on the rooftops, the hot water is produced locally and the supplementary tank is unnecessary.

A second reason to add a supplementary tank to a system with low-temperature borefield storage and central solar collectors for DHW, is that a low-temperature borefield allows for cooling to be integrated in the system (see below: *Fourth Incompatibility Rule*). This means that during the summer months, the district heating network is used to provide cooling to the dwellings. Hence, hot water cannot be provided from a central position via the existing district heating network and a supplementary tank with a separate heating network is required.

Figure 3.3 illustrates the first incompatibility rule for a tank STES in the system matrix. The following incompatibility rules can be understood from the system matrix in a similar way.

System Components		**Possibilities**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Incompatibility Rule

Figure 3.3: System matrix with the first incompatibility rule.

Second Incompatibility Rule

"If the solar collectors for DHW are centralised, the technology of these collectors should be the same as technology of the solar collectors for space heating."

The second incompatibility rule concerns the technology variable for the *Solar Collectors DHW*. This rule prescribes that if the location of the solar collectors for DHW is centralised, the technology of these collectors should correspond to the technology of the solar collectors for space heating. Indeed, in that case all the collectors are centralised at the solar collector field and based on the ease of maintenance, the same collector technology is considered for all of them. In other words, centralised flat plate collectors for DHW are incompatible with centralised vacuum tube collectors for space heating. The same goes for centralised vacuum tube collectors for DHW and flat plate collectors for space heating.

Third Incompatibility Rule

"A decentralised location of the solar collectors for DHW is incompatible with vacuum tube collector technology."

This rule of thumb concerns the decentralised location of solar collectors for DHW. The collectors on the rooftops of the houses are always flat plate collectors, since this technology turns out to be the most beneficial for local hot water production. This is concluded from calculating the Net Present Value (NPV) for two types of local hot water production units, each with a different collector technology. The calculation is provided in Appendix A.2.

Fourth Incompatibility Rule

"In case of tank storage or high-temperature borefield storage, cooling cannot be integrated in the heating network."

The fourth incompatibility rule considers cooling in a system with STES at higher temperatures, i.e. tank and high-temperature borefield storage. If either of these storage technologies is part of the overall district heating system, it will be impossible to provide cooling via the district heating network since the temperatures in the storage are too high during the summer months. Hence, the option "integrated" for cooling is incompatible with these STES options and a separate cooling system must be added. Following from this, only a system with low-temperature borefield storage is able to provide integrated cooling via the district heating network.

Fifth Incompatibility Rule

"A district heating network with a supply temperature of 15°C is incompatible with seasonal tank storage or high-temperature borefield storage."

This next rule of thumb concerns the district heating network with an average supply temperature of 15°C. Such a low transport temperature is incompatible with STES at higher temperature, i.e. tank and high-temperature borefield. Indeed, the exergy present in the STES would be destroyed for low-temperature transport and this temperature level would require upgrading of the heat in each dwelling both for space heating and hot water. The district heating network at 15°C is therefore only considered in combination with STES at lower temperatures, i.e. a low-temperature borefield.

Sixth Incompatibility Rule

"A district heating network with a supply temperature of 55°C is incompatible with a low-temperature borefield and with the option of decentralised solar collectors for DHW."

The sixth incompatibility rule first prescribes that the district heating network with a supply temperature of 55°C is incompatible with a low-temperature borefield. This is due to the district heating network only providing space heating, as hot water is either centrally stored in a supplementary tank or locally produced in each dwelling. Since space heating through an underfloor heating system is possible with temperatures slightly above room temperature, a supply temperature of 55°C is unnecessary. It is therefore said to be incompatible.

Furthermore, the district heating network at 55°C is incompatible with the option of decentralised solar collectors for DHW. The same reasoning is applied here: since in this case the district heating network only provides space heating and hot water is produced locally, a supply temperature of 55°C is unnecessary for the underfloor heating in each dwelling.

Seventh Incompatibility Rule

The seventh and last incompatibility rule consists of a set of rules concerning the supplementary heating systems.

Firstly, the rule of thumb prescribes that a large central heat pump for space heating is incompatible with a district heating network with an average supply temperature of 15°C. This follows directly from the low transport temperature, which can be provided by the STES, i.e. a low-temperature borefield in this case. Hence, no supplementary heating is required before transport.

Secondly, a small decentral heat pump is incompatible with a district heating network at 35 or 55°C. This is due to the fact that dwellings are equipped with an underfloor heating system, which allows direct space heating with these temperatures. A small decentral heat pump will therefore only provide supplementary space heating if a network at 15°C is used.

Thirdly, the option of no supplementary space heating is incompatible with both low- and high-temperature borefields. For a low-temperature borefield, this follows from the limited storage temperatures below 25°C. As the temperatures throughout the year will be too low to provide space heating, some form of supplementary heating is necessary to attain higher temperatures. For a high-temperature borefield, supplementary space heating is necessary for at least the first couple of years, when the borefield has not reached sufficiently high temperatures yet.

Fourthly, a micro booster heat pump is incompatible with a district heating network with 55°C supply temperature. This is because of the transport temperature of 55°C being sufficiently high to directly provide hot water in the dwellings. Furthermore, the micro booster heat pump is incompatible with decentralised solar collectors for DHW, since this means that each dwelling is equipped with a local production unit that foresees hot water. The district heating network will in that case only provide space heating and adding a micro booster heat pump is unnecessary.

Fifthly, the option for a domestic hot water (DHW) heat pump is incompatible with centralised solar collectors for DHW. Only when decentralised solar collectors are used for DHW, as part of a local production unit for hot water, a DHW heat pump is added to this local system to provide supplementary heating.

Lastly, the option of no supplementary heating for DHW is incompatible with heating networks at 35 and 15°C. Not adding supplementary heating for DHW to the system is only an option in case of a district heating network at 55°C. Indeed, hot water production requires temperatures around 55°C, which can only be directly provided with the network at 55°C.

This concludes the part on the rules of thumb, consisting of three excluding rules and seven incompatibility rules. The next step is to form the final concepts of the district heating system.

3.3 Final Concepts

A total of 26 concepts is formed by making a choice in each row of the matrix and meanwhile respecting all the excluding and incompatibility rules. These concepts can be interpreted as 13 practical concepts, each with two versions: a and b. The only difference between these a and b versions is the solar collector technology used to capture heat for space heating. The a versions have flat plate collectors while the b versions have vacuum tube collectors. For all the other parts of the system, these versions are exactly the same. Recall that if the solar collectors for domestic hot water are placed centrally, the same collector technology as for the solar collectors for space heating should be used (see *Second Incompatibility Rule*). In other words, if the solar collectors for domestic hot water are placed centrally, these collectors are flat plate collectors in version a and vacuum tube collectors in version b.

This section gives an overview of the resulting 13 concepts, each with versions a and b. The concepts can be divided into three subgroups depending on the seasonal thermal storage technology that is used. This gives a group of six concepts with tank storage, a group of four concepts with low-temperature borefield storage (<25°C) and a group of three concepts with high-temperature borefield storage (>25°C). In what follows a description of each of these concepts is given, as well as a schematic overview of the system and the corresponding filled-in system matrix. To limit the number of figures, only the system matrix of version a is given for each concept.

3.3.1 Concepts with Tank Storage

There are six concepts in which a tank storage is used. Each with two versions a and b, depending on their solar collector technology for space heating. The concepts with an uneven number (concepts 1, 3 and 5) use a heat exchanger to transfer heat from the tank to the district heating network. In the concepts with an even number (concepts 2, 4 and 6) a central heat pump is placed between the tank and the district heating network. In this way, it is possible that the temperature of the water in the central storage tank can drop below the supply temperature of the district heating network. As a result of this, a smaller storage tank can be used compared to the equivalent concepts without central heat pump. However, this comes at the cost of a large central heat pump and its corresponding electricity use. Note that as long as the temperature in the central storage tank remains above the supply temperature of the heating network, this central heat pump is not used. In that case, still regular heat exchange occurs between the central storage tank and the heating network.

Concepts 1 and 2

Along with a centralised tank storage, these two concepts have centralised solar collectors for domestic hot water, an individual cooling system in each dwelling and a district heating network with a supply temperature of 55°C and a return temperature of 25°C. The difference between concepts 1 and 2 is the placement of a heat pump in between the central storage tank and the heating network. As mentioned above, the addition of a central heat pump can allow for a smaller central storage tank. Remember that for concepts 1b and 2b, the only difference with 1a and 2a is the choice of solar collector technology for space heating. In concepts 1b and 2b, vacuum tube collectors are used instead of flat plat collectors. In concepts 1 and 2 the solar collectors for domestic hot water are also placed centrally, so these should also be flat plate collectors in version a and vacuum tube collectors in version b.

Concepts 1a and 1b There is no supplementary heating for space heating or domestic hot water. This means that the tank storage should provide all the necessary heat throughout the year. Therefore, it is probably necessary to slightly oversize these systems to cope with extremely cold years or summers with low solar irradiation. After all, if not enough heat is stored during summer or if the heat demand is exceptionally large, there is no system that can supply this heat. This is a major disadvantage of these concepts.
Figures 3.4 and 3.5 respectively show a schematic overview for concepts 1a and 1b and the system matrix of concept 1a.

Concepts 2a and 2b Concept 2 is very similar to concept 1. The only difference between these two concepts, is the use of a heat pump in between the central storage tank and the district heating network instead of only a heat exchanger. As stated above, this allows for a smaller central tank, as the temperature in the tank can drop below the supply temperature of the district heating network. Moreover, the central heat pump acts as a supplementary heating system. Therefore, there is no need to oversize these systems to cope with extremely cold winters or low irradiation summers. The difference between 2a and 2b, is again the collector technology.
Figures 3.6 and 3.7 respectively show a schematic overview of the concepts and the system matrix of concept 2a.

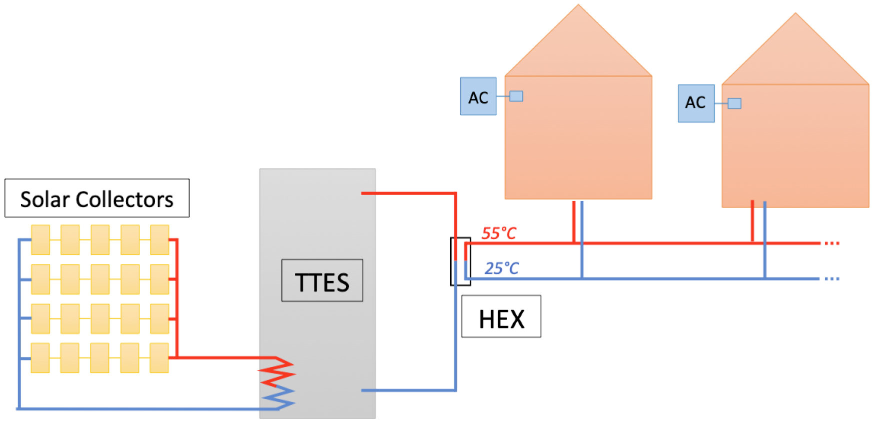


Figure 3.4: Schematic system overview of concepts 1a and 1b (HEX: Heat exchanger, TTES: Tank Thermal Energy Storage, AC: Air Conditioning).

System Components		Concept 1 a			
Seasonal Thermal Energy Storage (STES)	Technology	**Tank**	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept   Excluding Rule   Incompatibility Rule

Figure 3.5: System matrix of concept 1a.

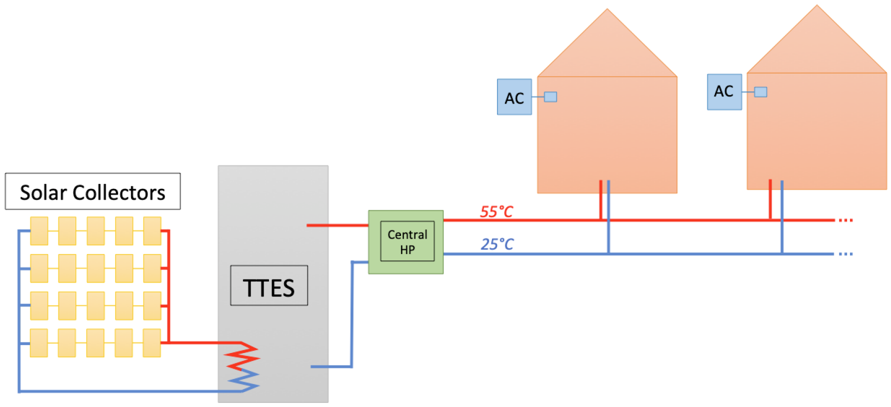


Figure 3.6: Schematic system overview of concepts 2a and 2b (TTES: Tank Thermal Energy Storage, AC: Air Conditioning).

System Components		Concept 2 a			
Seasonal Thermal Energy Storage (STES)	Technology	**Tank**	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.7: System matrix of concept 2a.

Concepts 3 and 4

Concepts 3 and 4 differ from concepts 1 and 2 in the temperature of the district heating network. In concepts 3 and 4, a district heating network with a supply temperature of 35°C instead of 55°C is used. This temperature is too low for domestic hot water, for which a minimum temperature of 55°C is necessary. Therefore, a micro booster heat pump is added in each dwelling. This micro booster heat pump is a water-to-water heat pump connected to a small, local domestic hot water tank. It uses the district heating network as a source to provide the necessary heat for domestic hot water supply. The small domestic hot water tank is the heat sink of this heat pump.

As in concepts 1 and 2, all solar collectors are centralised and each dwelling has its own cooling system. As was the case for the difference between concepts 1 and 2, the difference between concept 3 and 4 is also the placement of a central heat pump in between the central storage tank and the heating network.

Concepts 3a and 3b Concept 3 only has a heat exchanger in between the central tank and the heating network. So, there is no supplementary heating system for space heating. The highest temperature of the central storage tank should thus always remain above the network temperature of 35°C. For domestic hot water, the micro booster heat pump acts as a supplementary heating system.
Figures 3.8 and 3.9 respectively show a schematic system overview of concepts 3a and 3b and the system matrix of concept 3a.

Concepts 4a and 4b Concept 4 has a central heat pump in between the central storage tank and the heating network. Again, this gives the possibility to lower temperatures in the tank below the temperature of the supply temperature of the heating network, allowing for a smaller storage tank. This also means that there is now a supplementary heating system for both the domestic hot water with the micro booster heat pump and for space heating with the central heat pump. Oversizing is thus not necessary.
Figures 3.10 and 3.11 respectively show a schematic system overview of concept 4 and the system matrix of concept 4a.

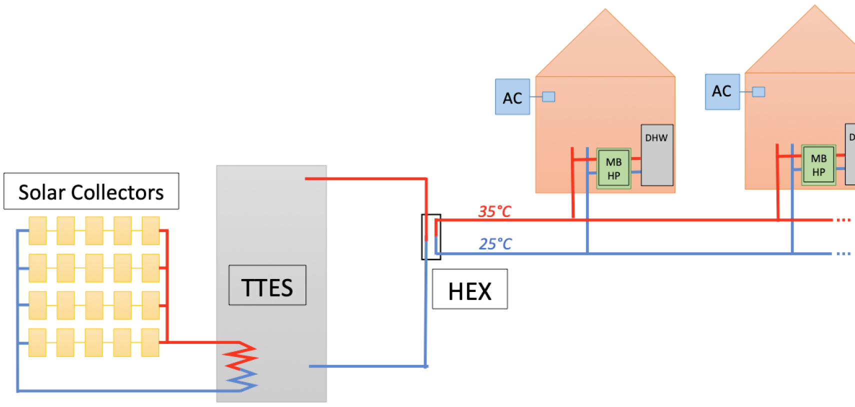


Figure 3.8: Schematic system overview of concepts 3a and 3b (HEX: Heat exchanger, TTES: Tank Thermal Energy Storage, AC: Air Conditioning, MB HP: Micro Booster Heat Pump).

System Components		**Concept 3 a**			
Seasonal Thermal Energy Storage (STES)	Technology	**Tank**	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept   Excluding Rule   Incompatibility Rule

Figure 3.9: System matrix of concept 3a.

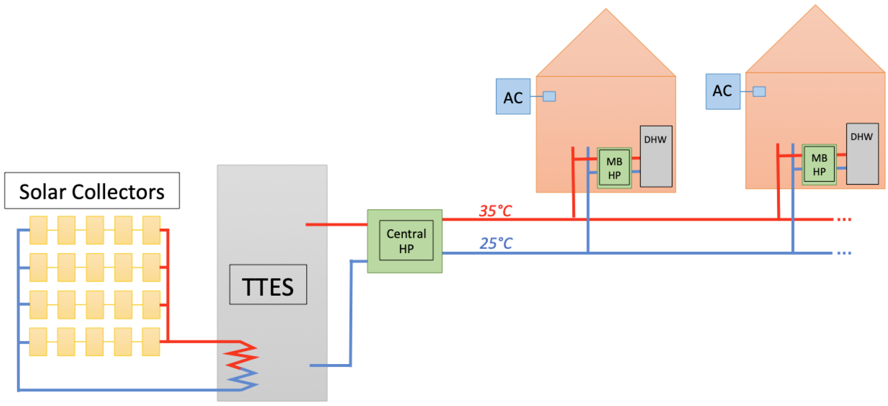


Figure 3.10: Schematic system overview of concepts 4a and 4b (TTES: Tank Thermal Energy Storage, AC: Air Conditioning, MB HP: Micro Booster Heat Pump).

System Components		**Concept 4 a**			
Seasonal Thermal Energy Storage (STES)	Technology	**Tank**	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.11: System matrix of concept 4a.

Concepts 5 and 6

In concepts 5 and 6, the supply temperature of the district heating network remains at 35°C. Each dwelling still has an individual cooling system as well. However, the difference with concepts 3 and 4 is that in concepts 5 and 6, the supply of domestic hot water is handled by an individual hot water production unit in each dwelling. Each dwelling has its own solar collectors coupled with a domestic hot water storage tank, which in turn is connected to an air-to-water DHW heat pump. During summer, most of the domestic hot water demand is covered by these decentral solar collectors, whereas during winter, the DHW heat pump has to operate. The supply of heat for space heating is still done via the heating network, which is connected to the central storage tank with either a heat exchanger or a central heat pump.

An important feature of these concepts with a completely individual system for domestic hot water is worth mentioning. During summer, it is possible that the temperature in these local storage tanks reaches near the boiling point of water. This results from the abundantly available solar energy and the limited tank volume. Since boiling has to be avoided, further heat addition to the tanks is not allowed at these moments, regardless of the ability of the collectors to still collect heat. At moments of solar energy abundance, not using or not collecting all solar heat is a possibility. However, another possibility is to still collect this heat and to transfer it to the central seasonal storage tank. After all, these moments of heat surplus occur in summer when there is no heat demand for space heating. Hence, the district heating net is not being used and allows transferring heat from these decentral collectors to the central seasonal storage tank. In this way, all heat can still be collected without boiling in the domestic hot water tanks and as a consequence, probably less centralised solar collectors for space heating are necessary.

Concepts 5a and 5b In concept 5, only a heat exchanger connects the central storage tank with the district heating network. This again means no supplementary heating system for space heating is present and oversizing might be necessary. As already mentioned, concept 5 has individual cooling systems in each dwelling, a district heating net with a supply temperature of 35°C and completely individual systems for domestic hot water.
Figures 3.12 and 3.13 respectively show a schematic system overview of this concept and the system matrix of concept 5a.

Concepts 6a and 6b Concept 6 is almost the same as concept 5, except again for the central heat pump that is placed between the central seasonal storage tank and the heating network. As both space heating and domestic hot water have a supplementary heating system, oversizing is not necessary. Moreover, concept 6 also has individual cooling systems in each dwelling, a district heating net with a supply temperature of 35°C and completely individual systems for domestic hot water.
Figures 3.14 and 3.15 respectively show a schematic system overview of this concept and the system matrix of concept 6a.

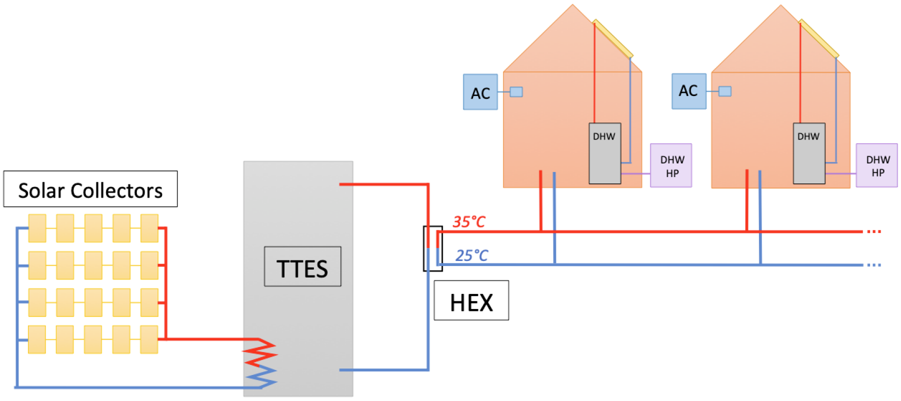


Figure 3.12: Schematic system overview of concepts 5a and 5b (HEX: Heat exchanger, TTES: Tank Thermal Energy Storage, AC: Air Conditioning, DHW HP: Domestic Hot Water Heat Pump).

System Components		**Concept 5 a**			
Seasonal Thermal Energy Storage (STES)	Technology	**Tank**	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.13: System matrix of concept 5a.

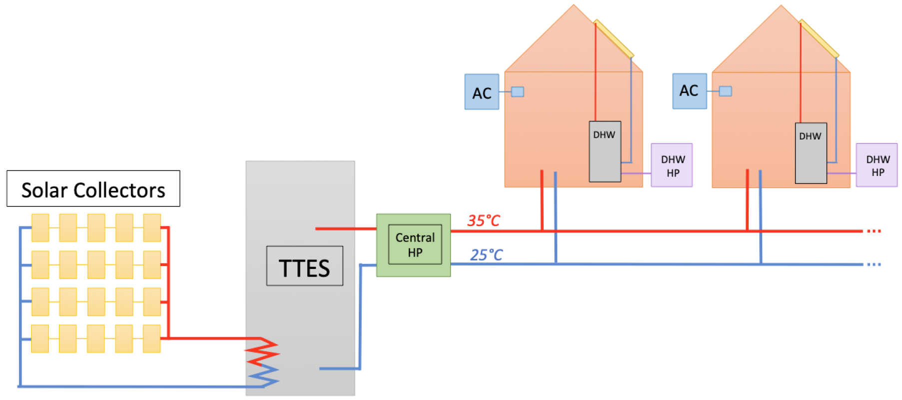


Figure 3.14: Schematic system overview of concepts 6a and 6b (TTES: Tank Thermal Energy Storage, AC: Air Conditioning, DHW HP: Domestic Hot Water Heat Pump).

System Components		Concept 6 a			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept   Excluding Rule   Incompatibility Rule

Figure 3.15: System matrix of concept 6a.

3.3.2 Low-temperature Borefield Concepts

There are four concepts in which a borefield with soil temperatures below 25°C is used. For sake of simplicity, these concepts are referred to as low-temperature borefield concepts. Thanks to these low temperatures, also cooling can now be provided by the central system. Although a heat pump is still necessary to provide this cooling. There is no need anymore for the individual cooling systems in each dwelling. Along with this cooling possibility, a buffer tank has to be provided to limit peak loads on the borefield. This tank, for example, buffers the heat coming from the solar collectors or the district heating network. This gives the possibility to better control and limit the loads on the borefield.

As prescribed by the first incompatibility rule (see Section 3.2.2), heat for domestic hot water is not stored in the borefield if a borefield at low temperatures is used. So, only heat for space heating is supplied using the borefield. The heat stored in the borefield is either heat coming from the central solar collectors or heat coming from the heating network. For the domestic hot water a separate system has to be provided, as this heat is not stored in the borefield. There are two possibilities for these separate systems.

A first option is to provide a supplementary seasonal storage tank solely for domestic hot water. In that case a second heating network is provided to supply the heat from this central tank to the dwellings. This second heating network has a supply temperature of 55°C. In this first option, only the case without a supplementary heating system for the domestic hot water is considered. Hence, it is assumed that no heat pump is provided for domestic hot water. If this would turn out to be an interesting concept, the assumption can be refined and supplementary heating systems for domestic hot water can potentially be added. This first option is considered in concepts 7 and 8.

A second option is to provide individual domestic hot water systems in each dwelling, similarly to concepts 5 and 6. In that case, each dwelling has its own solar collectors, heat pump and small storage tank solely for domestic hot water. Since now also cooling is provided by the network, there is no possibility anymore to send the abundant solar energy from the decentral collectors to the central storage, as was the case for concept 5 and 6. After all, the network is used for cooling during summers. This second option is applied in concepts 9 and 10.

The heat for space heating is stored in the borefield. Since this heat is stored at temperatures below 25°C, a heat pump is always necessary to increase this temperature to 35°C. Either a large central heat pump is used or each dwelling is provided with its own heat pump. Note that both these heat pumps are also used to provide cooling during summers. In the case of a large central heat pump, the heating network has a supply temperature of 35°C in heating mode and a temperature of 16°C in cooling mode. In the case of individual heat pumps in each dwelling, the heating network has a fluctuating supply temperature in heating mode, depending on the borefield temperature, with an estimated average around 15°C. However, these supply temperatures are low and close to the borefield temperature, as only the buffer tank separates the borefield and the heating network.

Concepts 7 and 8

Concepts 7 and 8 both make use of a supplementary seasonal storage tank to provide the heat for domestic hot water. Recall that this also means that a second heating network is required, solely for the supply of heat for domestic hot water. Compared to the concepts with tank storage, cooling is integrated in the heating network and no individual cooling systems are provided. The difference between concept 7 and 8 is the placement of the heat pump that provides space heating and cooling.

Concepts 7a and 7b In concept 7, there is one large central heat pump that provides space heating and cooling. It also has a central storage tank for domestic hot water. Figure 3.16 shows the schematic overview of concepts 7a and 7b. Note that in the schematic overview, only the heating mode of the district heating system is shown. Again the difference between a and b is the solar collector technology for space heating. The system matrix of concept 7a is shown in Figure 3.17.

Concepts 8a and 8b As stated above, the difference between concept 7 and concept 8 is the placement of the heat pump that provides space heating and cooling. In concept 7, this heat pump is placed centrally, while in concept 8 this heat pump is placed in each dwelling separately. This means the district heating network has a temperature of more or less 15°C in heating mode. Concept 8 also has a central storage tank for domestic hot water with an accompanying district heating network at 55°C. The schematic overview of concepts 8a and 8b is shown in Figure 3.18. Note that in the schematic overview, only the heating mode of the district heating system is shown. Again the difference between a and b is the solar collector technology for space heating. The system matrix of concept 8a is shown in Figure 3.19.

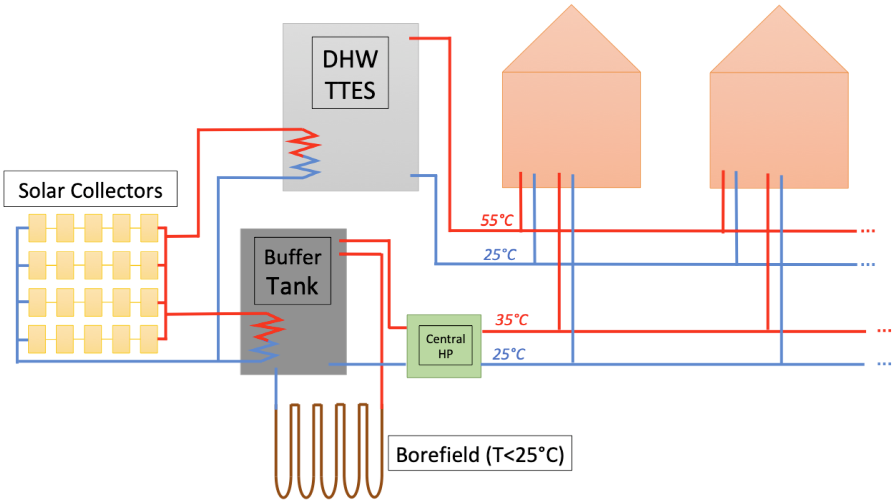


Figure 3.16: Schematic system overview of concepts 7a and 7b (TTES: Tank Thermal Energy Storage).

System Components		**Concept 7 a**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	**Borefield (<25°C)**	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept　Excluding Rule　Incompatibility Rule

Figure 3.17: System matrix of concept 7a.

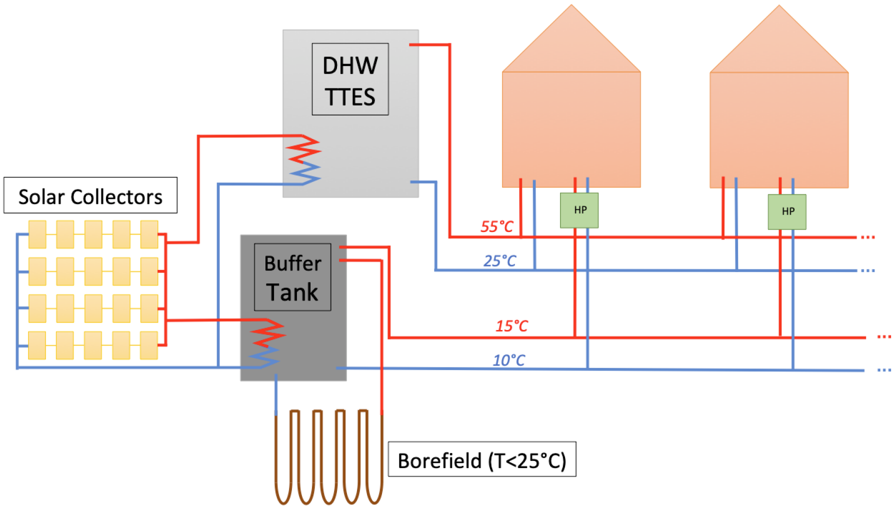


Figure 3.18: Schematic system overview of concepts 8a and 8b (TTES: Tank Thermal Energy Storage).

System Components		**Concept 8 a**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	**Borefield (<25°C)**	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.19: System matrix of concept 8a.

Concepts 9 and 10

As was the case for concepts 7 and 8, also here the difference between concept 9 and concept 10 is the placement of the heat pump for space heating and cooling. Concept 9 has the large central heat pump, while concept 10 has heat pumps in each dwelling. The difference with concepts 7 and 8 is that concept 9 and 10 use a completely decentralised approach for domestic hot water. Thus, each dwelling has its own hot water production unit that combines solar collectors, an air-to-water heat pump and a storage tank. Note that for concept 10 consequently two heat pumps are present in each dwelling. One air-to-water heat pump for the domestic hot water and one water-to-water heat pump for space heating and cooling.

Concepts 9a and 9b In concept 9, a central heat pump is used to provide space heating and cooling. For domestic hot water, each dwelling has its own hot water production unit. The schematic overview of concept 9 is shown in Figure 3.20, while the system matrix of concept 9a is shown in Figure 3.21. Note that in the schematic overview, only the heating mode of the district heating system is shown.

Concepts 10a and 10b In concept 10, each dwelling has its own heat pump that provides space heating and cooling. Moreover, each dwelling has another heat pump as part of the above mentioned hot water production unit. The schematic overview of concept 10 is shown in Figure 3.22, while the system matrix of concept 10a is shown in Figure 3.23. Note that in the schematic overview, only the heating mode of the district heating system is shown.

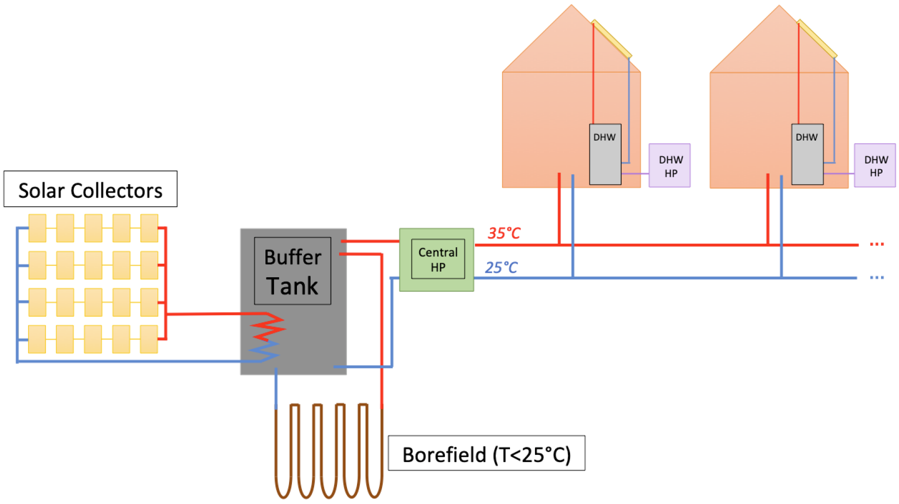


Figure 3.20: Schematic system overview of concepts 9a and 9b (DHW HP: Domestic Hot Water Heat Pump).

System Components		**Concept 9 a**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	**Borefield (<25°C)**	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.21: System matrix of concept 9a.

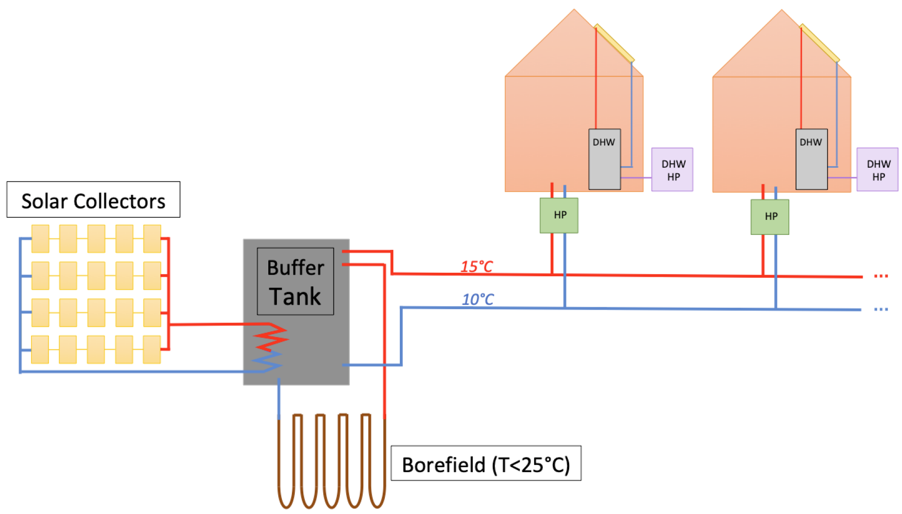


Figure 3.22: Schematic system overview of concepts 10a and 10b (DHW HP: Domestic Hot Water Heat Pump).

System Components		**Concept 10 a**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	**Borefield (<25°C)**	Borefield (>25°C)	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.23: System matrix of concept 10a.

3.3.3 High-temperature Borefield Concepts

Three concepts have a borefield in which soil temperatures can rise above 25°C. For sake of simplicity, these concepts are referred to as high-temperature borefield concepts. As prescribed by the fourth incompatibility rule (see Section 3.2.2), cooling can not be provided anymore by the central system. Again, individual cooling systems are provided in each dwelling. The three concepts with a high-temperature borefield storage are very similar to concepts 2, 4 and 6 in which a seasonal storage tank is used instead of a borefield at high temperatures. Conceptually the three borefield concepts are the same as these three tank concepts, except thus their seasonal storage technology. As was the case for borefield at low temperatures, here also a buffer tank is required to control and limit the loads on the borefield.

Concept 11 Concept 11, which is similar to concept 2, has a large central heat pump, a district heating net with a supply temperature of 55°C and centralised collectors for both space heating and domestic hot water. Since a supplementary heating system is present, there is no need for oversizing of the collectors or the borefield.

Concept 12 Concept 12, which is similar to concept 4, has a large central heat pump, a district heating net with a supply temperature of 35°C and only centralised collectors. Since a temperature of 55°C is necessary for domestic hot water, each dwelling is equipped with a micro booster heat pump connected to a small domestic hot water tank. Again, thanks to the supplementary heating systems there is no need to oversize the borefield or the collectors.

Concept 13 Concept 13, which is similar to concept 6, also has a large central heat pump and a district heating network with a supply temperature of 35°C. However, for the domestic hot water it does not use a micro booster heat pump, but a completely decentralised system in each dwelling. Again these decentralised domestic hot water systems consist of solar collectors, an air-to-water heat pump and a small storage tank. Since cooling is provided by separate systems in each dwelling, the heating network is again available during summers to send the surplus of heat from the decentral collectors to the central storage system.

Figures 3.24, 3.26 and 3.28 respectively show the schematic overviews of concepts 11, 12 and 13. The system matrices of concepts 11a, 12a, and 13a can be respectively found in figures 3.25, 3.27 and 3.29.

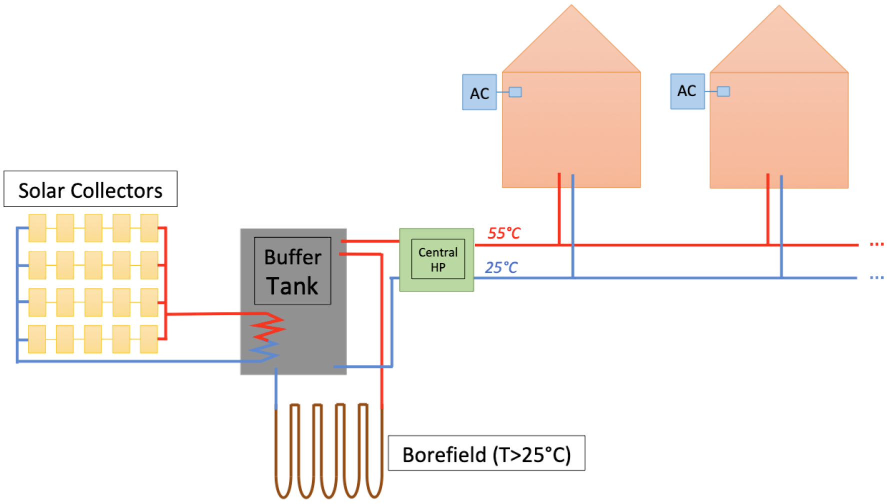


Figure 3.24: Schematic system overview of concepts 11a and 11b (AC: Air Conditioning).

System Components		Concept 11 a			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	**Borefield (>25°C)**	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.25: System matrix of concept 11a.

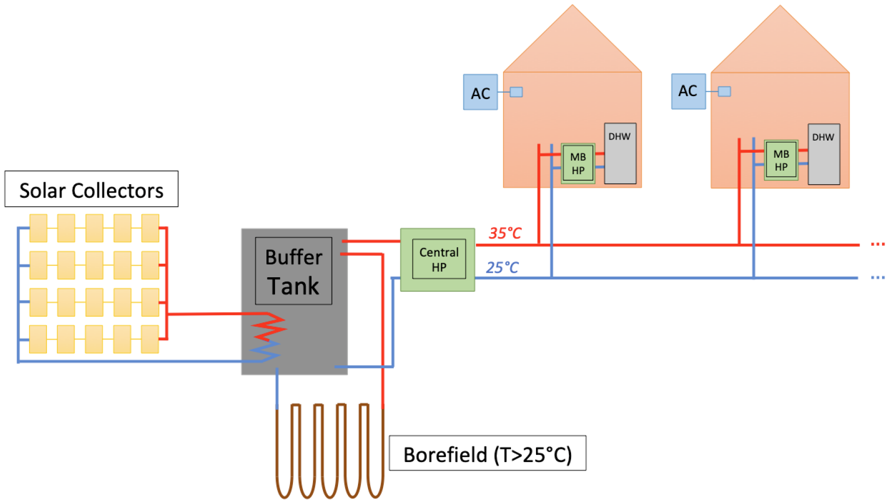


Figure 3.26: Schematic system overview of concepts 12a and 12b (AC: Air Conditioning; MB HP: Micro Booster Heat Pump).

System Components		**Concept 12 a**			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	**Borefield (>25°C)**	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept   Excluding Rule   Incompatibility Rule

Figure 3.27: System matrix of concept 12a.

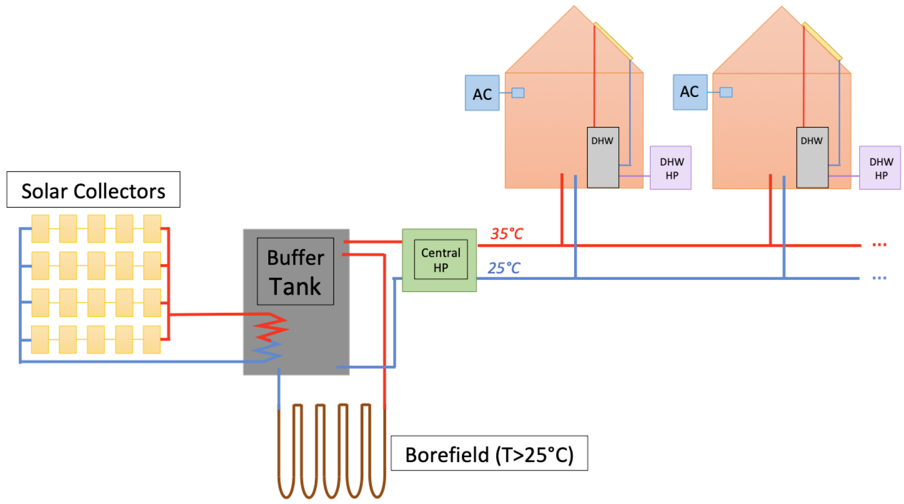


Figure 3.28: Schematic system overview of concepts 13a and 13b (AC: Air Conditioning; DHW HP: Domestic Hot Water Heat Pump).

System Components		Concept 13 a			
Seasonal Thermal Energy Storage (STES)	Technology	Tank	Borefield (<25°C)	**Borefield (>25°C)**	
	Location	Centralised	Decentralised		
	Supplementary STES	Tank	None		
Solar Collectors Space Heating	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Solar Collectors DHW	Location	Centralised	Decentralised		
	Technology	Flat plate collectors	Vacuum tube collectors		
Cooling	Option	Integrated	Separate		
District Heating Network	Working Temperatures	SP: 55°C RT: 25°C	SP: 45°C RT: 25°C	SP: 35°C RT: 25°C	SP: 15°C RT: 10°C
Supplementary Heating System	Space Heating	Large central HP	Small decentral HP	No suppl. heating	
	DHW	Micro-booster HP	DHW HP	No suppl. heating	
Emission System	Technology	Underfloor heating			

Concept Excluding Rule Incompatibility Rule

Figure 3.29: System matrix of concept 13a.

3.4 Conclusion

This concludes Chapter 3 on the design of concepts. Following from the system matrix and the rules of thumb, a total of 13 concepts was found, each with an a and b version. In the a version, flat plate solar collectors are used in the central solar field, whereas in the b version, vacuum tube solar collectors are used in the central solar field. These 13 concepts are divided into three groups, based on the STES technology that is applied. This leads to six concepts with tank storage, four concepts with low-temperature borefield storage and three concepts with high-temperature borefield storage.
With these results, the first part of this thesis is concluded, answering the question '*How does the selection and interconnection of different components lead to different concepts of a district heating system with STES?*'. In the next part, the assessment of the concepts is considered.

Part II

Concept Assessment

Chapter 4

Methodology and Basic System Specifications

This chapter gives an overview of the methodologies that are used to model the different system components. Moreover, basic specifications about these system components are given as well. The methodologies and basic system specifications are applied in Chapters 5 and 6 to size the system components in the different concepts. The first system component that is handled in this chapter is the STES, with the methodology for tank and borefield storage being discussed in Sections 4.1 and 4.2 respectively. This is followed by the methodology for the solar collectors in Section 4.3. Subsequently, Sections 4.4, 4.5 and 4.6 respectively handle the methodologies for cooling, the district heating network and supplementary heating systems. Finally, Section 4.7 discusses the method of calculating the Net Present Value (NPV), which is later used in the evaluation of the concepts. Note that the time horizon used throughout this thesis is 40 years. This period is in line with the recommendations of the BRE (British Building Research Establishment).

4.1 Seasonal Storage Tank

This section describes the methodology that is used for sizing the seasonal storage tank. As mentioned in Section 2.1.1, sensible heat storage can be improved by applying stratification of the storage. However, realistic modelling of a stratified tank is complex due to the changing thermocline in between the hot and cold zone of the tank. This thesis therefore considers two ideal cases for seasonal storage in tanks:

- A fully mixed tank, i.e. the entire tank volume has the same temperature and no stratification is present in the tank.
- A perfectly stratified tank, i.e. the thermocline in between the hot and cold zone of the stratified tank is infinitely small.

These two cases give two extremes between which a realistically stratified tank is situated. Modelling of a fully mixed tank and a perfectly stratified tank is dealt with separately in Sections 4.1.1 and 4.1.2.

4.1.1 Fully Mixed Tank

In a fully mixed tank, the entire water volume inside the tank is at the same temperature and no stratification is present. Water in the tank can reach a maximum temperature of 98°C to avoid boiling of the water. The minimum temperature of water in the tank corresponds to 10°C. However, this minimum temperature can only be reached in a fully mixed tank if a large central heat pump is placed in between the storage tank and the district heating network. Indeed, the heat pump allows the temperature in the tank to drop below the supply temperature of the network. Once this occurs, the heat pump is switched on to be able to deliver the required supply temperature of the network. This is the case for concepts 2, 4 and 6, as explained in Section 3.3.1. On the other hand, in the concepts without a large central heat pump (1, 3 and 5), the temperature in the tank always has to remain above the supply temperature of the network.

The seasonal tank is modelled as a cylinder. The depth and the diameter of the cylinder are optimized for a given tank volume by minimizing the area-to-volume ratio of the tank. This in turn minimizes the heat loss from the tank to the surrounding soil. The temperature of this surrounding soil is assumed constant at 10°C. It is important to note that this assumption is rather pessimistic, since the tank is buried underground. Storing hot water in the tank leads to an increase of the temperature of the ground that surrounds the tank [3]. However, modelling the temperature evolution of the ground surrounding the storage tank is complex and outside the scope of this thesis. Therefore, in all calculations, the ground temperature is kept constant at 10°C.

The temperature in a fully mixed tank T_{tank} is related to the energy content of the tank $Q_{tank,mixed}$ [Wh] through the following equation:

$$Q_{tank,mixed} = \rho_w \cdot V \cdot c_{p,w} \cdot (T_{tank} - T_{ref})/3600 \tag{4.1}$$

Here, ρ_w is the density of water, V is the tank volume in m^3, $c_{p,w}$ is the specific heat capacity of water and T_{ref} is an arbitrary reference temperature. Numerical values for ρ_w and $c_{p,w}$ can be found in Table 4.1. The reference temperature is set at the minimum temperature that can be reached in the tank, i.e. 10°C. In that way, if the tank temperature is at this minimum level, $Q_{tank,mixed} = 0$ Wh. Recall that the minimum temperature in the tank can only be reached if a large central heat pump is used in the concept (as for concepts 2, 4 and 6). Hence, for the concepts without a central heat pump (1, 3 and 5), $Q_{tank,mixed}$ never reaches zero and a certain amount of energy always remains in the tank.

Throughout the year, energy is added to and extracted from the storage tank,

Parameter	Value
Water density ρ_w	1000 $\frac{kg}{m^3}$
Water specific heat capacity $c_{p,w}$	4185 $\frac{J}{kg \cdot K}$
Overall HTC of the top U_{top}	0.1 $\frac{W}{m^2 \cdot K}$ [47]
Overall HTC of the bottom U_{bottom}	0.3 $\frac{W}{m^2 \cdot K}$ [47]
Overall HTC of the walls U_{walls}	0.3 $\frac{W}{m^2 \cdot K}$ [47]

Table 4.1: Properties of water and overall heat transfer coefficients (HTC) of the tank.

changing both its energy content and temperature. The energy balance for the seasonal storage tank is given on an hourly scale by:

$$Q_{tank,mixed}(i+1) = Q_{tank,mixed}(i) + \Delta Q_{tank,mixed} \tag{4.2}$$

with i the hour and $Q_{tank,mixed}(i)$ defined as in Equation 4.1, where $T_{tank}(i)$ is now time dependent as well. Similarly, $Q_{tank,mixed}(i+1)$ is related to $T_{tank}(i+1)$ via Equation 4.1. The change in energy content of the tank $\Delta Q_{tank,mixed}$ [Wh] is calculated with the following equation:

$$\begin{aligned}\Delta Q_{tank,mixed} = Q_{collectors,mixed}(i) + W_{heatpumps,mixed}(i) \\ - Q_{demand}(i) - Q_{loss,transport}(i) - Q_{loss,mixed}(i)\end{aligned} \tag{4.3}$$

with i the hour and the terms in this equation as explained below. Note that these terms are originally expressed in W. Since the demand and solar irradiance profiles are expressed per hour (see Chapter 1), the energy in Wh is simply obtained by multiplying these terms in W with one hour.

- $Q_{collectors,mixed}(i)$ corresponds to the heat that is absorbed by the solar thermal collectors in hour i. It is calculated by multiplying the heat absorbed by a single collector (see Equation 4.18) with the total number of collectors.

- $W_{heatpumps,mixed}(i)$ is the work added by the heat pumps in the system in hour i. This is calculated with Equation 2.4.

- $Q_{demand}(i)$ is the heat demand of 50 dwellings in the district in hour i. This includes the space heating demand and potentially also the domestic hot water demand. The latter is only the case if the district heating network provides heat for domestic hot water to the dwellings, as for concepts 1 to 4 (see Section 3.3.1). If domestic hot water is provided in each dwelling separately, as for concepts 5 and 6, $Q_{demand}(i)$ only includes the space heating demand.

- $Q_{loss,transport}(i)$ is the heat loss that occurs during transport of heat to the dwellings in hour i. As explained in Section 4.5, either single pipes or twin pipes can be used for the district heating network. The heat loss per unit length of the network is calculated with Equation 4.31 for single pipes and with Equation 4.34 for twin pipes. $Q_{loss,transport}(i)$ is then determined by multiplying this heat loss per unit length with the total length of the district heating network.

- $Q_{loss,mixed}(i)$ is the heat loss of the fully mixed tank to the surroundings in hour i. This heat loss is driven by the temperature difference between the water inside the tank and the ground surrounding the tank. It is calculated by the following equation:

$$Q_{loss,mixed}(i) = U \cdot A \cdot (T_{tank}(i) - T_{ground}) \tag{4.4}$$

  with U the overall heat transfer coefficient and A the surface area of the tank. Regarding the overall heat transfer coefficient U, a distinction is made between the top of the tank on the one hand and the bottom and walls of the tank on the other hand. In general, more insulation is applied on the top of the tank compared to the other sides, resulting in a lower heat transfer coefficient [47]. The numerical values for the heat transfer coefficients are shown in Table 4.1.

Equations 4.2 and 4.3 allow to determine the total number of solar collectors that is necessary in the system for a given volume of the storage tank. This number of solar collectors is such that the tank is balanced over a year.
However, in sizing the storage tank, the tank volume is unknown and therefore an iteration is performed over different tank volumes. For each tank volume, making the energy balance in Equation 4.2 for the entire year, results in the total number of solar collectors that is needed in combination with this specific tank volume. Hence, a number of different solution pairs for the tank volume and number of solar collectors is found. Subsequently, the optimal tank volume and corresponding number of collectors has to be selected from these different solution pairs. The optimization method that allows selecting the optimal solution pair, differs depending on the sizing method that is applied. Section 5.2.1 explains the sizing method in the Simplified Dynamic Assessment (SDA), whereas Section 6.2.1 explains the sizing method in the Detailed Dynamic Assessment (DDA).

Lastly, the efficiency of tank storage can be determined. The storage efficiency is defined as the ratio between the energy that is yearly discharged from the storage and the energy that is yearly charged to the storage [2]. For a fully mixed tank, it is calculated with the following equation:

$$\eta_{storage,mixed} = \frac{\sum_{i=1}^{8760} Q_{collectors,mixed}(i) - \sum_{i=1}^{8760} Q_{loss,mixed}(i)}{\sum_{i=1}^{8760} Q_{collectors,mixed}(i)} \tag{4.5}$$

with $Q_{collectors,mixed}(i)$ the energy that is charged to the storage by the solar collectors per hour i and $Q_{loss,mixed}(i)$ the heat that is lost in the tank storage per hour i.

4.1.2 Perfectly Stratified Tank

A perfectly stratified tank considers a stratified tank with an infinitely small thermocline in between the hot and cold zone of the tank. This means that the tank is divided into two perfectly separated volumes, each at a fixed temperature. The hot zone of the tank is at a fixed high temperature T_H of 98°C, which avoids boiling of

water inside the tank. The cold zone of the tank is at a fixed low temperature T_L of 10°C.
For a fully mixed tank, it was seen that the minimum temperature of 10°C can only be reached if a large central heat pump is placed in between the storage tank and the district heating network (concepts 2, 4 and 6). The heat pump starts operating once the temperature in the tank drops below the supply temperature of the network (see Section 4.1.1).
For a perfectly stratified tank on the other hand, heat is always present at a fixed high temperature of 98°C. By definition of a perfectly stratified tank, the temperature of this hot zone never drops below the supply temperature of the network. Hence, no operation of the large central heat pump is required in concepts 2, 4 and 6 for a perfectly stratified tank.

Similarly to a fully mixed tank, a perfectly stratified tank is modelled as a cylinder. The dimensions of the cylinder (diameter and depth) are again optimized by minimizing the area-to-volume ratio and hence the heat loss. The ground surrounding the tank is at a constant temperature of 10°C as well. Regarding this assumption, the same remark as for a fully mixed tank holds: it is a rather pessimistic assumption due to the temperature increase of the ground surrounding the tank (see Section 4.1.1).

A perfectly stratified tank consists of a hot and a cold zone, respectively at fixed temperatures T_H of 98°C and T_L of 10°C. Throughout the year, the volumes of these zones change, indicating that the storage tank is being charged or discharged. The hot volume in the tank V_H increases while charging the tank, corresponding to an increase in the energy content. On the other hand, the hot volume in the tank V_H decreases while discharging the tank, corresponding to a decrease in the energy content. In a fully charged tank, the entire tank volume is at high temperature T_H, whereas in a fully discharged tank, the entire tank volume is at low temperature T_L. The energy content of a perfectly stratified tank $Q_{tank,stratified}$ [Wh] is related to the hot volume in the tank V_H by the following equation [61]:

$$Q_{tank,stratified} = \rho_w \cdot V_H \cdot c_{p,w} \cdot (T_H - T_L)/3600 \tag{4.6}$$

with ρ_w the density of water and $c_{p,w}$ the specific heat capacity of water. The numerical values of these parameters are shown in Table 4.1.
The state of charge (SoC) of the tank is defined as [61]:

$$SoC = \frac{Q_{tank,stratified}}{Q_{max,stratified}} = \frac{V_H}{V} \tag{4.7}$$

The hot volume in the tank V_H changes throughout the year while energy is being added or extracted from the storage tank. The energy balance of a perfectly stratified tank can be described on an hourly scale, similarly to that of a fully mixed tank:

$$Q_{tank,stratified}(i+1) = Q_{tank,stratified}(i) + \Delta Q_{tank,stratified} \tag{4.8}$$

with i the hour and $Q_{tank,stratified}(i)$ defined as in Equation 4.6, where $V_H(i)$ is now time dependent as well. Similarly, $Q_{tank,stratified}(i+1)$ is related to $V_H(i+1)$ via

Equation 4.6. The change in energy content of the tank $\Delta Q_{tank,stratified}$ [Wh] is calculated with the following equation:

$$\begin{aligned}\Delta Q_{tank,stratified} = Q_{collectors,stratified}(i) + W_{heatpumps,stratified}(i)\\ - Q_{demand}(i) - Q_{loss,transport}(i) - Q_{loss,stratified}(i)\end{aligned} \tag{4.9}$$

Again, note that these terms are originally expressed in W. Since the demand and solar irradiance profiles are expressed per hour (see Chapter 1), the energy in Wh is simply obtained by multiplying these terms in W with one hour. Comparing the terms in this equation to those in Equation 4.3 for a fully mixed tank gives the following:

- $Q_{demand}(i)$ and $Q_{loss,transport}(i)$ are identical to the terms in Equation 4.2.
- $Q_{collectors,stratified}(i)$ and $W_{heatpumps,stratified}(i)$ are similar to the 'mixed' terms in Equation 4.2, in the way that they are calculated with the same equations as mentioned in Section 4.1.1. However, the value of these terms will be different. For $Q_{collectors,stratified}(i)$, this is due to a different number of collectors. For $W_{heatpumps,stratified}(i)$, this is due to different operating times of the heat pumps.
- $Q_{loss,stratified}(i)$ differs from the 'mixed' term in Equation 4.2. This term still represents the heat loss of the tank to the surroundings but is calculated using a different formula:

  $$Q_{loss,stratified}(i) = U \cdot A_{hot}(i) \cdot (T_H - T_{ground}) \tag{4.10}$$

  with U the overall heat transfer coefficient and A_{hot} the surface area of the hot volume in the storage tank. As for a fully mixed tank, the overall heat transfer coefficient U is divided into different coefficients for the top of the tank on the one hand and the bottom and walls of the tank on the other hand (see Table 4.1).

In analogy with Section 4.1.1, Equations 4.8 and 4.9 are used to optimize the total number of solar collectors in the system for a given tank volume. The necessary number of solar collectors for a given tank volume is again obtained by guaranteeing the yearly energy balance. Iterating over a set of different tank volumes results in multiple solution pairs for the tank volume and number of solar collectors. The optimal solution is found by the method described in Sections 5.2.1 and 6.2.1.

Furthermore, the efficiency of a perfectly stratified tank storage is determined again by the ratio between the energy that is yearly discharged from the storage and the energy that is yearly charged to the storage:

$$\eta_{storage,stratified} = \frac{\sum_{i=1}^{8760} Q_{collectors,stratified}(i) - \sum_{i=1}^{8760} Q_{loss,stratified}(i)}{\sum_{i=1}^{8760} Q_{collectors,stratified}(i)} \tag{4.11}$$

with $Q_{collectors,stratified}(i)$ the energy that is charged to the storage by the solar collectors per hour and $Q_{loss,stratified}(i)$ the heat that is lost in the tank storage per hour.

This concludes Section 4.1 on the methodology for the seasonal tank storage. Section 4.2 continues with the methodology for the other STES technology, i.e. borefield storage.

4.2 Borefield

In this section, the methodology used for sizing the borefield storage is explained. First, the general methodology to model a borefield is explained in Section 4.2.1. Subsequently, numerical values for different borefield parameters are mentioned in Section 4.2.2. Finally, the more specific sizing methods for a low-temperature borefield and high-temperature borefield are respectively discussed in Sections 4.2.3 and 4.2.4.

4.2.1 General Equations

Modelling the behaviour of the underground is a complex three dimensional heat diffusion problem. To model this behaviour, a method based on g-functions is used in this thesis. A g-function is a function that gives *"the normalised temperature response to a heat load as a function of normalised time"* [53]. In other words, a g-function shows how the ground temperature responds to a certain heat load. These g-functions or ground-response-functions depend on the borefield configuration and in this thesis, they are calculated using the Python package 'pygfunction' [8]. Figure 4.1 shows an example of a g-function for different borefield configurations.

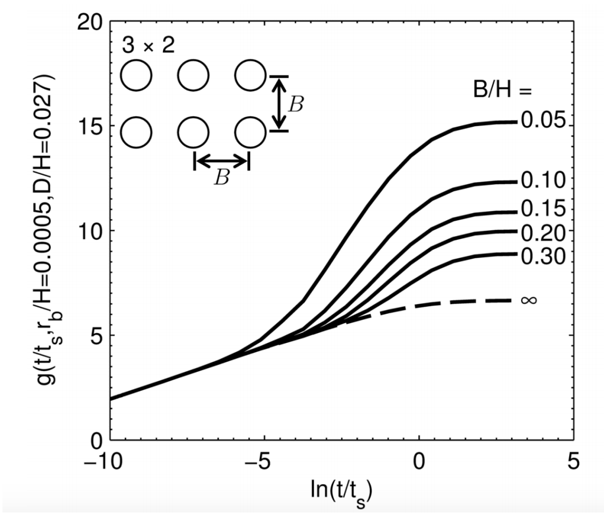


Figure 4.1: Example of a g-function for different borefield configurations [4] (t_s: a normalised time, H: borehole depth, B: borehole spacing, r_b borehole radius).

Once the g-function for a certain borefield configuration is known, the evolution in time of the average fluid temperature in the borefield $T_f(t)$ can be calculated, based on the following equation [52]:

$$T_f(t) = T_b(t) + Q(t) \cdot R_b^* \tag{4.12}$$

with $T_f(t)$ the evolution of the average fluid temperature between in- and outlet of the borefield, $T_b(t)$ the evolution of the borehole wall temperature, R_b^* the equivalent borehole resistance and $Q(t)$ the evolution of the heating or cooling load. Note that the heating load of the borefield includes both the heating demand of the dwellings and the transport losses, i.e. the heat extracted from the borefield. The cooling load on the other hand includes both the cooling demand of the dwellings and the heat captured by the solar collectors, i.e. the heat injected in the borefield.
The value of R_b^* can be both measured and theoretically calculated. However, the theoretical derivation of R_b^* is outside the scope of this thesis. The practical values for R_b^* that are used in this thesis are later introduced in Chapter 5.
Along with R_b^*, the calculation of the average fluid temperature T_f at time t requires the borehole wall temperature T_b. It is at this point that g-functions come into play, since this borehole wall temperature is calculated using g-functions in combination with a so-called temporal superposition. Indeed, a g-function gives the response of the underground for a certain thermal pulse q_i [W/m]. In case multiple thermal pulses are spread in time, a temporal superposition of the responses to these different thermal pulses is necessary. By applying this temporal superposition, the evolution of the borehole wall temperature $T_b(t)$ can be calculated with the following equation [9]:

$$T_b(t) = T_g - \frac{1}{2\pi k_s} \sum_{i=1}^{n} (q_i - q_{i-1}) g(\frac{t_n - t_{i-1}}{t_s}) \tag{4.13}$$

with T_g the undisturbed ground temperature, q_i the thermal load per meter in interval i, t_i the time at instance i and k_s the ground thermal conductivity. After this temporal superposition, the evolution of the average fluid temperature $T_f(t)$ can then be calculated using Equation 4.12.

Three different types of thermal pulses q_i are considered each month to calculate the evolution of the fluid temperature:

- A peak thermal pulse, i.e. the maximum load on the borefield for a duration of 6 hours.
- An average thermal pulse, i.e the average load on the borefield during that month.
- A yearly thermal imbalance pulse i.e. the yearly imbalance between heating and cooling load.

Moreover, these peak and average pulses are calculated for both the cooling and heating load. Hence, a total of four pulses is considered each month (a peak cooling load, a peak heating load, an average cooling load and an average heating load) and one pulse is considered each year (yearly imbalance). Once the evolution of the borehole wall temperature $T_b(t)$ is known, the evolution of the average fluid temperature $T_f(t)$ is determined for each of the four monthly loads by evaluating $Q(t)$ in Equation 4.12 for each load. This leads to four different curves for the evolution of the fluid temperature in the borefield, i.e. $T_f(t)$ for peak heating, peak cooling, average heating and average cooling. Figure 4.2 shows an example of the temperature evolution of a borefield. Note that in the figure, the "base" curves correspond to the average thermal pulses, while the "peak" curves correspond to the peak thermal pulses.

Recall that in all concepts with a borefield a buffer tank is provided to buffer the heat that is injected in and extracted from the borefield. To model the effect of this buffer on the loads on the borefield, the heating and cooling loads from the dwellings and solar collectors are transformed to an average load. In other words, each day the net sum of heating and cooling load from the dwellings and/or collectors is divided by 24 and this average load is applied to the borefield every hour. The buffer is then sized in such a way that it can perform this averaging behaviour on the day with the highest load. The volume of the buffer tank is calculated with the following equation:

$$V_{buffer} = \frac{Q_{max}}{\rho_w \cdot c_{p,w} \cdot \Delta T_{max}} \tag{4.14}$$

with ρ_w the density of water, $c_{p,w}$ the specific heat capacity of water, ΔT_{max} the maximum allowed temperature difference and Q_{max} the maximum amount of heat in the buffer tank. This Q_{max} is easily calculated using an energy balance of the buffer. Here, ΔT is the difference between the highest peak fluid temperature in the borefield and the lowest peak fluid temperature in the borefield in a specific year. ΔT_{max} is the value for the year where this difference is the largest.

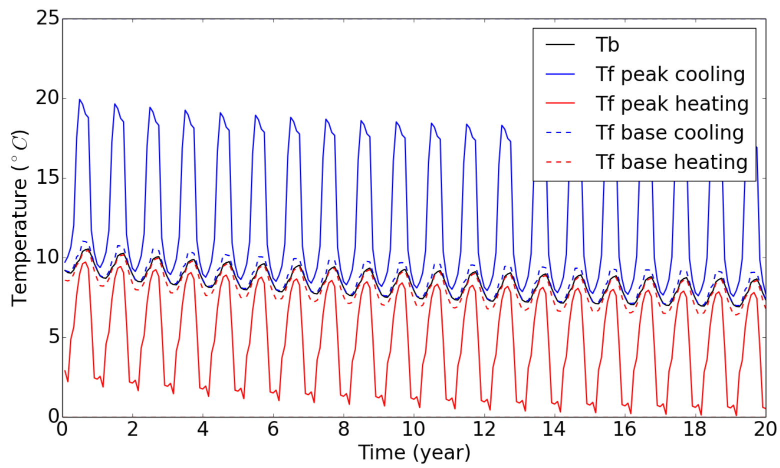


Figure 4.2: Example of a temperature evolution of a borefield.

The method explained in this section forms the basis of the methods used to size the borefield storages, both at low and high temperature. The specific sizing methods for these storages are explained in Sections 4.2.3 and 4.2.4. In the following section, the practical values for the borefield parameters used in this thesis are shown.

4.2.2 Numerical Values

Practical values for borefield, ground and fluid parameters are necessary both for calculating the g-functions and evaluating Equations 4.12 and 4.13. Table 4.2 gives an overview of the practical values used in this thesis for different borefield and ground parameters. Note that not all the necessary borefield parameters are given in this table. This is because some of the parameters change depending on the type of calculation method that is applied, as will be explained in Chapter 5. The parameters mentioned in this table however remain constant regardless of the calculation method.

Ground Parameters	
Ground thermal conductivity	1.8 and 2.4$\frac{W}{m \cdot K}$
Undisturbed ground temperature	10°C
Borefield Parameters	
Configuration	14x14 rectangle
Borehole spacing	3 and 6m

Table 4.2: Practical values of ground, fluid and borefield parameters used in this thesis.

4.2.3 Low-temperature Borefield

For a borefield at low temperature, a novel method developed by Peere et al. is used in this thesis [53]. The basic formula of this sizing method (and of many other methods) is the following [52]:

$$L = \frac{\sum_{i=1}^{N} q_i R_i + q_h R_b^*}{T_f - (T_g + T_p)} \tag{4.15}$$

Here, L is the length of the borefield, T_f is the average fluid temperature, T_g is the undisturbed ground temperature, T_p is a temperature penalty, R_i the thermal resistances, R_b^* the equivalent borehole resistance, q_i the thermal loads and q_h the thermal pulse. The temperature penalty T_p is necessary to take into account the effect of interactions between different boreholes [1]. However, the resistances in Equation 4.15 can be redefined using g-functions such that the temperature penalty can be dropped [1]. This is done in the method of Peere et al. The newly defined resistances now take into account the borehole-to-borehole interactions and are defined as follows [1]:

$$R_{t,g} = \frac{g(t_r) - g(t_r - t)}{2\pi k_s} \tag{4.16}$$

with k_s the ground thermal conductivity and t_r a reference time larger than t. The method of Peere et al. also uses the four thermal pulses that were explained in Section 4.2.1 as inputs.

The main idea of the method of Peere et al. is that the fluid temperature in the borefield should remain between 0°C and 25°C. In Figure 4.2, this means that the peak heating curve is not allowed to drop below 0°C, whereas the peak cooling curve is not allowed to exceed 25°C. Finally, the method iteratively searches for the minimum borefield length L that satisfies this condition.

4.2.4 High-temperature Borefield

For a high-temperature borefield, a somewhat different approach is used. Again, the fluid temperature evolution is calculated based on the four thermal pulses each month using g-functions. However, the temperature limits are not 0°C and 25°C anymore as for a low-temperature borefield, but 0°C and 98 °C, since the temperature is now allowed to exceed 25°C. The limit of 98 °C is necessary to avoid boiling of the fluid through the boreholes. Along with these temperature limits, a second condition is added when sizing a borefield at high temperature, i.e. the peak heating curve and thus also the fluid temperature should remain above 35°C after 20 years of operation. This condition is added to make sure that an imbalance is created and to make sure that space heating can be provided without the use of a heat pump after 20 years of operation.

Based on the four thermal pulses for heating and cooling, the method for a high-temperature borefield determines the temperature curves as shown in Figure 4.2 for a range of borefield depths between 20 and 200 m. Subsequently, it checks whether the two conditions are satisfied. The result of this method is a set of depths that satisfy the two conditions. A corresponding set of borefield lengths is found by multiplying the borehole depths with the total number of boreholes. Finally, the optimal solution for the borefield length is determined from this set as explained in Sections 5.2.2, 5.2.3 and 6.2.2.

This concludes the section on the methodology for borefield storage. Both the methodology for a low- and high-temperature borefield was explained. In the next section, the solar collectors are considered.

4.3 Solar Collectors

This section describes the methodology that is used to model the behaviour of the solar collectors. The general equations of the methodology are explained in Section 4.3.1. Moreover, Section 4.3.2 gives some practical considerations that have to be taken into account when sizing systems with solar collectors. Subsequently, Section 4.3.3 lists the numerical values that are used in this thesis. Finally, Section 4.3.4

discusses the solar collectors that are used in a local production unit for domestic hot water in each dwelling.

4.3.1 General Equations

To model the behaviour of the solar collectors, the most important equation is the one given in Section 2.2.3 and repeated here:

$$\eta_{collector} = \eta_0 - k_1 \frac{T_{f,col} - T_{amb}}{G} - k_2 \frac{(T_{f,col} - T_{amb})^2}{G} \tag{4.17}$$

This equation calculates the efficiency of a solar collector for a particular solar irradiance G [$\frac{W}{m^2}$], an ambient air temperature T_{amb} [K] and an average fluid temperature $T_{f,col}$ [K]. As already explained in Section 2.2.3, the values for η_0, k_1 and k_2 are part of the technical specifications of a certain type of collector. If the collector efficiency is defined for the gross area of the collector, the absorbed heat in one collector can be calculated as follows:

$$Q_{abs,col} = \eta_{collector} \cdot G \cdot A_{gross} \tag{4.18}$$

In this equation, $Q_{abs,col}$ is the thermal power absorbed by the working fluid that flows through the solar collector, expressed in W; $\eta_{collector}$ is the collector efficiency defined for the gross area; G is the solar irradiance in $\frac{W}{m^2}$ and A_{gross} is the gross area of the collector in m^2. Recall that $Q_{abs,col}$ can be easily converted to the energy that is absorbed in Wh by multiplying this term by one hour, as the solar irradiance G is given per hour.
Nevertheless, Equation 4.18 alone does not suffice to calculate the heat absorbed by the working fluid. After all, the value of $\eta_{collector}$ has to be known as well. To calculate this value, Equation 4.17 is used. Combining these two equations yields:

$$Q_{abs,col} = \left(\eta_0 - k_1 \frac{T_{f,col} - T_{amb}}{G} - k_2 \frac{(T_{f,col} - T_{amb})^2}{G} \right) \cdot G \cdot A_{gross} \tag{4.19}$$

Along with values for η_0, k_1 and k_2, this equation also needs values for the ambient air temperature T_{amb} and the solar irradiance G. As explained in Chapter 1, the evolution of the ambient air temperature and the solar irradiance is known for a typical Belgian year. Moreover, the average temperature of the working fluid $T_{f,col}$ ($= \frac{T_{out}+T_{in}}{2}$) has to be known. The value of $T_{f,col}$ depends on both the inlet temperature T_{in} and the outlet temperature T_{out} of the solar collector. For a certain inlet temperature, e.g. 20°C, the outlet temperature can be calculated as follows:

$$T_{out} = T_{in} + \frac{Q_{abs,col}}{\dot{m} \cdot c_p} \tag{4.20}$$

In this equation, $\dot{m}$ is the mass flow rate of the working fluid that flows through the collector, expressed in $\frac{kg}{s}$ and c_p is the isobaric specific heat of the working fluid, expressed in $\frac{J}{kg \cdot K}$. This equation clearly illustrates that the outlet temperature T_{out} and the heat absorbed by the working fluid $Q_{abs,col}$ mutually influence each other.

Therefore, an iterative solution method is applied when solving Equations 4.19 and 4.20. The mass flow rate in Equation 4.20 can be calculated as follows:

$$\dot{m} = \rho \cdot \dot{V} \tag{4.21}$$

ρ is the density of the working fluid in $\frac{kg}{m^3}$ and $\dot{V}$ is the flow rate of the working fluid through the collectors in $\frac{m^3}{s}$

4.3.2 Practical Considerations

To control the efficiency and outlet temperature of the solar collector, the main control variable is the mass flow rate through the solar collector. A small mass flow rate corresponds to a higher outlet temperature and lower collector efficiency, whereas a high mass flow rate corresponds to a lower outlet temperature and a higher collector efficiency (see Equations 4.19 and 4.20). Therefore, keeping the outlet temperature as low as possible, yields the highest possible solar collector efficiency . Nevertheless, there are limits to this:

- First of all, the outlet temperature should be sufficiently high such that the working fluid can still exchange heat with the storage medium. If, for example, the outlet temperature of the working fluid is only 30°C and the storage medium has a temperature of 55°C, the working fluid cannot transfer its heat to the storage medium. In other words, the outlet temperature should always be higher than the temperature of the storage medium.
- A second limit is imposed by the limited range of the mass flow rate. The company Viessmann prescribes a maximum flow rate of 25 liters per hour per gross collector area at a pump rate of 100 percent [66]. The lowest value of flow rate that is allowed, is 25 percent of this maximum value [66].

Another practical consideration is that sometimes, it might be necessary to place multiple collectors in series. This is the case when the storage medium is at a relatively high temperature compared to the inlet temperature. For a certain inlet temperature, a certain ambient air temperature and a certain solar irradiation, there is a limit in the outlet temperature that can be achieved per collector. The reason for this is that it is not allowed to keep lowering the mass flow rate to increase the outlet temperature. A simple method to resolve this is to place multiple solar collectors in series. Each of these collectors then has the outlet temperature of the previous collector as its inlet temperature. Note that for every collector that is added in series, the average fluid temperature in this collector is larger than the average fluid temperature in the previous collectors, but the mass flow rate remains the same. Consequently, each collector that is put in series has a lower efficiency than the previous collectors.

4.3.3 Numerical Values Used for Calculations

To solve Equations 4.19 and 4.20 and thus to determine the absorbed heat per collector, numerical values for T_{amb}, G, η_0, k_1, k_2, $\dot{V}$, c_p and ρ are necessary.

The evolution of the ambient temperature T_{amb} and the solar irradiance G throughout a typical Belgian year is known (see Chapter 1).
For the values of η_0, k_1 and k_2, two practical real-world collector types are used:

- For a flat plate collector, this thesis considers the *VITOSOL 200-F type SV2D* of the company Viessmann [68]. The technical specifications of this flat plate collector are shown in Table 4.3.
- For a vacuum tube collector, this thesis considers the *VITOSOL 300-T type SP3B* of the company Viessmann [67]. The technical specifications of this collector can be found in Table 4.4.

Note that for both of these collectors, the values for η_0, k_1 and k_2 are based on the gross area definition of the collector efficiency.

VITOSOL 200-F type SV2D	
A_{gross} $[m^2]$	2.51
$A_{absorber}$ $[m^2]$	2.32
$A_{aperture}$ $[m^2]$	2.33
η_0 [%]	75.7
k_1	3.28
k_2	0.021

Table 4.3: Technical specification of the *VITOSOL 200-F type SV2D* flat plate collector of the company Viessmann [68]. Note that η_0, k_1 and k_2 are defined for the gross area A_{gross}.

VITOSOL 300-T type SP3B	
A_{gross} $[m^2]$	4.61
$A_{absorber}$ $[m^2]$	3.03
$A_{aperture}$ $[m^2]$	3.19
η_0 [%]	53.3
k_1	0.655
k_2	0.005

Table 4.4: Technical specification of the *VITOSOL 300-T type SP3B* vacuum tube collector of the company Viessmann [67]. Note that η_0, k_1 and k_2 are defined for the gross area A_{gross}.

For the volume flow rate $\dot{V}$, the limits mentioned in Section 4.3.2 are used. Hence, the maximum volume flow rate corresponds to $25\frac{l}{s \cdot m^2}$ and the minimum flow rate corresponds to 25% of the maximum value. For the practical collectors under consideration, these limits can thus be calculated as follows:

- Maximum volume flow rate:

$$\dot{V}_{max} = 25\frac{l}{s \cdot m^2} \cdot A_{absorber} = 0.025\frac{m}{s} \cdot A_{absorber} \tag{4.22}$$

- Minimum volume flow rate:

$$\dot{V}_{min} = 0.25 \cdot 25 \frac{l}{s \cdot m^2} \cdot A_{absorber} = 0.00625 \frac{m}{s} \cdot A_{absorber} \tag{4.23}$$

Finally, the values of c_p and ρ depend on the working fluid that flows through the collectors. As temperatures can drop below 0°C during Belgian winters, water can not be used as working fluid. To avoid freezing of the working fluid in winter, a water-glycol mixture is often used [66]. In this thesis, a mixture of 50% water and 50% glycol is used as the working fluid. The corresponding c_p and ρ values are shown in Table 4.5.

Specific heat capacity c_p	$3600 \frac{J}{kg \cdot K}$
Density ρ	$1070 \frac{kg}{m^3}$

Table 4.5: Density and specific heat capacity of a 50/50 water-glycol mixture [18].

Using all these values for T_{amb}, G, η_0, k_1, k_2, $\dot{V}$, c_p and ρ, the heat absorbed by the working fluid can now be calculated. This is achieved by iteratively solving Equations 4.19 and 4.20. To solve these equations however, still the inlet temperature, the mass flow rate and the type of collector (flat plat or vacuum tube) have to be specified. These parameters are selected in the following chapters.

4.3.4 Solar Collectors in Local Production Unit for DHW

A local production unit can be used in each dwelling to provide domestic hot water, as is the case for concepts 5, 6, 9, 10 and 13 (see Section 3.3). This unit consists of a local storage tank for hot water, a heat pump and roof mounted solar collectors. The solar yield of these collectors is determined using the methodology that was explained above. The gross solar collector area A_{gross} of the collectors in this local unit for hot water is set at 5 m^2, following a general rule of thumb for a solar boiler system [16]. This corresponds to two of the flat plate solar collectors used in this thesis. To make a fair comparison with vacuum tube collectors, the same collector area of 5 m^2 is used for these collectors as well.

In this section, the methodology for the solar collectors was discussed. The general equations, as well as some numerical values that are used in this thesis were mentioned. In the following section, the next system component is discussed, i.e. cooling in the system.

4.4 Cooling

This section briefly explains how the separate cooling system is modelled. Recall that this individual cooling system is used when cooling cannot be provided by the district heating network (see Section 3.1.4). In that case, each dwelling has its own air-to-air heat pump to provide cooling. This air-to-air heat pump is often referred to as an air conditioning system. As mentioned in Section 2.4, the efficiency of a heat pump in cooling mode is given by its COP_R. The value of this COP_R depends on the specific temperature difference between heat source and sink, as well as on the specific heat pump that is used.

The cooling demand is known throughout the year (see Chapter 1). Hence, also the maximum cooling power that this air-to-air heat pump has to provide is known. This is 1.9 kW per dwelling. The maximum required power can be used to select a suitable air-to-air heat pump. After all, the air-to-air heat pump should be powerful enough to cover this peak demand. Based on this criterion, the selected heat pump for this thesis is the *Daikin FTXP + RXP 20K3* [15]. This system consists of an outdoor unit (RXP) and 5 indoor units (FTXP). The specifications of the heat pump are shown in Table 4.6. The table shows that the heat pump has a cooling capacity of 2 kW, making it capable of covering the peak cooling demand in each dwelling.

Daikin FTXP + RXP 20K3	
Cooling capacity	2.00 kW
Power input	0.50 kW
COP_R	4.02

Table 4.6: Technical specifications of the *Daikin FTXP + RXP 20K3* air-to-air heat pump [15].

4.5 District Heating Network

This section describes the methodology that is used for calculating the heat loss in the district heating network. The method is developed by P. Wallentén and determines the steady-state heat loss from insulated pipes [70]. The heat loss from the pipes is driven by the temperature difference between water in the pipes and the surrounding ground. In calculating the steady-state heat loss from the network, it is assumed that the temperature in the pipes is constant over the entire length of these pipes. Two options are considered for the two-pipe district heating network in this thesis, i.e. a configuration with two single pipes and a configuration with twin pipes. These options are discussed in Sections 4.5.1 and 4.5.2 respectively.

4.5.1 Single Pipes

The first possible configuration of a two-pipe district heating network is with single pipes and each of the pipes is individually insulated. One of the pipes acts as the supply line in the network and the other one as the return line. Figure 4.3 illustrates this configuration. The parameters in this figure are explained below.

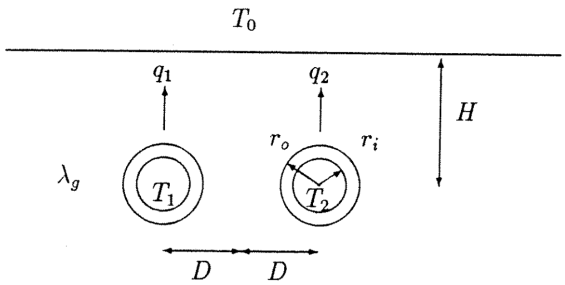


Figure 4.3: Single pipes for the district heating network [70].

H	Depth from the ground surface to the center of the pipes [m]
D	Half the distance between the center of the pipes [m]
r_o	Outer radius of the pipe [m]
r_i	Inner radius of the pipe [m]
λ_g	Thermal conductivity of the ground $[\frac{W}{m \cdot K}]$
λ_i	Thermal conductivity of the insulation $[\frac{W}{m \cdot K}]$
T_0	Temperature on the ground surface [°C]
T_1	Temperature in pipe 1 [°C]
T_2	Temperature in pipe 2 [°C]
q_1	Heat loss from pipe 1 per meter $[\frac{W}{m}]$
q_2	Heat loss from pipe 2 per meter $[\frac{W}{m}]$

The problem is to determine the total steady-state heat loss per unit length from these pipes, i.e. the sum of q_1 and q_2. To solve this problem, it can be separated into a symmetrical and anti-symmetrical problem, as shown in Figure 4.4.

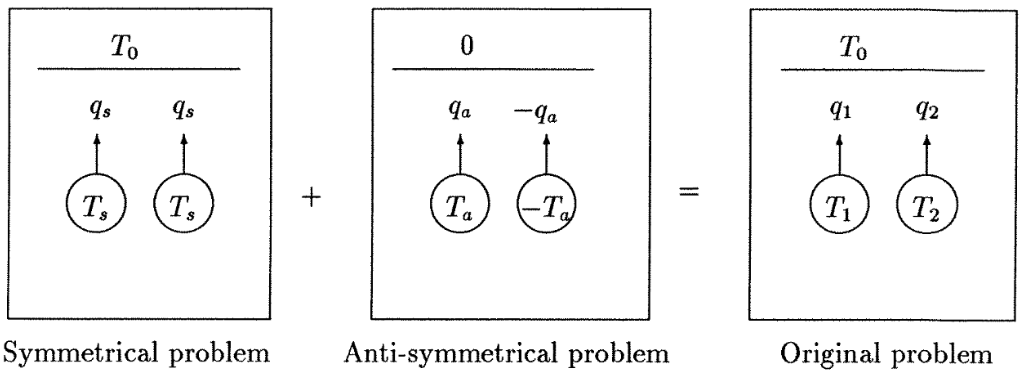


Figure 4.4: Superposition of the symmetrical and anti-symmetrical problem [70].

The temperature T_s in the symmetrical problem and the temperature T_a in the anti-symmetrical problem are defined as follows [70]:

$$T_s = \frac{T_1 + T_2}{2} \tag{4.24}$$

$$T_a = \frac{T_1 - T_2}{2} \tag{4.25}$$

Superposition of the symmetrical and anti-symmetrical problem leads again to the original problem. Therefore, heat losses q_1 and q_2 become [70]:

$$q_1 = q_s + q_a \tag{4.26}$$

$$q_2 = q_s - q_a \tag{4.27}$$

The total heat loss per unit length q depends only on the symmetrical part [70]:

$$q = q_1 + q_2 = 2 \cdot q_s \tag{4.28}$$

The heat loss q_s in the symmetrical problem is calculated with the following formula [70]:

$$q_s = 2 \cdot \pi \cdot \lambda_g \cdot (T_s - T_0) \cdot h_s \tag{4.29}$$

with h_s the dimensionless heat loss factor for the symmetrical problem. This heat loss factor is calculated with the so-called zero-order multipole formula [70]:

$$h_s^{-1} = \ln(\frac{2H}{r_o}) + \frac{\lambda_g}{\lambda_i} \cdot \ln(\frac{r_o}{r_i}) + \ln(\sqrt{1 + (\frac{H}{D})^2}) \tag{4.30}$$

In conclusion, the heat loss per unit length q $[\frac{W}{m}]$ from two single pipes is calculated with the following formula:

$$q = 4 \cdot \pi \cdot \lambda_g \cdot (T_{avg} - T_{ground}) \cdot h_s \tag{4.31}$$

with λ_g the thermal conductivity of the ground, T_{avg} the average of the supply and return temperature, T_{ground} the constant ground temperature and h_s as in Equation 4.30.

4.5.2 Twin Pipes

The second possible configuration of a two-pipe district heating network is with twin pipes, i.e. two pipes that are embedded in a circular insulation. Similarly to Section 4.5.1, one pipe acts as the supply line and the other one as the return line. The configuration with twin pipes is illustrated in Figure 4.5. The parameters in this figure are identical to the parameters for the single pipes, except that r_c corresponds to the radius of the circular insulation.

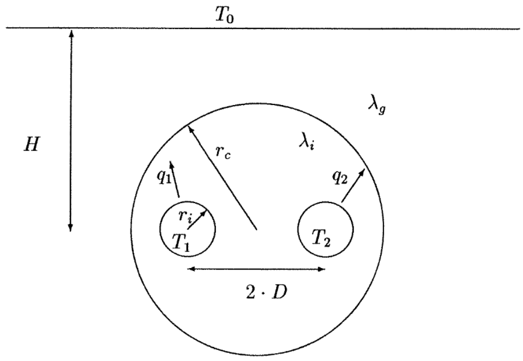


Figure 4.5: Twin pipes for the district heating network [70].

In analogy with Section 4.5.1, the problem of determining the steady-state heat loss per unit length from these pipes is separated into a symmetrical and anti-symmetrical problem. The same equations as in Section 4.5.1 hold and the total heat loss per unit length q only depends on the symmetrical part ($q = 2 \cdot q_s$). However, the heat loss q_s in the symmetrical problem is now calculated as follows [70]:

$$q_s = 2 \cdot \pi \cdot \lambda_i \cdot (T_s - T_0) \cdot h_s \tag{4.32}$$

with h_s the dimensionless heat loss factor for the symmetrical problem. This heat loss factor is now calculated with the following zero-order multipole formula [70]:

$$h_s^{-1} = \frac{2\lambda_i}{\lambda_g} \cdot \ln(\frac{2H}{r_c}) + \ln(\frac{r_c^2}{2Dr_i}) + \frac{\lambda_i - \lambda_g}{\lambda_i + \lambda_g} \cdot \ln(\frac{r_c^4}{r_c^4 - D^4}) \tag{4.33}$$

In conclusion, the heat loss per unit length q [$\frac{W}{m}$] from twin pipes is calculated with the following formula:

$$q = 4 \cdot \pi \cdot \lambda_i \cdot (T_{avg} - T_{ground}) \cdot h_s \tag{4.34}$$

with λ_i the thermal conductivity of the insulation, T_{avg} the average of the supply and return temperature and h_s as in Equation 4.33.

This concludes the methodology for the district heating network, in which both single pipes and twin pipes were discussed. The next section handles the methodology for the supplementary heating systems.

4.6 Supplementary Heating Systems

In chapter 3, it was explained that different types of heat pumps are used depending on the specific concept. More precisely, four different heat pumps can be found in the system matrix, i.e. a large central heat pump, a small decentral heat pump, a micro booster heat pump (MB HP) and a domestic hot water heat pump (DHW HP) (see Section 3.1.6). This section gives an overview of the specific real world heat pumps that are used in this thesis.

4.6.1 Large Central Heat Pump

The large central heat pump is a water-to-water heat pump that is located centrally in the district in between the seasonal storage and the district heating network. This heat pump provides supplementary heating for space heating and two practical heat pumps are considered:

- The first practical heat pump is the *VITOCAL 300-G PRO, type BW 302.D140* of the company Viessmann [64].
- The second practical heat pump is the *VITOCAL 350-HT- PRO, type BW 352.AHT.096* of the company Viessmann [65].

The first one is used if the district heating network has a supply temperature of 35°C, whereas the second one is used if the heating network has a supply temperature of 55°C. This distinction in use is necessary because the temperature of the heat source for the first heat pump is limited to 20°C [64]. If, for example, the central storage tank has a temperature of 40°C, then this temperature should be lowered to 20 °C to be able to use the first heat pump. The COP of the first heat pump would be 4.4 in that case. On the other hand, in the second heat pump, higher heat source temperatures are allowed [65]. For the same example, this means that if the tank is at 40°C, the temperature does not have to be lowered. The second heat pump can immediately upgrade the temperature to 55°C with a COP of 6. This example clearly illustrates that the second heat pump is more efficient in combination with a district heating network at 55°C.

However, if the supply temperature of the network is only 35°, it is more efficient to use the first heat pump. This heat pump is specifically designed to be efficient at a low temperature range. It has a COP of 7.26 if the temperature of the source is 20°C and the temperature of the sink is 35°C [64]. The COP of the second heat pump in the same temperature range is only 6.1 [65]. If, for example, the storage tank would have a temperature of 30°C, it is more beneficial (COP of 7.26) to lower the temperature to 20°C and use the first heat pump instead of directly using the second heat pump (COP of 5.5) and lowering the temperature after the heat pump.

The selection procedure for the specific type of the above two heat pumps is similar to the selection procedure of the air-to-air heat pump for cooling in Section 4.4. Since the heating demand is known (see Chapter 1) and transport losses can be calculated (see Section 4.5), the maximum heating load on the large central heat pump in each concept is known. The required power of the central heat pump can then be determined under the condition that it has to be able to provide this peak heating load. This procedure was applied to select the specific types of the heat pumps that were mentioned above.

4.6.2 Small Decentral Heat Pump

The small decentral heat pump is a water-to-water heat pump that is used in concepts 8 and 10 to provide space heating and cooling. The procedure to select this heat pump is similar as for the large central heat pump, i.e. a selection based on the peak demand the heat pump has to cover. By applying this procedure, the *VITOCAL 200-G, type BWC 201.B06* of the company Viessmann is chosen for the small decentral heat pump [63].

4.6.3 Domestic Hot Water Heat Pump

As mentioned in Section 3.1.6, the domestic hot water heat pump (DHW HP) is an air-to-water heat pump that is part of a local hot water production unit. The heat pump and storage tank used in this thesis are respectively the *DAIKIN Altherma ERWQ-02-AV3* and the *DAIKIN Altherma EKHWP-500PB* [14]. This is a standard system that the company Daikin provides for this application. The heat pump has a COP of 4.3. The storage tank has a volume of 477 liters.

4.6.4 Micro Booster Heat Pump

The micro booster heat pump (MB HP) is used in a number of concepts (3,4 and 12) to upgrade the temperature of the heating network from 35°C to 55°C to provide domestic hot water. Similarly to the DHW HP, the MB HP also has a small storage tank, although it is considerably smaller compared to the one that is used in combination with a DHW HP, i.e. the tank has a volume of 190 liters. The practical heat pump used in this thesis is the *NIBE Booster heat pump MT-MB21* [46]. This heat pump has a COP of 5.3 if the temperature of its heat source is 25°C and a COP of 6 if its heat source has a temperature of 40°C. In all concepts with a micro booster heat pump, the district heating network has a supply temperature of 35°C. Hence, to find the corresponding COP value for the MB HP, a linear interpolation is made between the COPs at both source temperatures. This results in a COP of 5.8 for a heat source temperature of 35°C, which is used in this thesis.

In this section, the methodology for the different heat pumps was explained. This concludes all the methodologies for the system components. In the next and final section, the calculation of a net present value is explained, as this will be used in the following chapters of this thesis.

4.7 Net Present Value

This section briefly shows how the Net Present Value (NPV) of the cost of the different concepts is calculated. This NPV of the concepts will be used to compare the concepts to each other. Note that a lower NPV of the cost is always preferred. The following costs are taken into account: investment costs, maintenance costs and energy costs. The following formula calculates the NPV:

$$NPV = \sum_{t=1}^{n=40} \frac{C_t}{(1+r)^t} \tag{4.35}$$

where C_t is the total cost of the system in year t and r is the interest rate. The total cost in year t, C_t, is determined by the sum of the costs of each of the technologies in year t. Note that the NPV is calculated for a study period of 40 years, in accordance with the recommendation of the BRE (British Building Research Estabilishment) that prescribes this period for research in this field of study. The interest rate r used in this thesis is calculated as follows:

$$r = \frac{R - R_i}{1 + R_i} \tag{4.36}$$

with R the market interest rate and R_i the inflation. The market interest rate R is set at 5.5%. Each year the effect of inflation on the different costs is taken into account. The inflation rate R_i is assumed at 2%. Both the inflation and the market interest rate are determined based on statistical data of the past 10 years in Belgium [5]. If these values for inflation rate and market interest rate are applied in Equation 4.36, the corresponding value for the interest rate r is 3.43%. The specific costs for each technology are discussed in the following sections.

Cost of Solar Collectors

The cost of solar collectors consists of the investment cost and the maintenance costs. For the initial investment cost, values of 600 $\frac{€}{m^2}$ and 900 $\frac{€}{m^2}$ are assumed for respectively flat plate collectors and vacuum tube collectors [5]. Note that the gross area of the collectors (see Section 2.2) is used to calculate the cost per collector. These values include both the purchase and installation of the solar collectors.
For the maintenance costs of solar collectors, a fixed percentage of the investment cost is assumed per year. This percentage is set at 1% of the investment cost [5].
Furthermore, a reinvestment cost of the solar collectors is considered as well, since solar collectors have an estimated lifetime of 20 years. The reinvestment cost in year 20 is simply the original investment cost in year 1 with the effect of inflation taken into account. This means the original investment cost is multiplied by $(1 + 0.02)^{20}$.

Cost of Tank Storage

The cost of tank storage also consists of the investment and maintenance costs. No reinvestment of tank storage has to be considered however, since the lifetime of tank storage is set at 40 years, which is equal to the study period. The annual maintenance cost corresponds to a fixed percentage of 0.5% of the initial investment cost. The initial investment cost is determined by multiplying the tank's specific investment cost with the total volume of the tank.
This specific investment cost of tank storage $[\frac{€}{m^3}]$ is a function of the volume. More precisely, it is a decreasing function of the volume, meaning that a smaller tank

has a higher specific investment cost compared to a larger tank. To determine the specific investment cost in function of the tank volume, a cost curve is drawn based on existing systems. These systems are divided into smaller systems that have been applied to date and larger systems from *Ecovat*. The company *Ecovat* designs large seasonal storage tanks with an innovative design, but none of these tanks has been applied in practice yet [17]. Figure 4.6 shows the specific investment cost of existing systems with tank storage. Note that the x-axis in this figure uses a logarithmic scale.

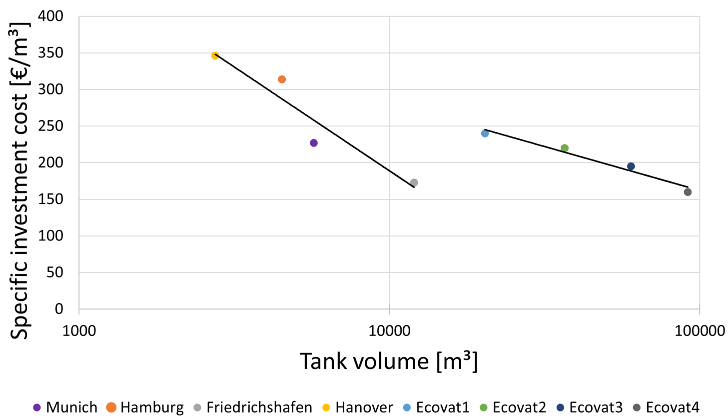


Figure 4.6: Specific investment cost of tank storage [30, 17].

Figure 4.6 shows two different trends in the specific cost of tank storage:

- For smaller systems, the specific investment cost c as function of the tank volume V is described by:

$$c = -123.2 \cdot \ln V + 1323.7 \tag{4.37}$$

- For larger systems, the specific investment cost c as function of the tank volume V is described by:

$$c = -52.17 \cdot \ln V + 762.6 \tag{4.38}$$

The transition between both cost curves is set at 20 300 m^3, corresponding to the smallest system from *Ecovat*. For smaller tanks, the existing technology of systems that have been applied to date is used and Equation 4.37 describes the specific cost of these tanks. For larger tanks, the technology of *Ecovat* systems is used and Equation 4.38 describes the specific cost of these tanks.

Cost of Borefield Storage

Similarly to solar collectors and tank storage, the cost of borefield storage includes the investment cost and maintenance costs. The value for the investment cost is expressed per unit length of the borefield ($\frac{€}{m}$) and depends on the assessment method

(See Sections 5.1.5 and 6.1.5). For the annual maintenance cost, again a percentage of the investment cost is assumed. This value is set at 0.5% [5]. No reinvestment of the borefield storage is considered, since the lifetime of the borefield storage is set at 40 years, i.e. the same as the study period.

Cost of the District Heating Network

The lifetime of the district heating network is also assumed to be 40 years and hence the cost of the district heating network only consists of the initial investment cost and maintenance costs. For the maintenance cost a percentage of 0.5% is assumed [5]. The initial investment cost depends on the supply temperature of the district heating network. For the district heating networks with a supply temperature of 55°C or 35°C, a value of $400\frac{€}{m}$ is assumed, while for a district heating network with a supply temperature of 15°C a value of $250\frac{€}{m}$ is assumed [5]. The difference in investment cost results from a more expensive insulation level that is necessary for higher temperatures. Note that this unit cost of the network is per meter supply and return line.

Cost of Heat Pumps

Depending on the concept, different types of heat pumps are used. Independently of the type of heat pump, the cost always consists of three components: the investment cost, the maintenance costs and the reinvestment cost. The initial investment cost of the different heat pumps can be found in Table 4.7. Recall that both the micro booster heat pump (MB HP) and the domestic hot water heat pump (DHW HP) consist of a heat pump coupled with a small storage tank. For a MB HP, its small storage tank is completely integrated with the heat pump and hence the initial investment cost of this MB HP is shown as one value in Table 4.7. However, for the DHW HP, this is not the case and the small storage tank has to be purchased separately. Therefore, the initial investment cost in Table 4.7 is split into two parts, i.e. the cost of the heat pump and the cost of the storage tank. Finally, for the air-to-air heat pump used in the individual air conditioning systems, the initial investment cost can be split into two parts i.e. the cost of the outdoor unit and the cost of the five indoor units (see Section 4.4).

For the annual maintenance cost, a percentage of the initial investment cost is again used. This percentage is set at 2% for all heat pumps, except for the air-to-air heat pump used for individual cooling, for which a percentage of 4% is applied [5]. These percentages are listed in Table 4.7 as well.

For the reinvestment costs, the lifetime of each heat pump has to be known. This lifetime can be either 20 years or 30 years, as listed in Table 4.7. The reinvestment cost is again calculated from the initial investment cost by applying inflation on this initial cost.

Energy Costs

Along with solar thermal energy, all concepts require electricity as a supplementary energy supply. Electricity is used for the supplementary heating and cooling systems in the different concepts. Recall that the electricity use of all auxiliary equipment such as pumps and control devices is neglected in this thesis (see Chapter 1). For the electricity cost, two types of electricity prices are considered, i.e. the household electricity price and the non-household electricity price. Based on these two prices, two different pricing scenarios are defined:

- In the first scenario, the electricity use of the heat pumps that are located in each dwelling are priced with the household electricity price. On the other hand, the electricity use of the large central heat pump is priced with the non-household electricity price. This scenario is named the '**traditional pricing scenario**'.
- In the second scenario, all electricity use is priced with the non-household electricity price. This scenario corresponds to possible energy communities in which people collaborate to buy their electricity collectively at non-household price. This scenario is named the '**energy community scenario**'.

The household electricity is set at a value of 0.2316€/kWh and the non-household electricity price is set at 0.1555€/kWh. These values correspond to the actual electricity prices in Belgium in 2020 [22, 23]. Note that in these values VAT and other recoverable taxes are excluded. Moreover, for both of the electricity prices an annual increase of 5.87% is assumed. This increase of energy prices is based on the Belgian Royal decree (24/07/2008) that prescribes a pessimistic increase for this value in feasibility studies.
Table 4.7 gives an overview of all the relevant cost data explained above.

This concludes Chapter 4 on the methodologies for the different system components and the methodology for calculating the NPV. These methodologies will be applied in Chapter 5 and 6 to size and assess the different district heating system concepts.

General Cost Data		
Inflation rate R_i	2%	
Market interest rate R	5.5%	
Interest rate r	3.43%	
Solar Collectors		
Initial investment cost	Flat plate Vacuum tube	600 €/m^2 900 €/m^2
Reinvestment	20 years	
Maintenance cost	1%	
Tank Storage		
Initial investment cost	see Section 4.7	
Reinvestment	/	
Maintenance cost	0.5%	
Borefield Storage		
Initial investment cost	see Section 5.1.5 or 6.1.5	
Reinvestment	/	
Maintenance cost	0.5%	
District Heating Network		
Initial investment cost	Supply T: 55°C/ 35°C Supply T: 15°C	400 €/m 250 €/m
Reinvestment	/	
Maintenance cost	0.5%	
Heat Pumps		
Initial investment cost	Large central HP Small decentral HP DHW HP MB HP air-to-air HP (AC)	50 000€ or 92 400€ 8661€ 2485€ (tank) 1026€ (HP) 2630€ 601€ (outdoor unit) 1655€ (indoor units)
Reinvestment	Large central HP Small decentral HP DHW HP MB HP air-to-air HP (AC)	30 years 30 years 20 years 20 years 20 years
Maintenance cost	Large central HP Small decentral HP DHW HP MB HP air-to-air HP (AC)	2% 2% 2% 2% 4%

Table 4.7: General overview of relevant cost data used to calculate the Net Present Value of the different concepts [12, 14, 29, 46]. (DHW HP: Domestic hot water heat pump, MB HP: Micro booster heat pump, AC: Air conditioning)

Chapter 5

Simplified Dynamic Assessment (SDA)

In this chapter, a first method to assess the different concepts of the district heating system is discussed. The method is named the Simplified Dynamic Assessment or SDA. It considers hourly variations of the energy flows in the system, but some simplifications are made in certain parts of the system to simplify the calculations and simulations. These simplifications correspond to general engineering simplifications that are made in a first analysis. In Chapter 6, a second method is discussed, i.e. the Detailed Dynamic Assessment (DDA). There, the simplifications introduced in the SDA are refined. The goal of applying both of these methods is to find out if both methods would come to the same conclusions. In this way, it can become clear whether a simplified method suffices to assess the kind of heating systems under consideration in this thesis.

This chapter begins with an overview of the system simplifications and parameters that are used in the SDA. This is explained in Section 5.1. Subsequently, the sizing methods in the SDA for the different concepts are discussed in Section 5.2. Finally, the results for the different concepts are shown in Section 5.3. These results include the dimensions of the system components and the electricity use in each concept. Moreover, the district heating system concepts are compared based on their Net Present Value (NPV) and CO_2 emissions.

5.1 System Simplifications and Parameters

This section gives an overview of the different system simplifications that are applied in the SDA. Furthermore, some specifications of the system components, that were not yet defined in Chapter 4 and that are dependent on the specific assessment method (i.e. SDA or DDA), are given as well. An example of such a specification is the equivalent borehole resistance (see Equation 4.12). The value of this parameter differs between the SDA and DDA. The system simplifications and parameters are explained for each system component in separate sections.

5.1.1 Solar Collectors

For the solar collectors, a major simplification is made in the SDA. Recall that for the calculation of the heat that is absorbed by the working fluid in a collector, the inlet temperature of the working fluid, the mass flow rate through the solar collector and the type of collector still had to be specified (see Section 4.3.3). In the SDA, both the mass flow rate and the inlet temperature of the working fluid have a fixed value. In other words, a fixed efficiency profile is applied for the solar collectors. Moreover, it is assumed that the outlet temperature of the working fluid is always sufficiently high such that it can transfer its heat to the storage medium.

The mass flow rate is set at its maximum value as explained in Section 4.3.3. The inlet temperature on the other hand is set at 25 °C. Note that these values correspond to optimistic values of the efficiencies. After all, this high mass flow rate corresponds to the lowest possible outlet temperature (for a certain solar irradiance) and thus the lowest possible temperature difference between the average working fluid temperature and ambient air temperature. In reality, probably multiple collectors have to be put in series to achieve sufficiently high outlet temperatures, which significantly lowers the efficiency. However, this effect is neglected in the SDA. The resulting efficiency evolution for a flat plate solar collector and vacuum tube solar collector are respectively shown in Figures 5.1 and 5.2. In these figures, it is clearly illustrated that under the above assumptions, a flat plate collector is more efficient than a vacuum tube collector. Only during the winter months, the vacuum tube collectors are more efficient.

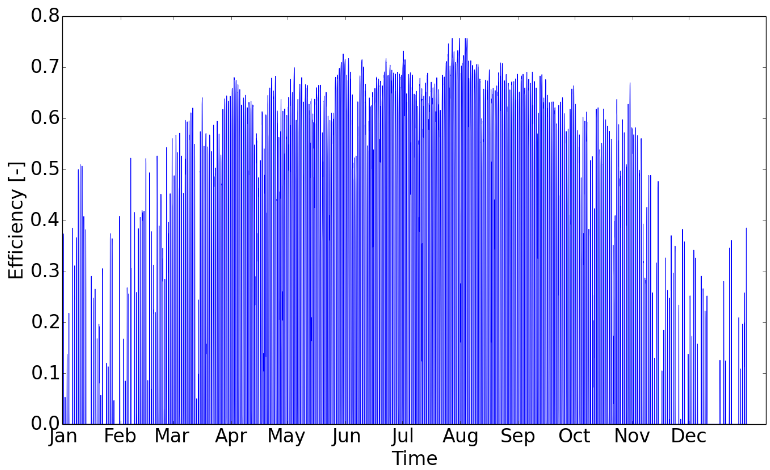


Figure 5.1: Efficiency evolution of a flat plate collector in the SDA.

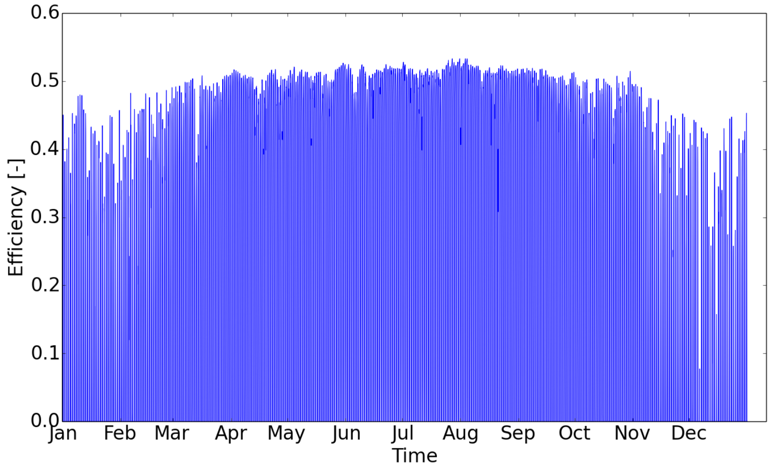


Figure 5.2: Efficiency evolution of a vacuum tube collector in the SDA.

Coupling this efficiency with the solar irradiance evolution (see Chapter 1) results in an amount of thermal energy that is absorbed by a solar collector per square meter per year. For a flat plate collector this is 553 $\frac{kWh}{m^2}$ per year, while for a vacuum tube collector this value is 480 $\frac{kWh}{m^2}$ per year. Note that these values are per unit gross area. Based on the higher efficiency and higher energy yield of the flat plate collector, the SDA only uses flat plate collectors in the different concepts. No vacuum tube collectors are considered here. Hence, only versions a of the district heating system concepts are considered in the SDA (see Section 3.3).

Figure 5.3 shows the evolution of the thermal energy absorbed in a flat plate collector throughout the year. Note that this evolution is calculated with the practical values of the *VITOSOL 200-F type SV2D* flat plate collector. The evolution in Figure 5.3 is the base for further calculations and simulations.

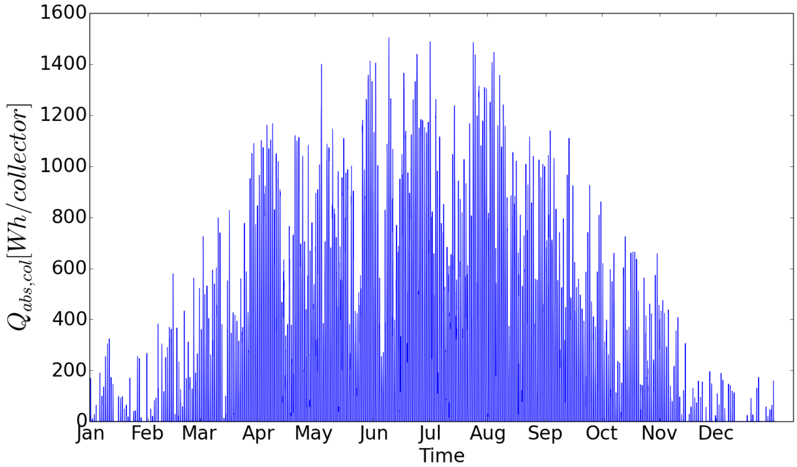


Figure 5.3: Evolution of the thermal energy absorption in the flat plate collector.

5.1.2 Supplementary Heating Systems

This section continues with the simplifications that are applied for the supplementary heating systems. As explained in Section 4.6, the small decentral heat pump, the micro booster (MB) heat pump and the DHW heat pump are modelled with a fixed COP and/or COP_R. For the MB heat pump and the small decentral heat pump this is a valid assumption, since their working temperatures do not change throughout the year. This is not the case for the DHW heat pump, since the outdoor temperature and thus its heat source temperature varies throughout the year. However, also for the DHW heat pump a fixed COP of 4.3 is considered. This COP corresponds to an outdoor air temperature of 7°C [14]. As this is only a few degrees less than the yearly average Belgian outdoor temperature, this value is assumed to be a good estimate for the real average COP. Furthermore, since the demand for domestic hot water does not change throughout the year, a yearly average COP is a good measure to calculate the yearly electricity use.

On the contrary, the large central heat pump (see Section 4.6) is subject to seasonal demand and temperature variations. Due to these variations, a fixed COP value of the heat pump at its yearly average source temperature is not a very realistic measure to calculate the electricity use. Nevertheless, in the SDA this assumption is made and fixed COP and COP_R values are used. The COP is assumed to be 5, while the COP_R is assumed to be 4. These values are chosen as a ball-park average of the COP evolution of the specific heat pumps under consideration in this thesis [64, 65]. The same values are assumed for the COP and COP_R of the small decentral heat pump.

In conclusion, in the SDA all heat pumps are modelled with a fixed COP and/or COP_R. Table 5.1 gives an overview of these different COP and COP_R values for the heat pumps.

Large Central Heat Pump	COP	5
	COP_R	4
Small Decentral Heat Pump	COP	5
	COP_R	4
Micro booster Heat Pump	COP	5.8
	COP_R	/
Domestic Hot Water Heat Pump	COP	4.3
	COP_R	/

Table 5.1: COP- and COP_R-values for different heat pumps in the SDA.

5.1.3 District Heating Network

In this section, the system simplifications for the district heating network are considered. In the SDA, the heat loss in the district heating network is calculated based on the equations from Section 4.5. As mentioned in that section, two possible configurations for the two-pipe district heating network are considered, i.e. a configuration with single pipes and a configuration with twin pipes.

For a district heating network with single pipes, Equation 4.31 determines the heat loss per unit length of the network. This equation can be transformed into the following equation, describing the heat loss per unit length and temperature of the network $U_{loss,transport}$ $[\frac{W}{mK}]$:

$$U_{loss,transport} = 4 \cdot \pi \cdot \lambda_g \cdot h_s \tag{5.1}$$

with λ_g the thermal conductivity of the ground in $\frac{W}{mK}$ and h_s according to Equation 4.30 (with parameters as described in Section 4.5.1).

Similarly, for a district heating network with twin pipes, the heat loss per unit length and temperature of the network $U_{loss,transport}$ $[\frac{W}{mK}]$ is obtained by transforming Equation 4.34:

$$U_{loss,transport} = 4 \cdot \pi \cdot \lambda_i \cdot h_s \tag{5.2}$$

with λ_i the thermal conductivity of the insulation in $\frac{W}{mK}$ and h_s according to Equation 4.33 (with parameters as described in Section 4.5.2).

Hence, the unit heat loss $U_{loss,transport}$ is obtained from Equation 5.1 for single pipes and from Equation 5.2 for twin pipes. This unit heat loss is calculated for a range of pipe diameters. The data for the dimensions of the district heating pipes is obtained from the company *Logstor* [38]. The SDA method applied in this chapter uses data of the 'Bonded pipe system' with steel pipes for both the single pipes and twin pipes. Furthermore, three different insulation series (1, 2 and 3) are considered, with an increasing degree of insulation. Values for the parameters in Equations 5.1

and 5.2 are listed in Table 5.2.
Figure 5.4 shows the heat loss per unit length and temperature $U_{loss,transport}$ as a function of the pipe diameter. This figure clearly illustrates that twin pipes have a lower unit heat loss than single pipes, making them more interesting for the district heating network. Moreover, increasing the insulation level can further decrease the heat loss. Therefore, twin pipes with insulation series 3 are chosen for the network. The pipe diameter of the district heating network is estimated at 50 mm. For this pipe diameter, the corresponding unit heat loss for twin pipes (series 3) is 0.1578 $\frac{W}{mK}$. This value is used in the SDA method for calculating the total heat loss in the district heating network.

Parameter	Value
Thermal conductivity of the ground λ_g	2 $\frac{W}{mK}$
Thermal conductivity of the insulation λ_i	0.025 $\frac{W}{mK}$
Depth of the pipes H	1 m
Ground temperature T_{ground}	10°C
Length of the district heating network L_{net}	1600 m

Table 5.2: Parameters in the district heating network heat loss calculation.

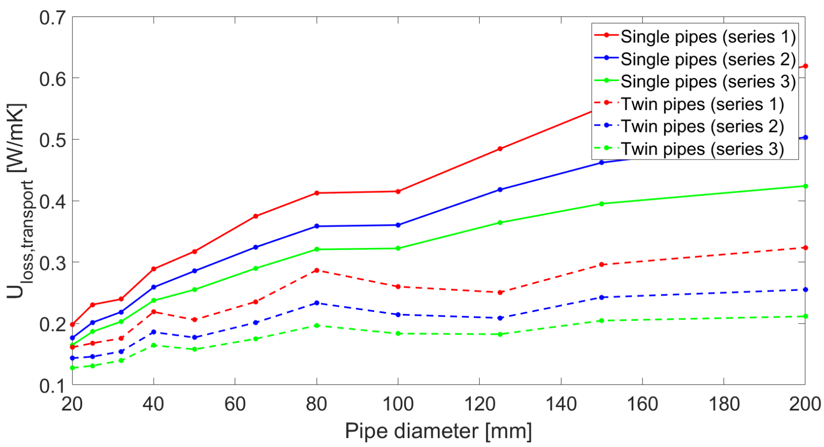


Figure 5.4: Unit heat loss in the district heating network in the SDA.

The total heat loss in the district heating network $Q_{loss,transport}$ [W] is given by:

$$Q_{loss,transport} = U_{loss,transport} \cdot L_{net} \cdot (T_{avg} - T_{ground}) \tag{5.3}$$

with $U_{loss,transport} = 0.1578\frac{W}{mK}$, L the total length of the district heating network, T_{ground} the constant ground temperature and T_{avg} the average of the supply and return temperature in the network. Hence, for the network at 55/25°C, T_{avg} corresponds to 40°C, whereas for the network at 35/25°C, T_{avg} corresponds to 30°C. Note that for the network with an average supply temperature of 15°C, it is assumed

that heat loss during transport is negligible.
Values for the ground temperature T_{ground} and the length of the network L_{net} are mentioned in Table 5.2. The assumed length of the network of 1600 m is based on the length of the existing district heating network in 'Drake Landing Solar Community' in Canada [43]. This district heating network connects 52 residential dwellings, which is similar to the 50 dwellings that are considered in this thesis.

5.1.4 Seasonal Storage Tank

For the seasonal storage tank, no simplifications are made and no further parameters have to be specified. The methodology and specifications as explained in Section 4.1 are used to model the behaviour of the seasonal tank storage. Nevertheless, a short calculation is performed regarding the overall heat transfer coefficient U, for which values are used that are mentioned by Ochs et al. [47]. This simplified calculation is provided in Appendix B and confirms that the values for U correspond to realistic insulation levels of the tank.

5.1.5 Borefield Storage

This section considers the borefield storage in the SDA. Along with the fluid and ground parameters that were set in Section 4.2.2, a number of borehole parameters still have to be specified for the borefield storage. For the borehole parameters, Section 4.2.2 already mentioned the configuration and borehole spacing of the borefield. Other borehole parameters that are of importance for sizing the borefield differ between the SDA and DDA. Table 5.3 shows all the relevant ground, fluid and borehole parameters that are used in the SDA. Note that for both the ground thermal conductivity and the borehole spacing two values are shown in this table. For the ground thermal conductivity, the values of 1.8 and 2.4 $\frac{W}{mK}$ correspond to the minimum and maximum values for the Flemish region in Belgium respectively. For the borehole spacing, the value of 3m is used for the borefield at high temperature, while a spacing of 6m is used for the borefield at low temperature. This smaller value for the borefield at high temperature is chosen to facilitate a faster temperature increase of the borefield. Except for these different parameters, no simplifications are considered for the borefield.

Ground Parameters	
Ground thermal conductivity	1.8 and 2.4$\frac{W}{m \cdot K}$
Undisturbed ground temperature	10°C
Fluid Parameters	
Thermal conductivity	0.513$\frac{W}{m \cdot K}$
Specific heat capacity	4000$\frac{J}{kg \cdot K}$
Density	1060$\frac{kg}{m^3}$
Viscosity	0.0079$\frac{kg}{m \cdot s}$
Freezing point	0°C
Flow rate per borehole	0.35 $\frac{l}{s}$
Borefield Parameters	
Configuration	14x14 rectangle
Borehole spacing	3 and 6m
Borehole installation	Single U-pipe
Borehole radius	75mm
Inner radius U-pipe	13mm
Outer radius U-pipe	16.7mm
center-to-center distance	62mm
U-pipe thermal conductivity	0.4 (W/mK)
Filling thermal conductivity	1.0 (W/mK)
Contact resistance pipe/filling	0 (mK/W)
Equivalent borehole resistance	0.2(mK/W)
Investment Cost	30 €/m

Table 5.3: Practical values of ground, fluid and borefield parameters used in the SDA.

5.2 Sizing Methods of the Concepts

In the previous section, different simplifications of the system were explained. These simplifications are applied to simplify the sizing of the different concepts. From the solar collector simplifications, it was concluded that flat plate collectors are the better option and hence, only the a versions of the concepts are further considered. In this section, the exact sizing strategies of the concepts are explained. These strategies are divided according to the seasonal storage that is used in the concepts, i.e. tank storage, low-temperature borefield storage or high-temperature borefield storage.

5.2.1 Tank Concepts

This section discusses the sizing strategy for the tank concepts in the SDA. It was explained in Section 4.1 that for tank storage a distinction is made between a fully mixed tank and a perfectly stratified tank. Moreover, it was also explained that for both options, solution pairs (tank volume - number of solar collectors) can be calculated based on the overall energy equation of the tank. In Figure 5.5, such solution pairs are plotted as an example. The figure corresponds to concept 1 with a fully mixed tank (see Section 3.3.1). In this figure, it is clearly visible that for larger tank volumes, the corresponding number of solar collectors decreases. This results from the limited heat storage capacity of smaller tanks. Indeed, in smaller tanks, the storage capacity is too limited to store enough heat during the summer to meet the heating demand in the winter. As a result, more solar collectors have to be added such that the heating demand in the winter can be met. For larger tanks on the other hand, more heat is seasonally stored and less collectors are required to meet the heating demand in the winter.

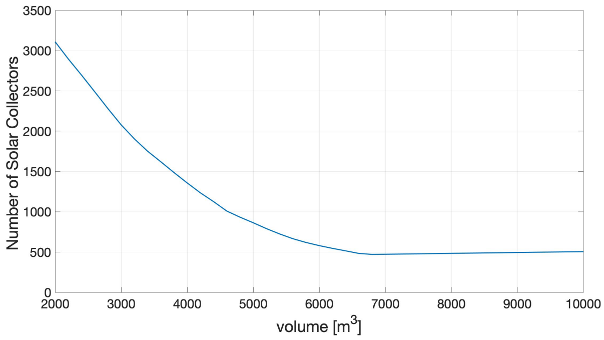


Figure 5.5: Example of the necessary number of collectors in function of the tank volume for concept 1.

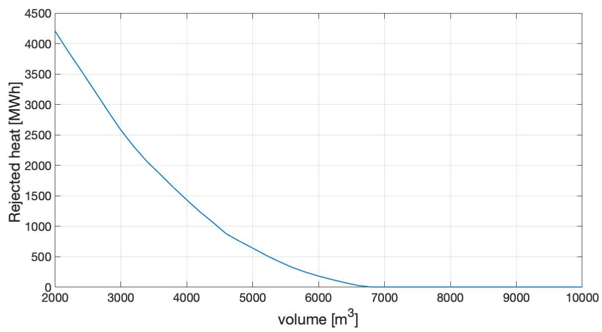


Figure 5.6: Example of rejected heat of collectors during summer in function of the tank volume for concept 1.

The use of a small storage tank with addition of a large amount of solar collectors is rather inefficient. After all, the tank is already saturated during summer, i.e. the energy content of the tank is at is maximum level. As a result, a lot of the additional collectors cannot be used in the months when their solar energy yield is the largest. Increasing the volume of the tank can solve this problem, since the tank can store more heat in that case and gets saturated less quickly. This is reflected in Figure 5.6, which shows the total rejected heat in a year as a function of the tank volume. Heat from the solar collectors is rejected if the tank is saturated, which clearly occurs more often for solution pairs with smaller tanks and a large amount of solar collectors.

The tank volume can be increased until the point the tank no longer becomes saturated during summer. Consequently, no more heat is rejected at this point. This point is the optimal point of operation of the tank storage in the SDA. That is, all the heat supplied by the solar collectors is used and the tank volume is completely used. The tank still reaches a temperature of 98°C, but only for a short amount of time. This is shown in Figure 5.7. In this figure, the temperature evolution of a fully mixed tank throughout the year is shown for three different tank volumes: a smaller tank volume, the optimal tank volume and a larger tank volume. For the smaller tank volume of 2000 m^3, it is clearly visible that it reaches the maximum temperature early in the year and the tank is saturated during several months. Recall that during these months, heat from the solar collectors is rejected, i.e. it cannot be added to the tank. This tank is said to be undersized. For the larger tank volume of 10 000 m^3 on the other hand, the tank never becomes saturated as it no longer reaches the maximum temperature of 98°C. However, this is not the optimal point of operation as the same number of solar collectors is required compared to smaller tank volumes (see Figure 5.5). Therefore, the tank with a volume of 10 000 m^3 is said to be oversized. The optimal volume of the tank corresponds to 6800 m^3, with the temperature just reaching 98°C and without saturating the tank.

To summarize, in the SDA the optimal tank volume corresponds to the volume where no heat is rejected and where the temperature in the tank still reaches 98°C. Note that this reasoning also holds for a perfectly stratified tank. In that case however, the evolution of the hot volume is considered instead of the evolution of the temperature. Of course, both the evolution of the tank temperature for the fully mixed approach and the evolution of the hot volume in tank for the perfectly stratified approach correspond to an evolution of heat content in the tank.

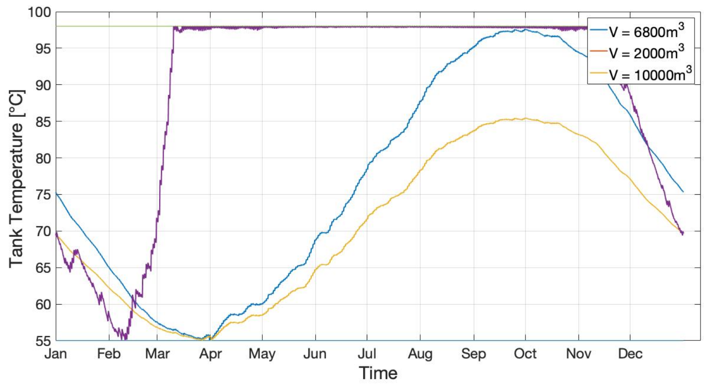


Figure 5.7: The temperature evolution in the tank of concept 1 for three different volumes.

5.2.2 Low-temperature Borefield Concepts

This section explains the sizing strategy for the low-temperature borefield concepts in the SDA. In Section 4.2.3, the sizing method for a low-temperature borefield was explained. This sizing method takes the heating and cooling loads on the borefield as inputs and gives the resulting length of the borefield as output. As the load on the borefield depends on the number of collectors, an iteration is performed over a relevant range of the total number of solar collectors. For each number of solar collectors, the method calculates the corresponding depth of the borefield. The pair (number of solar collectors - borefield depth) that results in the lowest initial investment cost, is selected as the optimum. In this way, both the number of collectors in the central solar collector field and the depth of the borefield are sized. Figure 5.8 shows an example of the results of this method for concept 9 (see Section 3.3.2). Figure 5.9 shows the corresponding initial investment cost. In this figure, a minimum is clearly visible, which is chosen as the optimal solution pair in the SDA. Note that the initial investment cost shown in Figure 5.9 only includes the investment of the solar collectors and the borefield. For the borefield, a cost of 30€/meter is used (see Table 5.3). Also note that this initial investment cost is calculated for flat plate collectors, since only this collector type is under consideration in the SDA (see Section 5.1.1). Similar figures are made for concepts 7, 8 and 10 and the optimal depth of the borefield and the optimal number of solar collectors are found in the same way.

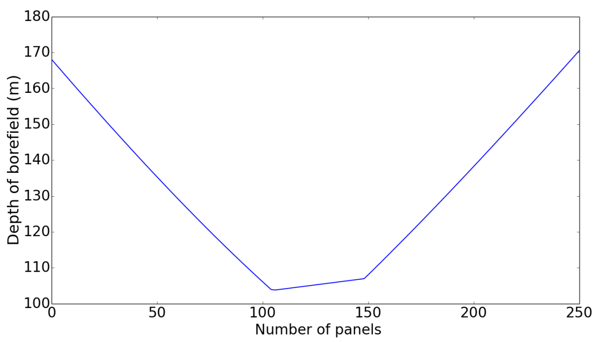


Figure 5.8: Example of the evolution of the borefield depth as a function of the number of panels for concept 9.

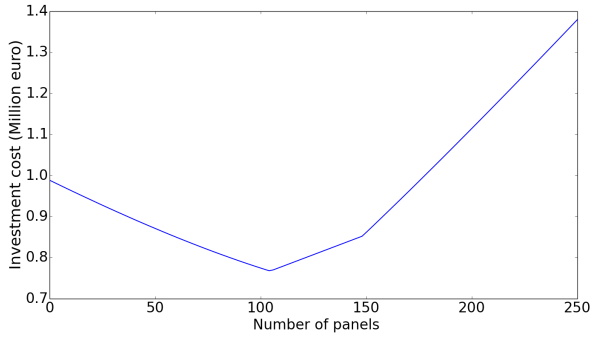


Figure 5.9: Example of the evolution of the initial investment cost as a function of the number of panels for concept 9. (Note that the investment cost only includes the investment of solar collectors and borefield.)

5.2.3 High-temperature Borefield Concepts

For a high-temperature borefield, a different sizing method is applied compared to a low-temperature borefield. After all, the method of Peere et al., which was applied for sizing of a low-temperature borefield, cannot be used anymore for a high-temperature borefield (see Section 4.2.3). Hence, for a high-temperature borefield, the method explained in Section 4.2.4 is used. Recall that for a certain heating and cooling load on the borefield, the method gives all borefield depths that satisfy two conditions. The first condition is that the fluid through the borefield has to remain between 0°C and 98 °C. The second condition states that the fluid temperature in the borefield has to remain above 35°C after 20 years of operation.

Just as for a low-temperature borefield, the load on a high-temperature borefield depends on the number of solar collectors that are connected with the borefield. Therefore, an iteration is again performed over a relevant range of the total number of solar collectors. Note that a double iteration is performed in this method, i.e. an iteration over the borefield depth and an iteration over the number of solar collectors. The result of this method is a set of solution pairs (borefield depth - number of collectors) that each satisfy the above two conditions. Similarly to the low-temperature borefield concepts, the solution pair with the lowest initial investment cost is selected.

This concludes Section 5.2 on the sizing methods for the different concepts in the SDA. The results of applying these sizing methods to the different concepts are shown in the next section.

5.3 Results

This section shows the results that are found in the SDA for the different concepts. Recall that only the a versions of the concepts using flat plate collectors are considered in the SDA. First, the solution pairs consisting of the dimensions of the seasonal storage and the number of solar collectors are listed for each concept, as well as the electricity use in each concept. These results are discussed in separate sections according to the type of seasonal storage that is applied in the concept, i.e. tank storage, low-temperature borefield storage or high-temperature borefield storage. Finally, the Net Present Value (NPV) and CO_2 emissions of each concept are calculated. Based on these parameters, a comparison between the different concepts is made.

5.3.1 Tank Concepts

The sizing of the tank concepts follows the method that was explained in Section 5.2.1. The result of this method is an optimal solution pair for the tank volume and the number of solar collectors. Recall that for tank storage, a distinction is made between two extreme cases, i.e. a fully mixed tank and a perfectly stratified tank. Using the above sizing method for both provides a range for the tank volume and the number of solar collectors. As the limits of this range are the two extremes, a realistically stratified tank lies somewhere in between. The results for the tank concepts, i.e. concepts 1 to 6 (see Section 3.3.1), are listed in Table 5.4.

	Concept 1		**Concept 2**	
	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	6800	2600	2300	2600
Storage efficiency [%]	57	81	78	81
Number of solar collectors [-]	468	328	298	328
Solar collector area [m^2]	1175	823	748	823
Electricity use [$\frac{kWh}{y}$]	18750	18750	67870	18750
	Concept 3		**Concept 4**	
	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	3900	2500	2300	2500
Storage efficiency [%]	67	80	77	80
Number of solar collectors [-]	360	301	283	301
Solar collector area [m^2]	904	756	710	756
Electricity use [$\frac{kWh}{y}$]	39379	39379	71267	39379
	Concept 5		**Concept 6**	
	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	3400	2200	2100	2200
Storage efficiency [%]	62	76	73	76
Number of solar collectors [-]	361	307	294	307
Solar collector area [m^2]	906	771	738	771
Electricity use [$\frac{kWh}{y}$]	30900	30900	55974	30900

Table 5.4: Results for the tank concepts in the SDA.

In this table, 'Mixed' corresponds to the system with a fully mixed tank and 'Stratified' corresponds to the system with a perfectly stratified tank. Since in the SDA the solar collector type is always set at flat plate, the total solar collector area is found by multiplying the number of solar collectors with the gross area of a single flat plate collector, i.e. 2.51 m^2 (see Section 4.3.3). The storage efficiency in Table 5.4 is calculated with Equation 4.5 for a fully mixed tank and with Equation 4.11 for a perfectly stratified tank.

In general, it can be observed from Table 5.4 that a perfectly stratified tank has a higher storage efficiency compared to a fully mixed tank. For the concepts without a central heat pump, i.e. the odd concepts, this leads to a smaller tank volume and fewer solar collectors in the system with a perfectly stratified tank. For the concepts with a central heat pump, i.e. the even concepts, the tank volume and the number of solar collectors are however smaller in the system with a fully mixed tank. This is due to the operation of the central heat pump, which provides supplementary heating to the system and therefore requiring less solar energy captured by the solar collectors. Recall from Section 4.1.2 that the central heat pump does not operate with a perfectly stratified tank, since the tank always provides heat at a temperature above the supply temperature of the heating network. Hence, the results for a perfectly stratified tank in the even concepts are also identical to the results of the odd concepts.
A more detailed interpretation of the results, explaining the nuances between the concepts, is provided in Appendix C.1.

Based on the results in Table 5.4, the initial investment cost of each concept can be determined with the following equation:

$$Investment = c(V) \cdot V_{tank} + 600\frac{€}{m^2} \cdot A_{collectors} + 400\frac{€}{m} \cdot L_{network} + Cost_{HPs} \quad (5.4)$$

with V_{tank} the tank volume and $A_{collectors}$ the solar collector area as mentioned in Table 5.4. Furthermore, c(V) corresponds to the specific investment cost of the seasonal tank storage (see Equations 4.37 and 4.38). 600 $\frac{€}{m^2}$ is the unit cost of the flat plate solar collectors and 400 $\frac{€}{m}$ is the unit cost of the district heating network at 55°C and 35°C (see Section 4.7). Recall that the unit cost of the network is per meter supply and return line and hence the length of the network $L_{network}$ corresponds to half the total length, i.e. 800 m. Finally, the relevant cost data of the heat pumps is mentioned in Table 4.7. Note that for the large central heat pumps always two heat pumps are placed centrally where one serves as a back-up and that local heat pumps always come in numbers of 50. This will also be the case for the borefield concepts in the following sections.

Evaluating Equation 5.4 for the six tank concepts leads to the investment costs that are shown in Figure 5.10. The figure illustrates that for concepts 1, 3 and 5, the investment cost is higher for the system with a fully mixed tank compared to the system with a perfectly stratified tank. For concepts 2, 4 and 6 on the other hand, the investment cost is slightly higher for the system with a perfectly stratified tank. This is in line with the interpretation of the results from Table 5.4. Figure 5.10 also shows that concept 2 has the lowest investment cost for a fully mixed tank, whereas concept 1 has the lowest investment cost for a perfectly stratified tank. However, the results for all concepts are close to each other and therefore the NPV is calculated in Section 5.3.4 to draw conclusions for these tank concepts.

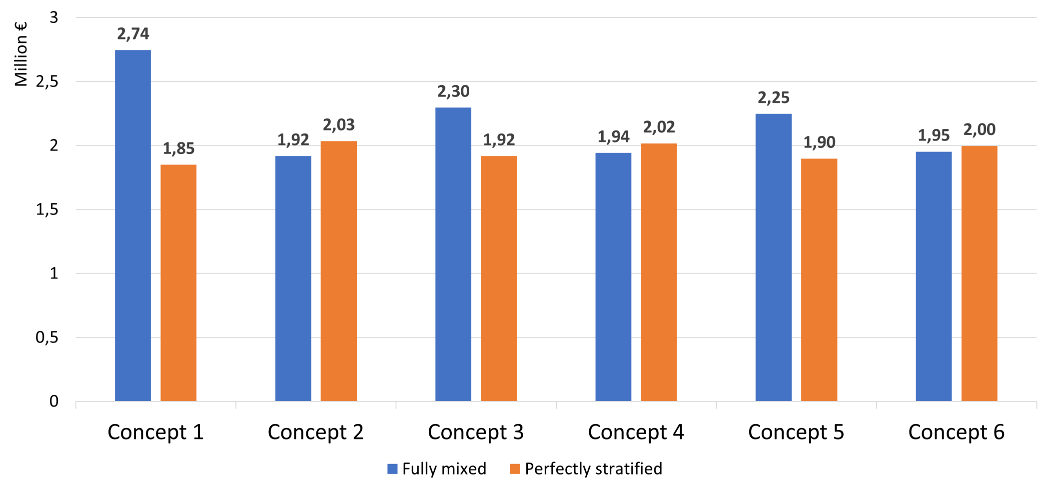


Figure 5.10: Investment cost of the tank concepts in the SDA.

5.3.2 Low-temperature Borefield Concepts

The sizing of the low-temperature borefield concepts follows the method that was explained in Section 5.2.2. The result of this method is an optimal solution pair for the borefield depth and the number of solar collectors. Recall that for borefield storage, two extreme values for the thermal conductivity of the ground are considered, i.e. 1.8 and 2.4 $\frac{W}{mK}$. The above sizing method is applied in both cases, providing a range for the borefield depth and the number of solar collectors. The results for the low-temperature borefield concepts, i.e. concepts 7 to 10 (see Section 3.3.2), are listed in Table 5.5.

	Concept 7		Concept 8	
	k_s= 1.8	k_s= 2.4	k_s= 1.8	k_s= 2.4
Borefield depth [m]	106	118	101	114
Number of boreholes [-]	196	196	196	196
Borehole spacing [m]	6	6	6	6
Borefield length [m]	20747	23108	19796	22364
Buffer tank volume [m^3]	45	45	43	43
Number of solar collectors [-]	106	32	88	10
Solar collector area [m^2]	266	80	221	25
Electricity use [$\frac{kWh}{y}$]	68699	68699	63749	63749
	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	2100	900	2100	900
Number of solar collectors [-]	202	144	202	144
Solar collector area [m^2]	507	361	507	361
	Concept 9		**Concept 10**	
	k_s= 1.8	k_s= 2.4	k_s= 1.8	k_s= 2.4
Borefield depth [m]	106	118	101	114
Number of boreholes [-]	196	196	196	196
Borehole spacing [m]	6	6	6	6
Borefield length [m]	20747	23108	19796	22364
Buffer tank volume [m^3]	45	45	43	43
Number of solar collectors [-]	206	132	188	110
Solar collector area [m^2]	517	331	472	276
Electricity use [$\frac{kWh}{y}$]	80849	80849	75899	75899

Table 5.5: Results for the low-temperature borefield concepts in the SDA.

Recall that for a low-temperature borefield only a borehole spacing of 6m is considered (see Section 5.1.5). The configuration of the borefield is always 14x14, resulting in a total of 196 boreholes. The borefield length is found by multiplying the amount of boreholes with the borefield depth.
Furthermore, Table 5.5 mentions the tank volume of the seasonal storage tank that is used in concepts 7 and 8, as well as the required number of solar collectors. Recall that this tank is used to centrally store heat for the production of domestic hot water. Both a fully mixed tank and a perfectly stratified tank are considered and give a range for the tank volume and the number of solar collectors.

In general, the results in Table 5.5 show that the borefield dimensions are identical in respectively concepts 7 and 9 and concepts 8 and 10. This is due to the borefield only storing energy for space heating, while energy for the production of hot water is either supplied by a central seasonal storage tank combined with a separate heating network (concepts 7 and 8) or supplied by a local hot water production unit (concepts 9 and 10). Furthermore, it is observed from this table that for each individual concept, the borefield depth (and length) is larger for a thermal conductivity of the ground of 2.4

$\frac{W}{mK}$ compared to 1.8 $\frac{W}{mK}$, whereas the number of solar collectors is smaller. This is a result of the optimisation criterion that is used, i.e. optimisation towards minimum cost instead of borefield depth.
A more detailed explanation of the nuances in the results of Table 5.5 is given in Appendix C.2.

Similarly to the tank concepts, the initial investment cost of each concept can be determined based on the results in Table 5.5 with the following equation:

$$Investment = 30\frac{€}{m} \cdot L_{borefield} + 600\frac{€}{m^2} \cdot A_{collectors} + 400/250\frac{€}{m} \cdot L_{network} \\ + 1000\frac{€}{m^3} \cdot V_{buffer} + Cost_{HPs}(+c(V) \cdot V_{tank}) \quad (5.5)$$

with $L_{borefield}$ the borefield length, $A_{collectors}$ the total solar collector area and V_{buffer} the buffer tank volume as mentioned in Table 5.5. Furthermore, 30 $\frac{€}{m}$ is the unit cost of the borefield, 600 $\frac{€}{m^2}$ is the unit cost of the flat plate solar collectors and 1000 $\frac{€}{m^3}$ is the unit cost of the buffer tank. The unit cost of the district heating network is 400 $\frac{€}{m}$ for the networks at 55 and 35°C and 250 $\frac{€}{m}$ for the network at 15°C (see Section 4.7). The length of the district heating network $L_{network}$ corresponds to 800 m supply and return line. The relevant cost data of the heat pumps is again mentioned in Table 4.7. The final term in Equation 5.5 is only added for concepts 7 and 8 with a seasonal storage tank, where V_{tank} is the volume of this tank and c(V) the unit cost according to Equations 4.37 and 4.38.

Evaluating Equation 5.5 for the four low-temperature borefield concepts leads to the investment costs that are shown in Figure 5.11. The figure shows that the investment cost is always lower for the systems with thermal conductivity of the ground of 2.4 $\frac{W}{mK}$. This means that although the borefield depth is larger for k_s= 2.4, the decrease in number of solar collectors compensates for this. Indeed, the decrease in solar collector cost outweighs the increase in the cost of the borefield. Note that for concepts 7 and 8 the worst case (most expensive investment) corresponds to the case with a fully mixed storage tank for DHW combined with a k_s value of 1.8 $\frac{W}{mK}$ for the borefield, whereas the best case (least expensive investment) corresponds to a perfectly stratified tank combined with a k_s value of 2.4 $\frac{W}{mK}$ for the borefield. Finally, Figure 5.11 also illustrates that concept 9 has the lowest investment cost and seems the most interesting out of these four concepts, based on the investment.

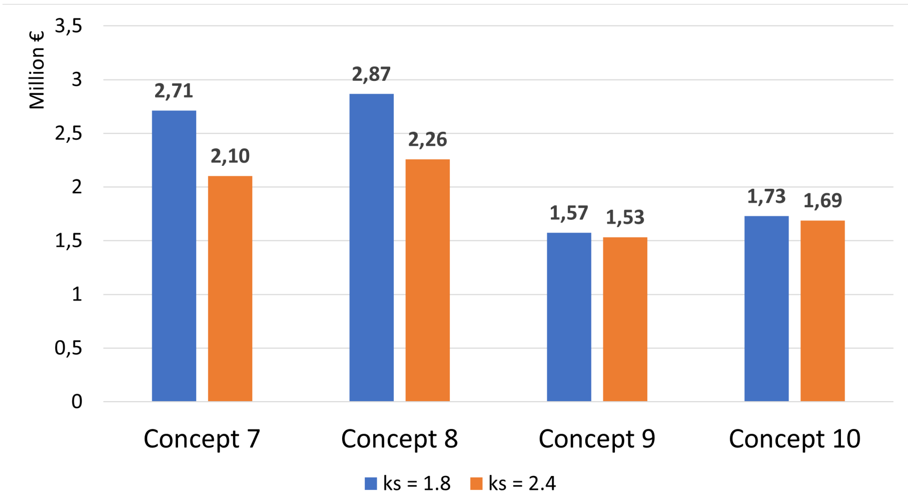


Figure 5.11: Investment cost of the low-temperature borefield concepts in the SDA.

5.3.3 High-temperature Borefield Concepts

The sizing of the high-temperature borefield concepts follows the method that was explained in Section 5.2.3. The result of this method is an optimal solution pair for the borefield depth and the number of solar collectors. Similarly to low-temperature borefield concepts, two extreme values for the thermal conductivity of the ground are considered, i.e. 1.8 and 2.4 $\frac{W}{mK}$. The above sizing method is applied in both cases, providing a range for the borefield depth and the number of solar collectors. The results for the high-temperature borefield concepts, i.e. concepts 11 to 13 (see Section 3.3.3), are listed in Table 5.6.

	Concept 11		**Concept 12**		**Concept 13**	
	k_s= 1.8	k_s = 2.4	k_s= 1.8	k_s = 2.4	k_s= 1.8	k_s = 2.4
Borefield depth [m]	125	115	120	110	110	100
Number of boreholes [-]	196	196	196	196	196	196
Borehole spacing [m]	3	3	3	3	3	3
Borefield length [m]	24500	22540	23520	21560	21560	19600
Buffer tank volume [m^3]	62	71	59	68	51	61
Solar collectors [-]	890	1030	850	990	830	990
Solar collector area [m^2]	2234	2585	2134	2485	2083	2485
Electricity use [$\frac{kWh}{y}$]	56402	56402	74062	74062	55875	55875

Table 5.6: Results for the high-temperature borefield concepts in the SDA.

Similarly to the low-temperature borefield concepts, the configuration of the borefield is always 14x14, resulting in a total of 196 boreholes. The borefield length is found by multiplying the amount of boreholes with the borefield depth. Recall that for a high-temperature borefield only a borehole spacing of 3m is considered (see Section 5.1.5).

In general, the results from Table 5.6 show that within each concept, the borefield length decreases and the number of solar collectors increases for k_s= 2.4 $\frac{W}{mK}$, compared to k_s= 1.8 $\frac{W}{mK}$. The increase in number of solar collectors results from the higher heat loss that occurs in the borefield with a ground thermal conductivity of 2.4 $\frac{W}{mK}$. Again, a more detailed interpretation of the results is provided in Appendix C.3.

The initial investment cost of each concept can be determined based on the results in Table 5.6 with the following equation:

$$Investment = 30\frac{€}{m} \cdot L_{borefield} + 600\frac{€}{m^2} \cdot A_{collectors} + 400\frac{€}{m} \cdot L_{network} + 1000\frac{€}{m^3} \cdot V_{buffer} + Cost_{HPs} \quad (5.6)$$

with $L_{borefield}$ the borefield length, $A_{collectors}$ the total solar collector area and V_{buffer} the buffer tank volume as mentioned in Table 5.6. The unit costs are identical to those in Equation 5.5. Recall that the length of the district heating network $L_{network}$ corresponds to 800 m supply and return line. Finally, the relevant cost data of the heat pumps is again mentioned in Table 4.7.

Evaluating Equation 5.6 for the three high-temperature borefield concepts leads to the investment costs that are shown in Figure 5.12. The figure shows that the investment cost is always lower for the systems with thermal conductivity of the ground of 1.8 $\frac{W}{mK}$. This is in line with the results in Table 5.6, which showed that fewer solar collectors are required in that case. Although the borefield length (and cost) slightly increases, the savings on the solar collectors compensate for this. Finally, Figure 5.12 also illustrates that concept 13 has the lowest investment cost out of these three high-temperature borefield concepts, but they differ only slightly.

In the next section, the NPV and CO_2 emissions of all concepts are calculated and a comparison is made based on these parameters.

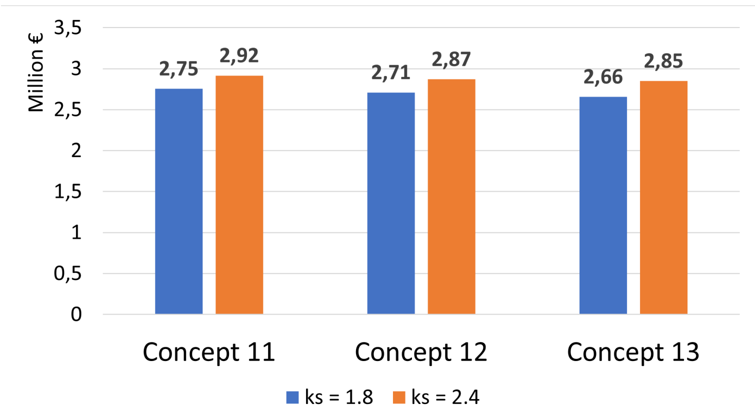


Figure 5.12: Investment cost of the high-temperature borefield concepts in the SDA.

5.3.4 Concept Comparison and Main Observations

In this section, a comparison between all concepts is made based on the net present value of their costs and on their CO_2 emissions. The NPV is calculated for a study period of 40 years and according to the method that is described in Section 4.7. Recall that for the calculation of the NPV, the (re)investment costs, maintenance costs and energy costs are taken into account and that a lower NPV of the costs is preferred. Along with the cost data provided in Section 4.7, the NPV calculation uses the results of the individual concepts, as listed in the previous sections (see Tables 5.4, 5.5, 5.6).

In Section 4.7, two different scenarios were mentioned for the electricity prices, i.e. the 'traditional pricing scenario' and the 'energy community scenario'. In the first scenario, a distinction is made between a non-household electricity price for the central heat pumps and a household electricity price for all other heat pumps located in the dwellings. In the second scenario, only the non-household electricity price is applied. These two scenarios lead to two different NPV calculations for each concept.

Calculating the NPV of the costs for all concepts in the 'traditional pricing scenario', leads to the results that are shown in Figure 5.13. Here, the NPV of the concepts is plotted against the average electricity use in the concepts. Subsequently, the electricity use of the concepts on the x-axis can be transformed into the CO_2 emissions of the concepts by applying the CO_2-intensity of the electricity grid. Currently, this CO_2-intensity corresponds to 167 $\frac{gCO_2}{kWh}$ [21]. Applying this factor on the electricity use of the concepts leads to the corresponding CO_2 emissions, which can be read on the top axis.
Calculating the NPV of the costs for all concepts in the 'energy community scenario', leads to the results that are shown in Figure 5.14. Again, the NPV of the concepts is plotted against the electricity use and the corresponding CO_2 emissions in the concepts. The results of the 'traditional pricing scenario' are depicted in grey.

Note that the electricity use in these figures corresponds to the average electricity use in the concept, e.g. the average of the electricity use in the systems with a fully mixed tank and a perfectly stratified tank. Hence, the CO_2 emissions of the concepts are average values as well. Furthermore, note that for each concept a range is calculated for the net present values. The upper and lower value of these ranges respectively correspond to the worst and best cases of the specific concepts, e.g. fully mixed vs perfectly stratified. For a tank storage, this results in very large ranges. In reality, a high degree of stratification is always preferred and mostly also obtained [2]. Therefore, the ranges in Figures 5.13 and 5.14 are limited to their second quarter. A stratification level of 0% corresponds to a fully mixed tank, while a stratification level of 100% corresponds to a perfectly stratified tank. Hence, a stratification level between 50% and 75% is assumed.

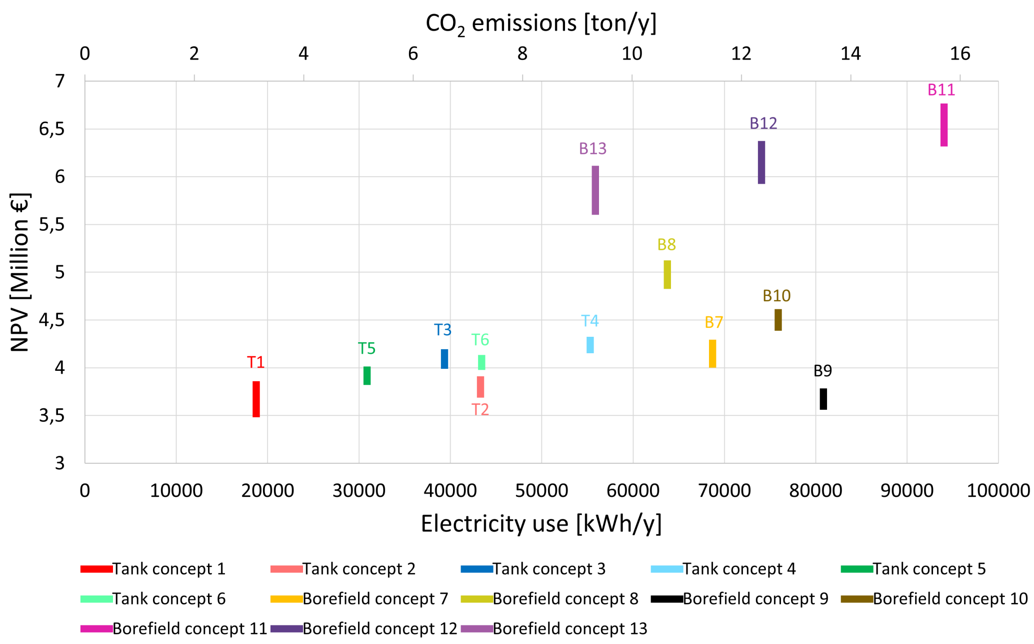


Figure 5.13: NPV of the concepts in the traditional pricing scenario in the SDA.

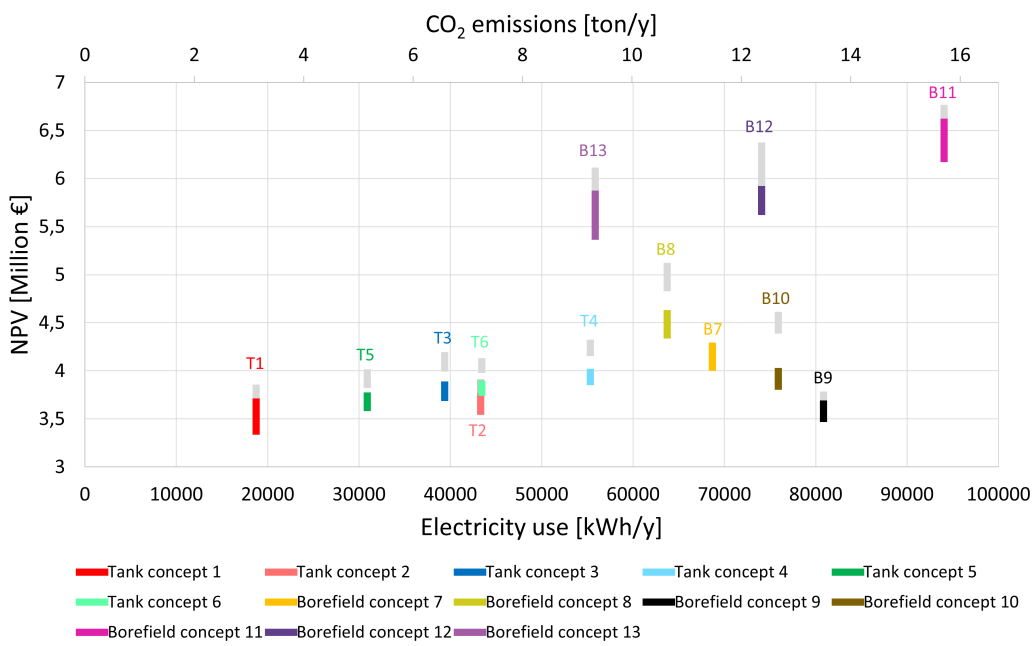


Figure 5.14: NPV of the concepts in the energy community scenario in the SDA.

The first observation from Figures 5.13 and 5.14 is that the different pricing scenarios in both figures do not influence the general trends that are visible. Nevertheless, the NPV of the costs in the 'energy community scenario' decreases compared to the 'traditional pricing scenario' due to the lower non-household electricity price that is applied to the electricity use of all heat pumps. As a result, the decrease in NPV is the largest for the concepts with decentral heat pumps, e.g. concept 10.

Secondly, it is observed from the above figures that the tank concepts can be considered superior to the borefield concepts. Indeed, these tank concepts show a similar range in NPV as the most interesting borefield concepts and their CO_2 emissions are considerably lower. From an NPV point of view, only concept 9 (and concept 10 in the energy community scenario) can still be considered competitive to the tank concepts. However, this comes at the cost of higher CO_2 emissions.

Thirdly, considering the tank concepts, Figures 5.13 and 5.14 show that the use of a large central heat pump in the concepts is not beneficial. Indeed, the concepts with a central heat pump, i.e. concepts 2, 4 and 6, have an NPV in the same range or higher than their corresponding concepts without a central heat pump, i.e. concepts 1, 3 and 5, while requiring more electricity for the operation of this central heat pump. Hence, the CO_2 emissions in these concepts are higher as well.
A second observation concerning the tank concepts is that concept 5 with a local hot water production unit outperforms concept 3 with micro booster heat pumps for DHW production. Indeed, concept 5 shows lower CO_2 emissions, as well as a lower range in NPV compared to concept 3. On the other hand, concept 1 without any supplementary heating systems for DHW in turn outperforms concept 5. Therefore, concept 1 can be considered the most interesting tank concept. However, an important remark has to be made regarding the robustness of this concept: since no supplementary heating systems are present, the tank storage has to provide all the necessary heat throughout the year. If not enough heat is stored, e.g. during a year with less solar irradiation, there is no system that can supply this heat and the heating demand cannot be met. Therefore, oversizing of this system might be necessary to account for changes in the solar yield, as well as in the heating demand. Of course, this would increase the NPV of the concept as well.

Fourthly, considering the low-temperature borefield concepts, concept 9 provides the option with the lowest NPV. From an NPV point of view, the use of a large central heat pump in concepts 7 and 9 is preferred over the use of small decentral heat pumps in concepts 8 and 10. Nevertheless, the use of decentral heat pumps can slightly reduce the CO_2 emissions compared to the use of a large central heat pump.
A second observation concerning the low-temperature borefield concepts is that CO_2 savings can be obtained by using a central seasonal storage tank for DHW in concepts 7 and 8 compared to using a local hot water production unit in concepts 9 and 10. However, this is accompanied by a significant increase in the NPV of the costs.

Finally, Figures 5.13 and 5.14 illustrate that the high-temperature borefield concepts are expensive compared to the other concepts. Moreover, these concepts still have considerable CO_2 emissions. Hence, from these figures, high-temperature borefield concepts can be considered uninteresting.

To summarize, the results in the SDA show that a district heating system with tank storage is more interesting than a system with borefield storage, considering its lower CO_2 emissions and costs. Concept 1 seems to outperform all other concepts but might turn out to be impractical due to the absence of supplementary heating systems, which leads to a lack of robustness. Concept 9 can be considered competitive cost-wise, but CO_2 emissions are significantly higher than in the tank concepts. In the next chapter, a new and detailed assessment method is applied to determine whether these conclusions still hold if the simplifications of the SDA are refined.

Chapter 6

Detailed Dynamic Assessment (DDA)

In this chapter, a second method to assess the different concepts of the district heating system is discussed. This second assessment method is named the Detailed Dynamic Assessment or DDA. As mentioned before, the simplifications made in the Simplified Dynamic Assessment (SDA) are refined in this chapter. The goal of applying both methods is to compare their ability to differentiate the interesting concepts and hence to decide whether general first analysis simplifications have a large impact on the final results.

This chapter first explains how the system simplifications of the SDA are refined in the DDA. This is done in Section 6.1. Subsequently, the new sizing methods of the DDA are explained in Section 6.2. Finally, in Section 6.3, the results of these sizing methods are shown. Similarly to the SDA, the results include for each concept the dimensions of the seasonal storage, the number of solar collectors and the annual electricity use. Furthermore, again the NPV and annual CO_2 emissions of the concepts are calculated and a comparison between the concepts is made.

6.1 Refinement of the System Simplifications of the SDA

In this section, it is explained how the system simplifications of the SDA are refined in the DDA. The most important refinements are made for the solar collectors, the large central heat pumps and the district heating network. Moreover, certain parameters of the borefield are changed as well. The refinements are discussed in separate sections for each system component.

6.1.1 Solar Collectors

In the SDA, a fixed efficiency profile was assumed for the solar collectors. This resulted from a fixed inlet temperature of 25°C and a fixed mass flow rate at its maximum value. A first consequence of these assumptions was a very optimistic thermal energy supply of the collectors, especially for the concepts with a tank storage. Moreover, vacuum tube collectors were found to be less efficient compared to flat plate collectors (see Section 5.1.1). However, these conditions are not very realistic. Indeed, the inlet and outlet temperature of the solar collectors are dependent on the temperature of the storage medium to which the working fluid transfers its heat. In the limit, the inlet temperature of the solar collector equals the temperature of the storage medium. In reality, of course the temperature is a bit higher to guarantee sufficient heat transfer from the working fluid to the storage medium.

Linking the solar collector temperatures to the storage medium temperatures particularly influences the concepts with a tank storage. For a fully mixed tank, temperatures reach near 98°C in the summer months and for a perfectly stratified tank, a collector outlet temperature of 98°C is always required. These high temperatures result in a lower efficiency of the solar collectors. Recall that the efficiency of a vacuum tube collector decreases less with increasing temperatures compared to the efficiency of a flat plate collector (see Section 2.2.3). Therefore, vacuum tube collectors are considered in the DDA as well, whereas only flat plate collectors were considered in the SDA. Hence, in the DDA, both the a and b versions of the district heating system concepts are considered (see Section 3.3).

The following sections briefly explain the exact methods to model the solar collectors for the different seasonal storages. In all cases, the efficiency of the solar collectors now varies throughout the year depending on the storage temperatures in a specific concept. This is in contrast to the SDA in which a static efficiency profile was assumed for all concepts.

Fully Mixed Tank

If solar collectors transfer their absorbed heat to a fully mixed tank, the inlet temperature of the collectors is set equal to the tank temperature. As the outlet temperature is always higher, heat can always be transferred to the tank and there is no need to put multiple collectors in series. Since the inlet temperature of the collectors follows the tank temperature, the efficiency of the solar collectors throughout the year depends on the temperature evolution of the tank. This temperature evolution is different in each concept and therefore each concept has its own efficiency profile. Moreover, the mass flow rate can be set at the maximum value for a high efficiency (see Section 4.3.3). Note that the efficiency evolution depends on the temperature evolution and vice versa, hence requiring an iterative approach.

Perfectly Stratified Tank

In a perfectly stratified tank, it is assumed that the hot volume always has a temperature of 98°C and the cold volume a temperature of 10°C. Therefore, the inlet temperature of the solar collectors is set equal to 10°C, while the minimum outlet

temperature of the solar collectors is set at 98°C. To achieve this outlet temperature, multiple collectors have to be put in series. It is assumed that the number of collectors that are placed in series can be varied each hour. Furthermore, it is assumed that no heat is transferred to the storage tank if the outlet temperature of 98°C is not achievable for a given solar irradiation. As the same temperatures are used in all concepts with a perfectly stratified tank, this results in the same efficiency profile of the solar collectors for all concepts. Nevertheless, the heat transferred to the tank can vary between concepts if the tank gets saturated, i.e. completely occupied with the hot volume.
Note that this behaviour is not in line with reality, as in reality heat can always be transferred to the cold volume. However, in this thesis only the two extreme situations (fully mixed - perfectly stratified) are modelled.

Low-temperature Borefield

For a borefield at low temperatures, the same efficiency as in the SDA is again used in the DDA. This can be done, since the temperature of the low-temperature borefield cannot exceed 25°C. Therefore, the assumption of an inlet temperature of the solar collectors of 25°C is justified. It can even be considered a pessimistic view on the efficiency, since the borefield mostly has a temperature lower than 25°C and hence lower inlet temperatures are possible.

High-temperature Borefield

For a high-temperature borefield, a similar approach as for a fully mixed tank is used to model the efficiency of the solar collectors. The temperature evolution of the borefield is set as the evolution of the inlet temperature of the solar collectors. After iteration, this again results in an evolution of the efficiency of the solar collectors and an evolution of the borefield temperature. Note that the borefield temperature does not reach the same high temperatures as in the tank. As a result, the efficiency decrease of the solar collectors in the DDA in comparison to the SDA is less pronounced for a high-temperature borefield than for a tank storage.

This concludes the refinements of the system simplifications for the solar collectors in the DDA. Next, the refinement of system simplifications for the supplementary heating systems is considered.

6.1.2 Supplementary Heating Systems

In the SDA, fixed COP and COP_R values were assumed for all heat pumps. For most heat pumps, this is a valid assumption and the assumption is still used in the DDA (see Section 5.1.2). The heat pumps with a fixed COP are the MB HP, the DHW HP and the small decentral heat pump. However, the large central heat pump is no longer modelled with a fixed COP in the concepts with a tank storage or a high-temperature borefield storage. Indeed, the temperature evolution of the storage has a significant impact on the COP of the large central heat pump. Therefore, the

COP-curves supplied by the manufacturer of these heat pumps are used to model this temperature dependency. Figures 6.1 and 6.2 show the COP-curves for the two large central heat pumps under consideration in this thesis. Recall that the *VITOCAL 300-G PRO, type BW 302.D140* is used in the concepts with a district heating network with a supply temperature of 35°C (see Section 4.6.1). Therefore, the relevant curve in Figure 6.1 is the 'B'-curve. The *VITOCAL 350-HT- PRO, type BW 352.AHT.096*, on the other hand, is used in concepts with a district heating network with a supply temperature of 55°C. The relevant curve in Figure 6.2 is thus the curve for an outlet temperature of 55°C. Note that an iterative method is used to model this temperature dependent *COP*, since the temperature evolution of the storage and the *COP*-evolution mutually influence each other.

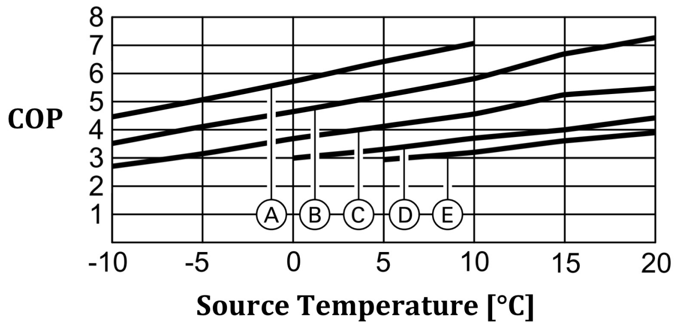


Figure 6.1: *COP* evolution of the *VITOCAL 300-G PRO, type BW 302.D140* in function of the source temperature for different outlet temperatures [64]. (A: T_{out} = 25°C, B: T_{out} = 35°C, C: T_{out} = 45°C, D: T_{out} = 55°C, E: T_{out} = 60°C)

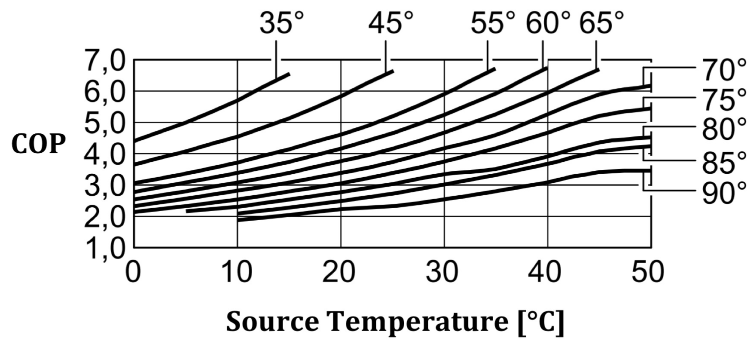


Figure 6.2: *COP* evolution of the *VITOCAL 350-HT- PRO, type BW 352.AHT.096* in function of the source temperature for different outlet temperatures [65].

For a low-temperature borefield storage, the temperature dependency of the *COP* is not taken into account in the DDA, similar to the SDA. The reason for this is that the temperature fluctuation throughout the year is more limited compared to the high-temperature STES options. Therefore, an average *COP* and COP_R are assumed in case of a borefield at low temperature. The values of *COP* and COP_R

are determined based on the manufacturer's data. A value of 5.8 for the COP is chosen, corresponding to a source temperature of 10°C and an outlet temperature of 35°C [64]. The value of the COP_R is also set at 5.8. This corresponds to a source temperature of 15°C and an outlet temperature of 35°C. Note that in the SDA, general values of 5 and 4 were set for respectively the COP and COP_R of the central heat pump, whereas here the exact values of the heat pump under consideration are used. The same holds for the small decentral heat pump, where now 6.1 and 5.9 are used as the respective values for the COP and COP_R [63]. Table 6.1 gives an overview of the COP- and COP_R-values used in the DDA for the different heat pumps.

Large Central HP	Tank	COP	see Figures 6.1 and 6.2
		COP_R	/
	High T Borefield	COP	see Figures 6.1 and 6.2
		COP_R	/
	Low T Borefield	COP	5.8
		COP_R	5.8
Small Decentral HP		COP	6.1
		COP_R	5.9
Micro booster HP		COP	5.8
		COP_R	/
Domestic Hot Water HP		COP	4.3
		COP_R	/

Table 6.1: COP- and COP_R-values for different heat pumps in the DDA.

6.1.3 District Heating Network

This section continues with refining the simplifications for the district heating network. In the SDA, two equations were derived to calculate the heat loss per unit length and temperature of the district heating network $U_{loss,transport}$. Equations 5.1 and 5.2 describe this unit heat loss for single pipes and twin pipes respectively. The same equations are used in the DDA.

Similarly to the SDA, the unit heat loss of the network $U_{loss,transport}$ is calculated for both single pipes and twin pipes, for a range of pipe diameters. The data for the dimensions of the district heating pipes is again obtained from the company *Logstor*. However, the DDA uses data of the 'PexFlextra' pipes instead of data of the 'Bonded pipe system', which was used in the SDA. This means that steel pipes are no longer considered but polymer (PEX) pipes are used instead. These polymer pipes allow average operating temperatures of 85°C compared to 120°C for steel pipes [38]. Since the maximum supply temperature of the network is only 55°C, polymer pipes are more suitable for the district heating network. The 'PexFlextra' pipes are insulated with PUR foam with a thermal conductivity λ_i of 0.022 $\frac{W}{mK}$ [38]. Other parameters that are necessary for the calculation of $U_{loss,transport}$ are identical to the parameters in the SDA and are listed in Table 5.2.

Figure 6.3 shows the heat loss per unit length and temperature $U_{loss,transport}$ as a function of the pipe diameter. Both single and twin pipes are considered and there is a choice between two degrees of insulation. Similarly to the SDA, the figure illustrates that twin pipes with a high insulation level (series 2) are the most interesting. Note that the curves are plotted over a limited range of pipe diameters due to the limited range of practical pipe dimensions available at *Logstor*.

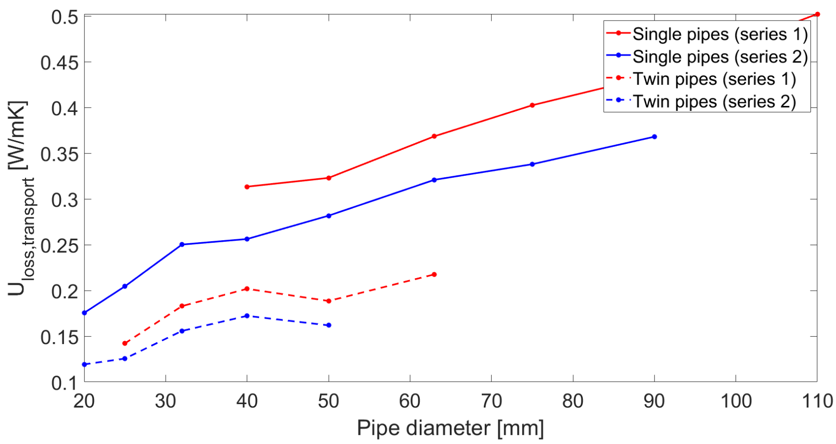


Figure 6.3: Unit heat loss in the district heating network in the DDA.

Compared to the SDA, the estimation of the pipe diameter is now refined. The mass flow rate through the district heating network $\dot{m}$ $[\frac{kg}{s}]$ is first estimated with the following equation:

$$\dot{m} = \frac{\dot{Q}}{c_p \cdot \Delta T} \tag{6.1}$$

with c_p the specific heat capacity of water (4185 $\frac{J}{kgK}$), ΔT the temperature difference between the supply and return line and $\dot{Q}$ the heat through the network in W. Secondly, the mass flow rate through the district heating network $\dot{m}$ is related to the pipe diameter d via the following equation:

$$\dot{m} = \rho_w \cdot \pi \cdot (\frac{d}{2})^2 \cdot v \tag{6.2}$$

with ρ_w the density of water (1000 $\frac{kg}{m^3}$) and v the velocity of water through the pipes in $\frac{m}{s}$.
Equations 6.1 and 6.2 are combined and solved for the pipe diameter d. They are evaluated for the peak heating demand $\dot{Q}$ of 160 kW (see Chapter 1) and a maximum velocity through the pipes of 2.5 $\frac{m}{s}$, as mentioned by *Logstor* [37]. This leads to the following results:

- For the network at 55/25°C: d = 0.025 m $\longrightarrow$ $U_{loss,transport}$ = 0.1255 W/mK

- For the network at 35/25°C: d = 0.044 m $\longrightarrow U_{loss,transport} = 0.1681$ W/mK

The results for $U_{loss,transport}$ are obtained from Figure 6.3 for twin pipes series 2. Knowing the unit heat loss $U_{loss,transport}$, the total heat loss in the district heating network $Q_{loss,transport}$ is calculated in the same way as in the SDA (see Equation 5.3).

In this section, the refinements for the district heating network in the DDA were explained. In the final two sections, refining the simplifications for the STES technologies is discussed.

6.1.4 Seasonal Storage Tank

In the SDA, there were no simplifications or unspecified parameters for the seasonal storage tank. The same holds for the DDA. The methodology and specifications as explained in Section 4.1 are used to model the behaviour of the seasonal storage tank.

6.1.5 Borefield Storage

In the SDA, certain borehole parameters were assumed. Recall that the borehole parameters, along with the fluid and ground parameters determine the g-functions of the borefield. In the SDA, single U-tube boreholes were considered. This single U-tube configuration had a pessimistic value of the equivalent borehole resistance R_b^* of 0.2 mK/W. The downside of this relatively high equivalent borehole resistance is that it results in higher peak fluid temperatures (see Equation 4.12). Consequently, the length of the borefield has to increase to satisfy the constraints in the sizing methods.

To take this effect in consideration, two new types of borefields are used in the DDA. The first one is a borefield configuration with a single U-tube and a relatively high value of the equivalent borehole resistance. This configuration more or less corresponds to the one used in the SDA. The second one is a borefield with double U-tubes and a relatively low value of the equivalent borehole resistance. Note that the mass flow rate is also doubled in the second configuration, but since it has a double U-tube, the mass flow rate per tube remains the same. Furthermore, also the heat carrier fluid is changed. In the SDA, the heat carrier fluid was water, while the DDA takes a water-glycol mixture as heat carrier fluid. Table 6.2 shows the practical values of the ground, heat carrier fluid and borehole parameters for both configurations. Also the initial investment cost of the borefield differs between the two types. The configuration with a single U-tube has a cost of 30 €/m, while the configuration with a double U-tube has a cost of 35 €/m [5].
Just as in the SDA, again two values of the ground thermal conductivity are considered. These correspond to the lowest and highest values in the Flemish region in Belgium. Moreover, two values are mentioned again for the borehole spacing: a spacing of 3m is used for the borefield at high temperature, while a spacing of 6m is used for the borefield at low temperature.

Ground parameters	Single U-pipe	Double U-pipe
Ground thermal conductivity	1.8 and 2.4$\frac{W}{m \cdot K}$	
Undisturbed ground temperature	10°C	
Fluid parameters		
Thermal conductivity	0.47$\frac{W}{m \cdot K}$	
Specific heat capacity	3930$\frac{J}{kg \cdot K}$	
Density	1033$\frac{kg}{m^3}$	
Viscosity	0.0079$\frac{kg}{m \cdot s}$	
Freezing point	-10°C	
Flow rate per borehole	0.35 $\frac{l}{s}$	0.70 $\frac{l}{s}$
Borefield Parameters		
Configuration	14x14 rectangle	
Borehole spacing	3 and 6m	
Borehole installation	Single U-pipe	Double U-pipe
Borehole diameter	130mm	
U-pipe diameter	32mm	
U-pipe thickness	2mm	
U-pipe shank spacing	74mm	
U-pipe thermal conductivity	0.42 (W/mK)	
Filling thermal conductivity	1.4 (W/mK)	
Contact resistance pipe/filling	0(mK/W)	
Equivalent borehole resistance	0.1858(mK/W)	0.0818(mK/W)
Investment cost	30 €/m	35 €/m

Table 6.2: Practical values of ground, fluid and borefield parameters used in the DDA for both the single U-tube configuration and double U-tube configuration.

6.2 Sizing Methods of the Concepts

In the previous section, the refinements of the simplifications for the different system components were explained. In this section, the changes in the sizing methods in the DDA compared to the SDA are discussed. In general, the sizing principles remain mostly the same. However, the difference with the sizing methods in the SDA is that the Net Present Value is the main optimisation criterion in the DDA. Again, the sizing methods are discussed for each storage technology separately.

6.2.1 Tank Concepts

For the concepts with a seasonal storage tank, the sizing method is again based on the overall energy equation of the tank (see Section 4.1). Both for a fully mixed tank and a perfectly stratified tank, solution pairs are obtained for the tank volume and the number of solar collectors. However, in contrast to the SDA, an additional step is done once the solution pairs are found. This step includes calculating the NPV of each of those solution pairs. The solution pair with the lowest NPV is then set as the optimum.

Figure 6.4 illustrates the evolution of the NPV of different solution pairs, represented by the tank volume. The figure corresponds to concept 2 with a fully mixed tank (see Section 3.3.1). The NPV in this figure includes (re)investment costs and maintenance costs of the tank and the solar collectors, as well as the electricity costs over a period of 40 years. Other costs in the concept, e.g. the cost of the district heating network, are fixed and will therefore not change the optimum. The figure illustrates that a clear minimum in NPV is present, which is set as the optimum. Note that the optimal solution pair differs between the systems with flat plate collectors and vacuum tube collectors.

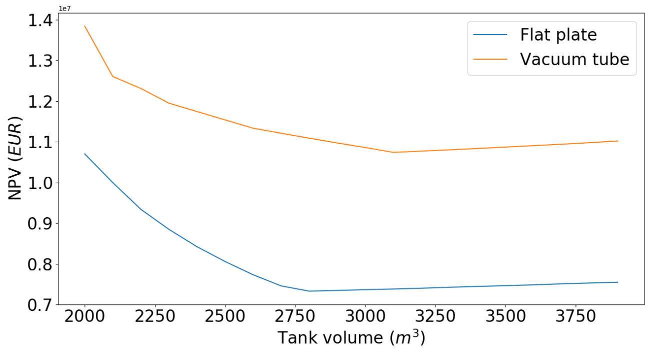


Figure 6.4: Example of the evolution of the NPV as a function of the tank volume for concept 2.

6.2.2 Borefield Concepts

For the concepts with a borefield storage, the sizing methods remain mostly the same as well. Here, the only difference with the SDA is that now the NPV is used as optimisation criterion instead of the initial investment cost. In the SDA, the initial investment cost was the criterion for both the low-temperature borefield and high-temperature borefield concepts (see Sections 5.2.2 and 5.2.3). The calculations and different steps in the respective methods do not change in the DDA. In the end, it is just the Net Present Value instead of the investment cost that selects the best solution pair.

6.3 Results

With the sizing methods that were explained in the previous section, new results are found for the different concepts in the DDA compared to the SDA. Recall that in the DDA, both a and b versions of the concepts are considered, respectively using flat plate collectors and vacuum tube collectors for the central solar field. First, the solution pairs consisting of the dimensions of the seasonal storage and the number of solar collectors are listed for each concept, as well as the electricity use in each concept. The results are discussed in separate sections according to the type of seasonal storage that is applied in the concept, i.e. tank storage, low-temperature borefield storage or high-temperature borefield storage. Finally, the NPV and CO_2 emissions of each concept are calculated and used to compare the different concepts.

6.3.1 Tank Concepts

The sizing of the tank concepts follows the method that was explained in Section 6.2.1. This method determines the optimal solution pair for the tank volume and the number of solar collectors, based on the minimum NPV. Table 6.3 lists the results for the tank concepts, i.e. concepts 1 to 6 (see Section 3.3.1), for both types of solar collectors.

	Concept 1				**Concept 2**			
	a: Flat plate		b: Vacuum tube		a: Flat plate		b: Vacuum tube	
	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	20200	3500	7500	2900	2800	3500	3100	2900
Storage efficiency [%]	42	78	54	80	77	78	81	80
Number of solar collectors [-]	5700	1810	2370	320	1530	1810	930	320
Solar collector area [m^2]	14307	4543	10926	1475	3840	4543	4287	1475
Electricity use [$\frac{kWh}{y}$]	18750	18750	18750	18750	80116	18750	87739	18750
	Concept 3				**Concept 4**			
	a: Flat plate		b: Vacuum tube		a: Flat plate		b: Vacuum tube	
	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	10400	3400	6500	2700	3300	3400	3800	2700
Storage efficiency [%]	60	77	65	79	78	77	81	79
Number of solar collectors [-]	2130	1720	1510	320	1340	1720	930	320
Solar collector area [m^2]	5346	4317	6961	1475	3363	4317	4287	1475
Electricity use [$\frac{kWh}{y}$]	39464	39464	39464	39464	70028	39464	73775	39464
	Concept 5				**Concept 6**			
	a: Flat plate		b: Vacuum tube		a: Flat plate		b: Vacuum tube	
	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	4400	2900	4400	2300	3300	2900	3700	2300
Storage efficiency [%]	58	74	61	76	75	74	79	76
Number of solar collectors [-]	3030	1320	1250	250	780	1320	660	250
Solar collector area [m^2]	7605	3313	5763	1153	1958	3313	3043	1153
Electricity use [$\frac{kWh}{y}$]	40276	40276	40276	40276	66464	40276	68533	40276

Table 6.3: Results for the tank concepts in the DDA.

In this table, 'Mixed' corresponds to the system with a fully mixed tank and 'Stratified' corresponds to the system with a perfectly stratified tank. Furthermore, the total solar collector area is found by multiplying the number of solar collectors with the gross area of a single collector, i.e. 2.51 m^2 for a flat plate collector and 4.61 m^2 for a vacuum tube collector (see Section 4.3.3). Recall that in concepts 5 and 6, a local production unit for DHW is used and each dwelling is equipped with two additional flat plate solar collectors (see Section 6.1.1).

In general, Table 6.3 shows that for flat plate collectors, the same observations are made as in the SDA. For the concepts without a central heat pump, i.e. the odd concepts, the perfectly stratified tank volume and its corresponding number of solar collectors are smaller due to the higher storage efficiency compared to the fully mixed tank. For the even concepts, the fully mixed tank volume and its corresponding number of solar collectors are smaller due to the operation of the central heat pump (see Section 5.3.1). On the other hand, different observations are made regarding the vacuum tube collectors. In these systems, the perfectly stratified tank volume and the number of solar collectors are always smaller compared to the fully mixed tank. This results from the efficient operation of the vacuum tube collectors with a perfectly stratified tank.
A more detailed interpretation of the results, explaining the nuances between the concepts, as well as similarities to the results found in the SDA, is provided in Appendix D.1.

Similarly to Section 5.3.1, the initial investment cost of each concept can be determined based on the results in Table 6.3 and using Equation 5.4. However, this equation only considers flat plate collectors with a unit price of 600 $\frac{€}{m^2}$. In the DDA, vacuum tube collectors are considered as well and in that case, the unit price corresponds to 900 $\frac{€}{m^2}$. All other parameters in the equation are identical as in Section 5.3.1.

Calculating the investment costs for the six tank concepts leads to the results that are shown in Figure 6.5. The figure illustrates that for a fully mixed tank, the investment cost of the system with flat plate collectors is always lower than the cost of the system with vacuum tube collectors. For a perfectly stratified tank on the other hand, the investment cost of the system with vacuum tube collectors is always lower than the cost of the system with flat plate collectors. Furthermore, Figure 6.5 shows that concept 6 has the lowest investment cost for a fully mixed tank, both for the systems with flat plate collectors and vacuum tube collectors. On the other hand, concept 5 has the lowest investment cost for a perfectly stratified tank in both cases.

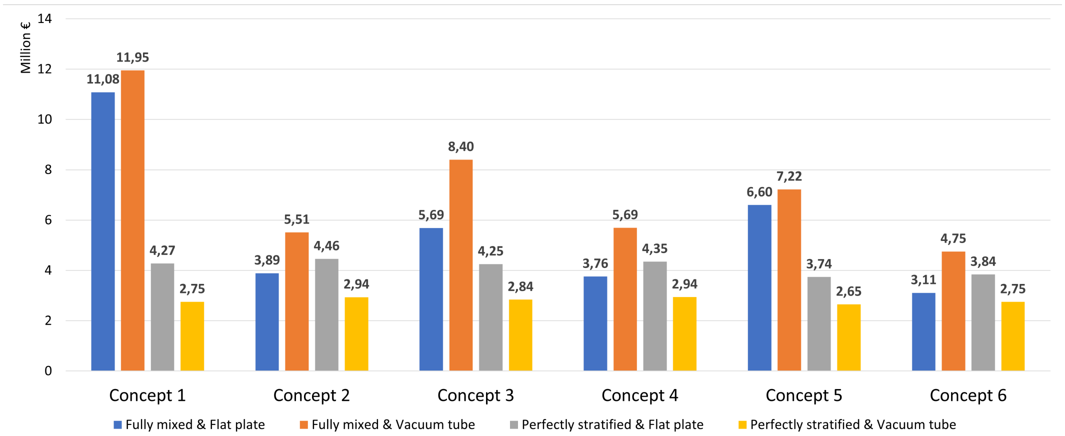


Figure 6.5: Investment cost of the tank concepts in the DDA.

6.3.2 Low-temperature Borefield Concepts

For the low-temperature borefield concepts, the sizing method was explained in Section 6.2.2. This method determines the optimal solution pair for the borefield depth and the number of solar collectors, based on the minimum NPV. Recall that in the DDA, two different configurations of the borefield are considered, i.e. single U-tube and double U-tube, resulting in different values for the equivalent borehole resistance (see Section 6.1.5). Moreover, two extreme values for the thermal conductivity of the ground k_s are again considered, i.e. 1.8 and 2.4 $\frac{W}{mK}$. The above sizing method is applied for all possible combinations: single U-tube and k_s= 1.8, single U-tube and k_s= 2.4, double U-tube and k_s= 1.8 and double U-tube and k_s= 2.4. Since in the DDA, both flat plate collectors and vacuum tube collectors are considered, this is done for both collector types in the system.

It is found that for the low-temperature borefield concepts, the combination of single U-tube and k_s= 1.8 results in the largest borefield dimensions, whereas the combination of double U-tube and k_s= 2.4 results in the smallest borefield dimensions. Therefore, these pairs of parameters are selected as respectively the upper and lower limit of the system's dimensions and hence the investment cost. Table 6.4 lists the results for the low-temperature borefield concepts, i.e. concepts 7 to 10 (see Section 3.3.2), for both types of solar collectors.

	Concept 7				Concept 8			
	a: Flat plate		b: Vacuum tube		a: Flat plate		b: Vacuum tube	
	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4
Borefield depth [m]	117	67	117	67	108	61	108	61
Number of boreholes [-]	196	196	196	196	196	196	196	196
Borehole spacing [m]	6	6	6	6	6	6	6	6
Borefield length [m]	22932	13132	22932	13132	21168	11956	21168	11956
Buffer tank volume [m^3]	32	32	32	32	32	32	32	32
Number of solar collectors [-]	0	0	0	0	0	0	0	0
Solar collector area [m^2]	0	0	0	0	0	0	0	0
Electricity use [$\frac{kWh}{y}$]	56270	56270	56270	56270	49596	49596	49596	49596
	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified	Mixed	Stratified
Tank volume [m^3]	6500	1200	2700	700	6500	1200	2700	700
Number of solar collectors [-]	3030	820	1070	220	3030	820	1070	220
Solar collector area [m^2]	7605	2058	4933	1014	7605	2058	4933	1014

	Concept 9				Concept 10			
	a: Flat plate		b: Vacuum tube		a: Flat plate		b: Vacuum tube	
	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4	Single U k_s= 1.8	Double U k_s= 2.4
Borefield depth [m]	117	67	117	67	108	61	108	61
Number of boreholes [-]	196	196	196	196	196	196	196	196
Borehole spacing [m]	6	6	6	6	6	6	6	6
Borefield length [m]	22932	13132	22932	13132	21168	11956	21168	11956
Buffer tank volume [m^3]	32	32	32	32	32	32	32	32
Number of solar collectors [-]	0	0	0	0	0	0	0	0
Solar collector area [m^2]	0	0	0	0	0	0	0	0
Electricity use [$\frac{kWh}{y}$]	77796	77796	77796	77796	71122	71122	71122	71122

Table 6.4: Results for the low-temperature borefield concepts in the DDA.

Table 6.4 mentions the results for both flat plate and vacuum tube collectors. Recall that in concepts 9 and 10, two additional flat plate collectors are added per dwelling, as part of a local hot water production unit (see Section 6.1.1).
Furthermore, Table 6.4 mentions the tank volume of the seasonal storage tank for domestic hot water that is used in concepts 7 and 8, as well as the required number of solar collectors.

What strikes in Table 6.4 is that for all concepts, the optimal solution does not include any solar collectors that are connected to the borefield. This results from the sizing method that is applied in the DDA, i.e. an optimisation based on the minimum NPV. It seems that this minimum NPV is obtained by excluding solar collectors in combination with a borefield. This means that the only heat that is injected in the borefield originates from the cooling of the dwellings. In other words, the temperature of the soil is slowly decreased throughout the years. Solar collectors are however still used in combination with the central storage tank in concepts 7 and 8, as well as in the local hot water production unit in concepts 9 and 10.
Other interpretations of the results in this table are similar to those in the SDA and are shortly repeated in Appendix D.2.

Similarly to Section 5.3.2, the initial investment cost of each concept can be determined based on the results in Table 6.4 and using Equation 5.5. However, both flat plate collectors with a unit price of 600 $\frac{€}{m^2}$ and vacuum tube collectors with a unit price of 900 $\frac{€}{m^2}$ are now considered. Moreover, the unit cost of the borefield equals 30 $\frac{€}{m}$ for a single U-tube configuration, whereas a unit cost of 35 $\frac{€}{m}$ is considered for a double U-tube configuration. All other parameters in Equation 5.5 remain the same.
The results for the investment costs of the different concepts are shown in Figure 6.6. Note that again the combination of a k_s value of 1.8 $\frac{W}{mK}$ with single U-tube configuration of the borefield is considered on the one hand and the combination of a k_s value of 2.4 $\frac{W}{mK}$ with double U-tube configuration on the other hand. Furthermore, for concepts 7 and 8 the worst case (most expensive investment) corresponds to the case with a k_s value of 1.8 $\frac{W}{mK}$ combined with a fully mixed tank, whereas the best case (least expensive investment) corresponds to a k_s value of 2.4 $\frac{W}{mK}$ combined with a perfectly stratified tank.
The figure shows that the investment cost is always lower for the systems with thermal conductivity of the ground of 2.4 $\frac{W}{mK}$, used in combination with a double U-tube configuration of the borefield. Furthermore, in concepts 7 and 8, the use of vacuum tube collectors leads to a lower investment cost compared to flat plate collectors. For concepts 9 and 10 on the other hand, there is no difference in investment cost between both collector types. This is due to the fact that the only collectors used in these concepts are the flat plate collectors in the local hot water production unit. Finally, Figure 6.6 clearly illustrates that the investment costs of the concepts with a local hot water production (9 and 10) are lower compared to the concepts with a central heat storage for DHW (7 and 8). Out of these concepts, concept 9 has the

lowest initial investment cost.

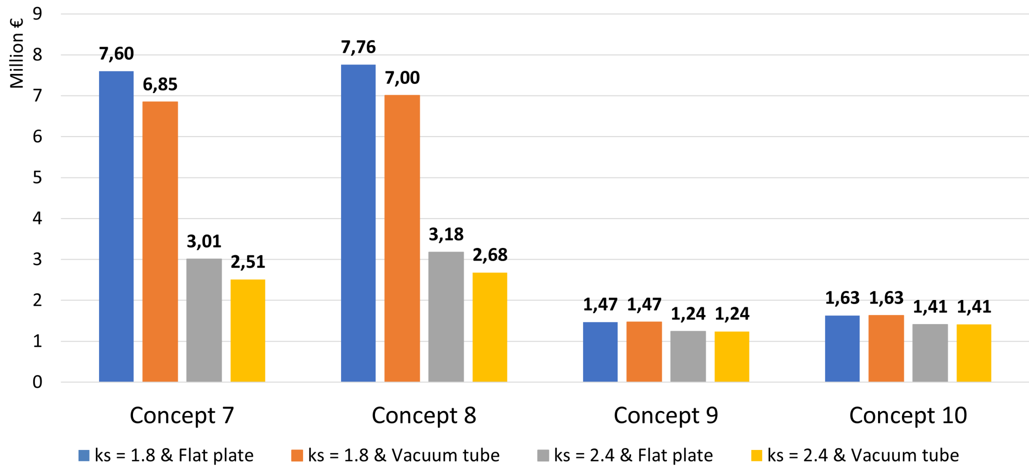


Figure 6.6: Investment cost of the low-temperature borefield concepts in the DDA.

6.3.3 High-temperature Borefield Concepts

The sizing of the high-temperature borefield concepts follows the method that was explained in Section 6.2.2, determining the optimal solution pair for the borefield depth and the number of solar collectors based on the minimum NPV. Again, all possible combinations between the single U-pipe and double U-pipe configurations on the one hand and the thermal ground conductivities of 1.8 and 2.4 $\frac{W}{mK}$ on the other hand, are considered. However, the combination of parameters that leads to the smallest dimensions of the system is the one between double U-pipe and k_s= 1.8. On the other hand, the combination that leads to the largest dimensions of the system is the one between single U-pipe and k_s= 2.4. This is the result of more heat loss in the borefield if the thermal conductivity of the ground is set at 2.4 $\frac{W}{mK}$. Hence, for the high-temperature borefield concepts, a thermal conductivity of 1.8 $\frac{W}{mK}$ is preferred. Recall that for the low-temperature borefield concepts, a higher thermal conductivity of 2.4 $\frac{W}{mK}$ was preferred because of the requirement for the temperature to remain above 0°C.
The above combinations of parameters that lead to the smallest and largest system dimensions are set as respectively the lower and upper limit of the sizing and hence the investment cost. Table 6.5 lists the results for the high-temperature borefield concepts, i.e. concepts 11 to 13 (see Section 3.3.3), both for flat plate and vacuum tube solar collectors.

Concept 11				
	a: Flat plate		b: Vacuum tube	
	Single U k_s= 2.4	Double U k_s= 1.8	Single U k_s= 2.4	Double U k_s= 1.8
Borefield depth [m]	107	67	85	92
Number of boreholes [-]	196	196	196	196
Borehole spacing [m]	3	3	3	3
Borefield length [m]	20972	13132	16660	18032
Buffer tank volume [m^3]	493	278	307	207
Number of solar collectors [-]	1310	1010	500	430
Solar collector area [m^2]	3288	2535	2305	1982
Electricity use [$\frac{kWh}{y}$]	84658	83668	85139	84210
Concept 12				
	a: Flat plate		b: Vacuum tube	
	Single U k_s= 2.4	Double U k_s= 1.8	Single U k_s= 2.4	Double U k_s= 1.8
Borefield depth [m]	115	74	90	70
Number of boreholes [-]	196	196	196	196
Borehole spacing [m]	3	3	3	3
Borefield length [m]	22540	14504	17640	16807
Buffer tank volume [m^3]	508	295	327	222
Number of solar collectors [-]	1370	1060	540	400
Solar collector area [m^2]	3439	2661	2489	1844
Electricity use [$\frac{kWh}{y}$]	59437	57398	57858	56271
Concept 13				
	a: Flat plate		b: Vacuum tube	
	Single U k_s= 2.4	Double U k_s= 1.8	Single U k_s= 2.4	Double U k_s= 1.8
Borefield depth [m]	102	67	86	66
Number of boreholes [-]	1240	980	590	440
Borehole spacing [m]	3	3	3	3
Borefield length [m]	19992	13132	16856	12936
Buffer tank volume [m^3]	408	192	262	178
Number of solar collectors [-]	1240	980	590	440
Solar collector area [m^2]	3112	2460	2720	2028
Electricity use [$\frac{kWh}{y}$]	54240	50560	54953	53436

Table 6.5: Results for the high-temperature borefield concepts in the DDA.

Recall that for a high-temperature borefield only a borehole spacing of 3m is considered (see Section 6.1.5). The configuration of the borefield is again 14x14, resulting in a total of 196 boreholes. The borefield length is found by multiplying the amount of boreholes with the borefield depth.
Furthermore, Table 6.5 mentions the results for both flat plate and vacuum tube

collectors. In concept 13, two additional flat plate collectors are added per dwelling to supply hot water (see Section 6.1.1).

In general, the results in Table 6.5 show that both the borefield length and the number of solar collectors decrease for k_s= 1.8 $\frac{W}{mK}$ compared to k_s= 2.4 $\frac{W}{mK}$. As mentioned before, a thermal conductivity of 1.8 $\frac{W}{mK}$ is preferred for the high-temperature borefield concepts, since it corresponds to less heat loss in the borefield.
A more detailed interpretation of the results is provided in Appendix D.3.

The initial investment cost of each concept can be determined based on the results in Table 6.5 and using Equation 5.6. However, this equation only considers flat plate solar collectors with a unit price of 600 $\frac{€}{m^2}$. Now, vacuum tube solar collectors with a unit price of 900 $\frac{€}{m^2}$ are considered as well. Moreover, the unit cost of the borefield in this equation (30 $\frac{€}{m}$) corresponds to a single U-tube configuration, whereas in the DDA also a double U-tube configuration is considered with a unit cost of 35 $\frac{€}{m}$. All other parameters in the equation are identical as in Section 5.3.3.

The investment costs for the three high-temperature borefield concepts are shown in Figure 6.7. Note that again the combination of a k_s value of 2.4 $\frac{W}{mK}$ with single U-tube configuration of the borefield is considered on the one hand and the combination of a k_s value of 1.8 $\frac{W}{mK}$ with double U-tube configuration on the other hand.
The figure confirms that the investment cost is always lower for the systems with thermal conductivity of the ground of 1.8 $\frac{W}{mK}$. Concerning the difference between systems with either flat plate collectors or vacuum tube collectors, it is concept-dependent which of these systems has the lowest investment cost. In concept 11 and 13, flat plate collectors seem more interesting, whereas in concept 12, vacuum tube collectors lead to a slightly lower investment cost.

In the next section, the NPV and CO_2 emissions of all concepts are calculated and a new comparison is made based on these parameters. The results are compared to the those of the SDA from the previous chapter.

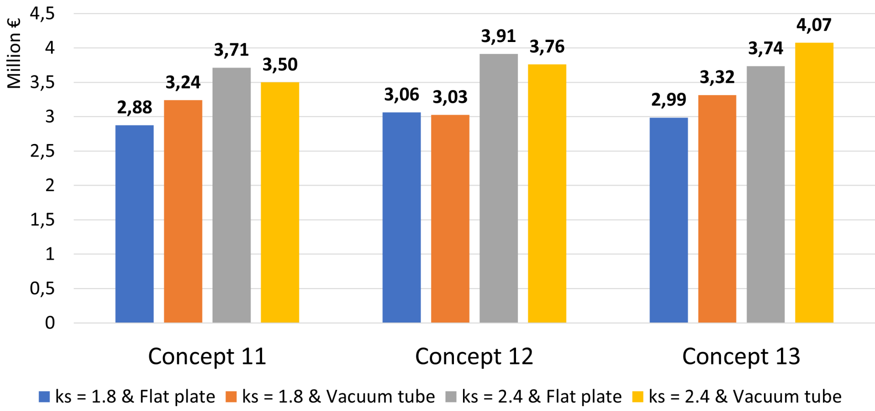


Figure 6.7: Investment cost of the high-temperature borefield concepts in the DDA.

6.3.4 Concept Comparison and Main Observations

In this section, the concepts are compared based on the NPV of their costs and on their CO_2 emissions. The NPV is calculated for a study period of 40 years and according to the method that is described in Section 4.7. Along with the cost data provided in that section, the NPV calculation uses the results of the individual concepts, as listed in Tables 6.3, 6.4 and 6.5.

Similarly to the SDA, two different scenarios for the electricity prices are considered, i.e. the 'traditional pricing scenario' and the 'energy community scenario'. In the first scenario, a distinction is made between a non-household electricity price for the central heat pumps and a household electricity price for all decentral heat pumps. In the second scenario, only the non-household electricity price is applied.

Since in the DDA both flat plate collectors and vacuum tube collectors are considered, corresponding to versions a and b of the concepts respectively, the NPV is calculated for each version and in both scenarios for the electricity price. The results are listed in Appendix E and the following conclusions are drawn:

- For the **tank concepts (1 to 6)**, the NPV range is lower in case vacuum tube collectors are used in the system.
- For the **low-temperature borefield concepts (7 to 10)**, vacuum tube collectors provide a lower range for the NPV of concepts 7 and 8, while for concepts 9 and 10 the NPV range is independent of the collector type. The latter is due to the absence of central collectors in these concepts (see Section 6.3.2).
- For the **high-temperature borefield concepts (11 to 13)**, the NPV range is lower in case flat plate collectors are used in the system.

Figure 6.8 shows the results for all concepts in the 'traditional pricing scenario'. For each concept, the optimal choice of collector type is applied. The NPV of the concepts is plotted against the average electricity use in the concepts. Subsequently, the electricity use of the concepts on the x-axis can be transformed into the CO_2 emissions of the concepts by applying the CO_2-intensity of the grid of 167 $\frac{gCO_2}{kWh}$ [21]. Figure 6.9 shows the NPV of the costs for all concepts in the 'energy community scenario'. Again, the NPV of the concepts is plotted against the electricity use and the corresponding CO_2 emissions in the concepts. The results of the 'traditional pricing scenario' are depicted in this figure in grey.

The same remarks as in the SDA hold. Firstly, the electricity use in these figures corresponds to the average electricity use in the concept, e.g. the average of the electricity use in the systems with a fully mixed tank and a perfectly stratified tank. Furthermore, a range is calculated for the net present values of the concepts. The upper and lower value of these ranges respectively correspond to the worst and best cases of the specific concepts, e.g. fully mixed vs perfectly stratified. For a tank storage, again a stratification level between 50% and 75% is assumed (see Section 5.3.4).

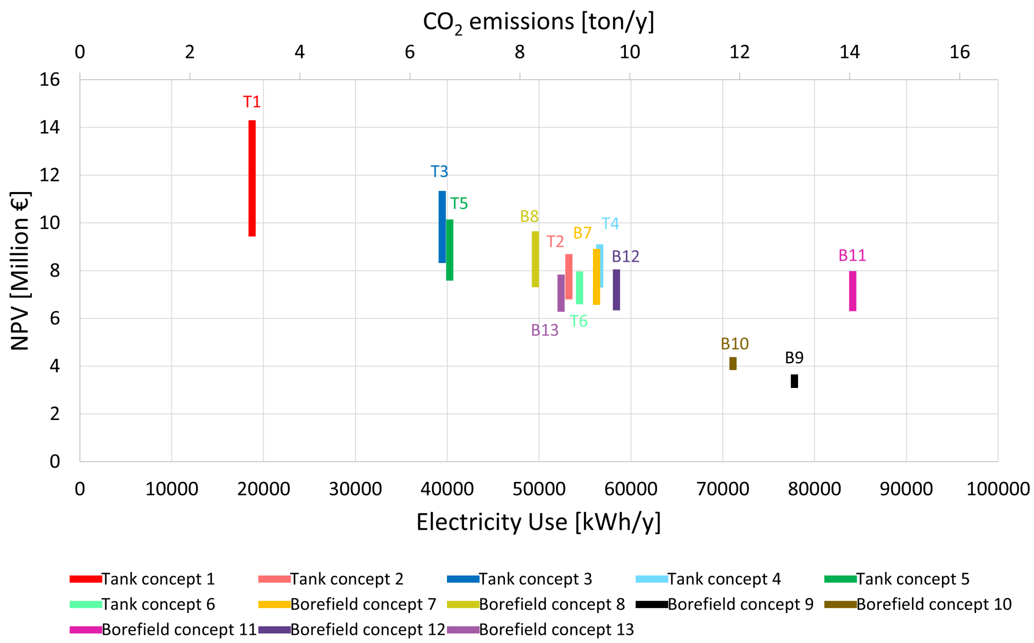


Figure 6.8: NPV of the concepts in the traditional pricing scenario in the DDA.

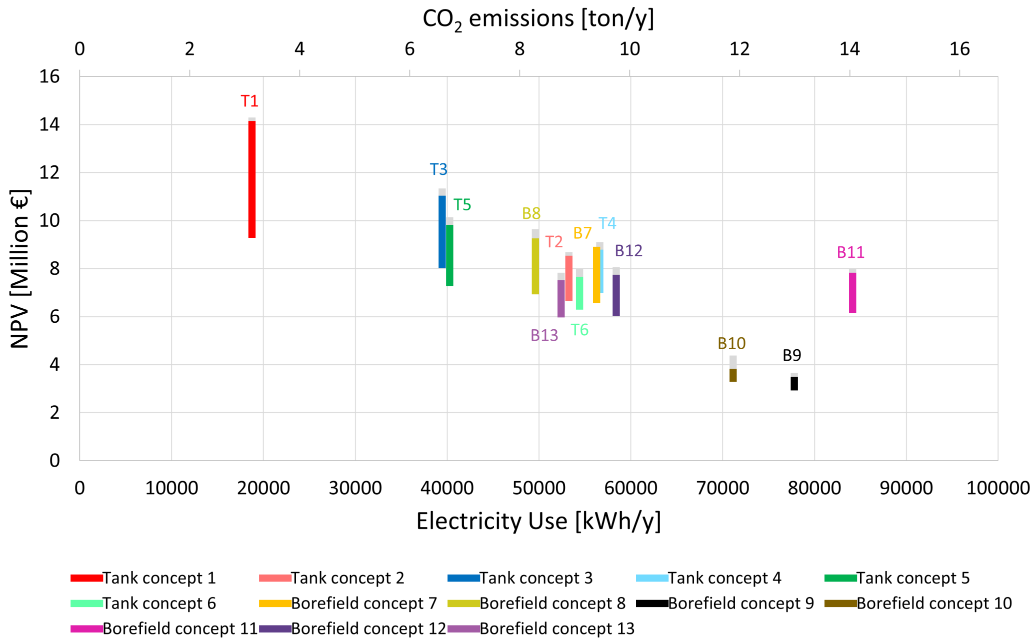


Figure 6.9: NPV of the concepts in the energy community scenario in the DDA.

The first and most important observation in the above figures is that the type of assessment method (SDA vs DDA) has a large influence on the final results. In the SDA, the concepts with a tank storage were more interesting because of a lower NPV and electricity use (see Section 5.3.4). In the DDA on the other hand, the concepts with a low-temperature borefield storage can be considered more interesting. More precisely, concepts 9 and 10 stand out. The Net Present Value of these concepts is at least 2 million euros less than the NPV of all other concepts. This is mainly due to a significant increase in the NPV of the tank concepts compared to the SDA. The NPV of the tank concepts now reaches the same level as for the high-temperature borefield concepts, corresponding to an increase of more than 1 million euros compared to the SDA. In other words, the simplifications that were made in the SDA favoured the tank concepts and led to an incorrect comparison between the concepts. Therefore, new observations are made based on the results in the DDA.

As said, for the concepts with a low-temperature borefield storage, concepts 9 and 10 outperform concepts 7 and 8. Their NPV is at least 2 million euros lower. This is the result of the expensive central storage tank and the additional heating network in concepts 7 and 8. Another observation for the low-temperature borefield concepts is that NPV-wise concept 9 is more interesting than concept 10. However, this lower NPV comes at the cost of more CO_2 emissions. The higher NPV of concept 10 results from the higher investment cost of the small decentral heat pumps in all dwellings, compared to the two central heat pumps in concept 9. The lower electricity use of concept 10, on the other hand, is the result of less heat loss in the heating network and a higher COP value of the heat pumps.

For the concepts with a tank storage, significant differences are observed between the SDA and DDA. Firstly, the NPV off all tank concepts has increased considerably. This is a direct result of refining the simplifications in the DDA. A very important refinement in that regard is the temperature dependent efficiency of the solar collectors (see Section 6.1.1). For the concepts with a borefield storage, the increase in NPV is less pronounced (high-temperature borefield concepts) or not present (low-temperature borefield concepts). Secondly, another observation within the tank concepts is that the concepts with a central heat pump have a lower NPV than their counterparts without a central heat pump (T2 vs T1, T4 vs T3 and T6 vs T5). The concepts with an individual domestic hot water production unit (T5 and T6) still outperform the concepts with a micro booster heat pump for domestic hot water (T3 and T4). However, in contrast to the SDA, the concepts with an individual hot water production unit now also have a lower NPV compared to the concepts with a centralised production of domestic hot water (T1 and T2). Note that concept 1 still has the lowest CO_2 emissions. However, these low CO_2 emissions come at the cost of a large NPV.

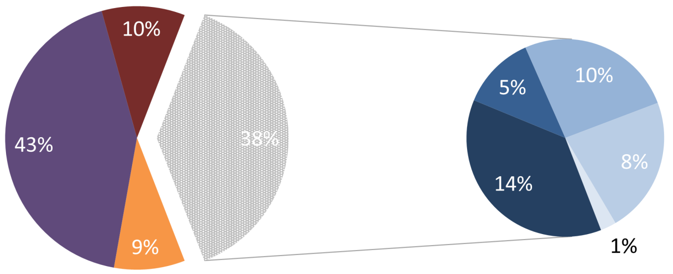
Concerning the concepts with a high-temperature borefield, concept 13 still outperforms concepts 11 and 12. Just as in the SDA, the NPV range of the high-temperature borefield concepts is similar, but the CO_2 emissions of concept 13 are lower. An important difference with the SDA however is that these high-temperature borefield concepts are no longer outperformed by all other concepts. Except for concept 11, they come at the same level as most of the tank concepts.

To summarize, the results in the DDA significantly differ from the results in the SDA, leading to the conclusion that the simplifications in the SDA can be considered too simple for a reliable comparison between the different concepts. New results in the DDA show that concepts 9 and 10 are the most interesting concepts, based on the considerably lower costs compared to the other concepts. In the next section, the results of these two concepts are further examined.

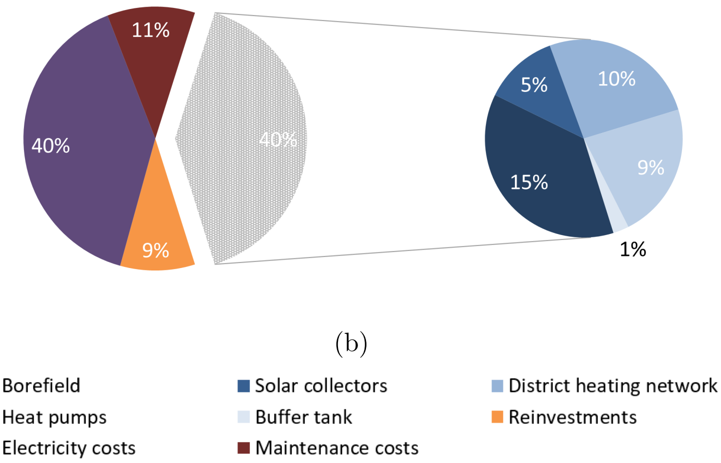
6.3.5 Further Assessment of the Interesting Concepts

As concluded in Section 6.3.4, concepts 9 and 10 are the most interesting concepts based on their lower NPV of their costs. The composition of the NPV of the costs of concepts 9 and 10 is illustrated below. For both concepts, the composition of the NPV is determined in the 'traditional pricing scenario', as well as in the 'energy community scenario'.

Figures 6.10a and 6.10b illustrate the composition of the NPV of concept 9 in the traditional pricing scenario and in the energy community scenario respectively. In each figure, the left diagram shows the different parts in the NPV, i.e. the investment cost, the reinvestment costs, the maintenance costs and the electricity costs. In the right diagram, the composition of the investment cost is illustrated. The figures show that in the traditional pricing scenario, the electricity costs have the largest contribution to the NPV, whereas in the energy community scenario, the electricity costs and the investment cost equally contribute to the NPV. In both scenarios, the largest part of the investment cost is made up by the cost of the borefield.



(a)



(b)

Figure 6.10: Composition of the NPV of concept 9 in (a) the traditional pricing scenario and (b) the energy community scenario.

Figures 6.11a and 6.11b illustrate the composition of the NPV of concept 10 in the traditional pricing scenario and in the energy community scenario respectively. Again, in the traditional pricing scenario, the electricity costs have the largest contribution to the NPV. However, in the energy community scenario, the contributions clearly change and the largest contribution originates from the investment cost. Moreover, in contrast to concept 9, the investment cost is dominated by the cost of the decentral heat pumps.

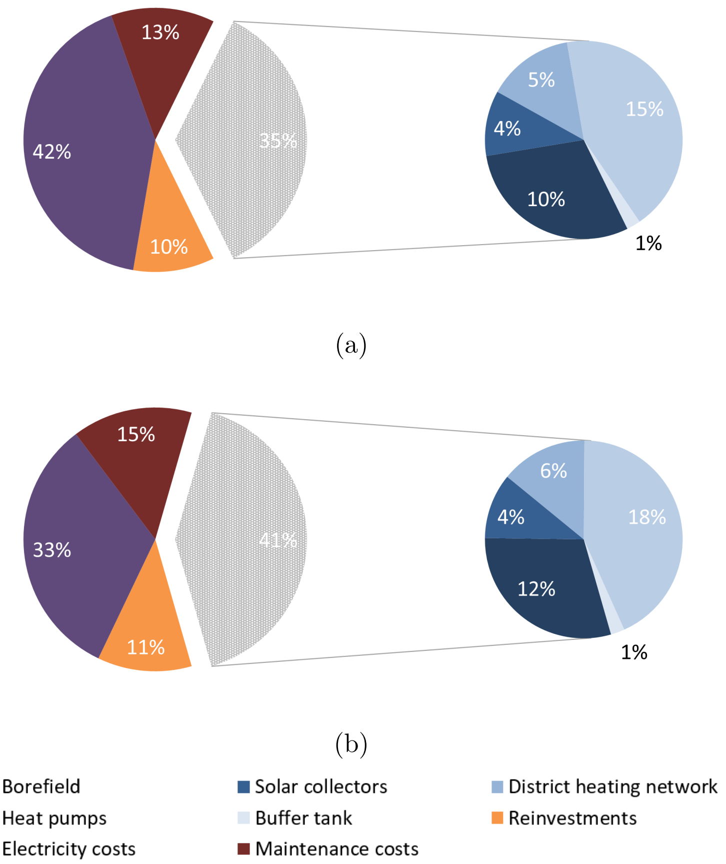


Figure 6.11: Composition of the NPV of concept 10 in (a) the traditional pricing scenario and (b) the energy community scenario.

6.4 Conclusion

This concludes Chapter 6 with a detailed assessment of the different concepts. It is shown that the results of the DDA differ significantly from the results of the SDA. Hence, the system simplifications that were applied in the SDA created a distorted picture and cannot be used for a reliable comparison between the concepts. The new, detailed assessment illustrated that concepts 9 and 10 are superior to the other concepts based on their lower costs.
These results also end the second part of this thesis, providing an answer to the question '*How do different concepts of a district heating system with STES compare to each other on an environmental (CO_2 emissions) and economic (Net Present Value) basis?*'. In the next part of the thesis, benchmarking of the concepts is considered.

Part III

Benchmarking of the Concepts

Chapter 7

Benchmarking

In this chapter, the results of the previous chapter are compared to two benchmark cases. Both benchmark cases consider a completely individual heating and cooling approach. In the first benchmark case, heating is provided by a gas condensing boiler, whereas cooling is provided by the same individual cooling system as in some of the concepts. The second benchmark case is a completely electrified case where heating and cooling are provided by an air-to-water heat pump. In this second benchmark case, photovoltaic (PV) solar panels are considered in each dwelling as well.

This chapter is divided into two main sections. In Section 7.1, the first benchmark case is explained in more detail and the primary energy use, the CO_2 emissions and the NPV of this system are calculated. Moreover, the results of the most interesting concepts of the previous chapter are compared to these benchmark numbers. Subsequently, in Section 7.2, this procedure is repeated for the second benchmark case.

7.1 First Benchmark Case with Gas Condensing Boilers

The first benchmark case considers a more or less 'traditional' heating approach, i.e. heating is provided in each dwelling by a gas condensing boiler. Cooling on the other hand is done with the same 'air-conditioning' system as introduced in some of the concepts. The same space heating, cooling and domestic hot water demand per dwelling as defined in Chapter 1 are considered.

The gas condensing boiler is modelled with a static efficiency. A range for this efficiency is applied from 92% to 98%, corresponding to modern gas condensing boilers [5]. Hence, the gas consumption can be calculated as follows:

$$g_c = \frac{Q_{demand}}{\eta_{gas}} \tag{7.1}$$

with g_c the gas consumption in kWh, Q_{demand} the heating demand in kWh and η_{gas} the efficiency of the gas condensing boiler.
For cooling, the same air-to-air heat pump is considered as in the previous individual cooling systems (see Section 4.4). This heat pump is modelled with a fixed COP_R-value of 4. The electricity use of the heat pump is calculated using Equation 2.5.

7.1.1 Primary Energy Use

In this section, the annual primary energy use of the benchmark with a gas condensing boiler is calculated. It consists of two parts: the first part is the combustion of the gas itself for space heating and domestic hot water, whereas the second part is the primary energy use related to the electricity use of the individual cooling system. To calculate the first part, Equation 7.1 can be used. To calculate the second part, the conversion factor of the Belgian electricity grid has to be known. In 2019, 54% of the Belgian electricity was produced by nuclear power plants, 27% was produced by gas power plants and 19% originated from renewables [10]. The efficiency of a nuclear power plant, a gas power plant and renewables are estimated at respectively 33%, 50% and 100%. Using this data, The conversion factor of the Belgian electricity grid is calculated as follows:

$$\eta_{grid} = \frac{54 \cdot 0.33 + 27 \cdot 0.5 + 19}{100} = 0.5032 \approx 0.5 \tag{7.2}$$

Hence, the primary energy use that corresponds to the electricity use of the cooling system can be calculated by dividing the electricity use by this conversion factor. The results of the above calculations are shown in Figure 7.1 for a district of 50 dwellings. This figure shows the primary energy use of the benchmark case with a gas condensing boiler in function of the efficiency of the gas condensing boiler. The figure shows that the primary energy use of the district ranges between 406 and 384 MWh. Note that this calculation is only done for the present time. No estimation is done about the future primary energy use, since this largely depends on the conversion factor of the electricity grid, for which the evolution is highly uncertain.

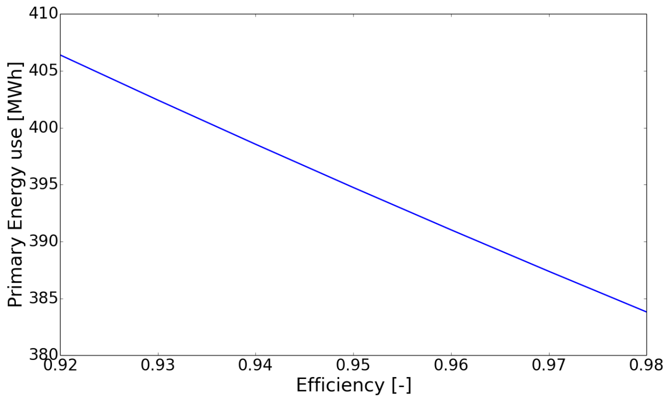


Figure 7.1: Primary energy use of the benchmark case with a gas condensing boiler in function of the gas condensing boiler efficiency.

7.1.2 CO_2 Emissions

In this section, the annual CO_2 emissions of the district of 50 dwellings in the gas benchmark case are calculated. The CO_2 emissions again consist of two parts: one part represents the CO_2 emissions of the combustion of the gas itself and the other part represents the CO_2 emissions related to the electricity use of the cooling systems.

For the first part, gas with a CO_2 intensity of 0.2 $\frac{kgCO_2}{kWh}$ is assumed [73]. Multiplying this CO_2 intensity with the gas consumption obtained from Equation 7.1, yields the CO_2 emissions corresponding to the heating demand. For the second part, the CO_2 intensity of the Belgian electricity grid has to be known. In 2019, this CO_2 intensity was 0.167 $\frac{kgCO_2}{kWh_{elec}}$ [21, 10]. In contrast to the conversion factor of the electricity grid for which no future estimate was done, future estimates are now made for the CO_2 intensity. The European green deal prescribes that the European electricity grid has to be carbon neutral in 2050. Therefore, two additional CO_2 intensities are considered. The first one assumes a carbon neutral electricity grid, i.e. 0 $\frac{kgCO_2}{kWh_{elec}}$. The second one is assumed to be at the halfway point between the current situation and carbon neutrality, i.e. 0.0835 $\frac{kgCO_2}{kWh_{elec}}$. Hence, three CO_2 emission scenarios are considered: the current situation, the halfway point and carbon neutrality. Figure 7.2 shows the results for the above data. It shows the CO_2 emissions of the district in function of the gas condensing boiler efficiency for the three different scenarios. Note that for the carbon neutral electricity grid scenario, all the CO_2 emissions result from the combustion of gas.

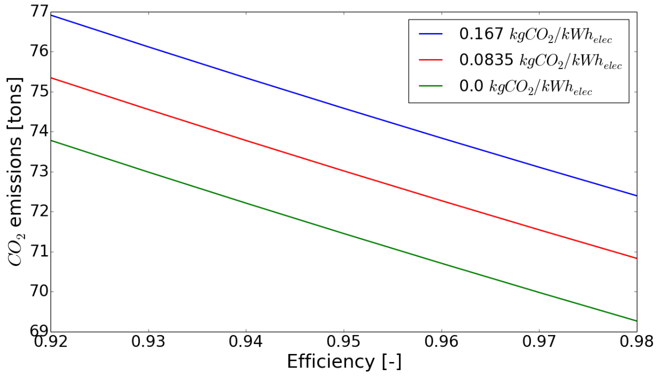


Figure 7.2: CO_2 emissions of the benchmark case with a gas condensing boiler in function of the gas condensing boiler efficiency.

7.1.3 Net Present Value

In this section, the NPV of the district of 50 dwellings with each an individual heating system with gas condensing boiler, is calculated. To calculate this NPV, the same procedure as explained in Section 4.7 can be used. Again, the (re)investment costs, maintenance costs and energy costs are considered over a time horizon of 40 years. To calculate the net present value of this benchmark case however, still some relevant cost data has to be known. An overview of this cost data is shown in Table 7.1. Note that two gas prices are taken into consideration. This is based on the relatively low gas price in Belgium compared to the EU average. In the future, a CO_2-related penalty can potentially increase the gas price and hence also a higher gas price, set at the current EU average, is considered. For the electricity price, only the price for households is relevant for this benchmark case as it considers an individual heating approach.

General Cost Data	
Inflation rate R_i	2%
Market interest rate R	5.5%
Interest rate r	3.43%
Gas condensing boiler	
Initial Investment cost [36]	€3950
Reinvestment [5]	20 years
Maintenance cost [36]	100 €/2 years
Cooling system (AC)	
Initial Investment cost [12]	€601 (outdoor unit) €1655 (indoor units)
Reinvestment [5]	20 years
Maintenance cost [5]	4%
Gas price	
Belgium (2020) [25] EU average (2020) [25] Annual price increase	0.0411 €/kWh 0.0544 €/kWh 5.87%
Electricity price	
Belgium (2020, household) [22] Annual price increase	0.2316 €/kWh 5.87%

Table 7.1: General overview of relevant cost data used to calculate the NPV of the benchmark case with gas condensing boilers.

Using the above cost data in combination with the primary energy use (see Section 7.1.1), the NPV of the benchmark case with gas condensing boilers can be calculated. Figure 7.3 shows the result of this calculation for the two gas prices under consideration. For the Belgian gas price, the net present value ranges between more or less 2.77 and 2.87 million euros, while for the EU average gas price, the net present value ranges between 3.25 and 3.38 million euros. The gas price clearly has a significant impact on the net present value.

Figure 7.4 shows the contribution of the different components to this net present value for both gas pricing scenarios. The figure clearly illustrates that the gas is the major cost component in both scenarios.

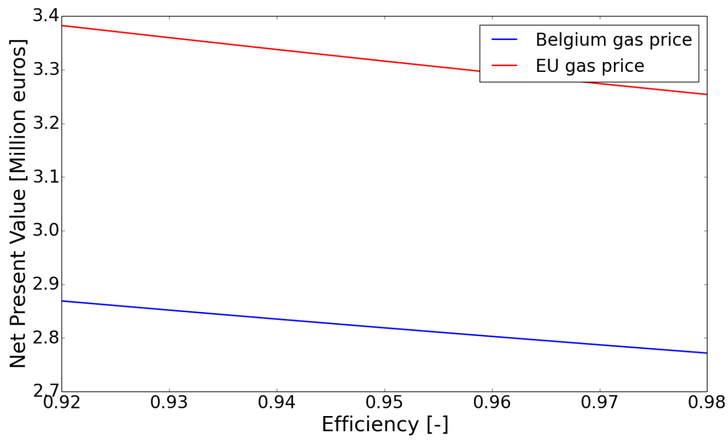


Figure 7.3: Net present value of the benchmark case with a gas condensing boiler in function of the gas condensing boiler efficiency.

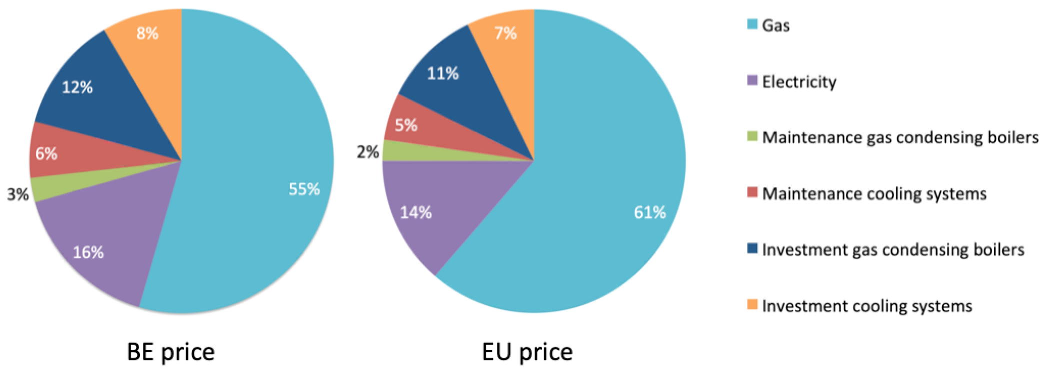


Figure 7.4: Composition of the NPV of the benchmark case for both the Belgian gas price scenario and the EU average gas price scenario.

7.1.4 Comparison Between Interesting Concepts and Benchmark Case with Gas

In this final section of the gas benchmark case, the results of the previous sections are compared to the most interesting concepts of the district heating system, i.e. concept 9 and concept 10. Table 7.2 gives an overview of the different results of the previous sections. Note that the three CO_2 emissions correspond to the three scenarios. A first important conclusion is that the primary energy use and the CO_2 emissions of concepts 9 and 10 are significantly lower than in the gas benchmark case. The difference in CO_2 emissions is even more outspoken if carbon neutrality of the electricity grid is reached and equals around 70 ton CO_2. For the net present value, the gas price is the deciding factor. If the gas price is relatively low (Belgian price), the NPV of the benchmark is lower than the NPV of concepts 9 and 10. However, for the EU average gas price, the NPV of the benchmark reaches the level of the NPV of concepts 9 and 10. This is an important observation. After all, it means that if the gas price is sufficiently high, concepts 9 and 10 are cost competitive with the benchmark case, while the CO_2 emissions and primary energy use in these concepts are significantly lower.

	Concept 9	Concept 10	Gas benchmark
Primary Energy use	155.6 MWh	142.2 MWh	384 - 406 MWh
CO_2 emissions	13 $tonCO_2$	11.9 $tonCO_2$	72.4 - 76.9 $tonCO_2$
	6.5 $tonCO_2$	5.95 $tonCO_2$	70.8 - 75.3 $tonCO_2$
	0 $tonCO_2$	0 $tonCO_2$	69.26 - 73.8 $tonCO_2$
Net present value	3.24 - 3.50 M€	3.98 - 4.23 M€	2.77 - 3.38 M€

Table 7.2: Comparison between concept 9, concept 10 and the gas benchmark case.

7.2 Second Benchmark Case with Full Electrification

The second benchmark case considers an individual and fully electrified heating approach, i.e. heating and cooling are provided by a heat pump in each dwelling. Moreover, photovoltaic solar panels are placed on the rooftop of each dwelling. Similarly to the first benchmark case, the primary energy use, the CO_2 emissions and the net present value are again calculated.

To calculate these three parameters, the benchmark case first has to be sized and modelled. The behaviour of the solar PV panels is modelled by using a monthly efficiency profile. This profile is shown in Figure 7.5. Note that the efficiency is obtained from a real-world solar PV installation by dividing the monthly electricity generation per unit area by the corresponding monthly average solar irradiation [26].

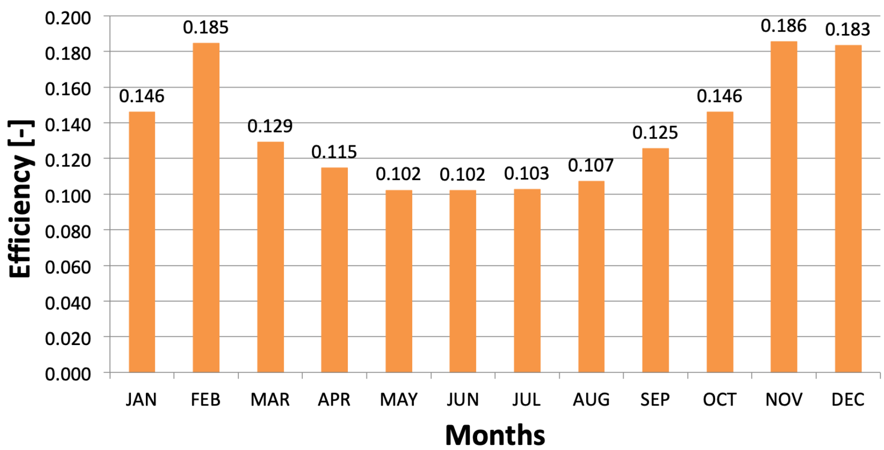


Figure 7.5: Monthly efficiency profile of the photovoltaic solar panels.

The efficiency profile is later used to determine the required area of solar PV panels. First, the specific heat pump that provides heating, cooling and domestic hot water has to be specified. For this, the *Daikin Altherma 3RF* system is used [13]. This is an all integrated air-to-water heat pump with an outdoor unit combined with an indoor unit, which includes an integrated tank for domestic hot water. The specific indoor and outdoor units are *EHVX04S18D6V* and *ERGA04DV*. An overview of the relevant specifications of this system is shown in Table 7.3. Note that the COP values are mentioned for two outdoor temperatures, i.e. -7°C and 7°C. This is done to provide a range for the real electricity use between an optimistic and a pessimistic value. Moreover, the COP values are mentioned for two outlet temperatures of the heat pump, i.e. 35°C and 55°C. These temperatures correspond to the required temperatures for space heating and domestic hot water respectively. For the COP_R, a single average value is used.

DAIKIN Altherma 3RF EHVX04S18D6V + ERGA04DV	
COP (7°C/35°C)	5.1
COP (-7°C/35°C)	3.01
COP (7°C/55°C)	2.65
COP (-7°C/55°C)	1.6
COP_R (18°C/35°C)	5.94
Investment cost	€6726

Table 7.3: Overview of the relevant specifications of the heat pump in the fully electrified benchmark case [13].

Once the specifications of the heat pump are known, the area of the solar PV panels can be determined. In this thesis, the solar PV area is sized such that the electricity generation of the solar PV panels covers the peak electricity demand following from the peak cooling demand. This peak electricity demand for cooling can be calculated by dividing the peak cooling demand by the COP_R value of the heat pump. For a COP_R value of 5.94, the peak electricity demand equals 0.32 kW for each dwelling. Using the efficiency profile of Figure 7.5 and the solar irradiance profile from Chapter 1, the required area is calculated as follows:

$$A_{PV} = \frac{E_{peak,cool}}{\eta_{PV} \cdot G} \tag{7.3}$$

with $E_{peak,cool}$ the peak electricity demand for cooling (0.32 kW), η_{PV} the efficiency of the solar PV panels and G the solar irradiance. The value of G corresponds to the peak solar irradiance on the most critical day, i.e. the day with the highest peak cooling demand and the lowest peak solar irradiance. Hence, the solar PV area is determined such that even on this critical day, the peak cooling demand can be met with solar PV. The result of this calculation is a required solar PV area of 6.98 m^2 for each dwelling. Subsequently, this area can be related to an installed power of the solar PV panels in Watt peak (Wp). To determine the installed power, an area of 1.74m^2 per panel ($A_{PV,panel}$) is assumed, based on modern real-world solar PV panels [34, 55]. Moreover, a maximum power output of 350 Wp is assumed for the solar PV panel, also in correspondence with real-world solar PV panels [57]. The necessary installed power $P_{installation}$ is then calculated as follows:

$$P_{installation} = \frac{A_{PV}}{A_{PV,panel}} \cdot 350Wp = \frac{6.98m^2}{1.74m^2} \cdot 350Wp = 1404Wp \tag{7.4}$$

Hence, the solar PV area of 6.98 m^2 per dwelling corresponds to an installed power of 1404 Wp. Once these values are known, the primary energy use, CO_2 emissions and net present value can be calculated. This is done in the following sections.

7.2.1 Primary Energy Use

The annual primary energy use of this benchmark case is completely determined by its electricity use. As explained above, the area of the solar PV panels is sized based on the cooling demand in the summer and equals 6.98 m^2. Combining the solar PV area and the solar PV efficiency profile with the solar irradiance profile and the hourly demand profiles (see Chapter 1), allows to calculate the residual electricity demand. This residual demand is the electricity use of the heat pump that is not provided by the solar PV panels and has to be delivered by the electricity grid. Once this value is known, the conversion factor of the Belgian electricity grid (see Equation 7.2) can again be used to calculate the corresponding primary energy use. The result of this calculation is shown in Table 7.4. The range in the primary energy use results from applying a pessimistic and optimistic value of the COP. Note that the results in the table correspond to the primary energy use of a single dwelling. For the NPV calculation in Section 7.2.3, the electricity use of the 50 dwellings is used. Recall that for the benchmark case with gas, no estimates were done for future conversion factors of the electricity grid due to its high uncertainty. The same applies here and only the current situation is considered to determine the primary energy use.

Conversion factor	0.5
Residual electricity demand	1561.2 - 5291.7 kWh
Primary energy use	3122.4 - 10583.4 kWh

Table 7.4: Annual primary energy use of one dwelling in the fully electrified benchmark case.

7.2.2 CO_2 Emissions

The CO_2 emissions of the fully electrified benchmark case only result from the residual electricity that is provided by the electricity grid. For the benchmark case with gas condensing boilers, three CO_2-intensity scenarios were considered: the current situation, the halfway point and carbon neutrality (see Section 7.1.2). The same scenarios are again considered here. Using the residual electricity demand of Table 7.4, the corresponding CO_2 emissions of these three scenarios can be calculated.

The result is shown in Table 7.5 for a single dwelling. Again, the range in CO_2 emissions is a result of the use of a pessimistic and optimistic value of the COP.

Scenario	CO_2 intensity	CO_2 emissions
Current situation	0.167 $kgCO_2/kWh_{elec}$	260.7 - 441.9 $kgCO_2$
Halfway point	0.0835 $kgCO_2/kWh_{elec}$	130.4 - 220.9 $kgCO_2$
Carbon neutrality	0.0 $kgCO_2/kWh_{elec}$	0.0 $kgCO_2$

Table 7.5: Annual CO_2 emissions of one dwelling in the fully electrified benchmark case.

7.2.3 Net Present Value

The NPV of this benchmark case consists of the (re)investment and maintenance costs of the heat pumps, the (re)investment and maintenance costs of the solar PV panels and the electricity costs. Table 7.6 gives an overview of the relevant cost data used to calculate the net present value. Note that the investment cost of the solar panels of €98281.6 corresponds to an investment cost of €1965.6 for each dwelling. This is calculated by multiplying the installed power of solar PV per dwelling (1404 Wp) with the cost per unit peak power. In this thesis, a value of 1.4 €/Wp is assumed. It is based on an analysis made on multiple installation offers in 2021 and is in correspondence with the installed power under consideration [57]. Furthermore, note that the electricity price is applied on the residual electricity that is provided by the grid. On the other hand, it is assumed that overproduction of electricity by the solar PV panels is not reimbursed. This assumption is based on the reasoning that moments of overproduction occur at similar times for all dwellings and hence lead to low value of the electricity.

General Cost Data	
Inflation rate R_i	2%
Market interest rate R	5.5%
Interest rate r	3.43%
Solar PV panels	
Initial Investment cost	€98281.6
Reinvestment [5]	20 years
Maintenance cost [5]	1%
Heat pumps	
Initial Investment cost [13]	6726€
Reinvestment [5]	20 years
Maintenance cost [5]	2%
Electricity price	
Belgium (2020, household) [22]	0.2316€/ kWh
Annual price increase	5.87%

Table 7.6: General overview of relevant cost data used to calculate the NPV of the fully electrified benchmark case.

The values in the above table are used to calculate the net present value. Again, the same method as before is used (see Section 4.7). This results in a net present value between €2 892 839 and €4 206 881, depending on the specific COP value that is used. Figure 7.6 shows the composition of this net present value for the optimistic and pessimistic value of the COP. This figure clearly shows that the electricity cost takes up a larger portion of the net present value in case of the pessimistic COP.

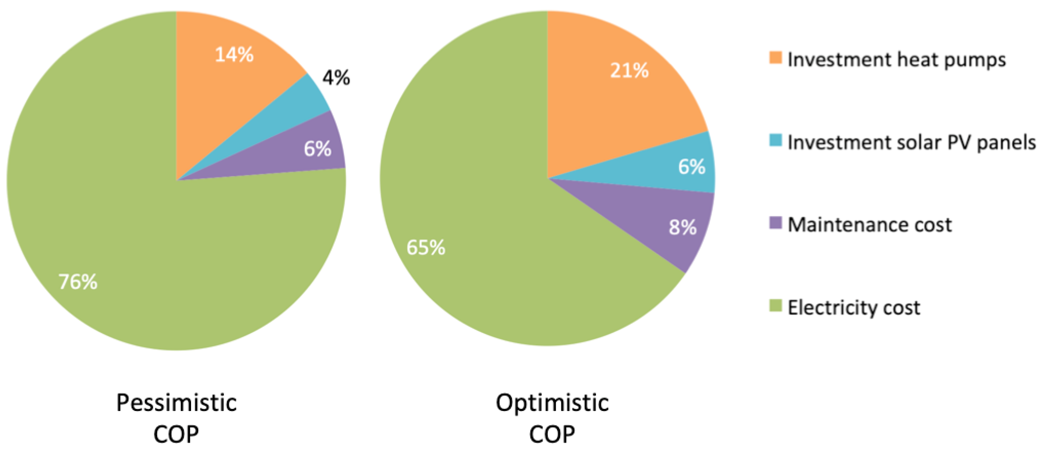


Figure 7.6: Composition of the NPV of the benchmark case for both the pessimistic and the optimistic COP value.

7.2.4 Comparison Between Interesting Concepts and Fully Electrified Benchmark Case

In this section, the results of the fully electrified benchmark case are compared to the most interesting concepts, i.e. concept 9 and concept 10. Table 7.7 gives an overview of the different results of the previous sections. Note that the three CO_2 emissions correspond to the three scenarios. A first observation is that the primary energy use of the benchmark case is larger than the primary energy use of concepts 9 and 10. Only for the optimistic COP value it is at the same level. The same holds for the CO_2 emissions. For all three scenarios, the CO_2 emissions of the benchmark are higher than those of concepts 9 and 10. Again only for the optimistic COP value, the same level is reached. In the carbon neutral scenario, of course the CO_2 emissions for both the benchmark case and the concepts drop to zero. A last observation is that the NPV of concepts 9 and 10 lies within the range of the NPV of the benchmark case. In other words, concepts 9 and 10 are more or less cost competitive with the benchmark. This is an important observation, since this means that lower primary energy use and CO_2 emissions can be achieved by concepts 9 and 10 without a higher cost.

	Concept 9	Concept 10	Fully electrified benchmark
Primary Energy use	155.6 MWh	142.2 MWh	156.1 - 529.2 MWh
CO_2 emissions	13 $tonCO_2$	11.9 $tonCO_2$	13 - 22.1 $tonCO_2$
	6.5 $tonCO_2$	5.95 $tonCO_2$	6.5 - 11 $tonCO_2$
	0 $tonCO_2$	0 $tonCO_2$	0 $tonCO_2$
Net present value	3.24 - 3.50 M€	3.98 - 4.23 M€	2.9 - 4.2 M€

Table 7.7: Comparison between concept 9, concept 10 and the fully electrified benchmark case.

7.3 Conclusion

To summarize, a comparison between the interesting concepts of the DDA (concepts 9 and 10) and two benchmark cases shows that the two concepts can achieve lower CO_2 emissions and primary energy use. The decrease in CO_2 emissions is significant compared to the first benchmark case with gas condensing boilers and reaches up to 70 tons of CO_2 per year. Cost-competitiveness of the concepts is realised in the first case if the gas price is sufficiently high, e.g. equal to the EU average. In the second case, the range of costs is similar for the concepts and the fully electrified benchmark case and they can be considered cost-competitive.
This concludes the third part of the thesis, answering the question '*How do interesting concepts of a district heating system with STES compare to two benchmark cases with individual heating systems?*'. The final part of this thesis mentions the most important conclusions, as well as recommendations for future research.

Part IV

Conclusion and Recommendations

Chapter 8

Conclusion and Recommendations

8.1 Conclusion

In this thesis, concepts of a district heating system with Seasonal Thermal Energy Storage were designed and evaluated. The district that is considered in this thesis consists of 50 low-energy dwellings with a floor area of 150 m^2. The overall lay-out of the system, as well as the components present in the system were introduced at the start of the design process.

The district heating system concepts were designed from a system matrix, which represents the system components and their variables. General rules of thumb were developed by reasoning and basic calculations to limit the number of practical concepts that can be formed. These rules of thumb consist of three excluding rules and seven incompatibility rules. Applying the rules of thumb on the system matrix led to a total of 26 concepts of the district heating system. These concepts were interpreted as 13 practical concepts with versions a and b, with the only difference between both versions the solar collector technology that is used in the central solar field. As a result, the design process led to six concepts with seasonal tank storage, four concepts with low-temperature borefield storage and three concepts with high-temperature borefield storage.

In the second part of the thesis, the different district heating system concepts were sized and compared according to two methods. In the first method, the Simplified Dynamic Assessment, simplifications were made in certain parts of the system. Subsequently, these simplifications were refined in the second method, the Detailed Dynamic Assessment. The results of both methods show significant differences. On the one hand, according to the SDA, the concepts with tank storage are the better option for a district heating system with STES, considering their lower costs and CO_2 emissions. On the other hand, according to the DDA, the concepts with low-temperature borefield storage can be considered more interesting. Therefore, it is concluded that a simplified method is not sufficient to assess the district heating system concepts in this thesis and a detailed method is required to obtain a reliable comparison between the concepts.

The results of the DDA show that concepts 9 and 10 outperform all other concepts from a cost perspective. The net present value of the costs of these concepts over a period of 40 years is at least 2 million euros lower compared to other concepts. Both concepts include a centralised low-temperature borefield for space heating and a local production unit for hot water in each dwelling. Moreover, cooling in both concepts is provided through the district heating network. They differ in the provision of supplementary heating in the system, with concept 9 using a large central heat pump and concept 10 using small decentral heat pumps. Remarkably, the optimal configuration of both systems does not include a central solar field connected to the borefield. This is the result from the sizing method, which optimizes the system towards a minimum NPV of the costs.

In the third part of the thesis, concepts 9 and 10 were compared to two benchmark cases. The first benchmark case considers an individual heating and cooling approach with a gas condensing boiler and an air-to-air heat pump in each dwelling. The comparison shows that concepts 9 and 10 offer significantly lower primary energy use and CO_2 emissions with a potential saving of 60 to 70 tons CO_2 per year. Currently, the benchmark case offers a cheaper solution compared to both concepts due to the low Belgian gas price. If the gas price would increase to the average price in the EU or higher, concepts 9 and 10 would become cost competitive with this benchmark case.

The second benchmark case considers an individual and fully electrified heating approach with a heat pump in each dwelling and solar PV on the rooftop. Compared to this benchmark case, concepts 9 and 10 offer lower primary energy use and CO_2 emissions with a maximal saving of 10 tons CO_2 per year if a pessimistic COP value of the heat pump is considered. For an optimistic COP value, the difference in CO_2 emissions is however negligible. The costs of concept 9 and 10 lie in a similar range compared to the costs of this benchmark case. Hence, concepts 9 and 10 are considered cost competitive with this benchmark case.

8.2 Recommendations for Future Research

All calculations in this thesis were performed for a district with a fixed number of 50 dwellings. Hence, the results and conclusions are valid for this specific case. A first recommendation for further research would be to examine the scalability of the system. Both districts with a smaller and larger amount of dwellings can be considered. For these different districts, the concepts that were designed in Chapter 3 can again be sized and a new comparison can confirm whether concepts 9 and 10 with low-temperature borefield storage still offer the lowest NPV of the costs. An additional refinement could be applied by distinguishing tank and pit storage, in particular for larger systems. Large district heating systems with pit storage have been realised for example in Denmark, and data regarding thermal losses and costs could be obtained from these systems [58]. This could prove interesting since for larger storages ($> 5000\ m^3$), a pit heat storage is considerably cheaper per m^3 compared to a tank storage [31].

A second recommendation would be to study the potential integration of solar PV in the district heating system concepts. Since most concepts use heat pumps, introducing solar PV could lower the electricity that is required from the grid and hence lower the corresponding CO_2 emissions. It could also prove economically interesting if the investment costs of solar PV are compensated by savings in the electricity costs over the study period of 40 years. In a next step, the potential of combined solar photovoltaic-thermal systems (PVT) could be examined as well. These systems convert solar irradiation into electricity and heat simultaneously [33]. These PVT panels could for example be applied in concept 10, using the heat for domestic hot water production and the electricity for the decentral heat pump that provides space heating.

Finally, a third recommendation would be to make a detailed model of the interesting concepts (concepts 9 and 10 in this case), for example in *Modelica*. Compared to the detailed assessment in Chapter 6, further refinements could be applied, taking into account pumps, heat exchangers and other control mechanisms in the system.

Appendices

Appendix A

Calculations for the System Matrix

This chapter explains some of the calculations regarding the excluding and incompatibility rules that are applied to the system matrix in Chapter 3.

A.1 Calculation for the Third Excluding Rule

In this section, the calculation is provided that explains the third excluding rule in Section 3.2.1. This rule excludes the option of a supply temperature of 45°C from the system matrix. This follows from a comparison between the supply temperatures of 45°C and 35°C, which both require supplementary heating with a micro booster heat pump for hot water production.

The comparison between both network supply temperatures is made on a primary energy basis. On the one hand, energy is lost during transport of heat to the dwellings. This heat loss is calculated with Equation 5.3 from Section 5.1.3, leading to the following losses for both networks:

$$Q_{loss,transport}(35/25°C) = 0.1578\frac{W}{mK} \cdot 1600m \cdot (30°C - 10°C) = 5050W \tag{A.1}$$

$$Q_{loss,transport}(45/25°C) = 0.1578\frac{W}{mK} \cdot 1600m \cdot (35°C - 10°C) = 6312W \tag{A.2}$$

Over the entire year, the total heat loss for the network at 35°is 44 MWh, whereas the total heat loss for the network at 45°is 55 MWh.

On the other hand, energy is required for the production of hot water with the micro booster heat pump. Depending on the source temperature of the heat pump, i.e. the supply temperature of the network, the heat pump operates with a different COP. A supply temperature of 35°C allows the heat pump to operate with a COP of 5.8. A supply temperature of 45°C on the other hand allows the heat pump to operate with a COP of 6.2. This COP is calculated from the data available for the micro booster heat pump that is used in this thesis, i.e. *NIBE Booster heat pump MT-MB21* [45]. The COP of the heat pump, along with the annual demand of domestic hot water of 115 MWh (see Chapter 1), allows calculating the annual electricity use of the heat pump:

$$W_{heatpump}(35/25°C) = \frac{115}{5.76} = 19.97MWh \tag{A.3}$$

$$W_{heatpump}(45/25°C) = \frac{115}{6.23} = 18.46MWh \tag{A.4}$$

To compare the electricity use to the heat loss, this electricity use is converted to primary energy using a conversion factor of 0.5 (see Section 7.1.1). Hence, for the supply temperature of 35°C, the electricity use corresponds to around 40 MWh primary energy, whereas for the supply temperature of 45°C, the electricity use corresponds to around 37 MWh primary energy.

In conclusion, the difference in heat loss between the two networks is 11 MWh per year, while the difference in primary energy for the use of the heat pumps is only 3 MWh per year. Hence, it is concluded that the network at 35°C is the better option from a primary energy perspective and the network at 45°C is excluded as an option.

A.2 Calculation for the Third Incompatibility Rule

This section provides the calculation related to the third incompatibility rule (see Section 3.2.2). This rule prescribes that the solar collectors that are part of the local production unit of hot water are always flat plate collectors. The rule is based on the calculation of the NPV of two types of domestic hot water production units. Each type consist of the DHW HP combined with a storage tank and solar collectors on the roof. The difference between the two types, is the technology of the solar collectors. The first type considers flat plate collectors, while the second type considers vacuum tube collectors. The NPV of these two types of installations is calculated for a range of gross collector area. This is done with the method described in Section 4.7 and the cost data in Table 4.7. Figure A.1 shows the result of this calculation. In this figure, it is clearly visible that the NPV of the installation with flat plate collectors has a lower NPV. This is also the case for the area under consideration in this thesis, i.e. 5 m^2 (see Section 4.3.4). Note that the NPV in this figure includes the electricity use of the DHW heat pump, the (re)investment of the heat pump, the (re)investment of the solar collectors and the maintenance costs.

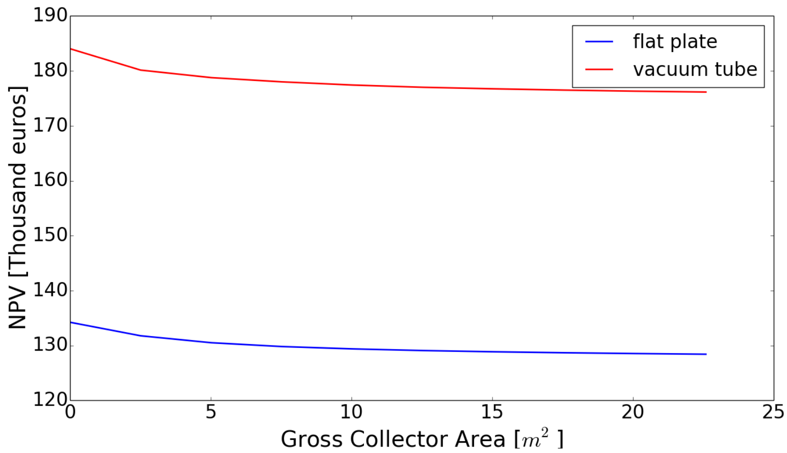


Figure A.1: Evolution of the NPV of the individual domestic hot water production units for a range of gross solar collector area.

Appendix B

Calculation for the Heat Transfer Coefficient

This chapter shows a simplified calculation to check the values for the overall heat transfer coefficient U. As described in the methodology in Section 4.1, the heat loss in the seasonal storage tank is calculated using Equations 4.4 and 4.10 for respectively a fully mixed tank and a perfectly stratified tank. Evaluation of these equations requires the overall heat transfer coefficient U, for which a distinction is made between the top of the tank on the one hand and the bottom and walls of the tank on the other hand. The values for U correspond with 0.1 $\frac{W}{m^2K}$ for the top and 0.3 $\frac{W}{m^2K}$ for the bottom and walls of the tank (see Table 4.1). These values are mentioned by Ochs et al. [47] and a short calculation is performed to check them.

The overall heat transfer coefficient U can be related to a certain insulation thickness x by the following equation:

$$\frac{1}{U} = \frac{x}{\lambda_{insulation}} \tag{B.1}$$

with $\lambda_{insulation}$ the thermal conductivity of the insulation in $\frac{W}{mK}$. This equation is a strongly simplified representation of the heat transfer through the walls of the tank. Indeed, an abstraction is made of the thermal conductivity of the liner material and the concrete walls since their thermal conductivity is considerably higher than the one of the insulation. Therefore, these terms are assumed to have a negligible effect on the overall heat transfer coefficient U.

For the insulation, foam glass with a thermal conductivity $\lambda_{insulation}$ of 0.054 $\frac{W}{mK}$ is used as an example [50]. This leads to the following insulation thicknesses x:

- For U = 0.1 $\frac{W}{m^2K}$ (top) $\longrightarrow$ x = 54 cm
- For U = 0.3 $\frac{W}{m^2K}$ (bottom and walls) $\longrightarrow$ x = 18 cm

Comparing these values to the insulation thicknesses that are used in existing systems with seasonal tank storage, shows that they lie in the range of these practical values. Ochs et al. [48] mentions possible insulation thicknesses of up to 1 m, with the ones used in Eggenstein between 50 and 90 cm. Reuss et al. [56] mentions an insulation thickness of 20 cm for the storage in Attenkirchen. Hence, it is concluded that the values for the overall heat transfer coefficient U are realistic and they are used for sizing the tank storage.

Appendix C

Interpretation of the Results in the SDA

This chapter provides a detailed interpretation of the results of the concepts following the methods in the SDA.

C.1 Results of the Tank Concepts in the SDA

This section gives a detailed interpretation of the results for the tank concepts in Section 5.3.1, that are listed in Table 5.4. These results lead to the following interpretations:

- Firstly, comparing a perfectly stratified tank to a fully mixed tank within each concept, leads to the conclusion that a perfectly stratified tank has a higher storage efficiency. In other words, less heat is lost in a perfectly stratified tank. This is a result of the difference in warm water volume in both tanks. In a perfectly stratified tank only the hot volume at 98°C undergoes heat losses, while in a perfectly mixed tank the complete tank volume undergoes heat losses and hence the heat transferring surface is smaller in a perfectly stratified tank. Of course, the temperature difference between the hot water and the ground is larger in a perfectly stratified tank. Nevertheless, this effect seems to be outweighed by the smaller heat transferring surface.
- Secondly, the comparison of a perfectly stratified tank to a fully mixed tank within each concept, shows that for the odd concepts, the perfectly stratified tank volume is considerably smaller than the fully mixed tank volume. Moreover, the number of solar collectors in these concepts is lower as well. This follows directly from the higher storage efficiency of the perfectly stratified tank. On the other hand, in the even concepts, the perfectly stratified tank volume is slightly larger than the fully mixed tank volume and the number of solar collectors used with the perfectly stratified tank is higher as well. This is due to the operation of the central heat pump in the even concepts. Indeed, Section 4.1.2 mentioned that the central heat pump does not operate with a perfectly

stratified tank, since the tank always provides heat at a temperature above the supply temperature of the network. As a result, for the perfectly stratified, more energy has to be captured by solar collectors, whereas for the fully mixed tank, the central heat pump provides supplementary heating and less solar thermal energy is required in that case.

- Thirdly, comparing the concepts with a central heat pump (2, 4 and 6) to their corresponding concepts without a central heat pump (1, 3 and 5) shows that for a fully mixed tank, the concepts with a central heat pump can use a smaller tank volume and less solar collectors. This is due to the higher storage efficiency in these concepts. For concepts 1, 3 and 5, the lower storage efficiency of the fully mixed tanks results from a restriction on its minimum temperature. Indeed, in these concepts no central heat pump is used and the minimum temperature in the tank has to stay above the supply temperature of the heating network (55 or 35°C). The addition of a heat pump allows the temperature to drop to 10°C and as a result, less heat losses and a higher storage efficiency are obtained. Note that for a perfectly stratified tank, the results of the odd concepts are identical to their even counterpart i.e. concepts 1-2, concepts 3-4 and concepts 5-6. After all the operation of the central heat pump is not required for a perfectly stratified tank, as the the tank always provide heat at a temperature above the network supply temperature (see Section 4.1.2).
- Finally, the electricity use in the concepts can be considered. In concepts 1 and 2, the electricity use originates from the heat pumps providing cooling in each dwelling. In concepts 3 and 4, the electricity use includes the one from the cooling devices and the one from the micro booster heat pumps. In concepts 5 and 6, the total electricity use is the sum of the electricity use of the cooling devices and the electricity use of the DHW heat pumps in each dwelling. For the systems with fully mixed tanks in concepts 2, 4 and 6, each time the electricity use of the large central heat pump is added as well.

C.2 Results of the Low-temperature Borefield Concepts in the SDA

This section provides a detailed interpretation of the results of the low-temperature borefield concepts in Section 5.3.2. The results are listed in Table 5.5 and lead to the following interpretations:

- Firstly, it can be seen that the results for the borefield dimensions are identical in respectively concepts 7 and 9 and concepts 8 and 10. This is due to the borefield only storing energy for space heating, while energy for the production of hot water is either supplied by a central seasonal storage tank combined with a separate heating network (concepts 7 and 8) or supplied by the local hot water production unit (concepts 9 and 10). In concepts 7 and 9, an identical central heat pump is placed in between the borefield storage and the district heating network, resulting in the same amount of energy that is stored in the borefield. In concepts 8 and 10 on the other hand, identical small decentral heat pumps are located in each dwelling, resulting again in the same amount of energy that is stored in the borefield.

- Secondly, for each individual concept, the borefield depth (and length) is larger for a thermal conductivity of the ground of 2.4 $\frac{W}{mK}$ compared to 1.8 $\frac{W}{mK}$, whereas the number of solar collectors is smaller. This is somewhat counter intuitive, since for a larger k_s value, a smaller depth is more expected. However, this is a result of the optimisation criterion used i.e. optimisation towards cost instead of depth. For a k_S of 2.4 $\frac{W}{mK}$ it seems to be more optimal to lower the number of collectors at the cost of a larger borefield, while for a k_s value of 1.8 $\frac{W}{mK}$ it is more optimal to keep the annual imbalance limited with a larger number of solar collectors and a smaller borefield.
- Thirdly, comparing the concepts with small decentral heat pumps (8 and 10) to their corresponding concepts with a large central heat pump (7 and 9), shows that the borefield depth and the number of solar collectors are smaller in the concepts with small decentral heat pumps. This results from the low supply temperature of the district heating network in these concepts, leading to less heat loss compared to concepts 7 and 9. Hence, in concepts 8 and 10, less solar collectors are required to account for the heat loss during transport and less energy has to be stored in the borefield.
- Fourthly, the electricity use of the small decentral heat pumps in concepts 8 and 10 is lower compared to the electricity use of the large central heat pump in concepts 7 and 9. Again, this is due to less heat loss during transport in these concepts. Recall that in the SDA the large central heat pump and the small decentral heat pumps are assumed to have the same fixed COP of 5. Furthermore, the electricity use in concepts 9 and 10 is larger compared to concepts 7 and 8 because of the DHW heat pump that is used in the local hot water production unit in these concepts.
- Finally, concerning the seasonal storage tank that is used in concepts 7 and 8, it is observed that the perfectly stratified tank volume is smaller than the fully mixed tank volume and less solar collectors are required in this system. This is in line with the results of the concepts with a tank storage.

C.3 Results of the High-temperature Borefield Concepts in the SDA

The results in Table 5.6 from Section 5.3.3 lead to the following interpretations:

- Firstly, comparing the results for k_s 2.4 to the results for k_s 1.8 within each concept shows that for all concepts, the borefield length decreases, while the number of solar collectors increases. This is the result from the higher heat loss that occurs in the borefield with a ground thermal conductivity of 2.4 $\frac{W}{mK}$. Considering that in the high-temperature borefield concepts the goal is to increase the ground temperature, a low thermal conductivity of 1.8 $\frac{W}{mK}$ is preferred. This indeed means less heat loss in the borefield and therefore fewer solar collectors are required. 5.2.3).

- Secondly, the electricity use in the concepts is considered. Note that the electricity use mentioned in Table 5.6 is the yearly average electricity use over a study period of 40 years. For the three concepts, it includes the electricity use of the separate cooling devices in each dwelling, as well as the electricity use of the large central heat pump. This central heat pump only operates during the first 20 years, since afterwards the fluid temperature in the borefield always remains above the supply temperature of the network, i.e. 35°C. In concept 12, the electricity use further includes the one for the micro booster heat pumps, whereas in concept 13 the electricity use of the DHW heat pumps is added to the total.

Appendix D

Interpretation of the Results in the DDA

This chapter provides a detailed interpretation of the results of the concepts following the methods in the DDA.

D.1 Results of the Tank Concepts in the DDA

This section gives a detailed interpretation of the results for the tank concepts in Section 6.2.1, that are listed in Table 6.3. These results lead to the following interpretations:

- Firstly, within each concept and for a specific type of solar collectors, it can be observed that the storage efficiency of the perfectly stratified tank is not always higher than the one of the fully mixed tank, as was the case in the SDA (see Appendix C.1). This is due the different sizing method that is applied in the DDA, based on the minimum NPV. In this method, it might be beneficial (from an NPV point of view) to oversize the tank, resulting in the tank not reaching its maximum energy content, i.e. not becoming saturated. This is always the case for the fully mixed tanks in the DDA and therefore, the temperature in these tanks does not reach 98°C. This results in less heat loss and can possibly lead to a higher storage efficiency compared to the perfectly stratified tank.
- Secondly, within each concept and for flat plate solar collectors, the same observations are made as in the SDA regarding the tank volume and the number of solar collectors. Indeed, for the odd concepts (without central heat pump), the perfectly stratified tank volume is smaller and its corresponding number of solar collectors is lower due to the higher storage efficiency compared to the fully mixed tank. For the even concepts, the perfectly stratified tank volume is larger and its corresponding number of solar collectors is higher due to the operation of the central heat pump (see Section 5.3.1). On the other hand, regarding the vacuum tube solar collectors, the perfectly stratified tank volume is always smaller and its corresponding number of solar collectors is

always lower compared to the fully mixed tank. This results from the efficient operation of the vacuum tube collectors with a perfectly stratified tank.

- Thirdly, comparing the concepts with a central heat pump (2,4 and 6) to their corresponding concepts without a central heat pump (1, 3 and 5) shows the same results as in the SDA. That is, for a fully mixed tank, the concepts with a central heat pump can use a smaller tank volume and less solar collectors due to the higher storage efficiency in these concepts. For a perfectly stratified tank, the results of the even concepts are again identical to their odd counterpart. These observations can be explained in the same way is in Appendix C.1.
- Finally, the total electricity use that is shown in the table can again be divided over the different heat pumps in each concept as explained in Appendix C.1.

D.2 Results of the Low-temperature Borefield Concepts in the DDA

This section provides a detailed interpretation of the results of the low-temperature borefield concepts in Section 6.3.2. The results are listed in Table 6.4 and lead to the following interpretations:

- Firstly, what strikes in this table, is that for all concepts, the optimal solution does not include any solar collectors that are connected to the borefield. This results from the sizing method that is applied in the DDA, i.e. an optimisation based on the minimum NPV. It seems that this minimum NPV is obtained by excluding solar collectors in combination with a borefield. This means that the only heat that is injected in the borefield originates from the cooling of the dwellings. Solar collectors are however still used in combination with the central storage tank in concepts 7 and 8, as well as in the local hot water production unit in concepts 9 and 10.
- Secondly, similar as in the SDA, it can be seen that the results for the borefield dimensions are identical in respectively concepts 7 and 9 and concepts 8 and 10 due to the borefield only storing energy for space heating (see Appendix C.2).
- Thirdly, the results show again that the borefield depth is smaller in the concepts with small decentral heat pumps (8 and 10) compared to the the concepts with a large central heat pump (7 and 9). The reason for this was already explained in Appendix C.2 and involves less heat loss during transport in concepts 8 and 10.
- Fourthly, the difference in transportation loss between the concepts with small decentral heat pumps and a large central heat pump explains the difference in the electricity use of their heat pumps as well (see Section Appendix C.2). Furthermore, in concepts 9 and 10, the electricity use is higher compared to concepts 7 and 8 because of the DHW heat pump used in the local hot water production unit in these concepts.

- Finally, concerning the seasonal storage tank that is used in concepts 7 and 8, the same observations are made as in the SDA (see Appendix C.2). Indeed, it is again observed that the perfectly stratified tank volume is smaller than the fully mixed tank volume and less solar collectors are required in this system. Moreover, the results in the DDA show that the system with vacuum tube collectors allows the smallest tank volume and number of solar collectors. This is in line with the results of concept 1 with tank storage, which uses the same temperature limits as in the seasonal storage tank considered here.

D.3 Results of the High-temperature Borefield Concepts in the DDA

The results in Table 6.5 from Section 6.3.3 lead to the following interpretations:

- Firstly, comparing the results for k_s 1.8 to the results for k_s 2.4 within each concept, shows that both the borefield length and the number of solar collectors decrease. As mentioned before, a thermal conductivity of 1.8 $\frac{W}{mK}$ is preferred for the high-temperature borefield concepts, since it corresponds to less heat loss in the borefield.

- Secondly, both the borefield length and the solar collector area are smaller for the systems with vacuum tube collectors compared to the systems with flat plate collectors. However, this does not necessarily mean that vacuum tube collectors are more interesting, since the unit cost of both collector types differ. Therefore, the investment cost of the systems has to be calculated.

- Finally, the electricity use that is mentioned in Table 6.5 is again the annual average electricity use over a study period of 40 years, as was the case in the SDA (see Appendix C.3). It includes the electricity use of the separate cooling devices in each dwelling, as well as the electricity use of the large central heat pump in each concept. In concept 12, the electricity use of the micro booster heat pumps is added as well, whereas in concept 13, the electricity use of the DHW heat pumps is added to the total.

Appendix E

Results for the NPV calculations in the DDA

This chapter lists all the results for the NPV calculations of the different concepts in the DDA. Table E.1 lists the results for the different concepts in the traditional pricing scenario and Table E.2 lists the results in the energy community scenario. The values in both tables are expressed in €.

	Concept 1		**Concept 2**		**Concept 3**		**Concept 4**		**Concept 5**		**Concept 6**	
	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube
50% Stratified	14591039	14145544,5	8347107,5	8533719,5	9622477,5	11184823	8351267	8944774,5	10402902,5	9986249	7204704,5	7815783
75% Stratified	113068920	9576164,75	8364384,25	6949710,75	9069394,25	8467601,5	8530898,5	7444686,75	8997428,75	7732152,5	7495439,25	6744029
	Concept 7		**Concept 8**		**Concept 9**		**Concept 10**		**Concept 11**		**Concept 12**	
$k_s = 1.8$	9683960	8762472,5	10415623,5	9494137,5	3502627	3502627	4227533	4227533	6445633	6918513	6481302	6496559
$k_s = 2.4$	7611377,5	6716947,25	8346419,75	7451991,25	3240203	3240203	3978624	3978624	7824534	7669300	7899882	7876736
	Concept 13											
$k_s = 1.8$	6423246	7148455										
$k_s = 2.4$	7676268	8600981										

Table E.1: NPV results for the different concepts in the traditional pricing scenario.

	Concept 1		**Concept 2**		**Concept 3**		**Concept 4**		**Concept 5**		**Concept 6**	
	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube	Flat plate	Vacuum tube
50% Stratified	14446714,5	14001220	8202783	8389394,5	9318710,5	10881056	8047500	8641007	10092885	9676232	6894687,5	7505765,5
75% Stratified	11162567,25	9431840,5	8220060	6805385,75	8765627,25	8163834,5	8227131,5	7140919,5	8687411,5	7422135,5	7185422,25	6434011,75
	Concept 7		**Concept 8**		**Concept 9**		**Concept 10**		**Concept 11**		**Concept 12**	
$k_s = 1.8$	9683960	8762472,5	10033867,5	9112380	3336934	3336934	3680084	3680084	6301308	6774188	6177535	6192792
$k_s = 2.4$	7611377,5	6716947,25	7964663,75	7070233,5	3074510	3074510	3431175	3431175	7680209	7524975	7596115	7572969
	Concept 13											
$k_s = 1.8$	6113228	6838437										
$k_s = 2.4$	7366251	8290963										

Table E.2: NPV results for the different concepts in the energy community scenario.

Bibliography

[1] M. Ahmadfard and M. Bernier. Modifications to ASHRAE's sizing method for vertical ground heat exchangers. *Science and Technology for the Built Environment 24(7)*, pages 803–817, 2018.

[2] B. Baeten. *Residential heating using heat pumps and hot water storage tanks - Tank sizing to minimize environmental impact in a renewable energy context*. PhD thesis, KU Leuven, 2017.

[3] Y. Bai, Z. Wang, J. Fan, M. Yang, X. Li, L. Chen, G. Yuan, and J. Yang. Numerical and experimental study of an underground water pit for seasonal heat storage. *Renewable Energy*, 150:487–508, 2020.

[4] M. Bernier. Borefield sizing: Theory and applications, 2018. In *Seminar - KTH, Stockholm, Sweden, May 28th 2015*. https://www.kth.se/polopoly_fs/1.574104.1550154719!/Bernier_KTH_final_for_web.pdf (22-2-2021).

[5] Boydens Engineering. Practical values used in the company, based on norms and experience. (Obtained from direct contact in the company Wouter Peere).

[6] L. F. Cabeza, L. Rincón, V. Vilariño, G. Pérez, and A. Castell. Life cycle assessment (LCA) and life cycle energy analysis (LCEA) of buildings and the building sector: A review. *Renewable and Sustainable Energy Reviews*, 29:394–416, 2014.

[7] J. Cho, B. Park, and T. Lim. Experimental and numerical study on the application of low-temperature radiant floor heating system with capillary tube: Thermal performance analysis. *Applied Thermal Engineering*, 163(January), 2019.

[8] M. Cimmino. pygfunction: an open-source toolbox for the evaluation of thermal response factors for geothermal borehole fields. In *Proceedings of eSim 2018, the 10th conference of IBPSA-Canada*, 2018. Montral, QC, Canada, May 9-10.

[9] M. Cimmino. Semi-analytical method for g-function calculation of bore fields with series- and parallel-connected boreholes. *Science and Technology for the Built Environment 25(8)*, pages 1007–1022, 2019.

[10] CREG. Nota over de opvallende evoluties op de Belgische groothandelsmarkten voor elektriciteit en aardgas in 2019. (January 2020). page 4.

[11] A. Dahash, F. Ochs, M. B. Janetti, and W. Streicher. Advances in seasonal thermal energy storage for solar district heating applications: A critical review on large-scale hot-water tank and pit thermal energy storage systems. *Applied Energy*, 239(January):296–315, 2019.

[12] Daikin Europe N.V. Prijslijst 2020-2021: Residentile lucht/lucht warmtepompen. page 5.

[13] Daikin Europe N.V. Verwarmingscatalogus residentile oplossingen 2020. pages 14-15.

[14] Daikin Europe N.V. Verwarmingscatalogus residentile oplossingen 2020. pages 139 and 164.

[15] Daikin Europe N.V. Residential air conditioners: Product catalogue for installers, 2017.

[16] W. D'haeseleer. Solar thermal energy, 2020. Lecture notes, Hernieuwbare Energie (B-KUL-H00S7A), KU Leuven.

[17] Ecovat Nederland B.V. Ecovat productinformatie. `https://www.ecovat.eu/productinformatie/`(20-03-2021).

[18] Engineering Toolbox. Ethylene Glycol Heat-Transfer Fluid. `https://www.engineeringtoolbox.com/ethylene-glycol-d_146.html` (22-10-2021).

[19] European Commission. 2050 long-term strategy. `https://ec.europa.eu/clima/policies/strategies/2050_en`. (12-10-2020).

[20] European Commission. New rules for greener and smarter buildings will increase quality of life for all europeans. `https://ec.europa.eu/info/news/new-rules-greener-and-smarter-buildings-will-increase-quality-life-all-europeans-2019-apr-15_en`. (12-10-2020).

[21] European Environment Agency. Greenhouse gas emission intensity of electricity generation in Europe. `https://www.eea.europa.eu/data-and-maps/indicators/overview-of-the-electricity-production-3/assessment`(19-04-2021).

[22] Eurostat. Electricity prices for household consumers - bi-annual data (from 2007 onwards). `https://ec.europa.eu/eurostat/databrowser/view/nrg_pc_204/default/table?lang=en`(21-04-2021).

[23] Eurostat. Electricity prices for non-household consumers - bi-annual data (from 2007 onwards). `https://ec.europa.eu/eurostat/databrowser/view/nrg_pc_205/default/table?lang=en`(21-04-2021).

[24] Eurostat. Energy consumption in households. `https://ec.europa.eu/eurostat/statistics-explained/index.php/Energy_consumption_in_households`. (12-10-2020).

[25] Eurostat. Gas prices for household consumers - bi-annual data (from 2007 onwards)). https://ec.europa.eu/eurostat/databrowser/view/nrg_pc_202/default/table?lang=en (21-04-2021).

[26] F. Deboosere. Zonnepanelen 2018. https://www.frankdeboosere.be/zonnepanelen/zonnepanelen2018.php (20-04-2021).

[27] E. Guelpa and V. Verda. Thermal energy storage in district heating and cooling systems: A review. *Applied Energy*, 252(June):113474, 2019.

[28] A. Hesaraki, S. Holmberg, and F. Haghighat. Seasonal thermal energy storage with heat pumps and low temperatures in building projects - A comparative review. *Renewable and Sustainable Energy Reviews*, 43:1199–1213, 2015.

[29] J. Hoogmartens and Viessmann Belgium BV. Personal email correspondence on 21-2-2021.

[30] IEA. Seasonal thermal energy storage - Report on state of the art and necessary further R + D. pages 1–48, 2015.

[31] IEA SHC. Task 55 Large Solar Heating & Cooling Systems: Seasonal pit heat storages - Guidelines for materials & construction. 2020. IEA SHC FACT SHEET 55.C-D2.

[32] J. M. Jebamalai, K. Marlein, and J. Laverge. Influence of centralized and distributed thermal energy storage on district heating network design. *Energy*, 202:117689, 2020.

[33] S. S. Joshi and A. S. Dhoble. Photovoltaic -Thermal systems (PVT): Technology review and future trends. *Renewable and Sustainable Energy Reviews*, 92(June 2017):848–882, 2018.

[34] LG Electronics. LG NeON2 Black datasheet. https://www.lg.com/nl/business/download/resources/CT20182061/LG02.0098_NeON2_Black_EN_2701_RZ[20210202_172154].pdf (2021-04-20).

[35] Livios. Vlaming wil niet inboeten op woonoppervlakte. https://www.livios.be/nl/bouwinformatie/woonwijzer/bouwen/bouwbudget-en-verzekeringen/vlaming-wil-niet-inboeten-op-woonoppervlakte/#:~:text=Dit%20laatste%20cijfer%20is%20opvallend,een%20open%20bebouwing%20140%20m%C2%B2 (2020-10-12).

[36] Livios. Wat kost een gascondensatieketel? https://www.livios.be/nl/bouwinformatie/technieken/verwarming-en-koeling/centrale-verwarming/wat-kost-een-gascondensatieketel/?authId=3e675a07-50c5-4e94-91d8-ac9ebc07a4f1{&}referrer=https{%}3A{%}2F{%}2Fwww.google.com{%}2F{&}referer=https{%}3A{%}2F{%}2Fwww.google.com{%}2F (2021-03-30).

[37] LOGSTOR A/S. Design with twinpipes version 2016.08. https://www.logstor.com/media/5355/design-with-twinpipes-201608.pdf(15-03-2021).

[38] LOGSTOR A/S. Product catalogue, district energy version 2020.09 (07-ib). https://www.logstor.com/media/6995/product-catalogue-uk-202104.pdf(30-10-2020).

[39] M. Lumbreras Mugaguren, R. Garay Martínez, V. Sánchez Zabala, K. Korsholmstergaard, and M. Caramaschi. Triple function substation and high-efficiency micro booster heat pump for Ultra Low Temperature District Heating. *IOP Conference Series: Materials Science and Engineering*, (5), 2019.

[40] H. Lund, S. Werner, R. Wiltshire, S. Svendsen, J. E. Thorsen, F. Hvelplund, and B. V. Mathiesen. 4th Generation District Heating (4GDH). Integrating smart thermal grids into future sustainable energy systems. *Energy*, 68:1–11, 2014.

[41] R. Lund, D. S. Østergaard, X. Yang, and B. V. Mathiesen. Comparison of low-temperature district heating concepts in a long-term energy system perspective. *International Journal of Sustainable Energy Planning and Management*, 12:5–18, 2017.

[42] X. Masip, E. Navarro-Peris, and J. M. Corberán. Influence of the Thermal Energy Storage Strategy on the Performance of a Booster Heat Pump for Domestic Hot Water Production System Based on the Use of Low Temperature Heat Source. *Energies*, (24):6576, 2020.

[43] L. Mesquita and Drake Landing Solar Community. Iea shc task 55, d-d3: Identification and preparation of best practice examples. Document received via e-mail contact.

[44] A. Neave. *Heat pumps and their applications*. Reed Educational and Professional Publishing Ltd, second edition edition, 2002.

[45] NIBE Energietechniek B.V. Handleiding nibe mt-mb21. page 11.

[46] NIBE Energietechniek B.V. Nibe prijscatalogus 2020. page 24.

[47] F. Ochs, A. Dahash, A. Tosatto, and M. Bianchi Janetti. Techno-economic planning and construction of cost-effective large-scale hot water thermal energy storage for Renewable District heating systems. *Renewable Energy*, 150:1165–1177, 2020.

[48] F. Ochs, J. Nußbicker, R. Marx, H. Koch, W. Heidemann, and H. Müller-Steinhagen. Solar assisted district heating system with seasonal thermal energy storage in Eggenstein-Leopoldshafen. *Conference Proceedings*, 2008.

[49] ODE-Vlaanderen. Warmte uit zonlicht, 2013. https://www.olino.org/wp-content/uploads/2008/articles/potentie_zonne-energie_nederland_brochure_zonlicht.pdf(30-10-2020).

[50] Owens Corning Foamglas. Foamglas one insulation. https://www.foamglas.com/en-us/applications-and-solutions/storage-tanks,-spheres,-vessels/water-containment/fresh-water-tanks/fresh-water-tanks(08-03-2021).

[51] T. Pauschinger, T. Schmidt, P. Alex Soerensen, D. Aart Snijders, R. Djebbar, R. Boulter, and C. Jeff Thornton. Integrated Cost-effective Large-scale Thermal Energy Storage for Smart District Heating and Cooling - Design Aspects for Large-Scale Aquifer and Pit Thermal Energy Storage for District Heating and Cooling. *International Energy Agency Technology Collaboration Programme on District Heating and Cooling including Combined Heat and Power*, 2018(September), 2018.

[52] W. Peere. Methode voor economische optimalisatie van geothermische verwarmings- en koelsystemen. Master's thesis, KU Leuven, 2020.

[53] W. Peere, D. Picard, I. Cupeiro Figueroa, W. Boydens, and L. Helsen. Validated combined first and last year borefield sizing methodology. *In Proceedings of International Building Simulation Conference 2021 (2021)*. Brugge (Belgium), 1-3 September 2021. Abstract accepted.

[54] P. Pinel, C. A. Cruickshank, I. Beausoleil-Morrison, and A. Wills. A review of available methods for seasonal storage of solar thermal energy in residential applications. *Renewable and Sustainable Energy Reviews*, 15(7):3341–3359, 2011.

[55] REC GROUO. REC ALPHA SERIES product specifications). recgroup.com/sites/default/files/documents/ds_rec_alpha_series_en.pdf?t=1621420501 (2021-04-20).

[56] M. Reuss, W. Beuth, M. Schmidt, and W. Schoelkopf. Solar district heating with seasonal storage in Attenkirchen. *Proceedings of the IEA Conference ECOSTOCK, Richard Stockton College Pomona, New Jersey, USA*, (February), 2006.

[57] S. Goessens. Zonnepanelen prijzen in 2021. https://zonnepanelenenergie.be/prijzenp (2021-04-20).

[58] A. Sørensen and T. Schmidt. Design and Construction of Large Scale Heat Storages for District Heating in Denmark. *International Conference on Energy Storage*, (April), 2018.

[59] S. Suman, M. K. Khan, and M. Pathak. Performance enhancement of solar collectors - A review. *Renewable and Sustainable Energy Reviews*, pages 192–210, 2015.

[60] Y. Tian and C. Y. Zhao. A review of solar collectors and thermal energy storage in solar thermal applications. *Applied Energy*, 104:538–553, 2013.

[61] B. van der Heijde. *Optimal integration of thermal energy storage and conversion in fourth generation thermal networks.* PhD thesis, KU Leuven, 2019.

[62] A. Vandermeulen and L. Vandeplas. Floor heating in residential buildings: optimisation towards different objectives in a smart grid context. Master's thesis, KU Leuven, 2016.

[63] Viessmann Belgium BV. Planningshandleiding - vitocal 200-g - 2020.

[64] Viessmann Belgium BV. Planningshandleiding - vitocal 300-g pro - 2018.

[65] Viessmann Belgium BV. Planningshandleiding - vitocal 350-ht pro - 2017.

[66] Viessmann Belgium BV. Technical guide - Solar thermal systems, 2008.

[67] Viessmann Belgium BV. Vitosol Planningsaanwijzing, 2016.

[68] Viessmann Belgium BV. Vitosol 200-FM/-F Datenblatt, 2017.

[69] Viessmann Belgium BV. Technology brochure - Solar thermal systems Vitosol, 2018.

[70] P. Wallentén. Steady-state heat loss from insulated pipes, 1991. Byggnadsfysik LTH, Lunds Tekniska Högskola.

[71] WTCB i.s.m. A.G.T. n.v. en de Smart Geotherm werkgroep. Code van goede praktijk: Het ontwerp, de uitvoering en het beheer van kwo systemen in vlaanderen. `https://www.smartgeotherm.be/documents/2017/02/code-goede-praktijk-kwo.pdf/`(25-05-2021).

[72] M. Zhao, Z. L. Gu, W. B. Kang, X. Liu, L. Y. Zhang, L. W. Jin, and Q. L. Zhang. Experimental investigation and feasibility analysis on a capillary radiant heating system based on solar and air source heat pump dual heat source. *Applied Energy*, 185:2094–2105, 2017.

[73] P. Zijlema. Berekening van de standaard co_2-emissiefactor aardgas t.b.v. nationale monitoring 2020 en emissiehandel 2020. `https://www.rvo.nl/sites/default/files/2020/05/vaststelling-standaard-co2-ef-aardgas-jaar-nationale-monitoring-2020-en-ets-2020-def_0.pdf`(06-04-2021).