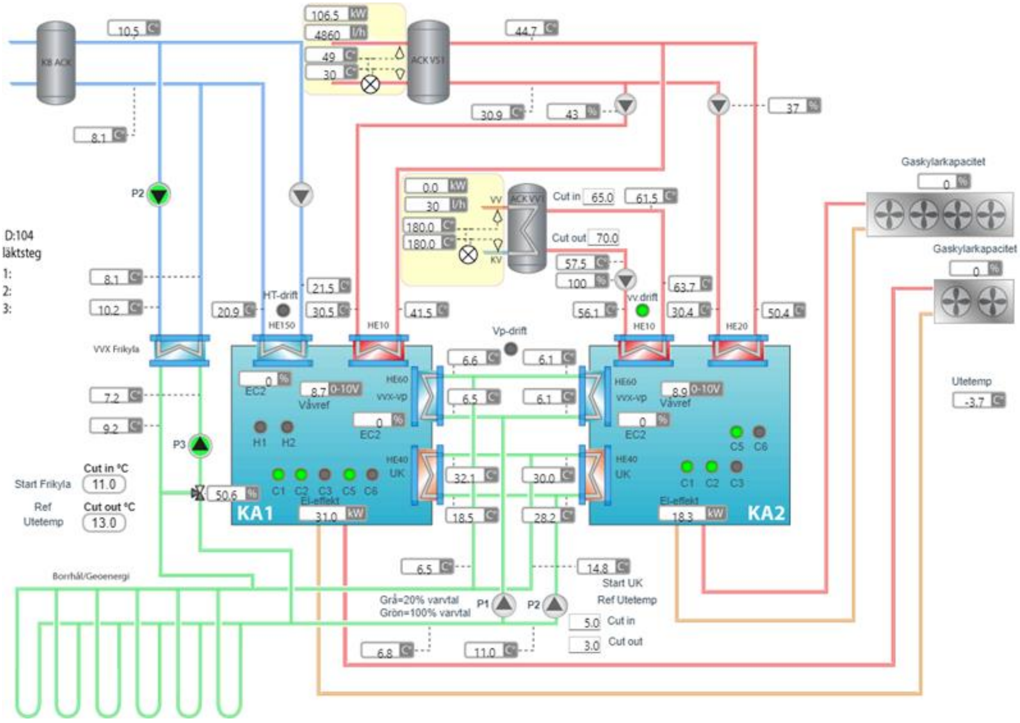
Practical and theoretical evaluation of CO2 refrigeration system with geothermal integration



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Jörgen Rogstam, Fabio Giunta, Juris Pomerancevs, Carlos Mateu-Royo, Simon Bolteau, Cajus Grönqvist

EKA - Energi & Kylanalys AB
Prästgårdsgränd 4, Älvsjö
www.ekanalys.se

Sammanfattning

Syftet med denna undersökning var att utvärdera funktionen vid befintliga installationer med CO2-kylsystem i kombination med geolagerlösningar. Vidare så har ofta befintliga systems instrumentering i praktiken begränsat möjligheten till verifiering av nyckelfunktionerna, varför en guide till rätt instrumentering har tagits fram. Genom att studera verkliga installationer och jämföra med modellering så har slutsatser och råd för optimal dimensionering tagits fram. Som grundläggande verktyg för lönsamhetsanalysen har LCC använts. I analysen har geolagerlösningar eller dessa i kombination med olika andra värme- och kyllösningar jämförts.

Som grund för utredningen har tre examensarbeten genomförts och denna rapport är baserad på en sammanställning av dessa tre vilka utfördes under åren 2017 till 2019. Mateu-Royo genomförde 2017 en analys av tio koldioxidaggregat installerade i livsmedelsbutiker vilken visade att alla saknade tillräckligt med mätpunkter för att bedöma systemens funktion fullt ut. Av denna anledning skapades en guide för instrumentering som också presenteras i denna rapport. Under åren mellan 2018 och 2019 fick detta projekt tillgång till flera ny installationer i primärt butiker. Bland dessa valdes en ut för en fördjupad analys. Utöver detta gjordes två fallstudier för att utvärdera den ekonomiska nyttan av geotermisk funktion kopplade till isbanor med CO2-kylsystem.

Driftstrategier för värmeåtervinning har undersökts på djupet. Den teoretiskt bästa reglerstrategin jämfördes med den som används i ett verkligt system och skillnaderna i den årliga energianvändningen utvärderades genom modellering. Resultaten visar att det finns praktiska alternativ till den bästa teoretiska strategin; värme kan tas från marken medan lågtemperaturvärme samtidigt avges via gaskylaren. En sådan strategi har intressanta tekniska fördelar, såsom att mer högvärdig värme produceras, med en försumbar ökning av (el-)energianvändningen.

En viktig skillnad mellan de två applikationer som primärt studerats i detta projekt är temperaturnivån vid vilken värme används. I många butiker är efterfrågan på tappvarmvatten marginell, medan den i t ex isbanor är hög. Det är utmanande att optimalt reglera värmeavgivningen och samtidigt respektera en hög önskad framledningstemperatur. Därför har en mer komplex värmeåtervinningsalgoritm utvecklats, testats och jämförts med den reglering som för närvarande används i en av fallstudierna. Beräkningarna visar att algoritmen leder till en minskning med 12% av den årliga energianvändningen för det bästa scenariot.

Fördelarna med integration av geolagerlösningar utvärderades genom att använda BIN-timmar-metoden och aktuella svenska energipriser. För en typisk ishall i Sverige tyder resultaten på att om ett CO2-kylsystem uppgraderas/kompletteras med en geolagerfunktion kan den spara mellan 1,7 och 6,8% av den årliga energianvändningen (el). Återbetalningstiden för denna geotermiska lösning skulle vara ca 16,4 år. Det ska noteras att detta bara gäller om anläggningens egna värmebehov ska täckas. I det fall man kan exportera värme så kan återbetalningstiden bli betydligt kortare.

När det gäller livsmedelsbutiker så har det visats att underkylning av kylsystemet genom en geolagerfunktion kan spara mellan 5 och 10% av den totala energianvändningen – givet ett Stockholmsklimat. Detta representerar större delen av den totala möjliga besparingen, eftersom den geolagerbaserade värmepumpsfunktionen endast används på vintern, för att hantera topparna i värmebehovet. Att bara förlita sig på besparingar från underkylning visade sig dock inte vara tillräckligt för att ekonomiskt motivera en geolagerlösning i butiker. Av detta skäl undersöktes flera scenarier för att kunna producera heltäckande resultat och känslighetsanalyser som syftar till att identifiera de fall där denna systemlösning är mest ekonomiskt motiverad. Bäst lönsamhet visar sig inte oväntat butiker med en högre värmeåtervinningsgrad,

HRR (Heat Recovery Ratio), än genomsnittet ha. Butiker som även levererar värme till exempelvis angränsande byggnader och på så sätt får en genomsnittlig HRR på minst 70% under vinterperioden, uppfyller normalt sett kraven på god lönsamhet.

Slutligen presenterar och sammanfattar rapporten några tekno-ekonomiska designtips som ska beaktas vid konstruktion och dimensionering av koldioxidkylsystem med integrerade geolager.

Rapportens huvudkällor

Som inledningsvis nämndes är denna rapport en sammanfattning av tre olika examensarbeten varför detta dokument utelämnat en del av de bakomliggande detaljerna. Den intresserade läsaren rekommenderas studera den enskilda rapporten som bäst täcker in hens intresseområde. Dessa rapporter listas nedan medan annan använd litteratur återfinns i referensavsnittet sist i rapporten.

1. CARLOS MATEU ROYO, **Field measurements and modelling analysis of CO2 refrigeration systems with integrated geothermal storage**
    - Master of Science Thesis, KTH School of Industrial Engineering and Management, Energy Technology EGI-2017-0047-MSC, Division of Applied Thermodynamics and Refrigeration, SE-100 44 STOCKHOLM
2. JURIS POMERANCEVS, **Geothermal function integration in ice rinks with CO2 refrigeration system**
    - Master of Science Thesis KTH School of Industrial Engineering and Management Energy Technology EGI-2019-TRITA-ITM-EX 2019:723 Division of Applied Thermodynamics and Refrigeration (ETT) SE-100 44 STOCKHOLM
3. FABIO GIUNTA, **Techno-economic assessment of CO2 refrigeration systems with geothermal integration, a field measurements and modelling analysis***
    - Master of Science Thesis KTH School of Industrial Engineering and Management Energy Technology EGI-2020-TRITA-ITM-EX 2020:3 Division of Applied Thermodynamics and Refrigeration (ETT) SE-100 44 STOCKHOLM

*Fabio Giuntas examensarbete vann 3:e pris i den europeiska organisationen ”eurammon’s” tävling om årets bästa examensarbete. Det ska tilläggas att priset ficks i konkurrens med arbeten på doktorsnivå. Läs mer: www.eurammon.com**.

***”eurammon is an association of companies, institutions and individuals with one goal: to encourage a sustainable approach in refrigeration engineering. eurammon has therefore been advocating the use of natural refrigerants since its foundation in 1996. The initiative sees its mission in providing a platform for information and knowledge sharing – be it for scientists and researchers, politicians, as well as the public at large.*”

Summary

This report is based on the summary of 3 master thesis performed in the years between 2017 and 2019. Royo, 2017 conducted a preliminary analysis of ten CO2 refrigeration units installed in supermarkets which showed a systematic lack of measurement points, necessary for an energy efficiency assessment. For this reason, an instrumentation guideline was created and presented in this report. In the years between 2018 and 2019, this project gained access to several newly built installations in supermarkets. Amongst these, one supermarket was selected for an in-depth analysis. Additionally, two case studies were utilized to evaluate the economic benefit of the geothermal function in ice rinks utilizing CO2 refrigeration ice rinks.

Heat Recovery operational strategies have been deeply investigated. The best theoretical operational strategy was compared to one that is used in real controllers and the differences in the annual energy usage were assessed through modelling. The results show that an alternative to the best theoretical operational strategy exists; heat can be extracted from the ground while low-temperature heat is rejected by the gas cooler. Such an alternative strategy has important technical advantages with a negligible increment of the energy usage.

An important difference between the two applications studied in this project is the temperature level at which heat is utilized. In many supermarkets, the demand for tap water is marginal, while in ice rinks high-temperature (70°C) heat is an important part of the demand. For these reasons, controlling the heat discharge while respecting the required forward temperature and high temperature demand is more challenging. This is why a more complex heat recovery control algorithm was developed, tested and compared with the control currently utilized in one of the case studies. The calculations show that the algorithm lead to a reduction of annual energy usage of 12% for the best scenario.

The benefits of geothermal integration were evaluated applying the BIN hours method and utilizing typical Swedish prices. For a typical ice rink in Sweden the results suggest that if a CO2 refrigeration system is upgraded with a geothermal function, it can save between 1.7 and 6.8% of the annual energy consumption. In the best case, this study suggests that the geothermal function would pay back the additional investment cost in 16.4 years.

In the case of supermarkets, it was calculated that subcooling through a geothermal storage can save between 5 and 10% of the total power consumption, in the Stockholm's climate. This represents almost the total amount of savings, since in winter the geothermal heat pump function is used only for peaks of heating demand. However, relying only on savings from subcooling was found to be not enough to economically justify a geothermal storage in a supermarket. For this reason, several scenarios were investigated to produce parametric curves and to perform a sensitivity analysis aimed at identifying the cases where this system solution is economically justified. These are supermarkets with a Heat Recovery Ratio (HRR) higher than the average. For examples, supermarkets supplying heat to the neighbouring buildings (considering the Stockholm's climate, systems with an average HRR of at least 70%, over the winter period).

Finally, some techno-economic considerations to be taken into account, when designing CO2 refrigeration systems integrated with geothermal storages, are presented and summarized in the conclusions.

Acknowledgement

Vi från projektet vill härmed tacka de som bidragit genom arbete och stöd så detta resultat kunnat dokumenterats. De som gjort huvuddelen av forskningsarbetet är Carlos Mateu-Royo (KTH), Fabio Giunta KTH/EKA) och Juris Pomerancevs (KTH/EKA).

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Ett extra stort tack riktas slutligen till huvudfinansiären Kylbranschens Samarbetsstiftelse, KYS. Utan ert bidrag skulle detta projekt aldrig kunna genomföras!

Stor tack till alla som bidragit!

Jörgen Rogstam

Projektledare, EKA

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1 Background information

During the last twenty-five years, the utilization of CO2 as a refrigerant has attracted the attention of academics and industrial companies. In 2016, there were roughly 11000 “CO2-only” installations worldwide and 8700 of them were in Europe (Shecco, 2016). Two years later, in 2018, the number of installations worldwide counted for roughly 18000 CO2-based refrigeration systems (Shecco, 2018). The market is expected to grow exponentially reaching 81000 units by 2030 (Shecco, 2016). This is not only due to the low Global Warming Potential (GWP) of CO2 but also to the good performance of this technology when heat is recovered from the refrigeration system (Sawalha, 2008).

The main CO2 refrigeration applications in Sweden are supermarkets and ice rinks. Indeed, CO2 trans-critical refrigeration system is considered the best solution to satisfy the heating and cooling demand in supermarkets located in cold climates (Karampour & Sawalha, 2017). Similarly, these refrigeration units have excellent performance if installed in winter sport facilities with high heating demand, such as ice rinks. Indeed, these have many functions that require heat, in particular, there is a great demand for relatively high-temperature heat (above 40°C).

It is worth to mention that there are several differences between the two applications. Refrigeration units for retailers are mass produced and more standardized compared with industrial applications. By contrast, CO2 systems for ice rinks are customized units with a larger variety of options. There are some obvious differences such as size and evaporation temperature, however, from an energy perspective, the most important difference is the heating demand. In a supermarket, comfort space heating is the main (often the only) heating demand to be satisfied, in fact, the tap water consumption is often minimal. On the other hand, in ice rinks there are many different heating demands ranging from 70 to 15°C. This has a huge impact on the heat recovery control strategy which in the case of supermarkets without geothermal integration is more straightforward and well documented while it is very complex and less investigated for ice rink applications.

Despite the good performance during winter, in warm periods (e.g. summer) the efficiency dramatically decreases due to high temperatures at the gas cooler outlet. For this reason, the system greatly benefits from sub-cooling the refrigerant (Sawalha, 2012)(Karampour et al., 2018). The integration of geothermal borehole heat exchangers (BHEs) enables the system to sub-cool the refrigerant. Moreover, in winter when there is a high heating demand and low cooling power absorbed, BHEs can be used as an additional source of heat. In other words, the boreholes field operates as geothermal seasonal storage.

To study the economics of this solution a key parameter is the heat balance in the ground. In other words, the amount of heat dumped in summer could be different from the amount of energy extracted in winter and vice versa. This could create a heat surplus/deficit which would lead to a constant increase/decrease of the geothermal field temperature over the years. To compensate for the temperature change, the size of the field needs to be increased, driving up the cost. The geothermal optimizations in this study aims at investigating the relationship between size/cost and efficiency improvements.

Regarding the geothermal storage, nowadays, the most common configuration consists of vertical or inclined ground heat exchangers (GHEs) connected in parallel. These are usually referred as boreholes. A borehole field is made up of single or multiple drills, with a diameter ranging from 0.075 to 0.180 m and with a depth between 40 and 200 m. The most spread types of heat exchanger are single and double U-shape plastic pipes. Finally, in Sweden, the heat carrier fluid is usually (70-75% of the heat pumps) an aqueous solution of 20-25 wt-% ethyl alcohol, with -15°C freezing point (Monzó, 2018).

As confirmed by (Gullo et al., 2018) and (Karampour & Sawalha, 2017), there is a shortage of studies dealing with field measurement analysis of CO2 refrigeration systems. In particular, such a scarcity is more pronounced regarding the integration of geothermal storages. Therefore, this study aims at filling partially this gap investigating the economical benefit of integrating CO2 refrigeration systems with a geothermal storage through field measurements and modelling.

1.1 Instrumentation guideline

(Royo, 2017) studied ten ground-coupled installations, concluding that not enough measurement points were available for a comprehensive evaluation, in those systems. Therefore, an instrumentation guideline was created in order to highlight the necessary measurement points for an energy efficiency analysis of this type of systems. Figure 1.1 and Table 1 give information about the positioning and the type of the necessary measurement equipment.

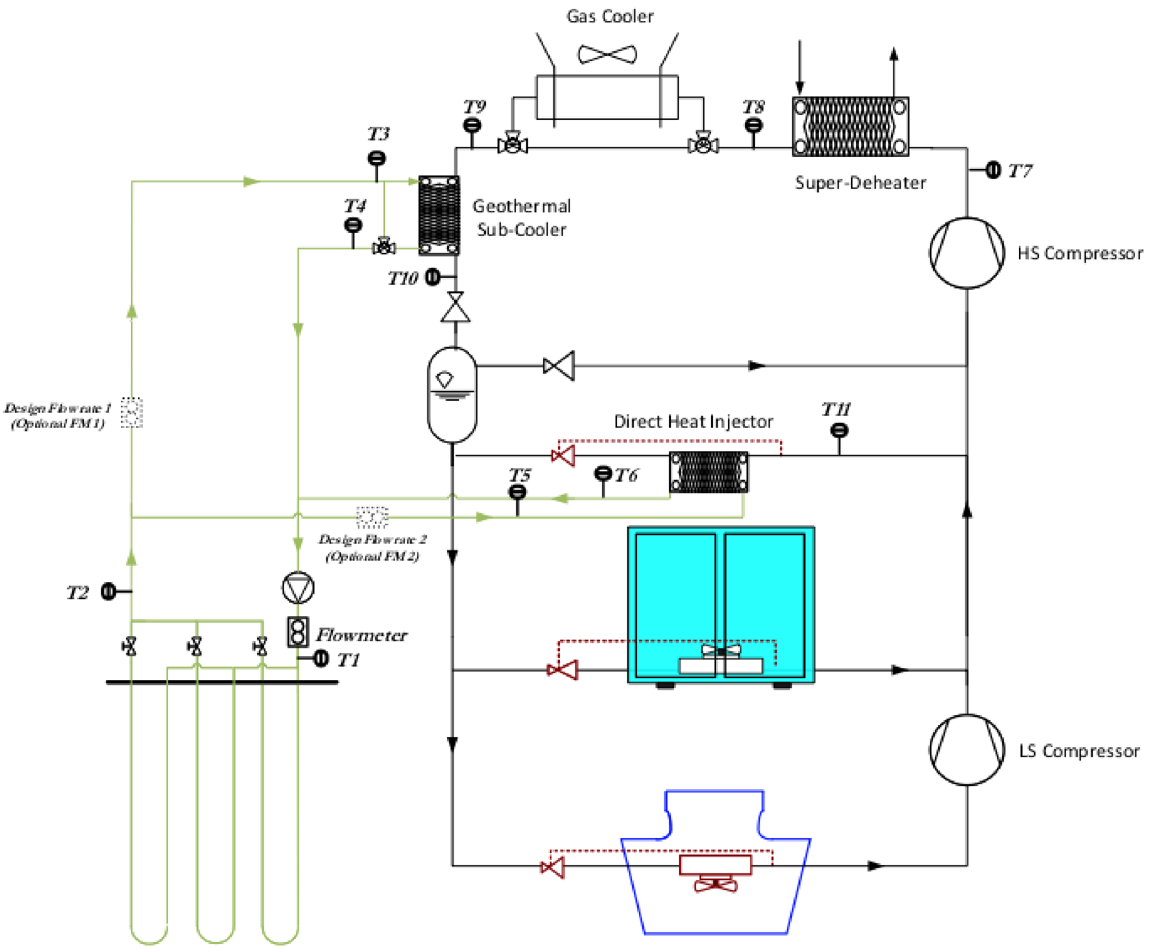


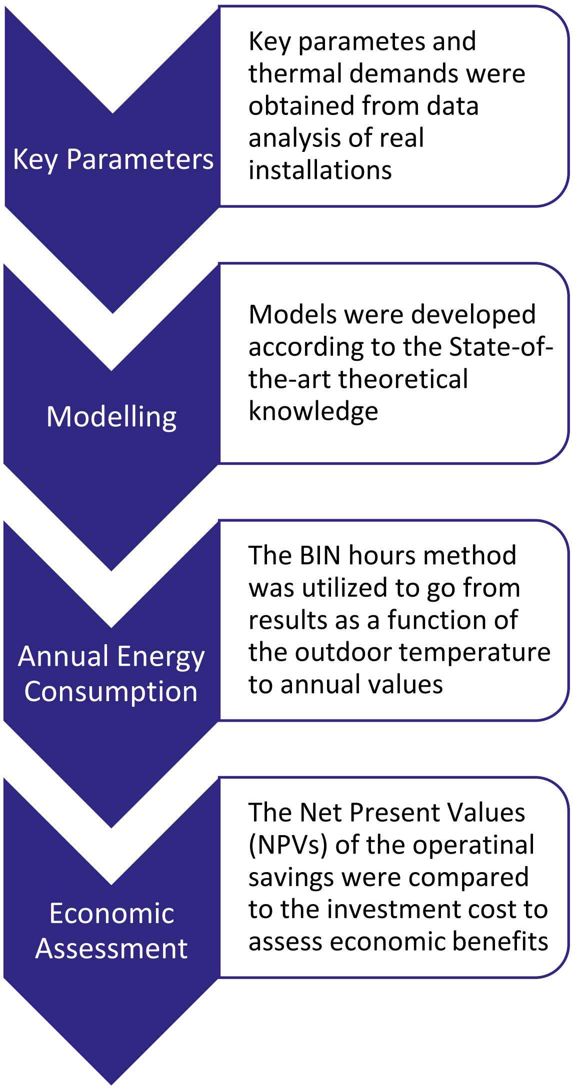
Figure 1.1 Scheme showing the positioning of the necessary measurement equipment for a complete energy efficiency analysis (Royo, 2017)

Table 1 Measurements for CO2 Refrigeration systems integrated with a geothermal storage (Royo, 2017)

	ID	DESCRIPTION
Geothermal Installation	**T1**	Inlet Ground Temperature
	T2	Outlet Ground Temperature
	T3	Inlet Geothermal Sub-cooler Temperature
	T4	Outlet Geothermal Sub-cooler Temperature
	T5	Inlet Geothermal Heat Extractor Temperature
	T6	Outlet Geothermal Heat Extractor Temperature
	FM	Flowmeter Geothermal Fluid
	FM1	Flowmeter Geothermal Fluid or Design Flow Rate
	FM2	Flowmeter Geothermal Fluid or Design Flow Rate
CO2 Refrigeration System	**T7**	Discharge Temperature HS Compressor
	T8	Inlet Gas cooler Temperature
	T9	Outlet Gas cooler Temperature
	T10	Outlet Geothermal Sub-cooler Temperature
	T11	Outlet Geothermal Heat Extractor Temperature
	EHS/ELS	Separate HS and LS Compressors Power Consumptions

2 Methodology

In this work a combination of modelling and field measurements was utilized. Firstly, theoretical models were created based on state-of-the-art knowledge of these systems. Then, data coming from field measurements of 3 different installations were utilized to validate the models, to back-up the assumptions and to generate the necessary inputs. The latter refers mainly to the thermal loads in ice rinks and supermarkets, as well as key control parameters. The BIN hours method was then applied to calculate the expected yearly energy consumption. Finally, the software Energy Earth Designer (EED) was used for simulating the geothermal boreholes.



2.1 Boreholes field sizing and optimization

Regarding the geothermal field, the load on the ground affects not only the size and configuration but also the outlet-temperature from the boreholes. The latter is directly related to the inlet temperature of the high-pressure expansion valve which, in turn, affects the heat injected into the ground. This correlation must be solved through an iterative process. In this work such a calculation was performed iterating between the geothermal simulation software EED and the refrigeration system model based either on a VBA or Python code.

3 Ice rinks

The Term "ice rink" gives just the minimum information about the specific installation. There is a variety of possible modifications. To start with, ice rinks can be either outdoor or indoor, and within the scope of this study only indoor ice rinks are considered. In Sweden there are more than 360 ice arenas with the most prevalent type being "Spectator arena C" with 500 to 1000 spectator seats – the size analyzed in this work.

According to an extensive statistical study in Sweden, where an audit of more than 200 ice rinks was conducted key conclusion is that more than 90% of the energy bill includes 5 systems – "the big five". The energy breakdown is illustrated in Figure 3.1, which suggests that refrigeration contributes to the largest portion with around 43% and heating demands come next with around 26%. The other functions are estimated with up to 10% share each (Jörgen Rogstam et al., 2014).

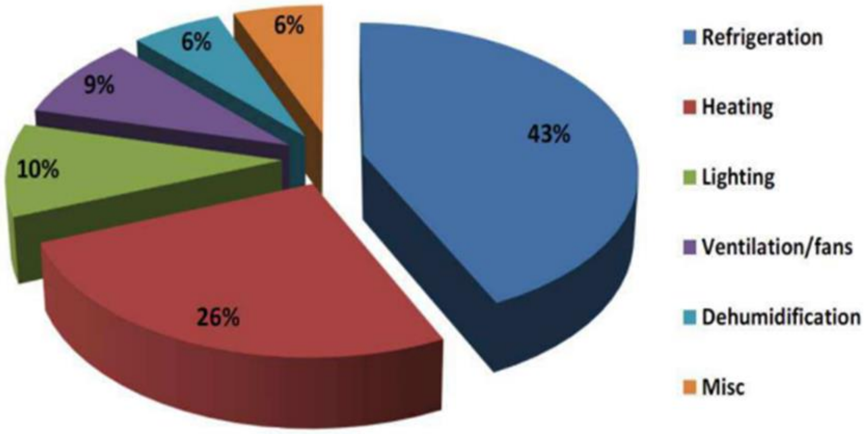


Figure 3.1. Main energy uses in a typical ice rink in Sweden.

When it comes to refrigeration system solution there are a number of options available. Looking into Swedish perspective, a study in 2011 concluded that the most prevalent type is indirect or partly indirect (97%) with ammonia as the primary refrigerant in about 85% of installations, the rest being HFC's (Makhnatch, 2011). The situation today is different due to the rapid penetration of CO2 technology in Swedish ice rinks, many renovations from ammonia to CO2 have been carried out since 2014. Nowadays about 10% of installations work with CO2 as the primary refrigerant (R744.com, 2018). In this project both a direct (case study 1) and indirect system (case study 2) have been investigated.

Cooling and heating demands can vary significantly from ice rink to ice rink which in turn affects the economic attractiveness of the geothermal function. This is why the first step of this section was a quantitative analysis of the thermal demands in an ice rink.

3.1 Case Study 1 - Thermal Demands in Ice Rinks

Without a doubt refrigeration system is the cornerstone of the ice rink, providing continuous cooling to maintain ice surface at a good quality.

But at the same time in an indoor ice rink there are several functions that utilise heating energy. To supply these two thermal demands accounts to about 70% of the total energy expenditures according to Swedish statistics (Jörgen Rogstam et al., 2014). Ice rink functions that need heat for their process are – hot water, dehumidification, radiators, resurfacer, ventilation, freeze protection and melting pit.

In order to estimate heating and cooling demands a field installation in operation is first analysed in this study. The ice rink under focus is from a small town Gimo, which since 2014 has a direct CO2 system with a single stage heat recovery that is designed to cover 100% of the heating demand of the ice rink and is even recently upgraded with a heat export function to reduce the energy bill of the neighbouring swimming pool. Out of the ice rinks with data available online, Gimo offers the most complete set of measurements to use for the analysis.

Figure 3.2. Pictures of Gimo ice rink.

3.1.1 Heating demand

The heating system in Gimo ice rink is designed to completely cover the heating demand, utilizing the heat recovered from the CO2 system. In order to maximize the potential the so-called “waterfall” principle is applied. The different functions are arranged in sequence, in order to achieve a forward-return temperature difference as high as possible and a low return temperature. The layout is shown in Figure 3.3.

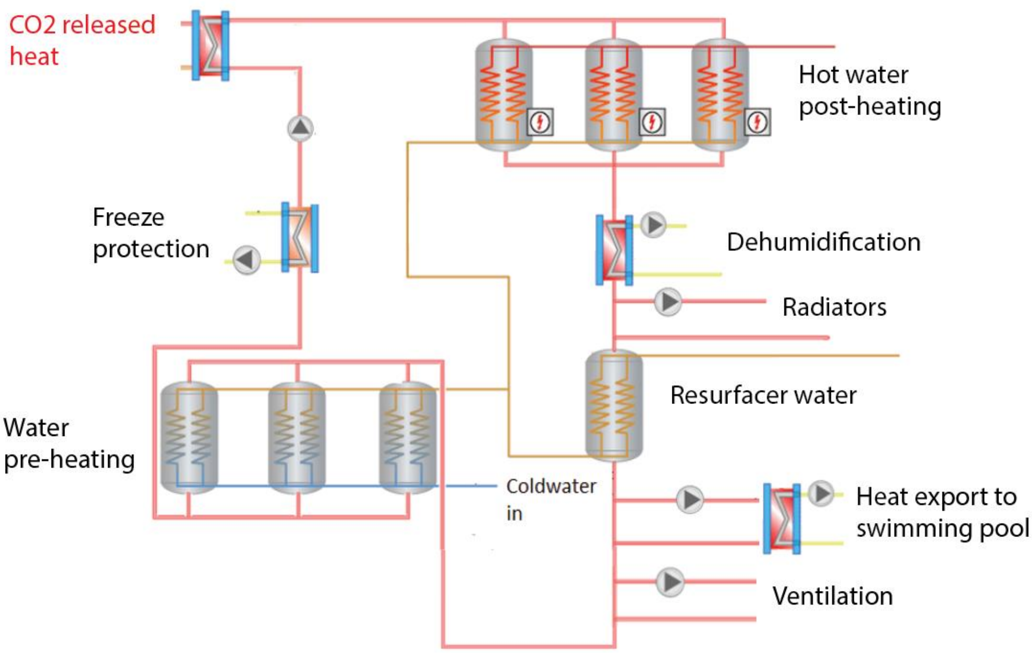


Figure 3.3. Heat recovery system layout in Gimo ice rink.

One of the cost-effective energy measures in an ice rink is use of recirculation in the ventilation system – it reduces heating demand in the winter and dehumidification demand in the summer. But even with recirculation, arena room space heating accounts to about 70% of the annual heating demand which is to large extent because of the convection between air and ice. In heated Public C arenas air temperature is maintained to about +8°C, while spaces around can be maintained at temperatures that are typical for indoor occupied premises. This creates several zones that interact with each other, like in Gimo ice rink as well.

Arena room air heating capacity data was compiled for a whole season (08/2016 – 03/2017). As also found in the literature, in real systems arena room temperature drops as it becomes colder outside. The key issue is under dimensioned heating system.

In the modelling part of the work, in order to analyse a system where rink space temperature is maintained at +8°C, the trendline is extrapolated as shown in Figure 3.4. This heating curve is then used in modelling, and for geothermal function it is crucial to define the heating demand as accurate as possible, in order to size the borehole field accordingly.

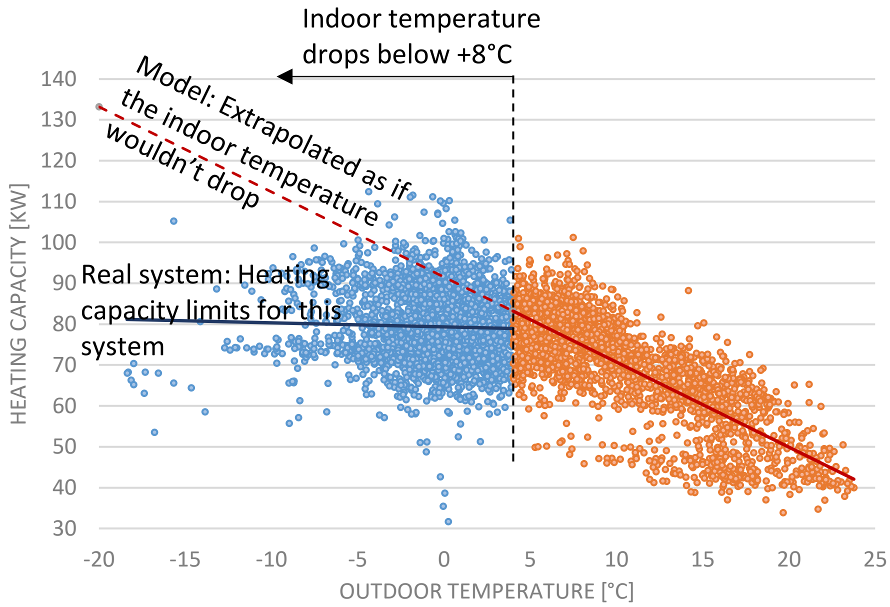


Figure 3.4. Space heating demand signature in Gimo arena room.

3.1.2 Cooling demand

Cooling demand in an indoor ice rink builds up from several load components – convection, radiation, condensation, ice resurfacing, lighting, ground conduction, pump work, headers, skaters. Several literature sources have been analysed. To explain how different location and time of the year can affect the specific design cooling load per area of ice, two examples are given - 80 W/m2 in Stockholm in spring and 135 W/m2 in Pittsburgh in summer. With these findings in the literature, theoretical magnitudes are to some extent known, but must be used with care due to sensitivity of assumptions. Furthermore, a model for hourly resolution is an extremely complex task to carry out since cooling demand has so many influential factors and is to a large extent interlinked with the space heating system and activity level.

Actual system analysis is needed to evaluate the cooling demand better. For the Gimo system, the cooling demand is shown in Figure 3.5. From the available IR ice-surface-temperature readings, it was possible to separate datasets when resurfacings took place and thus see the impact. As the data suggests there is a certain offset of increased cooling capacity once the warm water is laid on ice.

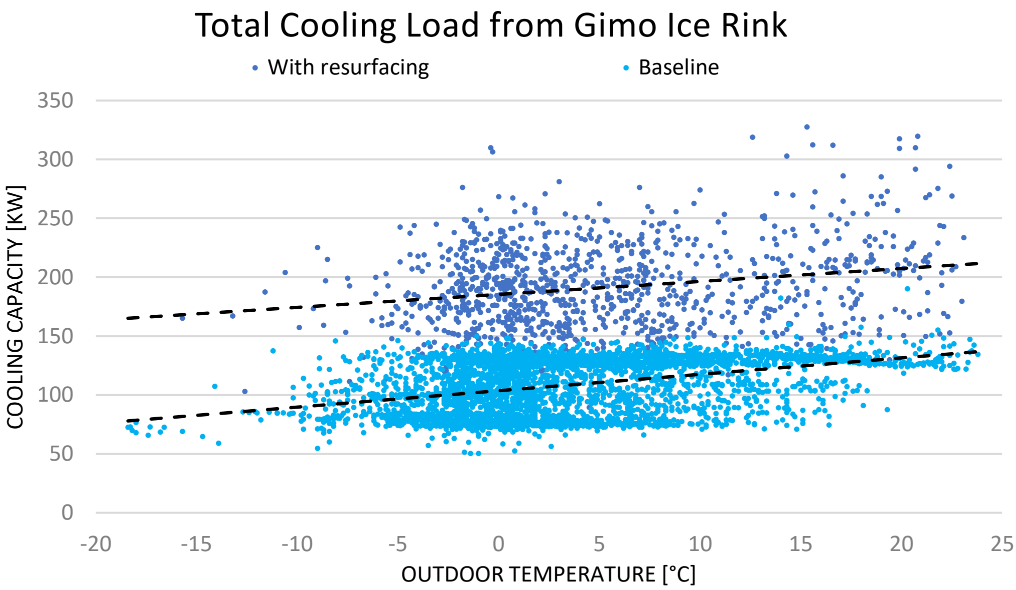


Figure 3.5. Cooling demand in Gimo ice rink. Data separated for cases with and without resurfacing.

In Gimo ice rink the cooling demand displays a reducing trend because the indoor temperature drops together with outdoor air temperature. In case the heating system would be able to maintain the indoor temperature to +8°C, the cooling demand trendline would theoretically flat out. It is important to note that both heating and cooling capacities fluctuate significantly over the day due to activities, which is another crucial aspect to consider when dimensioning a borehole field as it affects the heat recovery ratio (HRR).

3.2 Heat Recovery Control for CO2 refrigeration systems in Ice Rinks

The heat recovery potential of transcritical CO2 systems is remarkable. Nevertheless, there are many research works which are looking into further system optimization. One of the ways is to optimize the controls. A part of this project was to develop a control strategy to run CO2 systems at their maximum global COP at any given load conditions. It is important to stress out the key variables that make a CO2 refrigeration system with heat recovery so dynamic and why in practice it is a challenge to predict the best possible controls.

Table 3-1. Elements in a CO2 refrigeration system with heat recovery that are focal in optimum control.

Fluctuating influential variables	Controllable refrigeration system parameters	System controlling components
Heating demand	Discharge pressure	High pressure control valve
Heating system forward/return temperatures	Refrigerant temperature after gas cooler	Compressors
Cooling demand		Gas cooler fans, bypass valve
Ambient temperature		Heat recovery bypass valve

3.2.1 Modelling an Optimal Control Algorithm for Ice Rink Applications

In this study, a system with two stage heat recovery is evaluated. The system performance evaluation is done in Python programming language with RefProp thermodynamic properties library as an add-in.

The key for this simulation model is to identify the water temperature profile in the heat exchangers, in relation to the heat recovered from the refrigerant at the respective CO2 temperature along the heat exchanger. The method is based on the energy conservation law applied at heat exchanger sections.

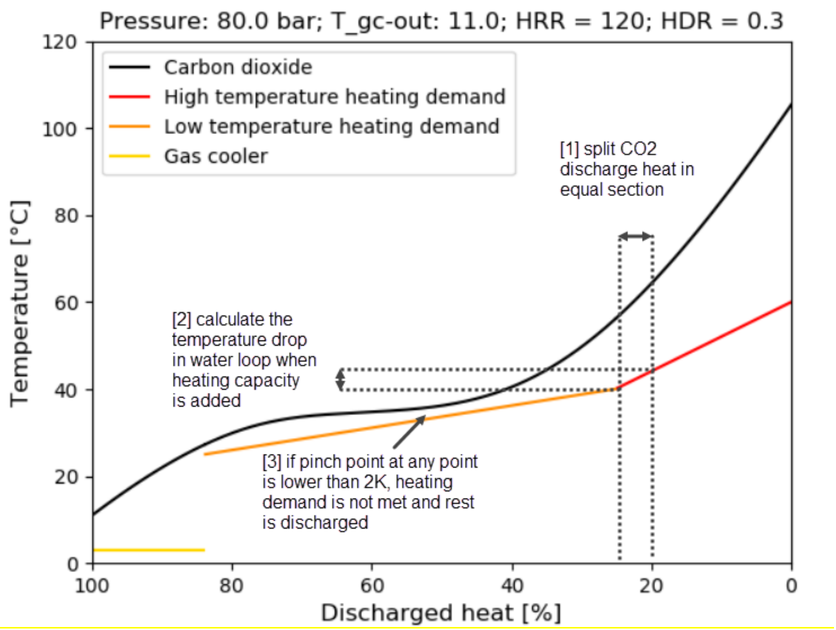


Figure 3.6. The underlying principle of heat recovery simulation. Example given for transcritical mode.

Model runs through numerous iterations for each hour of the year, with a goal to find the most optimum global COP running conditions respecting that the heating demand with its forward/return temperatures is satisfied. Equation for global COP is shown in Eq. 3.1 .

$$COP_{GL} = \frac{\dot{Q}_2 + \dot{Q}_{HR1} + \dot{Q}_{HR2}}{\dot{E}_{comp} + \dot{E}_{gc.fans} + \dot{E}_{geo.pump}} \quad (3.1)$$

More detailed information about the simulation model can be found in (Pomerancevs, 2019).

3.2.2 Case study 2 – Real Heat Recovery Control Strategy

A second case study was analysed to study the heat recovery control strategy for a real system. The refrigeration unit provides cooling to a single sheet ice rink. To better match the temperature profiles between water and CO2, in this ice rink heat recovery is performed through 2 heat exchangers. This is usually referred as 2-stage heat recovery.

The particular system is a partially indirect configuration with CO2 as the primary refrigerant and an NH3-water solution as the secondary fluid. The heat recovery system is designed in two stages – 40/20°C and 70/40°C (forward/return temperatures).

Table 3-2. Heat recovery temperature stages with their respective functions in the studied ice rink.

Medium temperature stage (40/20°C)	High temperature stage (70/40°C)
Rink space heating	Hot water post-heating
Water preheating	Dehumidification
Freeze protection	Resurfacing water heating
	Comfort space heating (radiators)

Firstly, the system is analysed according to the actual running conditions – to see how the given cooling, heating demands are satisfied in the actual system. Then identical demand conditions are modelled with the optimized control algorithm for the highest global COP.

Conclusions for the particular system are that more heat could have been rejected via gas cooler to improve the efficiency and still respecting the exact heating demand and having exactly the same conditions on the low-pressure side. As a result, for the period between mid-November and end-March, theoretically, the electricity use could have been reduced by roughly 12% (22 MWh in 4.5 months).

To understand better what makes the difference, CO2 transcritical heat rejection process should be analysed, shown in Figure 3.7. For the same discharge pressure, the CO2 temperature profile in the heat exchanger will depend on the CO2 temperature at the outlet of the gas cooler. Heat is released along the heat exchanger to the heating “sinks” – high temperature HR stage, medium temperature HR stage and rest is rejected to the ambient via gas cooler.

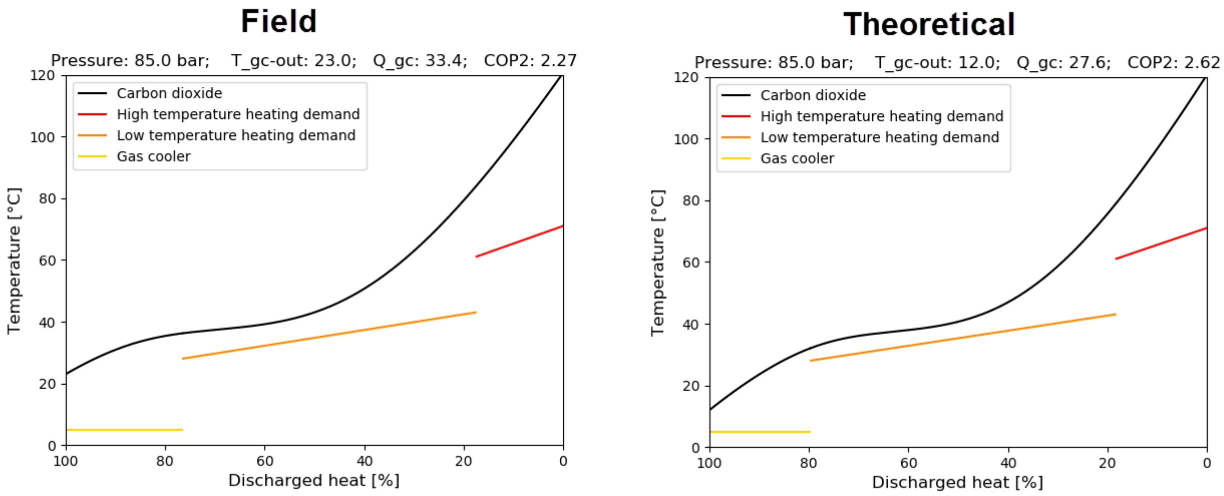


Figure 3.7. Heat exchange process between CO2 and heat "sinks" for example conditions from field data study. Gas cooler outlet temperature on the left +23°C, on the right +13°C.

In both cases, the same heating demand is covered, however, the difference is in the gas cooler outlet temperature – the lower it is, the lower the mass flow and consequently the better the COP of the system. In this case study, the gas cooler fans were not running for most of the time. However, theoretically, more heat could have been rejected to the ambient without compromising heating demand coverage. A follow-up of existing systems is essential to localize such control issues.

3.3 Geothermal storage integration with CO2 refrigeration systems in Ice Rinks

Modelling energy performance of a system is an excellent approach in order to evaluate the overall cost-effectiveness for different alternatives. In order to narrow the focus on the most relevant solutions in ice rinks, the analysis is done for CO2 refrigeration system with a 2-stage heat recovery, where forward-return temperatures are 70/40°C and 40/20°C for the first step and second steps respectively.

In this work two system solutions are compared – with and without a geothermal function. The system layout is shown in Figure 3.8, where the geothermal connection is highlighted in green. This is a direct expansion system which is more energy efficient due to a higher evaporation temperature in comparison with indirect configuration. The geothermal evaporator extracts heat from the ground during heat pump operation. This is operated at the same evaporation temperature of the main liquid receiver. The geothermal heat extraction could be more efficient, if the evaporator was connected at a higher-pressure level through parallel compressors (e.g. supermarket case study).

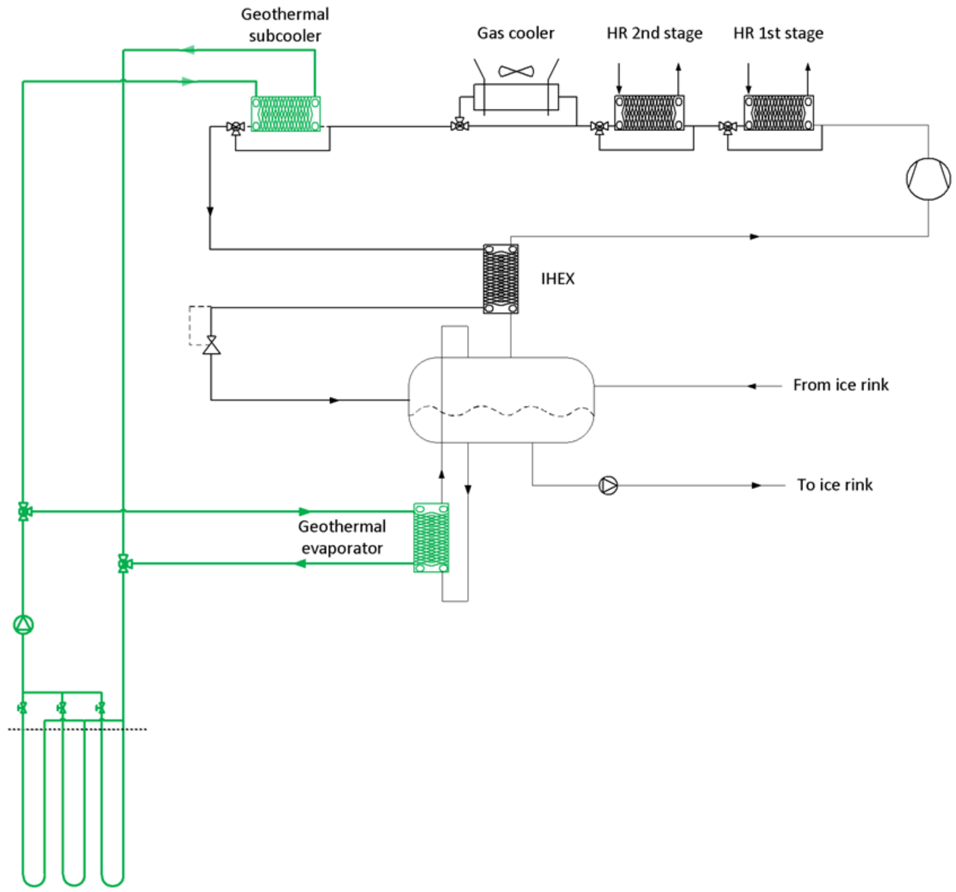


Figure 3.8. Ice rink system configuration that is modelled in this study.

3.3.1 Scenarios

An average size ice rink in Sweden falls under category "Public C", which means 500-1000 spectator seat capacity, with a single ice sheet and rink space is normally heated for +8 to +10°C air temperature. The defined normal size ice rink is specified in Figure 3.9. To diversify possible cases, activity level was varied between high and low activity ice rinks. Another crucial aspect is length of the season – the most typical is 8-

month season (august – march), but sometimes it can be year-round. These aspects affect the heating and cooling demands of the facility and the performance of different system solutions.

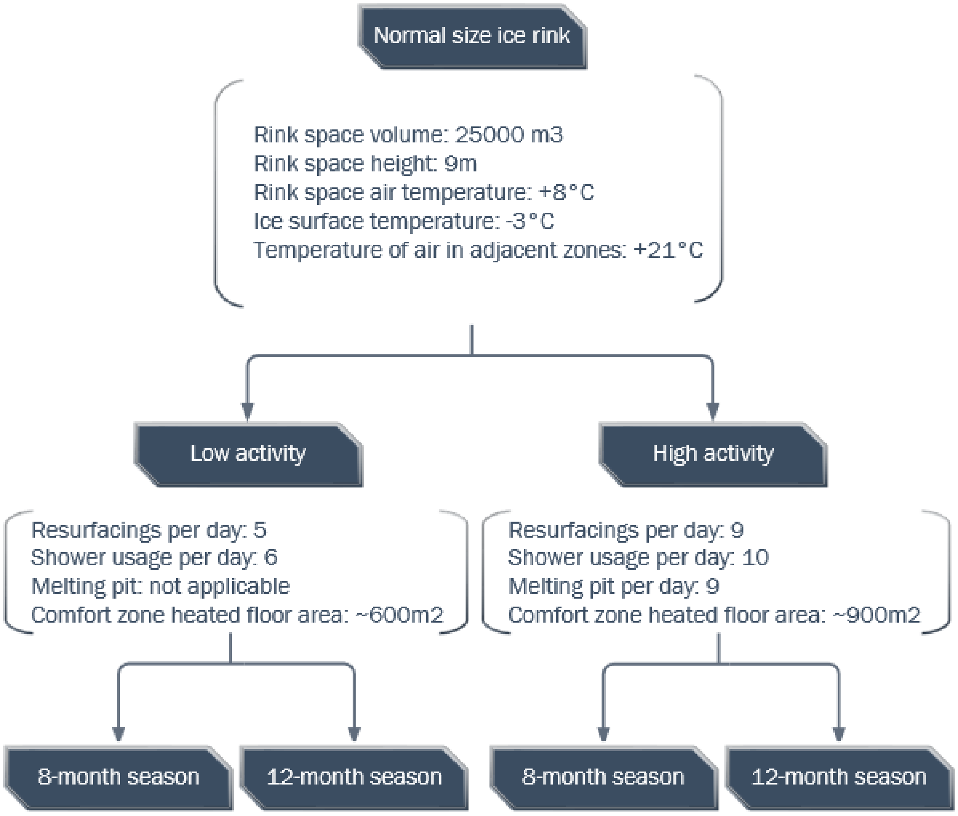


Figure 3.9. Classification of ice rinks in this study.

In the modelled scenarios HRR vary mainly due to the demand of high temperature heat. For instance, in high activity scenarios, there is more high-temperature load because of the melting pit and more shower usage. This implies that the average annual HRR is different according to the length of the season and level of activity. For example, an ice rink operating for 12-month per year has an average HRR of 86% for a low activity scenario and 100% in a high activity scenario, while a facility operating for a season of 8-month has an HRR of 92% and 106% respectively.

The average values give only the overall picture which is not sufficient to grasp the key reason for geothermal function. For this reason, the distribution of the HRR on the outdoor temperature is calculated and it is displayed in Figure 3.10. There is a noticeable shift at around 4°C ambient temperature, where the HRR is mainly between 100 and 120%, while above this temperature HRR is mostly below 100%.

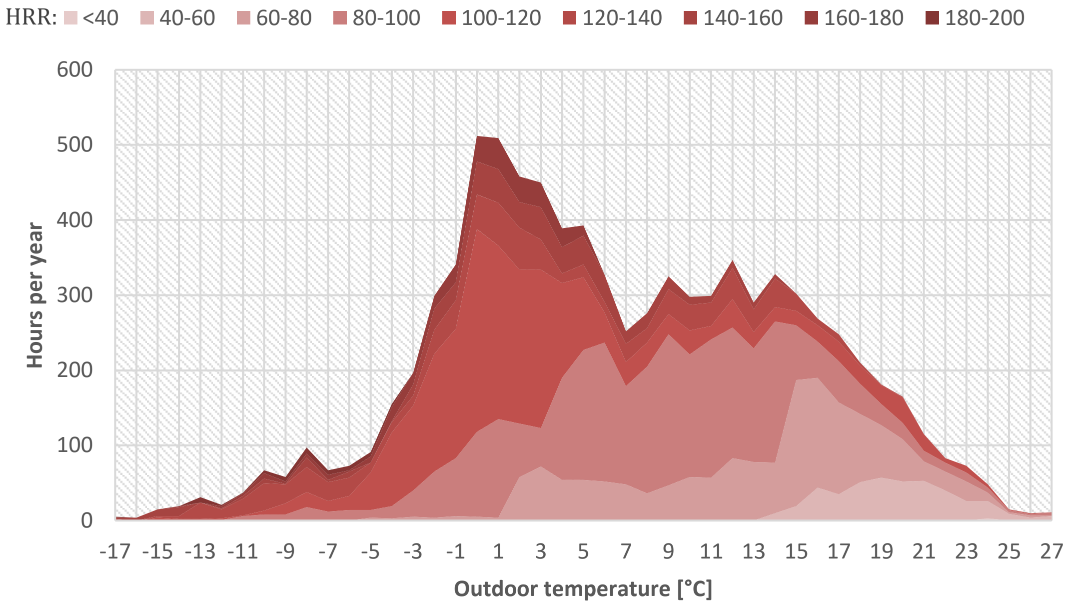


Figure 3.10. Hours of operation under certain HRR range. Example from results for high activity 12-month season scenario.

3.3.2 Need of a geothermal function in a CO2 system

HRR distribution is ice rink specific, so geothermal function has to be sized to the case. High activity 12-month season is a scenario that has a lot more dynamic demand profiles which is why this scenario is used for detailed result elaboration.

Heat recovery ratio for the system at the top can reach 200% when it is cold outside and coincides with no resurfacing. However even if it may seem exaggerated compared to previous research, it must be taken into account that this is an extreme scenario with more adjacent heated areas and also importantly the rink space temperature is maintained at +8°C which is not assumed in other investigations. Additionally, to clarify - HRR above 170% in this scenario is only 1.8% of the time, while for low activity scenario HRR never exceeds 160%.

To see the difference in global COP Figure 3.11 can be examined. HRR that is used here represents the demand ratio, but for the system with geothermal system HRR is up to 130%, because of the extra cooling demand imposed to the evaporator. Indeed, it can clearly be seen that geothermal function adds most of the value at high HRR's, where compressor, geothermal pump, gas cooler fans require additional energy, but no further pressure lift is needed by the compressor, instead refrigerant mass flow is increased.

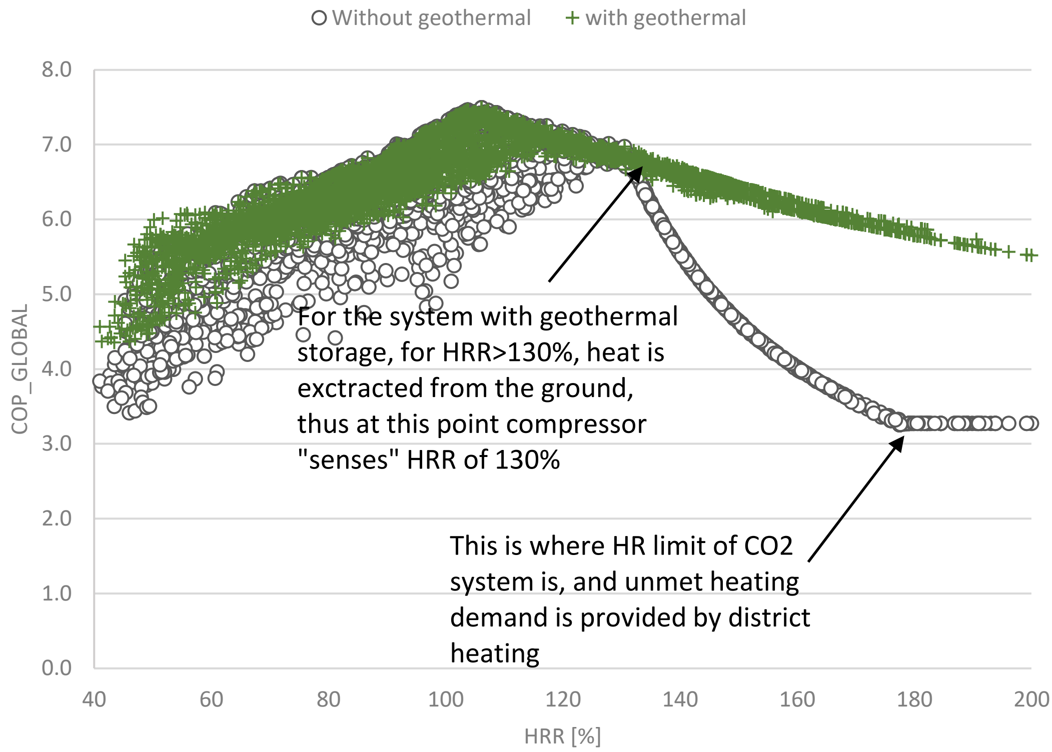


Figure 3.11. Global COP depending on the HRR comparing between systems with and without geothermal for high activity 12-month season scenario.

It is important to see the hourly electric power depending on the temperature outdoors, which is illustrated in Figure 3.12. There are several levels of energy use, which is dependent on whether there is resurfacing and also if hot water or melting pit is used, etc.

The climate is crucial dimension that defines the eventual cumulative savings, and bin hours of the respective ambient temperature are included in the graph as well. It can be seen that highest potential is at the extreme temperatures – when it is the coldest and warmest outside, but the bin hours suggest that the largest part of the year the ambient temperatures are between -2°C to +15°C, which does not play in favour to cumulative energy savings. Although between -2°C and +5°C outdoor temperature there are hours when about 80% savings can occur.

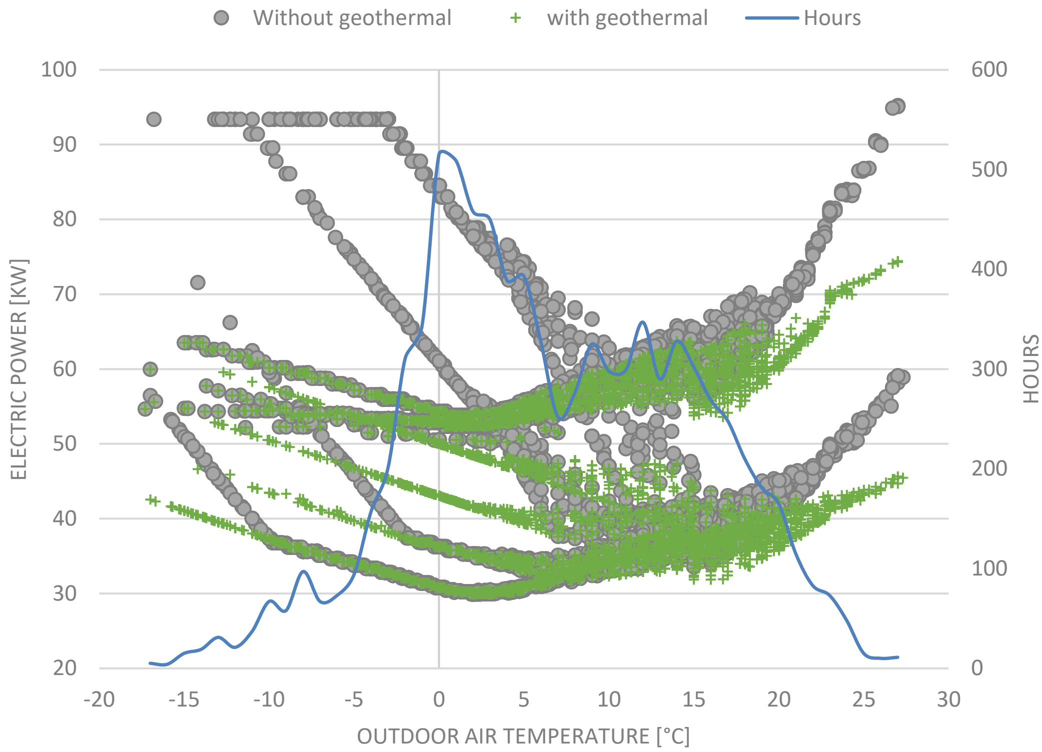


Figure 3.12. Hourly total electric power comparing between systems with and without geothermal for high activity 12-month season scenario.

3.3.3 Techno-economic evaluation

To show more clearly the cumulative energy savings, monthly electricity and supplementary heat savings at certain HRR range are visualized in Figure 3.13. Savings are calculated as a difference between system without and with geothermal function.

Even though timewise geothermal function is used more during the summer and also more thermal energy is injected than extracted annually, still most of the savings happen during the coldest months because there is a more significant savings effect at HRR's above 130%. The highest estimated energy saving is in January – about 4.3 MWh, while in the summer highest savings due to subcooling are in August – about 2.4 MWh.

The annual energy results for the two extreme scenarios are shown in Table 3-3. As it suggests in this investigation a CO2 system with two stage heat recovery and geothermal function in configuration shown in paragraph 3.3 can save between 1.7 to 6.8% of energy depending on season length and level of activity when compared to a system without a geothermal function.

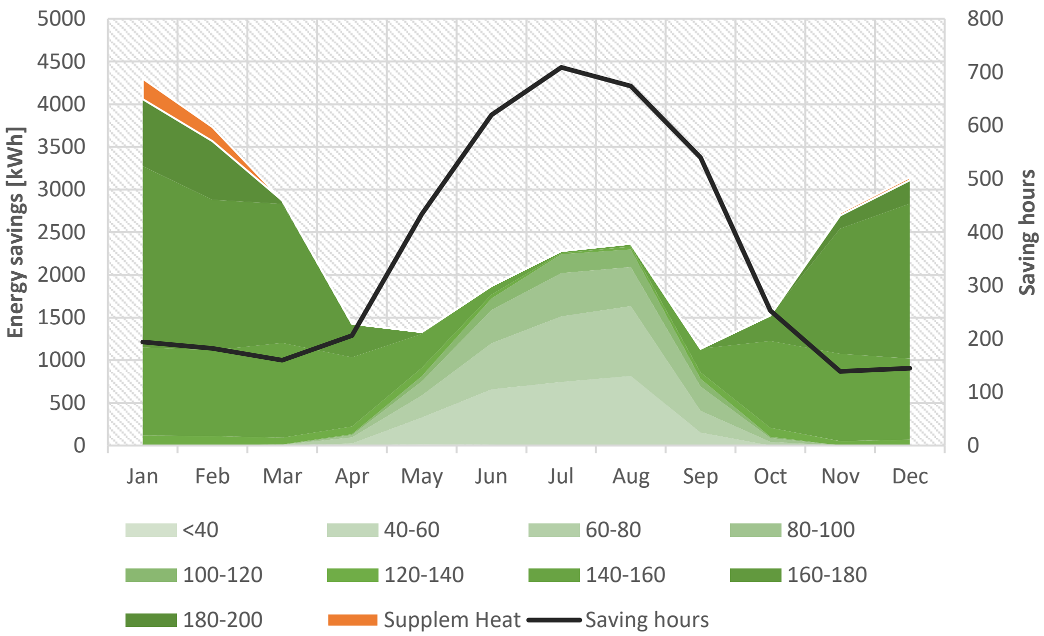


Figure 3.13. Monthly electricity and supplementary heat savings for high activity 12-month season scenario.

Finally, economic calculations are performed to show the feasibility of geothermal storage integration. Several commonly used economic indicators are used, i.e., discounted payback period, net present value and internal rate of return. The cost component that is considered in this analysis is the borehole drilling price of 250 SEK/m, which has a high uncertainty but nevertheless on the conservative side. The lifetime of investment is 20 years, which is a reasonable timeline commonly used in other investigations. The results are summarized in Table 3-3 and more detailed assumptions can be examined in (Pomerancevs, 2019).

Table 3-3. Economic evaluation results for the analysed scenarios.

	High activity 12-months	Low activity 8-months
El use without geothermal function [MWh]	414.0	212.4
El use with geothermal function [MWh]	385.7	208.4
Borehole depth [m]	161	157
Number of boreholes [-]	9	7
Price per m [SEK/m]	250	250
Investment [SEK]	362 250	274 750
Discount rate [%]	3	3
Discounted payback period [years]	**16.4**	**-**
NPV [SEK]	**57 911**	**-215 240**
IRR [%]	**4.7**	**-**

The results of this work suggest that in the best-case geothermal function would pay off in 16.4 years but for low activity ice rink there is even no payback within lifetime of the system.

4 Supermarkets

In this project, one new refrigeration unit integrated with geothermal boreholes was deeply investigated and the field data was utilized as input for a modelling analysis. In the following section the results of this in-depth analysis are presented.

4.1 Case Study 3 – All-In-One System for Supermarkets

The supermarket refrigeration system described in this session is installed in south of Sweden. The plant scheme is displayed in

Figure 4.1. This CO_2 trans-critical booster system provides cooling at two temperature levels, corresponding to cabinets and freezers. The heat is recovered from the refrigeration cycle to provide space heating during winter and tap water heating during the whole year.

The geothermal storage consists of 16 boreholes heat exchangers (BHEs) of 200-meter depth; these are used to satisfy the peaks of heating demand extracting heat from the ground. Additionally, the boreholes enable the sub-cooling of refrigerant during warm periods. This makes such a solution very promising in terms of cost-effectiveness. Summarizing, the only energy carrier bought externally is the electricity needed for compressors, pumps, ventilation, auxiliaries and lighting.

Four medium temperature (MT) compressors represent the heart of the system, recirculating refrigerant from the cabinets evaporator to the gas cooler. Additionally, four parallel compressors were implemented to take care of the flash gas generated at the outlet of the high-pressure expansion valve. These machines are also used to extract heat from the ground and to provide air conditioning. Finally, three low temperature (LT) compressors bring the refrigerant from the freezers evaporator to the suction line of MT compressors.

The heat recovery is done in two stages according to the different temperature levels of the heating demand, respectively tap water and space heating. Such a strategy increases the heat that can be recovered from the cycle since the temperature profiles of the secondary fluid in the heat exchangers can be adjusted to fit the CO_2 temperature profile (J. Rogstam et al., 2014). Then, the refrigerant goes through a gas cooler installed on the rooftop which, depending on the operational conditions, can also act as a condenser. Finally, a geothermal sub-cooler was implemented to dump the excess of heat into the ground during warm seasons.

Four internal heat exchangers (IHXs) were installed, these are mainly used to provide further sub-cooling of the refrigerant before entering cabinets and freezers (Kauko et al., 2016). The evaporator which is used to extract heat from the ground and to provide air conditioning works at an intermediate pressure between the liquid receiver and the medium temperature level. Obviously, this is the same suction pressure of the parallel compressors. In case the amount of flash gas does not reach the minimum mass flow elaborated by the parallel compressors, the flash gas is by-passed on the suction line of the MT compressors. The expansion valves on the different temperature levels are regulated to control the internal superheat in the evaporators. An essential component is the by-pass valve before the gas cooler which enables the regulation of the heat recovery.

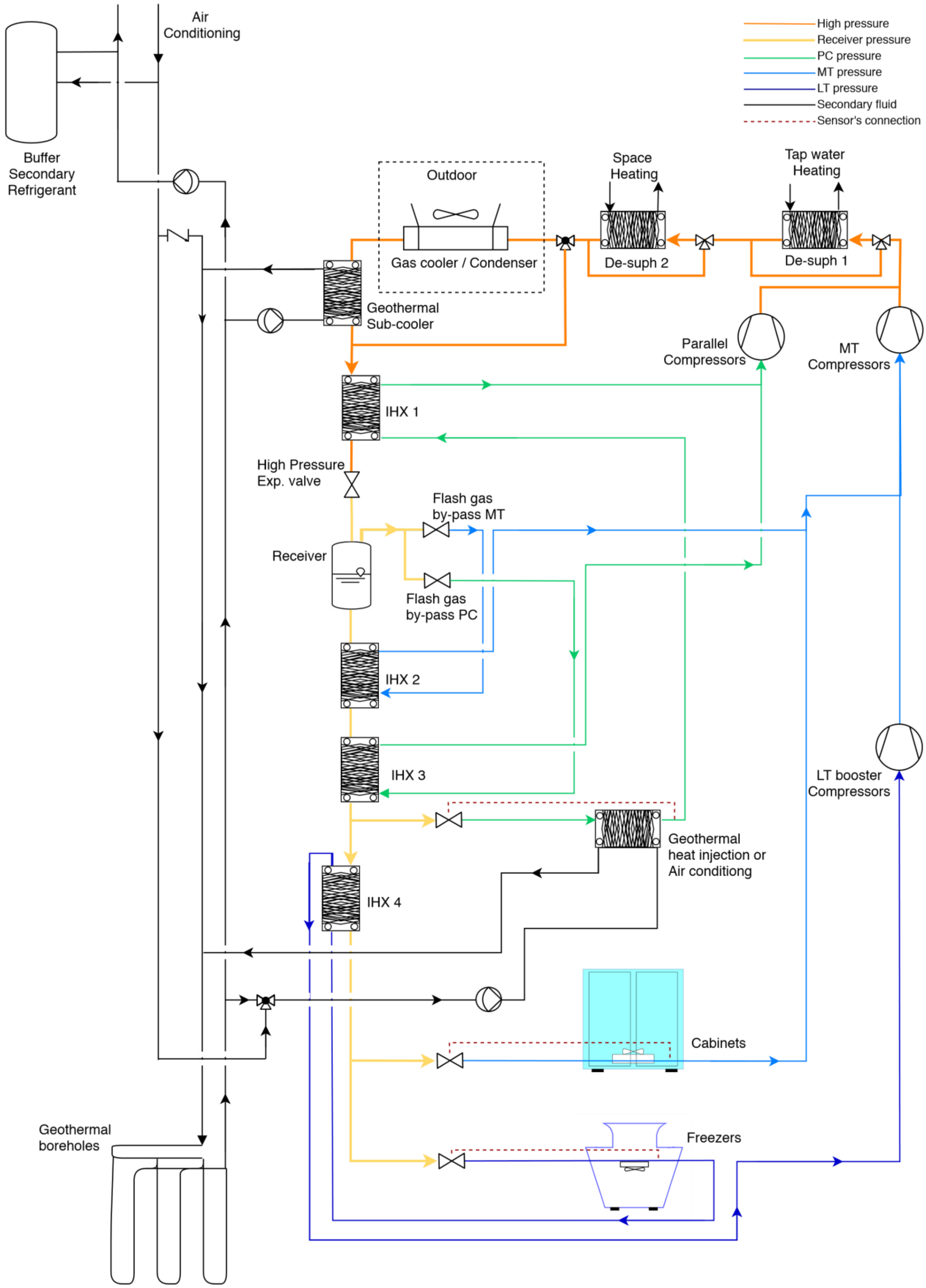


Figure 4.1 Scheme of the CO2 trans-critical booster system integrated with heat recovery, air-conditioning and geothermal boreholes.

4.1.1 Heating Demands

The studied system satisfies not only cooling demands but also two different heating demands, namely, high-temperature heating for tap water (TW) and space heating (SH).

High temperature heating for tap water

The system controls the discharge pressure to satisfy the space heating demand, recovering the available high temperature heat for tap water as a B-product. For this reason, the heat recovered for the tap water production follows the discharge pressure. An additional electric heater is usually utilized to satisfy the small amount of extra – energy needed for tap water. Figure 4.2 depicts the heat supplied in the first de-superheater. The scattered pattern at the edges is due to scarce amount of data for very low or very high outdoor temperatures.

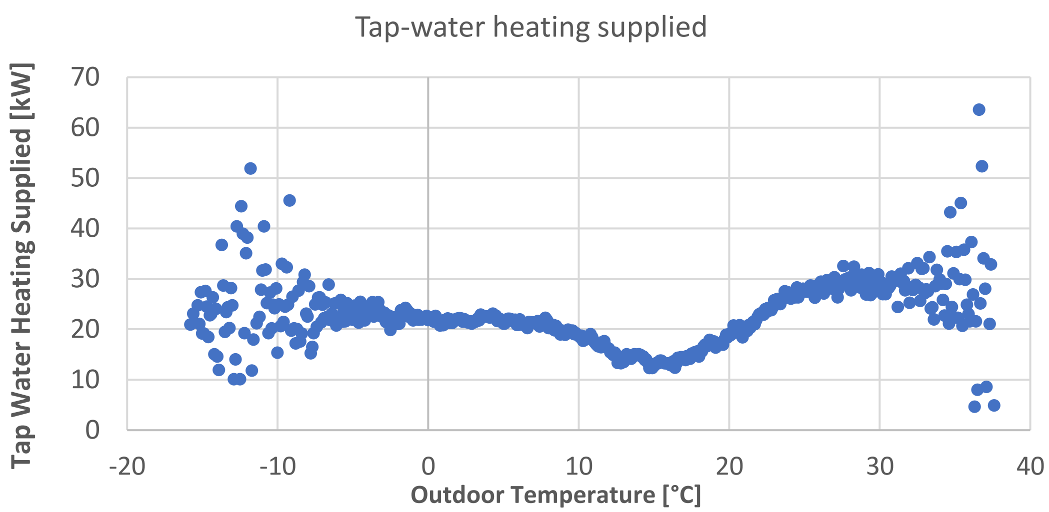


Figure 4.2 Field Measurements: Heat supplied to the tap-water loop.

Space heating

Being that the supermarket is self-sufficient in terms of heating, the heat supplied is representative of the space heating demand of the building (no external heat sources). This is shown in Figure 4.3.

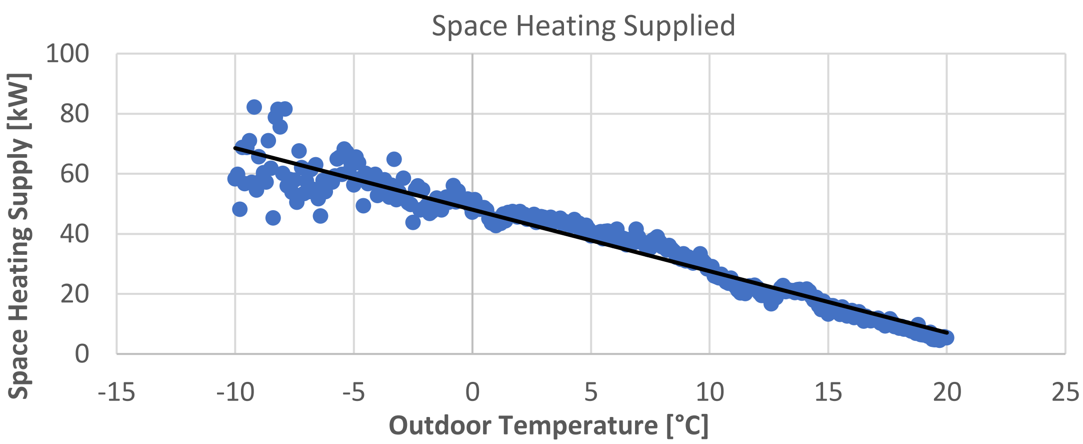


Figure 4.3 Field Measurements: Space Heating Supplied.

For an outdoor temperature between 13°C and 20°C, the heat is recovered as "waste heat" while for temperatures lower than 13°C the control system increases the refrigerant discharge pressure in order to satisfy the heating demand. It must be said that the heating demand not only varies with the outdoor temperature but also with the activities inside the supermarket.

4.1.2 Cooling Demands

The three cooling demand satisfied by the refrigeration system are: medium temperature refrigeration (MT), low temperature refrigeration (LT) and space cooling (AC).

Medium and Low temperature refrigeration

The refrigeration load is strictly dependent on the type of cabinets (with or without doors) and the outdoor absolute humidity. The latter varies proportionally with outdoor temperature, the warmer the weather, the higher the absolute humidity. Since the outdoor air flow is not dehumidified, a high outdoor absolute humidity is translated into high indoor relative humidity which, in turn, results in a higher frost formation rate inside the cabinets (Sawalha, 2012). When doors are installed a small difference can be seen between day and night demand and the effect of outdoor humidity is reduced. The examined installation is connected to cabinets and freezers equipped with doors. This explains the stable trend of the freezers cooling demand, displayed in Figure 4.4.

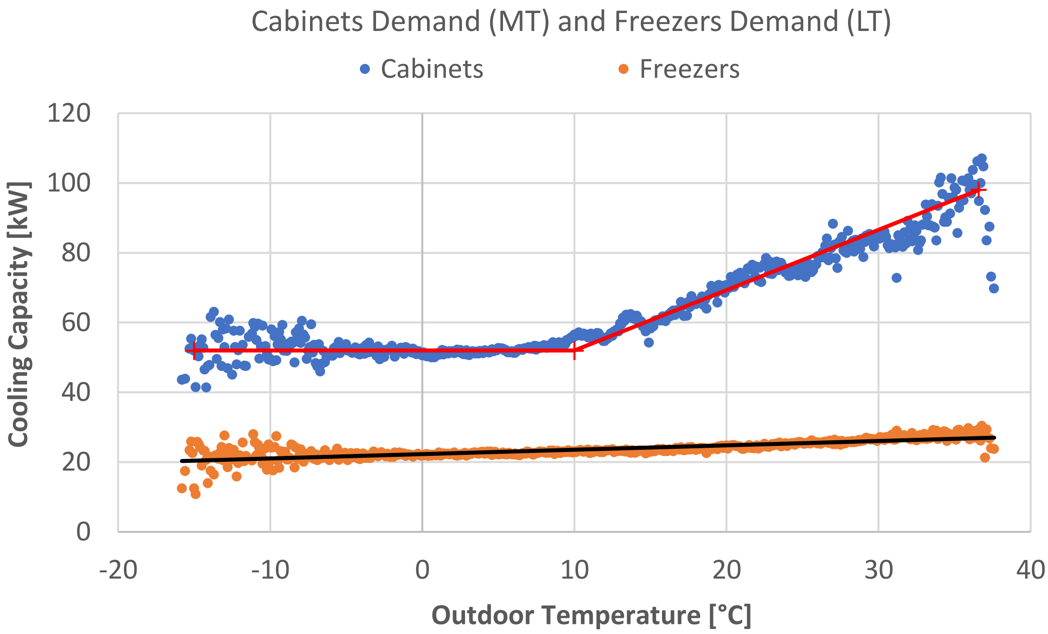


Figure 4.4 Field Measurements: Cooling demand for cabinets and freezers as a function of ambient temperature.

As can be seen, the MT demand (cabinets) is stable during the heating period (when the outdoor temperature is lower than 10°C) while it increases proportionally to the outdoor temperature when the indoor is not heated.

Air Conditioning

The Supply of cooling for the building is done through the same evaporator used to extract heat from the ground. Figure 4.5 shows the air conditioning supply curve, clearly, the activation set point is at 13°C outdoor temperature.

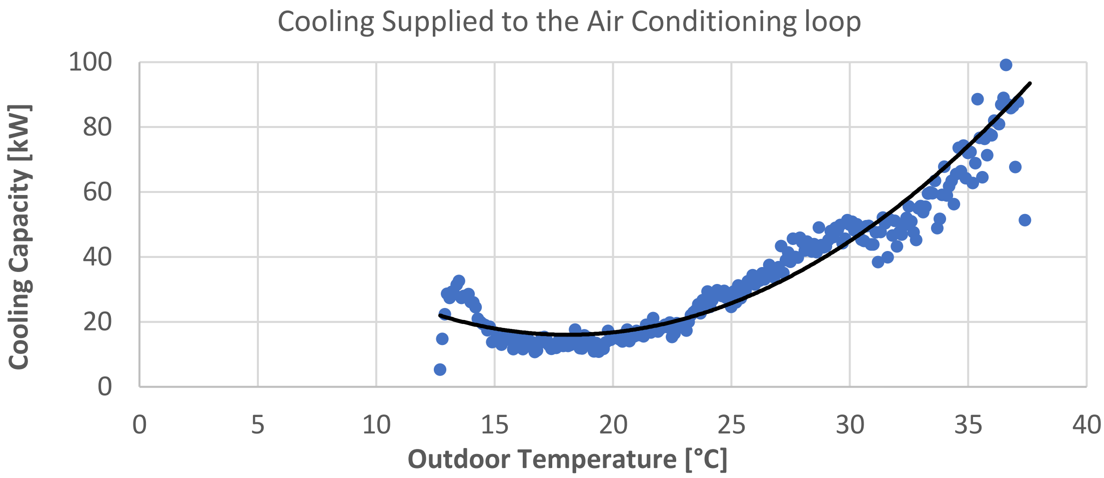


Figure 4.5 Field Measurements: Cooling Supplied to the Air Conditioning loop.

4.1.3 Electrical Power Consumption

In the installation studied during this project, two power meters were installed, one aggregating the MT compressors, the LT compressors and their auxiliaries, while another aggregating the parallel compressors (PC) and its auxiliaries. For this reasons, the individual compressors' powers were calculated from the manufacturer data using the thermodynamic properties at the discharge and suction point of the compressors. The breakdown of the calculated power consumption is shown in Figure 4.6 .

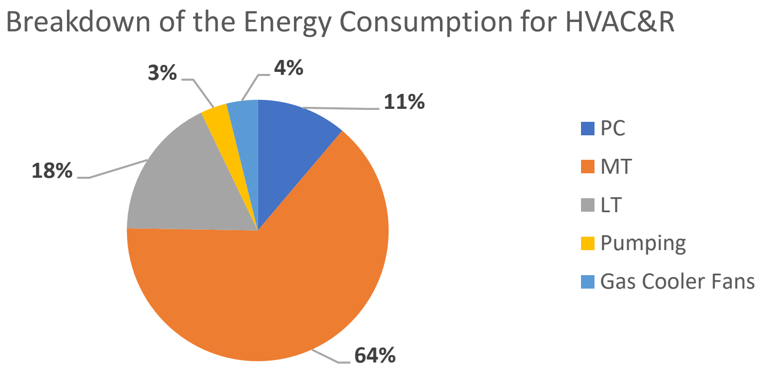


Figure 4.6 Breakdown of the Total Energy Consumption for HVAC&R.

The power absorbed by the system, averaged on the outdoor temperature, is shown in Figure 4.7 . The small step visible when passing from heat recovery mode to floating condensing is due to the activation of the air conditioning, which is connected to the load evaporator.

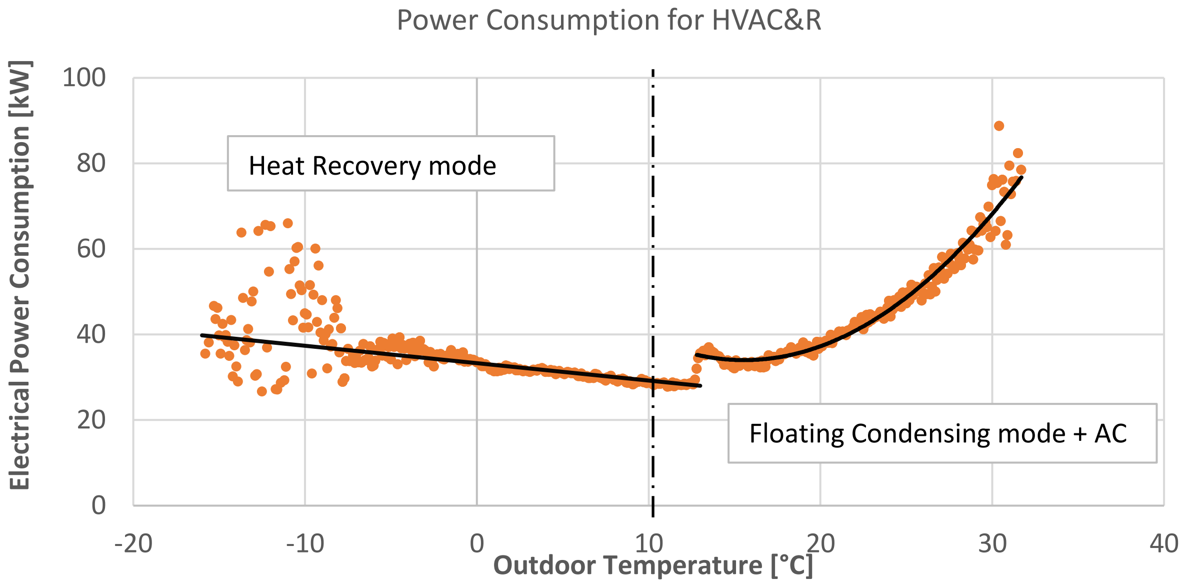


Figure 4.7 Total Power Consumption for HVAC&R.

4.2 Heat Recovery Control for CO2 refrigeration systems in Supermarket

4.2.1 Best Theoretical Heat Recovery Control Strategy

Concerning the control strategy, the theoretically best for a stand-alone installation (no additional systems e.g. boreholes or heat pumps) is described by (Sawalha, 2012). When geothermal boreholes are installed, some modification to the control strategy must be made as they represent an additional heat source. The latter is different from cabinets and freezers as it does not represent a cooling load that must be guaranteed. Such a control strategy is an evolution of the best theoretical identified by (Sawalha, 2012) and it was explained by (Karampour et al., 2018).The best theoretical heat recovery control strategy consists of five main steps which can be summarized as follow:

Step 1) The system is working in floating condensing mode with the lowest possible condensing pressure. As soon as the heating demand increases, the condensing pressure is raised.

Step 2) When the pressure brings the refrigerant in its supercritical region, the pressure should be gradually increased until it reaches the optimal value which is described in equation 4.1 (Sawalha, 2012).

$$P_{dis} = 2.7 * T_{desuperheater\ exit}[°C] - 6\ \ [bar] \tag{4.1}$$

$T_{desuperheater\ exit}$ is the outlet temperature from the second desuperheater. In case that the system is operating in transcritical mode but no heat is recovered (e.g. warm summer day), the value to be utilize should be the gas cooler outlet temperature.

Step 3) Once the pressure reaches the optimal value in the supercritical region, it is kept constant and the gas cooler fans' power starts to be regulated. When fans are switched off, CO2 continues to be circulated through the gas cooler which works in natural convection. The effect of step 3 is to increase the high-pressure expansion valve inlet-temperature. This leads to an increase of generated flash gas and subsequently to an increase of mass flow elaborated by the MT compressors. Indeed, they are controlled to satisfy to cooling loads (for the same amount of liquid refrigerant the total mass flow increases proportionally to the quality).

Step 4) The last step is to fully by-pass the gas cooler. In this case the high-pressure expansion valve inlet temperature would be almost equal to the desuperheater outlet. This is the condition where the system reaches the maximum Heat Recovery Ratio.

Step 5) A further growth in the heating demand could be satisfied by extracting heat from the ground through the load evaporator

One can encounter a potential limitation of this methodology if the heat recovery system is very efficient and the desuperheater outlet temperature is very low (< 30˚C).Indeed, the resulting optimal pressure could be too low, in some cases even lower than the critical pressure. Without considering the technical implications, If the pressure is too low the discharge temperature risks to be lower than tap water supply temperature.

This limitation does not really apply to supermarkets. These systems are usually controlled to satisfy the space heating demand, in case that additional heat is needed to increase the supply temperature of tap water some electric resistances can back-up the heat recovery system.

On the other hand, such a limitation is relevant for other applications such as ice rinks, where the high temperature demand is considerably higher than the other demands. Moreover, the outlet temperature in the last desuperheater can be as low as 10°C.

4.2.2 Alternative Heat Recovery Control Strategy

The heat recovery efficiency during step 1 and 2 is hardly comparable to any other heating apparatus since the system takes advantage from the cooling loads of cabinets and freezers. The real question is whether the heat extraction from the ground should be activated before or after by-passing the gas cooler. For this reason, it is possible to distinguish two different control strategies referred as "*Gas cooler capacity regulation*" and "*Geothermal capacity regulation*".

"*Gas Cooler capacity regulation*" (control strategy A) is the one described as the best theoretical heat recovery control strategy (previous section). *"Geothermal capacity regulation"* (control strategy B) follows the first two steps but, once that the optimal pressure is reached (Step 2), the parallel compressors connected to the boreholes are activated while the gas cooler is controlled to operate with the minimum outlet temperature (Step 3B). If the maximum amount of heat that can be extracted is reached, then, the system proceeds with the gas cooler capacity regulation (previously described as step 3) and its eventual by-pass (previously described as step 4).

Theoretically, (Karampour et al., 2018) demonstrated that the ground should be activated after that the gas cooler is by-passed. In case the ground was activated before by-passing the gas cooler, part of the heat extracted from the ground would be necessarily rejected through the gas cooler.

However, from a thermodynamic perspective, there is no difference in terms of COP between the two cycles. Indeed, both the heat rejected from the gas cooler and the heat extracted from the ground are considered not useful. What matters is the power consumption of the compressors. This does not change if the fluid is cooled in the gas cooler and then evaporated through the load evaporator or if it is expanded at "higher temperatures", producing flash gas, and then by-passed to the suction line of the parallel compressors.

Comparison between control strategies

To assess the differences in terms of the heat recovery performance between the two strategies, a modelling study was carried out. Figure 4.8 shows this comparison. The necessary inputs for the simulation were taken from field measurements.

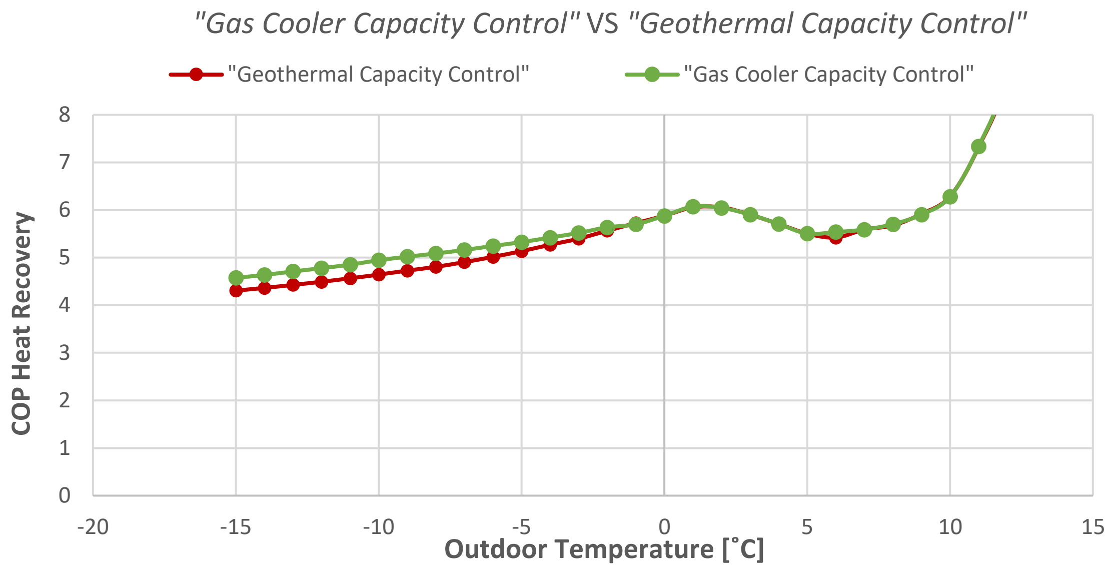


Figure 4.8 Comparison of COP heat recovery between theoretical control strategies.

Using the BIN hours method, it is possible to evaluate the average COP over the heating season. As can be noticed from the graph, the difference is minimal, in fact, the average COP goes from 6.1 (“Gas Cooler Capacity Control”) to 6 (“Geothermal Capacity control”).

To summarize, the only theoretical difference between control strategy A and B is the extra-power spent for pumping the secondary refrigerant through the geothermal loop. It is clear that the control strategy A is theoretically the most efficient, however, option B has some important co-benefits.

First of all, it helps to discharge the ground which is extremely unbalanced. Theoretically, this could be beneficial for the summer operations, increasing the subcooling potential. The second indirect advantage is related to technicalities. Control strategy A is more difficult to be implemented since regulating smoothly the inlet-temperature at the high-pressure expansion valve and the partial mass flow in the gas cooler is more challenging than regulating the capacity of the parallel compressors.

4.2.3 Real Heat Recovery Control Strategy

Going from the theory to the practice presents always some challenges and, as usual, not everything that is recommended by theoretical modelling is implemented in the real word. Analyzing the field measurements, the real control strategy and its effectiveness can be assessed.

The control device utilized is the Danfoss product AK-PC782A. The latter device converts a signal coming from a thermometer (Shr4 in Figure 4.9) on the supply line of the second desuperheater (Space heating supply) in a 0-10V output. Figure 4.9 shows a general scheme of this branch to better visualize which are the monitored parameters.

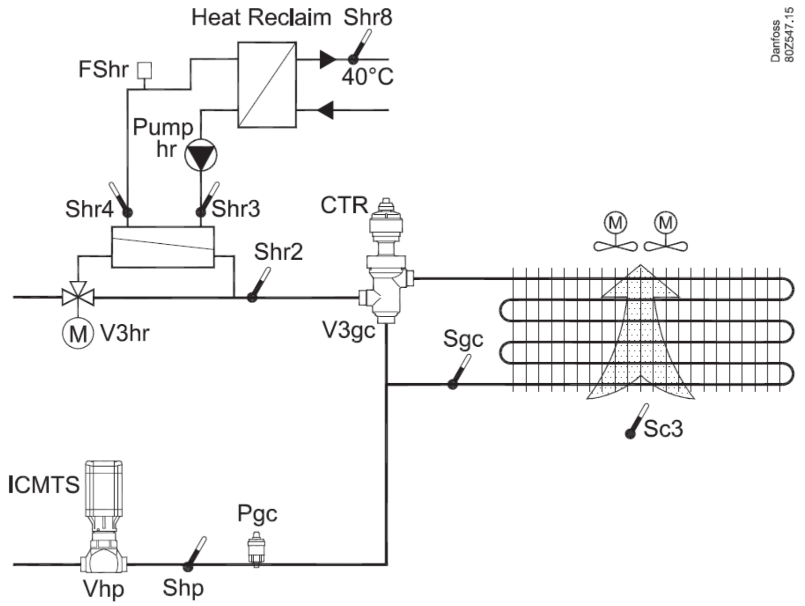


Figure 4.9 Measuring system for discharge pressure control. Source: (Danfoss, 2019)

As soon as the heat recovery system is activated (V3hr coupled up), the control increases the pressure to the minimum heat recovery value. Then, the 0 – 10V output is translated into a 0-100% “heat requested” which is the basis for the pressure, pump and gas cooler by-pass valve (V3gc) regulation. Figure 4.10 shows the principal of such a regulation. The “heat request” signal is shown on the x-axis. On the secondary fluid side, a variable speed pump is used to keep the temperature Shr4 close to the reference value. On the CO2 side, the pressure is regulated according to the “heat request” signal linked to the reference value (Shr4).

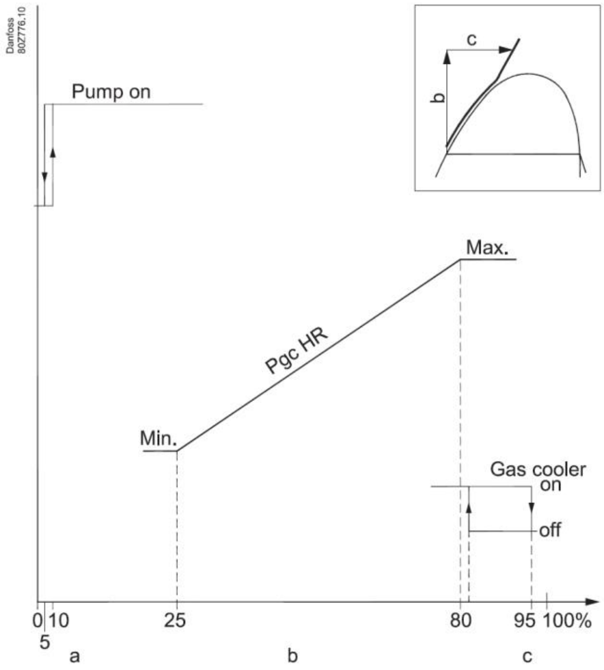


Figure 4.10 Discharge pressure regulation Danfoss controller Source:(Danfoss, 2019).

The set points and the corresponding minimum and maximum pressure can be adjusted in order to match the demand of the building. The concept at the basis of this process is that the supply temperature of the heating system should be regulated according to the outdoor temperature, the colder, the higher the supply temperature. Figure 4.11 displays data from field measurements showing the discharge pressure as a function of the space heating supply-temperature (Shr4). The similarities between the data and the control strategy described in the manual is glaring.

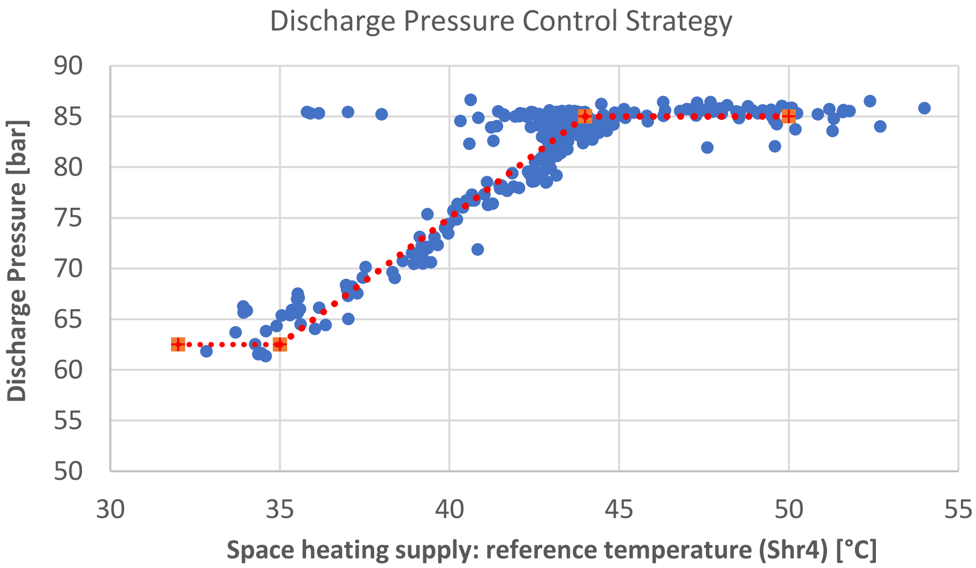


Figure 4.11 Field data: Discharge pressure as a function of space heating supply temperature (Shr4).

4.2.4 Real Heat Recovery Performance (COP heating)

Figure 4.12 displays the COP heat recovery (in orange) averaged on the outdoor temperature for the whole heating period (outdoor temperature <13°C). As expected, the COP is incredibly high for very low outdoor temperatures where a lot of waste heat is available from cabinets and freezers. At around 7.5 °C ambient temperature, the pressure exceeds the critical point and this why the loss of efficiency is less steep until -5°C where the maximum pressure is reached (on average). At this pressure level the system takes advantage of all the available heat. For a further increase in the supplied heat, the only option is to increase the refrigerant mass flow. This can be done either by-passing the gas cooler or activating the geothermal heat pump function. In this case the parallel compressors are activated, and the heat is extracted from the ground (control strategy B). The blue points represent the COP of the geothermal function.

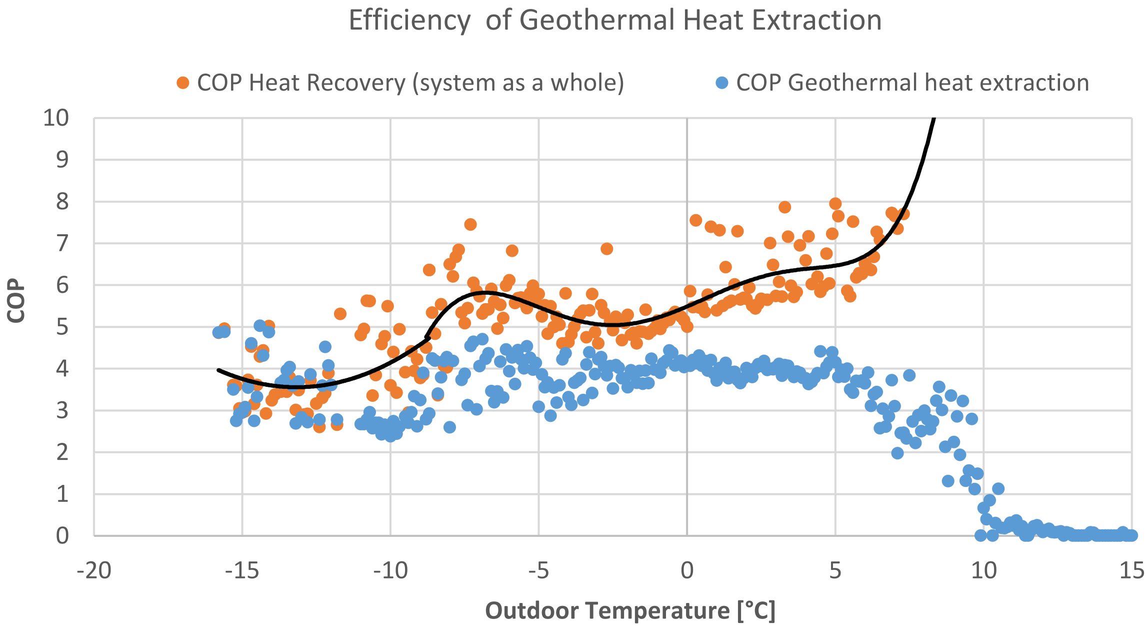


Figure 4.12 Comparison between COP heat recovery of the system as a whole and COP of the geothermal function.

The average COP heat recovery takes also into account moments when the boreholes are active for relatively high outdoor temperatures (e.g. +5°C). This means that the closer the two trends depicted in Figure 4.12, the more heat is extracted from the boreholes. For temperatures close to -10°C, the two efficiencies start overlapping demonstrating that the geothermal heat extraction starts supplying an important part of the heating demand. The average heating demand increases linearly with a decrease in the outdoor temperature. However, the peak loads' distribution is related to the activities (hours of the day) and these cannot be visualized if the values are averaged on the outdoor temperature.

For this reason, the effectiveness of the system was also studied in relation to the Heat Recovery Ratio, which is representative of the thermodynamic cycle and, therefore, the heat supplied by the refrigeration unit. Figure 4.13 shows the two aforementioned COPs averaged on the Heat Recovery Ratio (HRR), the values have been filtered in order to exclude values for outdoor temperatures lower than 13°C. Additionally, Figure 4.14 displays the control parameters affecting the COP.

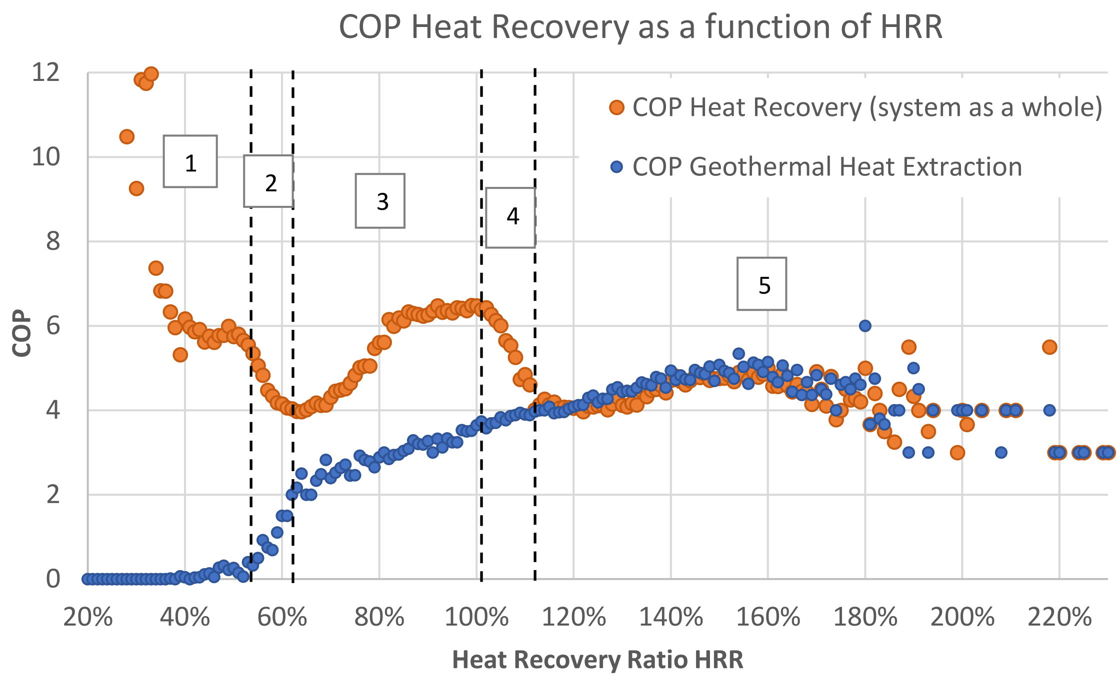


Figure 4.13 COP heat recovery and COP of geothermal heat extraction as a function of the Heat Recovery Ratio.

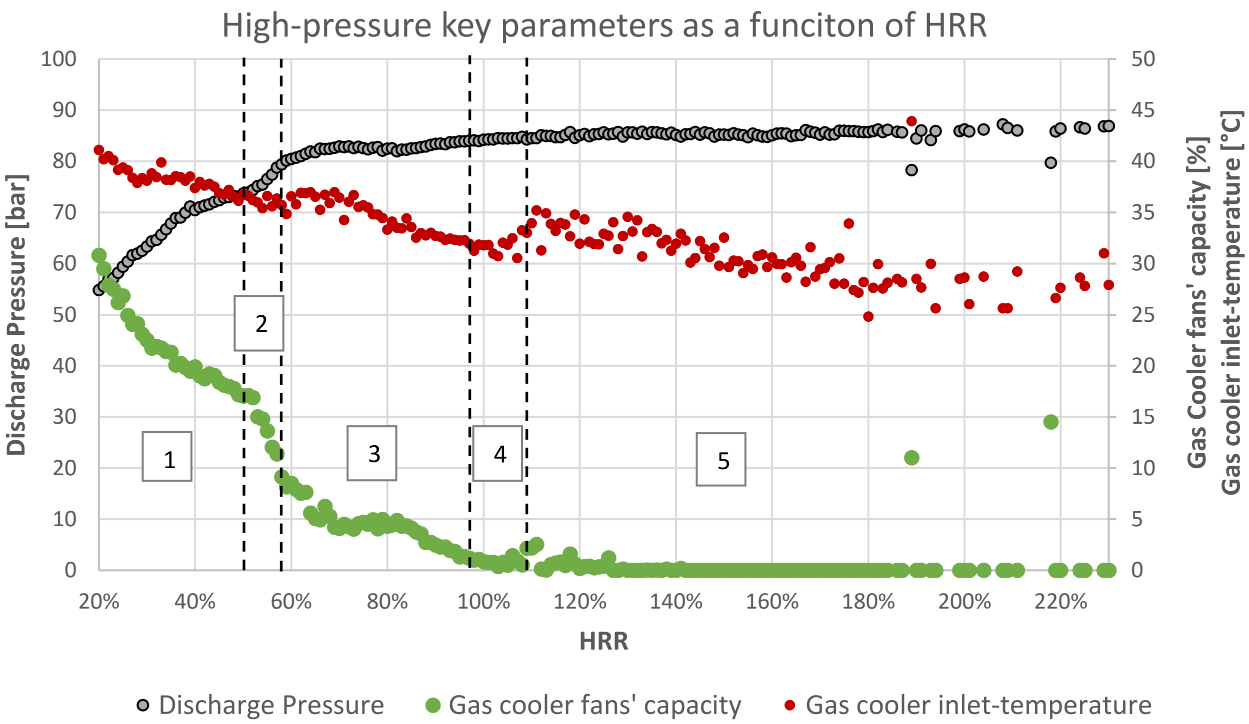


Figure 4.14 Key parameters of the high-pressure side as a function of the Heat Recovery Ratio.

In the first section of the graph, for an HRR between 20 and 52%, the system is operating a subcritical cycle. The COP is relatively high because the heating demand is satisfied almost completely by the waste heat of the floating condensing mode. The second section starts when the system switches to transcritical operation until it reaches a reasonable value for the discharge pressure.

Between 60 and 100% (section 3) the system increases the pressure slowly but steadily reaching the design pressure (roughly 85 bar). In this area, the COP is expected to slightly improve (due to the optimization of the pressure level) but it is not supposed to improve as much as it does. One of the reasons for such a peak is the simultaneous lowering of the return temperature coming from the space heating system (red points in Figure 4.14).

Region 4 is where the system starts extracting heat from the ground (geothermal extraction at its minimum). The losses of performance in this area are due to an increase of the return temperature at the gas cooler inlet. When the geothermal activation is needed, the capacity of the compressors connected to the load evaporator jumps from zero to its minimum value. Such a minimum capacity was found to be too high (in this installation) to ensure efficient heat recovery operations. In other words, when the heating demand needs a few kilowatts more, the parallel compressors are activated at the minimum, but this greatly exceeds the few kilowatts needed.

The last area, region 5, is where the geothermal integration works above the minimum and the gas cooler is not by-passed. Indeed, part of the heat is constantly rejected from the gas cooler (control strategy described in section 4.2.2). With a COP varying between 4.5 and 5, the geothermal heat pump function works in region 5 for almost 245 hours per year, satisfying only the peaks of heating demand. The high HRR reached in region 5 is characterized by an improvement of the COP. This is related to the decrease of the gas cooler inlet-temperature. This is due to a preheating of the outdoor air, used to ventilate the kitchen immediately after that the first and biggest portion of bakery products has been prepared. This causes a drop in the return temperature from the space heating system and subsequently a decrease of the gas cooler inlet temperature.

Finally, in Figure 4.13, it can be seen that the geothermal function is also working for low values of HRR. One can speculate that this happens when the thermodynamic cycle moves quickly from a high to low values of HRR and vice versa (e.g. low heating demand with sudden and short peaks). In these cases, the parallel compressors quickly reach the hourly start-and-stops limit, forcing the boreholes to operate even when the heating demand is low.

4.2.5 Technical considerations

The very first technical limitation when dealing with heat recovery in CO2 transcritical refrigeration units is due to the heat recovery control strategy. All the best theoretical strategies deal with controlling the pressure according to the heat supplied. In practice, there is no-device on the market which can implement such a strategy. The most modern controllers work with temperature sensors and, then, translate such an output in an equivalent heat request. This means that the controller needs to forecast a certain heating demand given the outdoor temperature. The difference between such a forecast and the real heating demand strongly affects the efficiency of the heat recovery. The inefficiencies due to the potential inaccuracy of the forecasted demand represents the main difference between theoretical and real performance.

Regarding the gas cooler, its potential capacity regulation is also very hard to implement in real installations. First of all, a rapid change in the temperature inside the gas cooler (if by-passed) causes a rapid pressure drop in the high pressure branch which needs to be compensated. This effect can be diminished utilizing a gas cooler with two separate internal circuits. In this way, it is possible to by-pass only one of them. Secondly, when the gas cooler is by-passed, a lot of refrigerant accumulated inside it moves into the liquid receiver. If this was not considered during the design phase (receiver volume), some liquid can arrive at the compressors' suction point. A third technical limitation lays in the difficulty of achieving a partial increase of the expansion valve inlet-temperature simply regulating the gas cooler fans power. This is due to the fact that the gas cooler is designed for the "worst" conditions achieved in summer. In other words, when the refrigerant mass flow is significantly higher than the winter conditions. For this reason, the gas cooler is usually over-dimensioned for the winter operations, making it difficult to control.

Considering all these technical constrains in the gas cooler regulation, the control strategy B, described in section 0, facilitates the control of the system. In other words, utilizing such a control strategy is possible to shift the challenge of regulating the gas cooler's capacity to the easier practice of regulating the parallel compressors' capacity. The necessary small extra-energy cost was quantitatively evaluated for the case study and it is presented in the next chapters.

4.3 Geothermal Storage Integration with CO2 Refrigeration Systems in Supermarkets

There are three ways in which a geothermal storage can bring economic benefits to these systems. The first has been extensively treated in the previous chapter and deals with extracting heat from the ground using the system as a geothermal heat pump. Secondly, free space cooling which could potential deliver promising savings. Unfortunately, the monitoring system installed was lacking the key measurement points, necessary for a detailed evaluation of the free space cooling effect. However, given the temperature of the secondary refrigerant circulated in the boreholes and the space cooling system installed in the supermarket, it is unlikely that the storage could be used for providing free space cooling. The temperature in the ground is too high. Finally, the third way of using the storage is to sub-cool the CO_2, during the warmest period of the year.

4.3.1 Subcooling

Figure 4.15 highlights the effect of the geothermal subcooling. The difference between the high-pressure expansion valve inlet-temperature and the gas cooler outlet-temperature is the clear effect of the geothermal subcooling.

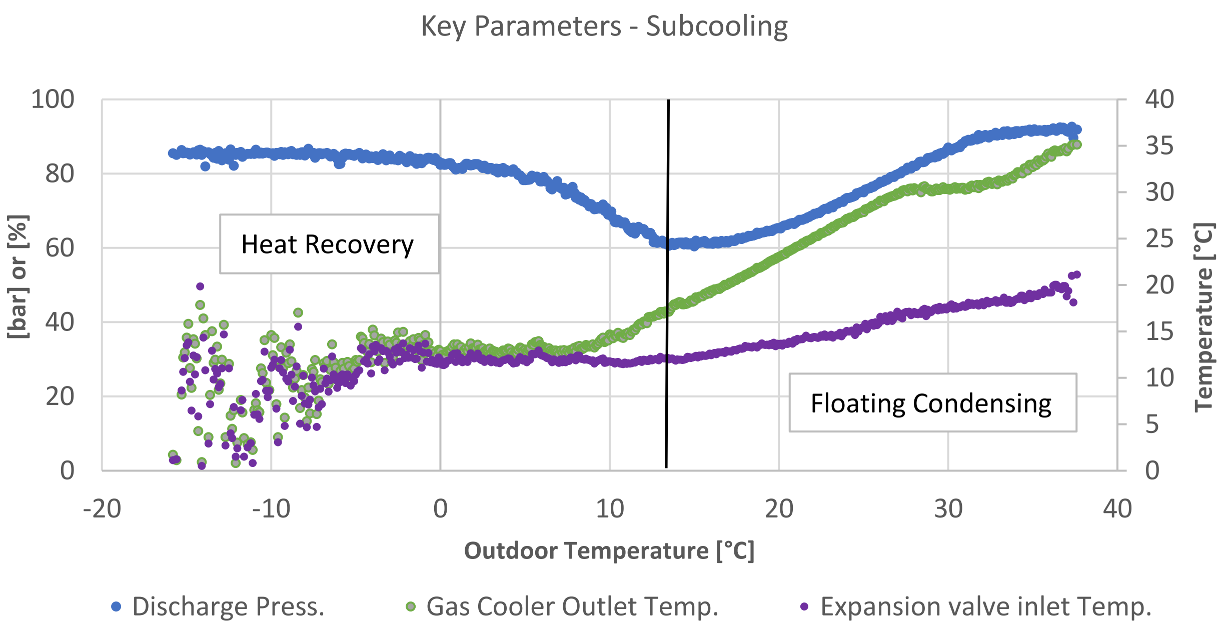


Figure 4.15 Subcooling key parameters plotted as a function of the outdoor temperature. Discharge Pressure (primary axis); Gas Cooler fans' capacity (primary axis) and high-pressure expansion valve inlet temperature (secondary axis).

Generally, the subcooling potential is limited by the boreholes mean fluid temperature. A temporary overheating of the cold source, due to the thermal power dumped into the ground, limits the subcooling potential, during the warmest period. This is why, the high-pressure expansion valve inlet-temperature raises with the increase of the outdoor temperature. It must be said that in the studied installation, this effect is very limited due to an over-dimensioning of the geothermal field.

Figure 4.16 shows the thermal power injected (red line) and extracted (blue line) from the ground as a function of the operational hours. The area delimited between the x-axis and the plotted function represents the total heat injected (roughly 73 MWh) or extracted (roughly 18.3 MWh), in kWh. In this installation, the design power of the heat exchanger used for the geothermal subcooling was 100 kW, however, for 95% of the time the subcooling power does not exceed 45 kW.

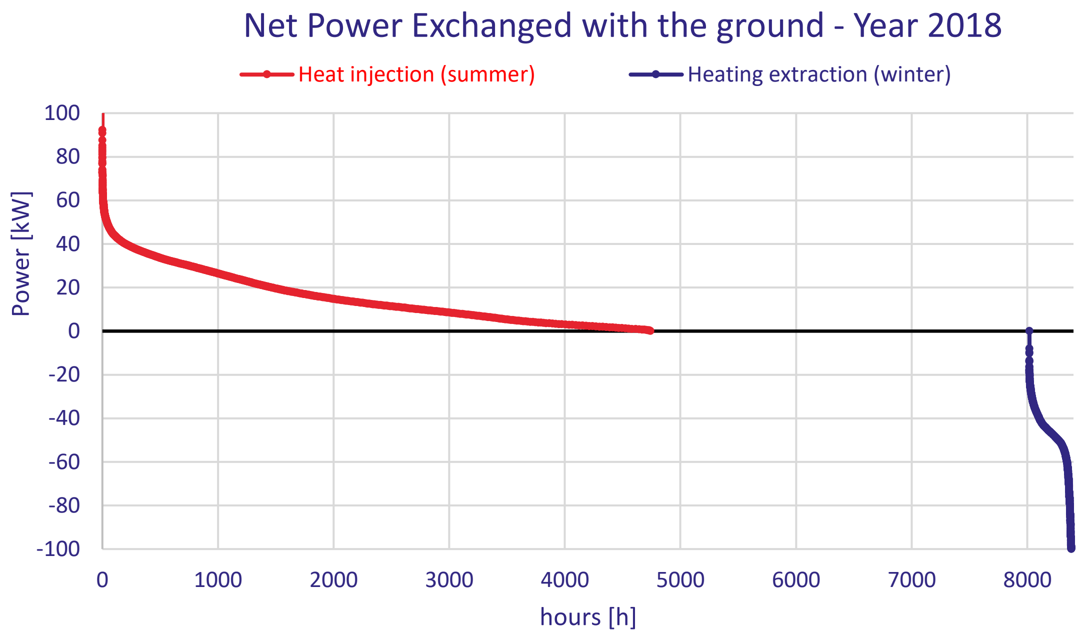


Figure 4.16 Net thermal power exchanged with the ground - Year 2018.

Looking at Figure 4.16, it is clear that the ground is strongly unbalanced. For this reason, in this system, the economic benefits of the geothermal integration are almost totally connected to the subcooling effect. Moreover, it must be said that for almost 3500 hours per year the ground is not utilized and that for roughly 1000 hours the power injected is lower than 5 kW. The usage rate of the boreholes is one of the most important factors when evaluating the economic benefits as will be discussed in the next chapters.

Evaluation of the savings

As well known, subcooling the refrigerant, before the high-pressure expansion valve, leads to savings of compressors' power. The savings were calculated estimating the extra refrigerant mass flow that would have been needed if the subcooling function was not implemented (Giunta, 2020)

The energy saved during the first year of operation is given in Table 4-1, highlighting its share of the total energy consumed by the installation.

Table 4-1 Energy savings due to subcooling.

Month	Electricity Consumption for HVAC&R [MWh]	Total Electricity Savings (due to subcooling) [MWh]	Savings as percentage of monthly electricity consumption [%]
Jan.	25.92	0.00	0%
Feb.	21.68	0.09	0%
Mar.	23.66	0.72	3%
Apr.	23.65	1.89	8%
May	23.36	2.37	10%
Jun.	20.07	2.40	12%
Jul.	35.98	7.51	21%
Aug.	28.97	4.30	15%
Sep.	23.24	3.23	14%
Oct.	23.06	2.02	9%
Nov.	21.06	0.00	0%
Dec.	22.63	0.02	0%
TOT	**293.28**	**24.55**	**8.4%**

4.3.2 Techno-economic Assessment

The aim of this chapter is to explain which are the parameters, in terms of demand profile and climate zone, that most affect the economics of a geothermal integration. With the developed models, it was possible to relate the power savings due to a geothermal integration (summer and winter) to an hourly profile of the ground load (heat injection and extraction). This represents the input for a borehole simulation software called Earth Energy Designer (EED). Through the software it is possible to identify the optimal borehole field size and configuration for a given load. Knowing the relationship between the load on the ground and the savings, a specific field size can be linked to its expected savings. Finally, assuming a reasonable lifetime of the installation and a discount rate, the annual savings can be expressed as a present value.

When evaluating the potential economic benefits of the boreholes' integration, one needs to take into account the difference in OPEX and CAPEX between having or not a geothermal storage. This difference can be referred as ΔNPV and if it is negative, the geothermal storage is the best option. The conceptual representation of the ΔNPV is shown in Figure 4.17. Due to time constrains it was not possible to evaluate the investment cost for the alternatives. Indeed, this strongly depends on the location and on the providers of the other technologies or energy vectors.

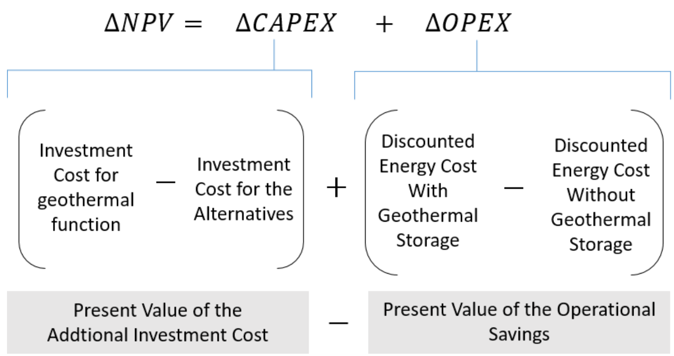


Figure 4.17 Methodology for the techno-economic assessment, Net Present Value of geothermal storage

The data related to thermal loads was obtained as output from the refrigeration system model. The ground properties were retrieved from Sveriges Geologiska Undersökning (SGU), the data for the borehole heat exchanger was not available for the studied supermarket, therefore, it was obtained from previous studies (Mateu-royo et al., 2018) and consultation with experts in this field. The heat carrier fluid is the same used in the real installation and the thermodynamic properties were taken from (Melinder, 2010). The simulation period was set to 15 years and the minimum mean fluid temperatures was set to -6˚C, while several maximum mean fluid temperatures were tested as explained in the next paragraph.

Borehole Field Optimization

For the studied supermarket, due to the geographical location and heating demands, most of the economic benefits of the boreholes are connected to the subcooling function. For this reason, the subcooling capacity was utilized as optimization parameter to find the optimal total length of the geothermal field. The different solutions with the respective savings are shown in Table 4-2. It is worth to notice how negligible is the amount of the extra heat that would be necessary from secondary sources.

Table 4-2 Modelling output: optimized geothermal field size for several subcooling capacities.

Design Subcooling Capacity [kW]	Max Mean Fluid Temperature [˚C]	Number of Boreholes	Spacing [m]	Depth [m]	Subcooling Savings [MWh/year]	Heat Saved from Secondary Sources [MWh/year]
43	13.5	11	11	207	12.0	0.76
40	16	10	11	211	11.5	0.76
33	20	4	10	193	10.2	0.76
25	25	3	5	160	9.1	0.76
15	30	1		188	7.2	0.76

The two types of energy savings need to be transformed into economic value to be summed. For doing this, the electricity price was assumed to be 1 SEK/kWh and 0.8 SEK/kWh was chosen as price for external heat supply (Karampour et al., 2018).

Present Value of the Operational Savings

Once having identified the optimal size of the geothermal field, as well as, the related yearly savings an economic analysis was carried out. The yearly savings were transformed into a present value, using the concept of Equivalent Annual Savings (EAS). When an annuity is constant and no investment is done, the NPV can be expressed as in Eq. 4.2. This formula was used to identify which would have been the present value of the operational savings. The latter represents the present value of 15 years of annual energy savings.

$$Present\ Value\ of\ Operational\ Savings = EAS * \frac{1-(1+r)^{-n}}{r} \tag{4.2}$$

In Eq. 4.2, n represents the expected lifetime of the installation in years, while r is the discount rate. The latter consists of the cost of borrowing capital and it is strictly dependent on the market (food retail), on the country (Sweden) and on the company (Supermarket chain). The life time of the installation (n) was assumed to be 15 years, while the discount rate (r) was assumed to be 7.4% (ICA Gruppen, 2018).

For each of the solutions presented in Table 4-2 a corresponding present value of the operational savings was estimated using the corresponding annual savings. The overall result is shown in Figure 4.18. Each of the solutions brings a label indicating the design subcooling capacity and the maximum mean fluid temperature constrain.

The more the subcooling capacity is reduced, the more beneficial the investment is. This is due to the fact that the "*boreholes' capacity factor*" increases at reduced designed capacities. On the other hand, decreasing the maximum mean fluid temperature enables the system to work at higher subcooling capacities for more hours, but the cost for the additional required length is higher than the additional EAS.

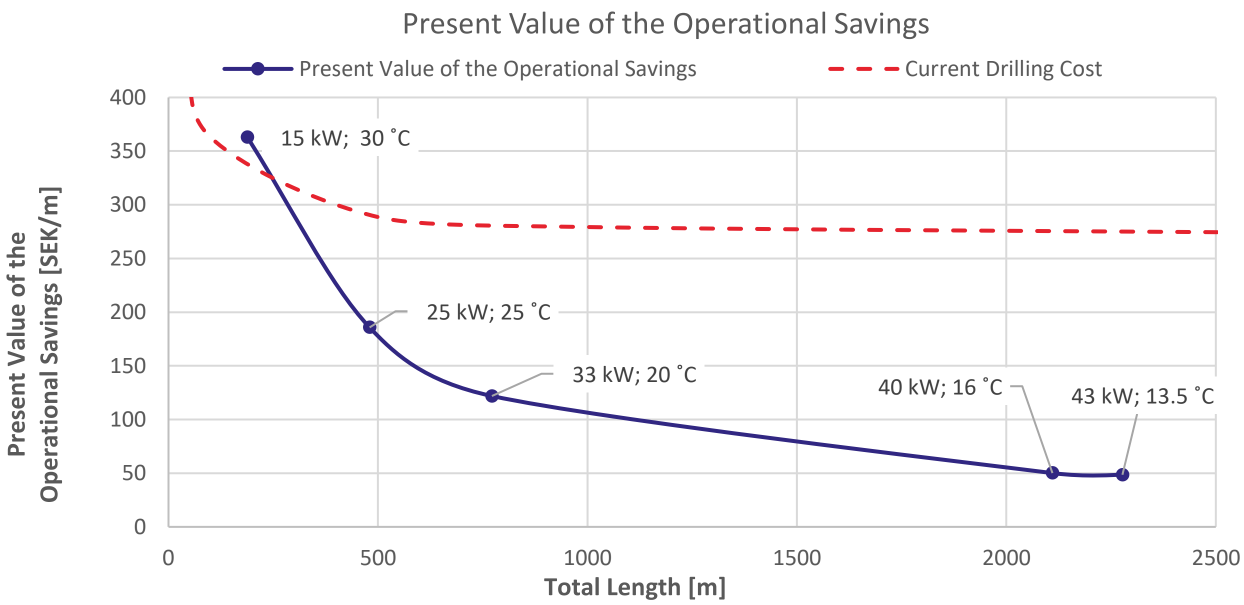


Figure 4.18 Results of the techno-economic assessment: present value of the operational savings for the case study

The cost per meter was chosen as an indicator since, this is the only parameter for determining the investment cost known from literature (Mazzotti et al., 2018) and (Karampour et al., 2018). The average cost per meter in Sweden is roughly 270 SEK/m (Mazzotti et al., 2018) and (Karampour et al., 2018) to this an additional fixed cost of 9300 SEK must be added. The latter is the cost for establishing a medium-size drilling site (Mazzotti et al., 2018).

From Figure 4.18, it is clear that, for the current installation, the only operational savings do not pay back the boreholes' investment cost. One of the reasons, is the ground unbalance. The system is injecting way more heat during summer than what it is extracting during winter. This drives the geothermal field size up. Indeed, this is one of the reasons why the solution with a smaller geothermal storage, are more profitable.

Sensitivity analysis

To scope of this sensitivity analysis, was to identify which parameter affect the results the most. The case study presented in the previous paragraph is used as reference case. The first parameter to be varied was the climate zone, therefore, the BIN hours. The solution for three different climates is presented in Figure 4.19.

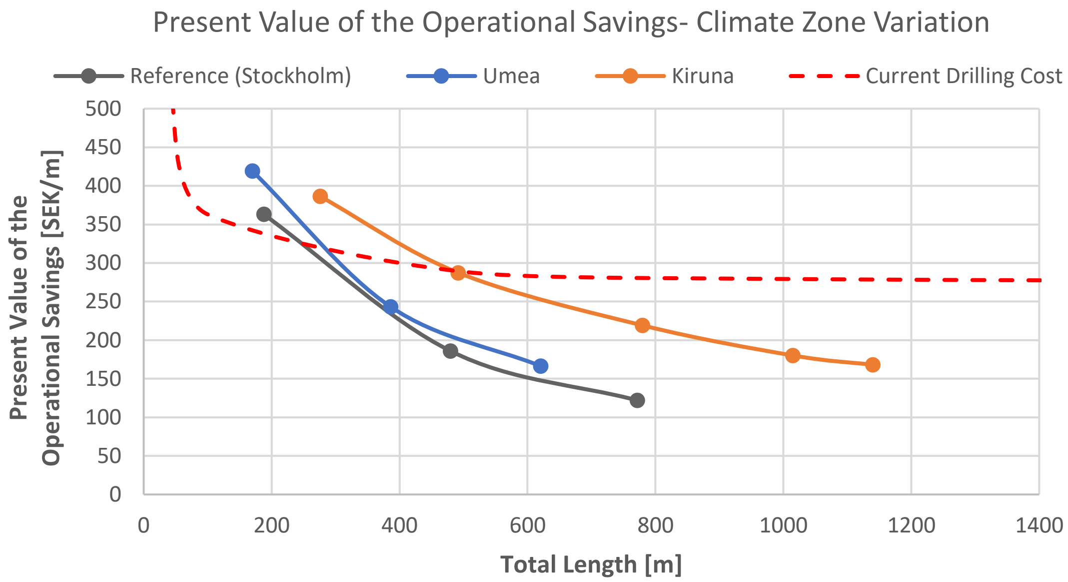
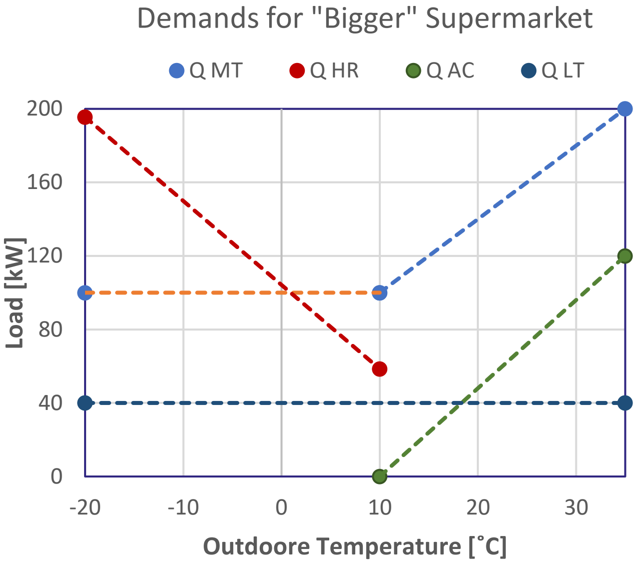


Figure 4.19 1st Sensitivity analysis: present value of the operational savings varying the climate zone

The sensitivity analysis does not take into account that the heating demand as a function of the outdoor temperature could be different for another climate zone. In other words, the study simulates what would be the result if the same building (e.g. same insulation, materials etc…) was virtually moved to another area.

The second sensitivity analysis was performed varying the cooling capacity. This was done, to study the influence of integrating a geothermal storage with a supermarket of a different size. In doing so, the same ratio between cooling and heating loads (HRR) was maintained. Two other systems, that were tested, were characterized by a higher and a smaller cooling demand (respectively displayed in Figure 4.20.

For the "bigger" system the tap water demand was kept the same as in the reference scenario (constant, 21.5 kW) and the space heating demand was set in order to obtain the same heat recovery ratio (HRR). The cooling loads were taken from (Karampour et al., 2018).



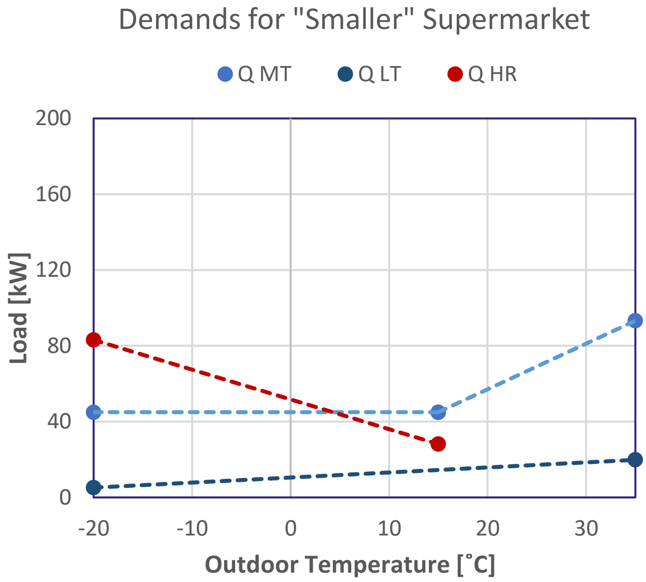


Figure 4.20 2nd Sensitivity analysis: Demands profile for a bigger and a smaller system

For the scenario simulating a smaller supermarket, the heat recovery ratio was kept the same (heating demand calculated accordingly) while the cooling demands were taken from a real system studied in another project. This system was equipped only with one heat exchanger for heat recovery and the tap water demand was almost negligible. Moreover, air condition was not provided by the refrigeration system. The resulting curves with the variation of the present value of the operational savings are shown in Figure 4.21.

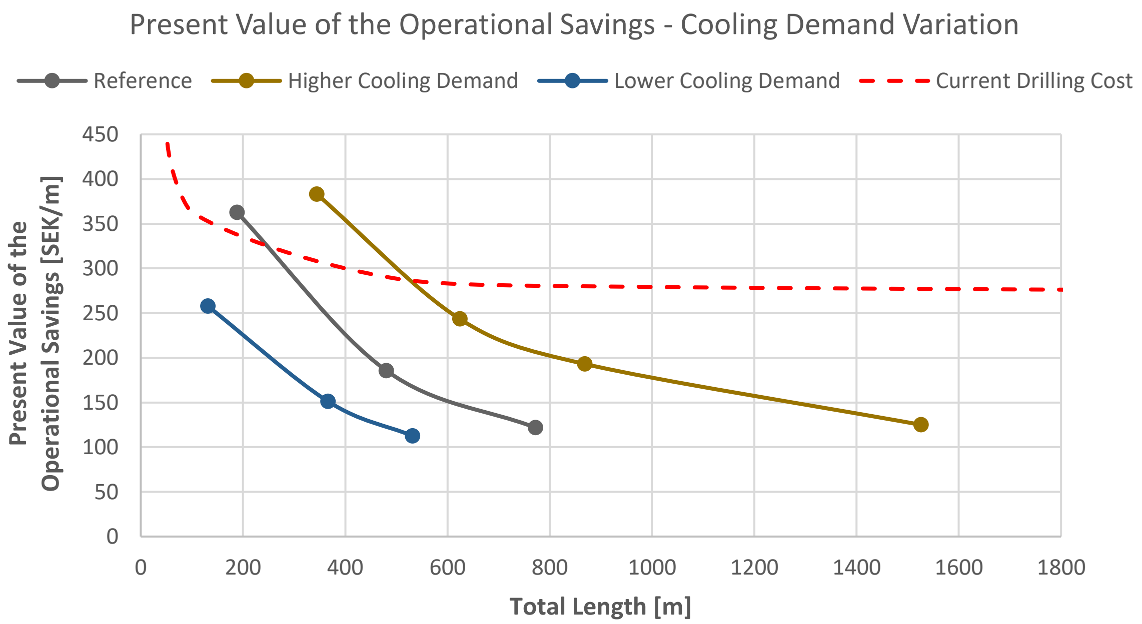


Figure 4.21 2nd Sensitivity analysis: present value of the operational savings varying the supermarket's size.

Finally, the third tested parameter was the Heat Recovery Ratio (HRR). To new scenarios were simulated keeping the same cooling demands but modifying the space heating demand. The percentage values, which are used to name the curves in Figure 4.22, represent the average annual HRR for each curve considering Stockholm's BIN hours.

It is worth to mention that the HRR for outdoor temperatures higher than 13˚C is technically not zero since part of the heat is always recovered for tap water production. However, in order to not consider the weight of the heat provided as waste heat when the system is controlled in floating condensing mode, the HRR was imposed to 0 in summer, to have comparable averages.

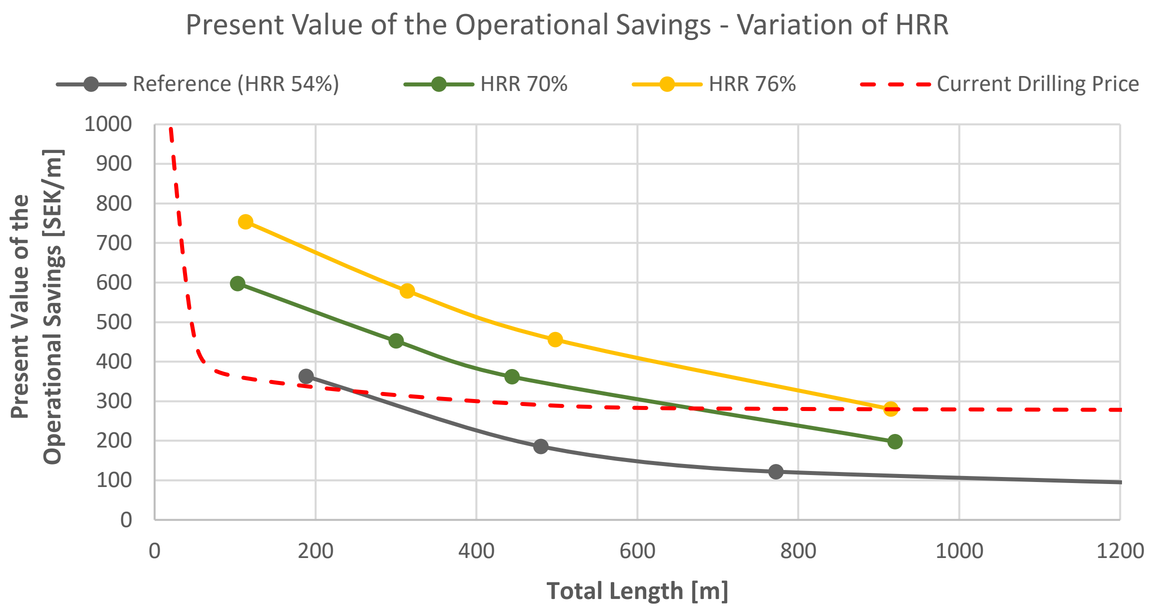


Figure 4.22 3rd Sensitivity analysis: present value of the operational savings varying the HRR's size.

Discussion of the results

The peak design capacity and the temperature constrain used for the maximum mean fluid temperature in the boreholes were found to be the most important design parameters which affects the optimal geothermal storage size.

For this case study, the techno-economic assessment shows that the operational savings obtained by the geothermal integration do not justify the storage investment cost. This is due to the ground unbalance and the small amount of operational hours of the geothermal field.

However, during the design phase, a system that satisfied the heating demand must be foreseen. If a system is equipped with boreholes, in theory it should not need any additional heating capacity. Therefore, the additional cost of the alternatives must be considered when evaluating the economic benefit of the geothermal storage (as shown in Figure 4.17). When taking into account the main alternatives, the heat coming from district heating is a good example. Therefore, the relative connection fee could be one of the cost for the alternatives that can be considered.

Moreover, when a refrigeration system is integrated with geothermal boreholes, the compressors taking care of the flash gas (parallel or MT) can be smaller. This is due to the fact that less flash gas is generated at the design conditions. The price difference between a system with higher compressors 'capacity and a smaller system integrated with boreholes, is a potential cost differential that can be summed to the cost of the alternatives.

Table 4-3 shows a minimum extra-price that could be paid for an alternative system solution (e.g. system with higher compressors ‘capacity connected to a district heating network). In case the cost of the alternative solution was less expensive than these values, the geothermal integration would not represent a favorable option.

Table 4-3 Results of the techno-economic analysis

Design Subcooling Capacity [kW]	Number of Boreholes	Spacing [m]	Depth [m]	Boreholes Investment Cost [SEK]	Present Value of the Operational Savings [SEK]	Minimum Cost for the Alternatives [SEK]
43	11	11	207	624’100	112’400	511’700
40	10	11	211	579’000	107’600	471’400
33	4	10	193	217’700	95’900	121’800
25	3	5	160	138’900	91’000	47’900
15	1		188	60’000	70’000	-10’000

A sensitivity analysis was performed to evaluate in which case, in terms of location and size, a supermarket should consider a geothermal integration. Within the limitations of this work, the outcome demonstrates that a variation of the cooling capacity of a system (for the same HRR) does not greatly affects the profitability of implementing a geothermal storage. In the same way, also the climate zone does not strongly affect the economics of this solution. The third parameter analyzed was the heat recovery ratio (HRR). The outcome demonstrate that this is the parameter affecting the economic benefits the most. In the Stockholm’s climate, financial profits are visible starting from a seasonal average HRR of approximately 70%. An example of a system with a high HRR is a supermarket embedded in a shopping mall which is providing heat to the whole building. The high profitability of this scenario was already mentioned by (Karampour et al., 2018).

4.4 Results of a modelling study comparing alternatives

(Royo,2017) compared three different system layouts to evaluate alternative solutions. The reference system was a stand-alone CO2 transcritical booster system (S1), while the second scenario modelled the same system with the addition of a geothermal storage without parallel compressors (S2). These two scenarios are shown in the left-hand scheme in Figure 4.23. The reference scenario is the system without the green connection, while the S2 layout includes the green lines.

The third configuration (S3) is the one shown on the right-hand side of Figure 4.23. In this case, the refrigeration unit is couple with a ground source heat pump (GSHP). The subcooling function works in the same way as in the S2 layout and in any other geothermally coupled refrigeration solution. However, in this configuration the heat pump increases the temperature of the water used for recovering heat from the refrigeration cycle. Clearly, the heat source for the GSHP is the geothermal storage. The main difference between S2 and S3 is that a higher mass flow can be pumped through the desuperheater in the S3 solution. Indeed, in the third scenario the heat recovery from the refrigeration cycle provides only water pre-heating.

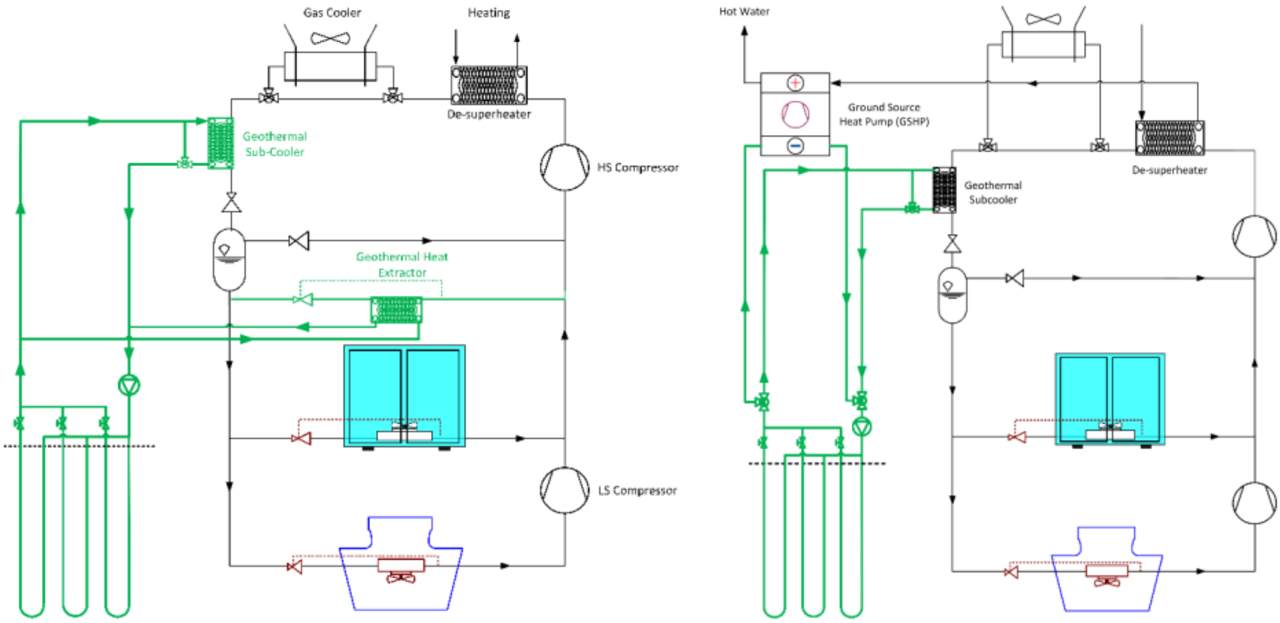


Figure 4.23 Left- Layout of CO2 refrigeration system with geothermal sub-cooler and heat extractor (S2). Right- Layout of hybrid system with CO2 unit and GSHP (S3) (Royo, 2017)

Looking at the differences in the results between the S1 and S2 configuration, the geothermally integrated system (S2) presented a lower annual energy use compared to the stand-alone configuration (S1). This was mainly due to the benefit of subcooling in summer, as confirmed also by the other studies presented in this report. Following, the hybrid solution (S3) showed 6.5% reduction in annual energy usage compared to the stand-alone configuration (S1). This was due to the high efficiency of the GSHP.

5 Conclusions

Monitoring Systems and Available Data

One of the first findings of Royo,2017 was a systematic lack of measurement points, necessary for an energy efficiency assessment. Despite this, a newly built supermarket with a reasonable amount of measurement points was identified and analysed in-depth by (Giunta,2019). To promote the implementation of a complete monitoring system, an instrumentation guideline was created and presented in this report. Regarding ice rinks, two field installations were studied, one of them serving as a good example of key parameters monitoring.

Heat Recovery Control and Optimization

During this project, the best theoretical heat recovery control function was compared with the strategy practically implemented. Such a comparison led to the development of a modified control strategy. According to the theory, only after that the gas cooler has been by-passed, the geothermal function should be activated. However, the modelling performed shows that, from a thermodynamic perspective, there is no-difference if the geothermal heat extraction is activated before increasing the gas cooler-outlet temperature. The only difference lays in the extra-energy spent for the gas cooler fans and for pumping the secondary fluid in the boreholes. It must be said that even if from an energy perspective there is a little difference, this alternative control strategy has important technical co-benefits. The main advantage is to shift from controlling the heat recovery through the gas cooler capacity's regulation to controlling through the compressors' capacity regulation. These considerations are valid for both commercial and industrial applications.

Looking at the performance, the average COP heat recovery ranges from 3 to 8 for outdoor temperatures lower than 7.5°C. For warmer periods, heat is available almost for free thanks to the heat wasted from the refrigeration cycle. In the case of supermarket, the average COP over the whole winter period was 7.5.

Through field data analysis and modelling, it was also demonstrated that existing CO2 systems can be controlled more efficiently. In the case of ice rinks, the developed model considered exactly the same demands and outdoor conditions as in one of the real systems, but with theoretically optimized controls. Using this model for the season between mid-November and the end of March, the electricity saving potential of a control optimization was found to be roughly 12%.

Techno-economics of the geothermal Integration

Overall, the geothermal integration becomes beneficial from a techno-economic perspective, when the winter heating demand is on average much higher than the cooling loads. The results suggest that this is valid for both, ice rinks and supermarkets.

A techno-economic evaluation of a geothermal function was performed for a CO2 system with 2-stage heat recovery in a normal size ice rink, in Sweden. Two different scenarios were evaluated – high activity 12-month season and low-activity 8-month season, representing the extreme cases. The energy savings on an annual basis are roughly 6.8% for a high-activity scenario and 1.7% for a low-activity scenario. In the most interesting scenario, the 12-month season, the savings share for heat pump mode is 71% while the remaining 29% is obtained thanks to subcooling. Moreover, the economic evaluation suggests that in the best-case, the discounted payback period is 16.4 years.

On the other hand, in supermarkets, for the given costs (Swedish market), loads (average size) and boundaries of the system (Swedish climate), the operational savings are found to not pay back the necessary investment for the boreholes. This is due to the fact that the boreholes lead to a negligible amount of energy savings during winter. In a stand-alone supermarket, the refrigeration unit can satisfy roughly 95% of the heating demand without using the ground. Indeed, the main operational savings, related to the geothermal integration in the studied supermarket, were due to the subcooling effect, during the warm season. Generally, for the Stockholm's climate, the energy savings due to subcooling were estimated to be in the order of 5-10% of the total power consumption.

Finally, the sensitivity analysis highlighted that this solution can be extremely beneficial if the system is providing heat to neighbouring facilities (e.g. supermarket embedded in a shopping mall). This is in agreement with the other findings since, in this case the winter heating demand would be much higher than the cooling loads, allowing the CO2 unit to efficiently extract heat from the ground.

It is worth to mention that the profitability of a geothermal storage strongly depends on the cost of the alternatives. The geothermal storage allows to decrease the total CAPEX of refrigeration and back-up system (e.g. district heating connection). These saving can partly or fully cover the borehole investment. Generally, the CAPEX savings are related to the subcooling function and to the heat pump function. Some examples are a lower compressors' size, thanks to the subcooling, and a lower or null district heating connection fee. In more than one case, it was found that avoiding the district heating connection fee was enough to justify the cost of the geothermal field.

Design Best Practice

In terms of designing best practices, there are two possible ways to proceed. The first is to design the geothermal storage to make the supermarket energy independent. This means, to design in a way that the geothermal heat pump function will totally satisfy the peaks of heating demand. The second option is to size the subcooling or heat pump function, optimizing the overall economy (depending on the climate and demands).

The first option is favourable when the alternatives to the geothermal storage are relatively expensive and the CAPEX savings, from these, can cover 70-100% of the boreholes' investment cost. In these cases, the designer does not need to rely on the operational savings to payback the investment.

The second option is favourable for systems where the alternatives are relatively cheap. Indeed, in this scenario, the operational savings (over the lifetime) must payback a great share of the investment cost. Otherwise, the geothermal connection would not be the best option from a techno-economic perspective. The calculation performed shows that an optimized geothermal storage can payback between 45% and 100% of the investment cost, if it is not designed to cover the peaks of heating demand or over-dimensioned with safety margins. Finally, it must be said that every system is unique, in fact, the numerical results presented in this document are system specific. Nevertheless, the concepts and best practices presented are generally valid.

6 Future Work

Due to time constraint several interesting aspects were not investigated. Therefore, they are left for future work.

Alternative artificial cooling loads

During winter the ground is cooled down to provide more heat to the building. In case boreholes were installed only to satisfy the peaks of heating demand, an alternative solution could be used. Indeed, a cheaper option is to utilize an air-to-CO2 load evaporator, after that the gas cooler has been by-passed. A similar solution and relative control strategy is described by (Heerup, 2019).

Techno-economics of parallel compression

More system configurations be investigated to identify the best configuration for connecting geothermal storages. A detailed techno-economic study should be performed to assess if the cost of the parallel compressors justifies higher performance in both heat pump and subcooling mode.

Heat export to district heating

Since the most favourable solutions are the ones with high HRR, an interesting idea is to use to boreholes to increase the heating capacity and inject heat into the district heating network. The efficiency of the heat production could be competitive with the efficiency of the district heating network, considering both production and distribution.

Free-space-cooling

Due to limited amount of data, the free-space-cooling potential was not deeply studied in this work. Boreholes could ideally provide space cooling to a building. Unfortunately, for the common system layouts and operational strategies, it seems that only a negligible amount of capacity can be utilized for free-space-cooling. However, the feasibility of some innovative solutions should be assessed. One of this could be to split the geothermal field, using only part of it for space heating and free-space-cooling. This could lead to a better match of the temperature levels needed for this application.

Heat Recovery Control Strategies

In section 4.24.2.5 the practical challenges of implementing a gas cooler by-pass are mentioned. A field measurements analysis of a system controlling the heat recovery, through gas cooler capacity's regulation, should be performed. Then, the heat recovery performance should be compared to the case (case study 3) presented in this work. Such a comparison would give a better understanding of the technical benefits of regulating the heating capacity only extracting heat from the ground and without by-passing the gas cooler.

In transcritical mode, CO2 rejects heat with a non-linear temperature change. Furthermore, the temperature distribution in a heat exchanger depends on pressure and gas cooler outlet temperature. This affects the location of the pinch point. Because the heating demand and forward-return temperatures are dynamic, finding the optimum CO2 discharge pressure and temperature is a multi-variable task, which is solved in this study with a sophisticated algorithm. The possibility to transform the developed control algorithm, for heat recovery in ice rinks, into a control strategy implementable in a real controller should be further investigated. Furthermore, more case studies are necessary to better quantify the economic benefit of retrofitting the system's controller.

Development of a new modelling method

A unique interconnected model should be developed integrating the geothermal storage model (in this study, performed with EED) and the refrigeration system model (in this study, based on python, VBA or EES). The commercial software TRNSYS is thought to be a good candidate for creating a unique modelling tool.

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