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Thermal Energy Storage Potential in Supermarkets

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Abstract

The objective of this research is to evaluate the potential of thermal energy storage in supermarkets with CO₂ refrigeration systems. Suitable energy storage techniques are investigated and the seasonal storage technology of boreholes is chosen to be the focus of the study. The calculations are done for five supermarket refrigeration systems with different combinations of heating systems and borehole thermal energy storage control strategies. The two heating systems analyzed are the ground source heat pump and the heat recovery from the supermarket's refrigeration system. The simulation results show that the introduction of thermal energy storage in the scenarios with heat pump can reduce the annual total energy by 6.3%. It is also shown that increasing the number of boreholes can decrease the life cycle cost of the system. Moreover, it is established that a supermarket system with heat recovery consumes 8.1% less energy than the one using heat pump and adding thermal energy storage on the heat recovery system further improves the energy consumption by 3.7% but may become costly.

Keywords: Supermarket, thermal energy storage, ground source heat pump, heat recovery, borehole

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Nomenclature

Abbreviations

BTES	Borehole thermal energy storage
COP	Coefficient of performance
CHR	Complete heat recovery
CO ₂	Carbon dioxide
FC	Refrigeration operating on floating condensing mode
HP	Heat pump
HPC	Heat pump, connected to the refrigeration system
HVAC	Heating, ventilation, and air conditioning
Inv	Investment
LCC	Life cycle cost
PCM	Phase changing material

Symbols

AE	Annual energy [kWh/year]
C	Price [SEK]
E	Electrical power [kW]
i	Interest rate
n	Number of years
p	Inflation rate
Q	Heating or cooling capacity [kW]
q	Heat extraction rate [W/h]
R	Thermal resistance [m.K/W]
r	Radius [m]
T	Temperature [°C]
t	Time [s]
α	Thermal diffusivity [m ² /s]
γ	Euler's constant
λ	Thermal conductivity [W/m.K]

Subscripts

AE	Annual energy
bh	Borehole
br	Brine
E	Energy costs
f	Low, freezing temperature level
gc	Gas cooler
hs	High stage
ls	Low stage
M	Maintenance costs
m	Medium temperature level
opt	Optimal
0	Undisturbed ground
1	Condensation level/Heating
2	Evaporation level/Cooling

Chapter 1

INTRODUCTION

1.1 Background

Supermarkets in Sweden are responsible for 3% of the total national electricity use (Kullheim, 2011). Almost half of the energy in supermarkets is used by the refrigeration system, whereas the heating needs can account to 15% of the total annual energy usage of a medium-sized supermarket (Arias, 2005). This intensive energy use urges to make improvements in the efficiency of supermarket refrigeration and heating systems.

The efficiency of refrigeration systems is limited by its boundary conditions. At cold climates the system runs at high efficiency due to low condensing temperatures; but the cold weather increases the heating demand.

Storing thermal energy may change the boundary conditions and adapt them to have an improved efficiency in refrigeration systems.

Figure 1 illustrates the effects of a thermal energy storage on a refrigeration system. On a 24 hour period, the cooling demand reaches a peak, which coincides with high ambient temperature leading to high condensation temperature. This results into a low efficiency operation. Without a thermal energy storage, Figure 1(a), the system has a high load during peak hours when the efficiency is low, whereas on off-peak hours – when the efficiency is high – the refrigeration system operates at minimum load. When a thermal energy storage is introduced, Figure 1(b), the system benefits from the high efficiency periods to charge the storage, which is later used during low efficiency periods to cover the cooling demand. As a result, the peak load is shifted to off-peak hours.

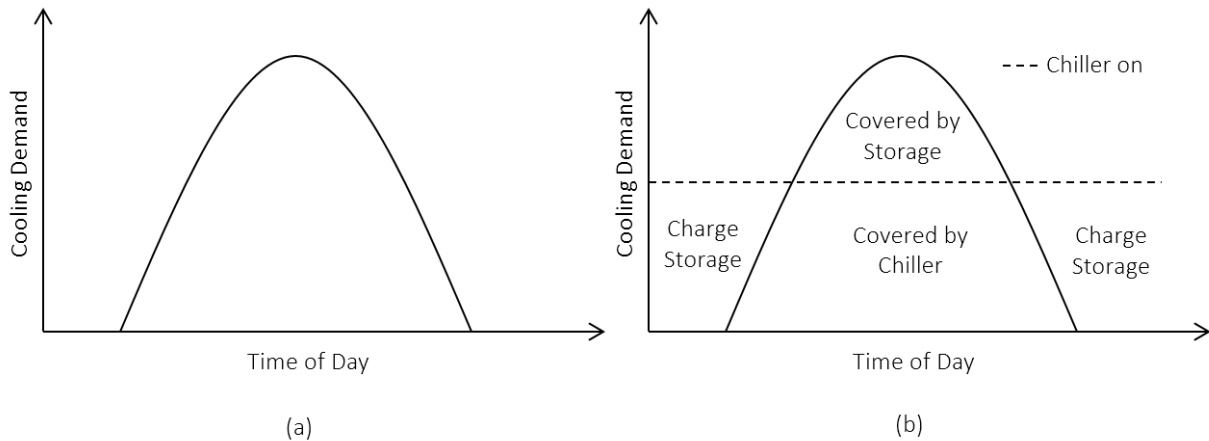


Figure 1: Daily cooling demand (a) without TES, (b) with TES

The same principle is applied with seasonal variations in loads. The winter period is utilized to run the refrigeration system with higher efficiency and charge the thermal storage with cooling energy, which is later used during summer, when the system efficiency is lower. Simultaneously, the thermal energy storage heats up when it is utilized during the summer and thus helps to cover the winter heating load more efficiently.

1.2 Aim of study

The objective of this study is to research the potential of thermal energy storage in CO₂ supermarket refrigeration systems, and to evaluate its influence on the electricity needed to provide cooling and heating.

1.3 Methodology

Several steps are carried out in order to achieve the aim of the study. These steps are:

First, different technologies and possibilities for thermal energy storage in supermarkets are investigated.

Second, several refrigeration system designs and control strategies are selected for study.

Third, a theoretical study is performed on different cases that combine the most suitable thermal energy storage with the selected systems designs and control strategies.

Fourth, the results of the different cases are compared with a reference scenario that operates without thermal energy storage.

Fifth, an economic analysis is carried out to analyze the life cycle cost of the systems with thermal energy storage.

1.4 Literature survey

Various thermal energy storage technologies are feasible to implement in supermarkets. This section examines the literature for both daily and seasonal storage designs that have been applied in supermarkets.

1.4.1 Daily storage

Refrigeration systems with daily storage utilize the low ambient air temperature during nighttime to charge the storage. The efficiently produced and stored cooling energy is later used during daytime when the ambient air temperature is higher and the system efficiency is lower.

Ure & Beggs (1997) suggested using an ice tank storage integrated with the refrigeration and the HVAC systems of the supermarket. The refrigeration produces ice during nighttime and this cooling energy is then used in the HVAC system during daytime. For further benefits, they suggest using ice slurry for storage and as a secondary refrigerant (Ure & Beggs, 1997).

Hägg (2005) studied the usage of ice slurry as a thermal energy storage in supermarkets. The refrigeration system produces the ice slurry which is used as a secondary refrigerant. The study compares the usage of ice slurry against other types of secondary fluids. The main benefits are further cooling provided by the ice slurry, thus the decrease in the power needed from the refrigeration system, the decrease in the refrigerant volume rate, and the decrease in the pipe dimensions (Hägg, 2005).

Calmac is an American company which provides three solutions for daily thermal energy storage in supermarkets. In the first method, during nighttime the medium temperature cycle of the refrigeration system charges an ice tank, where all the heat from the low temperature cycle is rejected during daytime. This method needs a huge volume of ice tank. In the second method, during nighttime the medium temperature cycle of the refrigeration system charges the ice tank, which is used during daytime to sub-cool the condenser output of the low temperature cycle. This method reduces the needed volume of ice tank. In the third method, instead of ice, a PCM tank is cooled by the medium temperature cycle during off-peak hours. The usage of PCM allows to store energy at lower temperatures than ice. During peak hours, the stored cooling energy provides all the cooling needed for the medium temperature cycle (Calmac Manufacturing Corporation, 2002).

Raeisi, et al. (2013) studied the usage of PCM in the freezing cabinets by two methods, pouches and honeycomb shaped. Both methods resulted into much lower compressor cycles. The PCM pouches increased the energy usage by 5%, due to the additional heat resistance created by the pouches, while the honeycomb PCM decreased the energy usage by 2% (Raeisi, et al., 2013)

1.4.2 Seasonal storage

Refrigeration systems with seasonal thermal energy storage utilize seasonal temperature variations to efficiently produce and store cooling energy during winter and use it during summer or efficiently produce and store heating energy during summer and use it during winter.

The main method of seasonal storage is the usage of borehole thermal energy storage (BTES) where the energy is stored in the ground. Titze, et al. (2012) designed a supermarket refrigeration system which is also used as a heat pump. The ground is thus cooled by the medium temperature refrigeration level during the winter and this stored thermal energy is used to sub-cool the condenser output during summer. Figure 2 illustrates the process. The system provides cooling in medium and low temperatures and heating through heat recovery. The two benefits of using a thermal energy storage with this design is the lower expansion valve input temperature, because of the sub-cooling provided by the ground during summer, and the higher heat recovery, because of the heat taken from the ground during winter (Titze, et al., 2012).

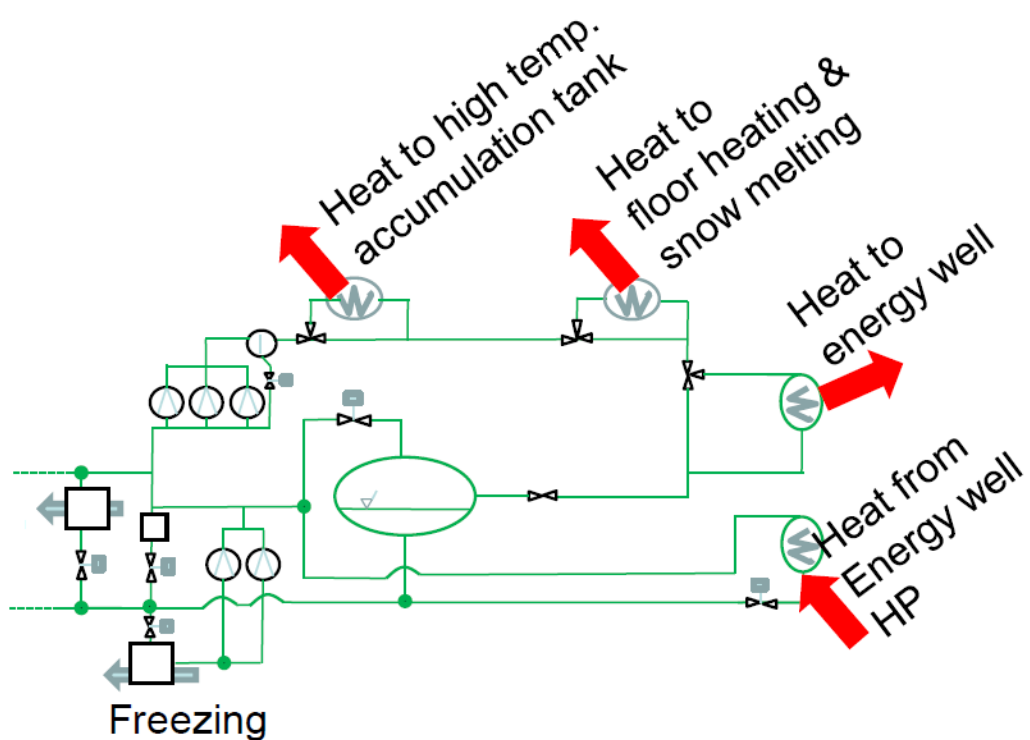


Figure 2: Refrigeration system integrated with BTES (Titze, et al., 2012)

Chapter 2

SIMULATION PARAMETERS

The aim of this study is to evaluate the potential of thermal energy storage in supermarkets. This is achieved by theoretical analysis of a refrigeration system with the suitable thermal energy storage. The theoretical simulation needs several input parameters, among which are the cooling and heating energy needs of the supermarket, the design of the thermal energy storage, the design of the refrigeration system, and the design of the heating system. This chapter describes the input parameters to be used in the simulation.

2.1 Energy profiles

The first input parameter for the simulation is the energy profiles of the supermarket. This includes the medium temperature cooling demand, the low temperature cooling demand, and the heating demand during a year.

The energy profiles are obtained from the CyberMart software, which simulates all types of hourly energy demands in a supermarket, depending on the location, the size, the refrigeration system, and the HVAC system designs (Arias, 2005).

The chosen supermarket is located in Stockholm, Sweden and is a very large one with an area of 14000 m².

Figure 3 shows the hourly cooling energy demand of the supermarket for both medium and low temperature levels as a function of the outdoor ambient temperature. The medium temperature cooling demand is 300 kW during winter and increases to 480 kW during summer. The low temperature cooling demand is rather constant around 65 kW. The points representing low cooling energy demand are caused by the defrosting of the supermarket at the opening and closing hours.

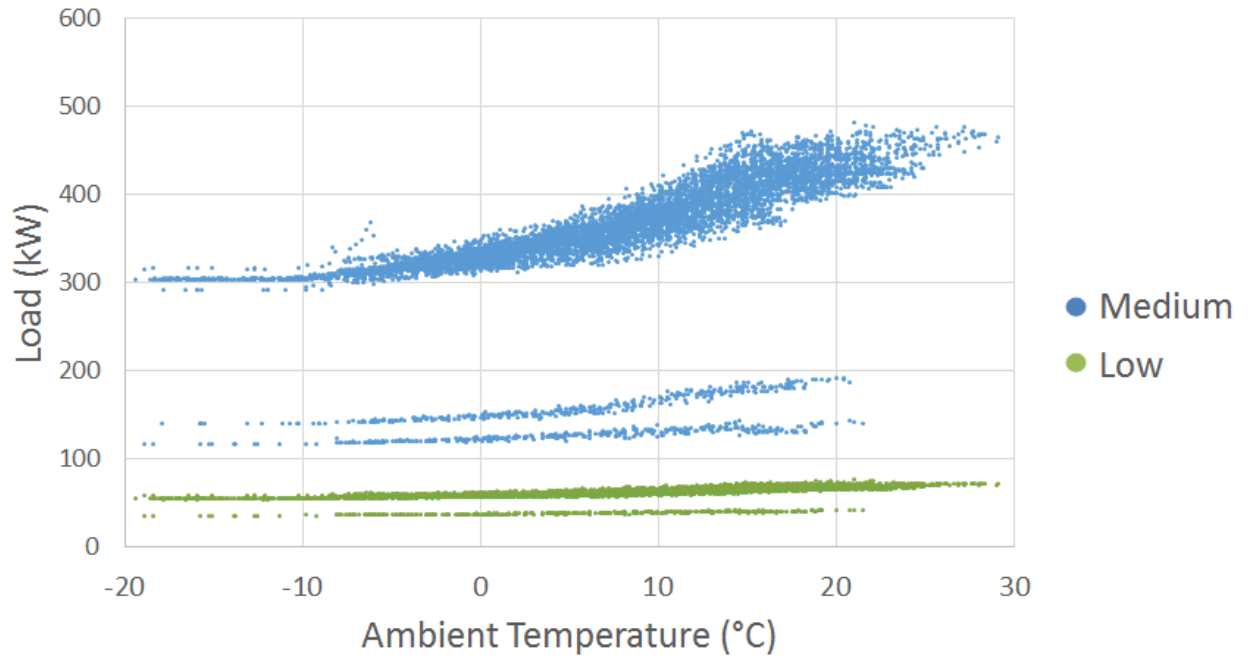


Figure 3: Medium and low temperature cooling energy demand as a function of ambient temperature

Figure 4 shows the hourly heating energy demand of the supermarket as a function of the outdoor ambient temperature. The demand starts when the outdoor temperature drops below 15 °C and increases up to 700 kW during the coldest hours of winter.

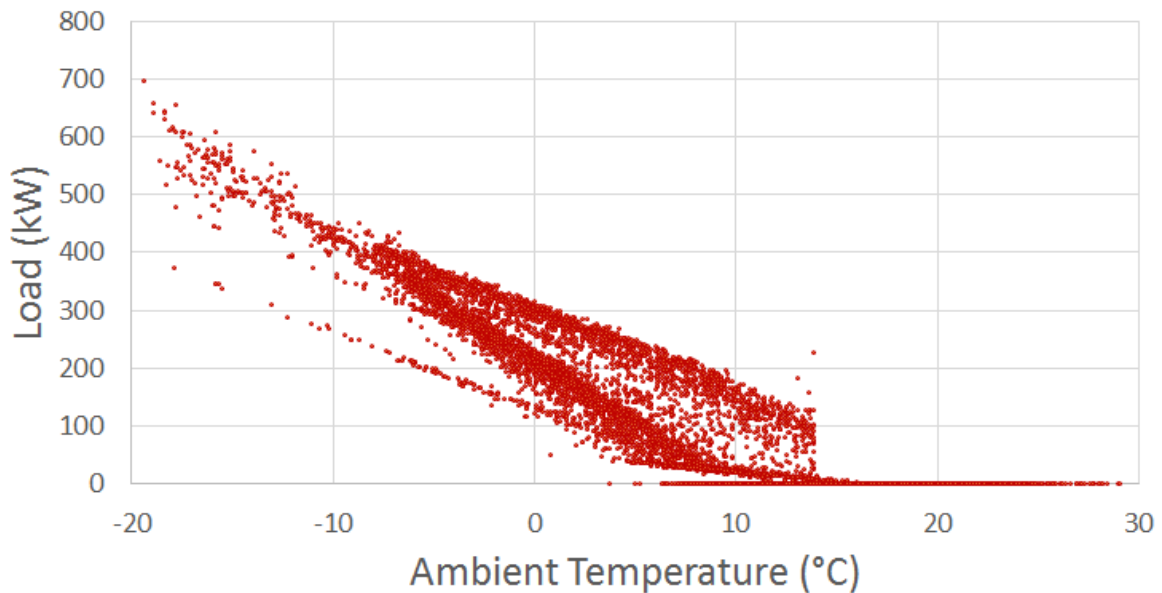


Figure 4: Heating energy demand as a function of temperature

2.2 Thermal energy storage

The study will be focused on seasonal storage, thus this section describes the design of borehole thermal energy storage, which includes the rock properties, the brine properties, and the borehole design properties.

The simulation location is Stockholm, therefore the rock properties of Stockholm are used, with rock thermal conductivity of 3.1 W/m.K, rock density of 2700 kg/m³, rock specific heat capacity of 830 J/kg.K, and an undisturbed ground temperature of 8.6 °C (Acuña, 2010)

The most common borehole brine used in Stockholm is the aqueous ethanol mixture, 25% by weight (Acuña, 2010), which has a freezing temperature of -15 °C (Melinder, 2007).

The borehole itself is filled with water with a thermal conductivity of 0.6 W/m.K, has an active depth of 200 m, and has a simple U-pipe, which has an outer diameter of 32 mm, a wall thickness of 3 mm, and a thermal conductivity of 0.42 W/m.K. As Figure 5 shows, the borehole has a diameter of 110 mm and a shank spacing of 70 mm, the distance between the centers of the pipes.

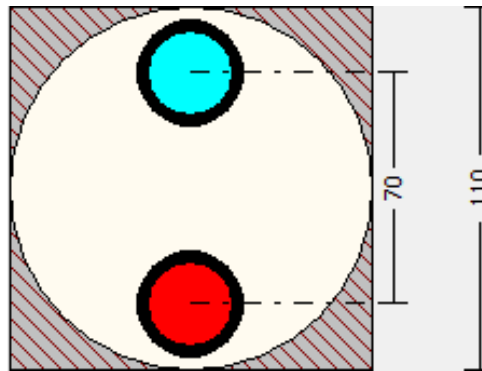


Figure 5: Cross sectional view of a borehole

The Earth Energy Designer software uses the above mentioned properties of the borehole to calculate the effective thermal resistance as 0.148 m.K/W. This resistance is defined between the brine and the borehole wall (Hellström, 1991).

Using the energy profiles described in section 2.1, Earth Energy Designer calculates the necessary number of boreholes to be 20. The limitation is the freezing point of the brine, as the brine temperature should not freeze.

The main variable calculated in the simulation is the brine temperature, which is given by (Eskilson, 1987):

$$T_{br}(t) = \frac{q}{4\pi\lambda_{rock}} \times \left(\ln \left(\frac{4\alpha_{rock}t}{r_{bh}^2} \right) - \gamma \right) + q \times R_{bh} + T_0 \quad (1)$$

In the above equation, q is the heat extraction rate, λ_{rock} is the rock thermal conductivity, α_{rock} is the rock thermal diffusivity, r_{bh} is the borehole radius, γ is Euler's constant, R_{bh} is

the effective borehole thermal resistance as described above, and T_0 is the undisturbed ground temperature.

The heat extraction rate, q , is not constant and is given by:

$$q(t) = \begin{cases} q_1, & t_1 < t < t_2 \\ q_2, & t_2 < t < t_3 \\ \dots & \dots \\ q_n, & t_{n-1} < t < t_n \end{cases} \quad (2)$$

The brine temperature is therefore given by (Monzó, 2011):

$$T_{br}(t) = \sum_{n=1}^N \left(\frac{q_n - q_{n-1}}{4\pi\lambda_{rock}} \times \ln(t - t_n) \right) + \frac{q_N}{4\pi\lambda_{rock}} \times \left(\ln \left(\frac{4\alpha_{rock}}{r_{bh}^2} \right) - \gamma \right) + q_N \quad (3)$$

$$\times R_{bh} + T_0$$

2.3 Refrigeration system

In supermarkets the refrigeration system generally has two levels, medium and low temperature cooling, keeping chilled and frozen products respectively at around 3 °C and -18 °C. Although during the early stages, CO₂ was used only as a secondary refrigerant, currently there exists refrigeration systems that cover both temperature levels, with CO₂ as the only refrigerant, in trans-critical operation. In Sweden, most current installations are designed with the two temperature levels in one single booster system (Sawalha, 2013).

Figure 6 illustrates a booster CO₂ refrigeration system that is used in the simulations of this study. The low temperature level has an evaporation temperature of -30 °C. The evaporator is assumed to have an internal superheating of 10 K and an external superheating of 15 K. The medium temperature level evaporates at -10 °C. The evaporator is assumed to have an internal superheating of 10 K and an external superheating of 10 K. The gas cooler/condenser has an approach temperature of 5 K between the incoming ambient air and the gas cooler output with a minimum condensation temperature of 10 °C and a minimum gas cooler output of 5 °C, after sub-cooling. After the gas cooler, the refrigerant is expanded to a level of 3 bar higher than the medium temperature evaporation pressure. A separator with flash gas bypass separates the liquid from the vapor, which goes directly to the high stage compressor (Sawalha, 2013).

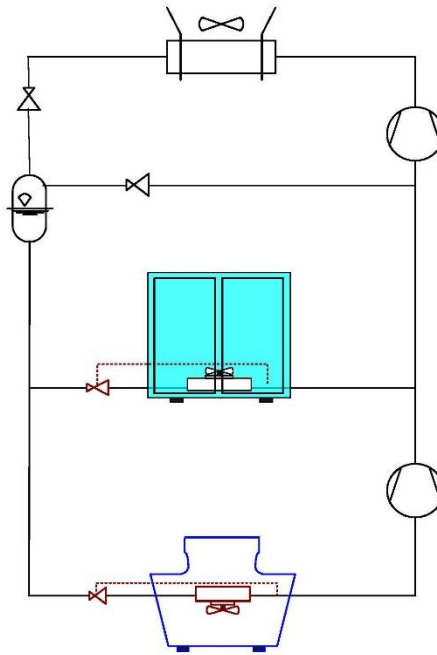


Figure 6: Two stage refrigeration system

In trans-critical operation, the discharge pressure is controlled to the optimal level by (Sawalha, 2008; Liao, et al., 2000):

$$P_{1,opt} = 2.7 \times T_{gc,exit} - 6 \quad (4)$$

The pressure in the above equation is given in bar, whereas $T_{gc,exit}$ is the gas cooler exit temperature in °C.

The compressor total efficiency is assumed to be a function of the pressure ratio, and it follows the curve shown in Figure 7 (Kullheim, 2011).

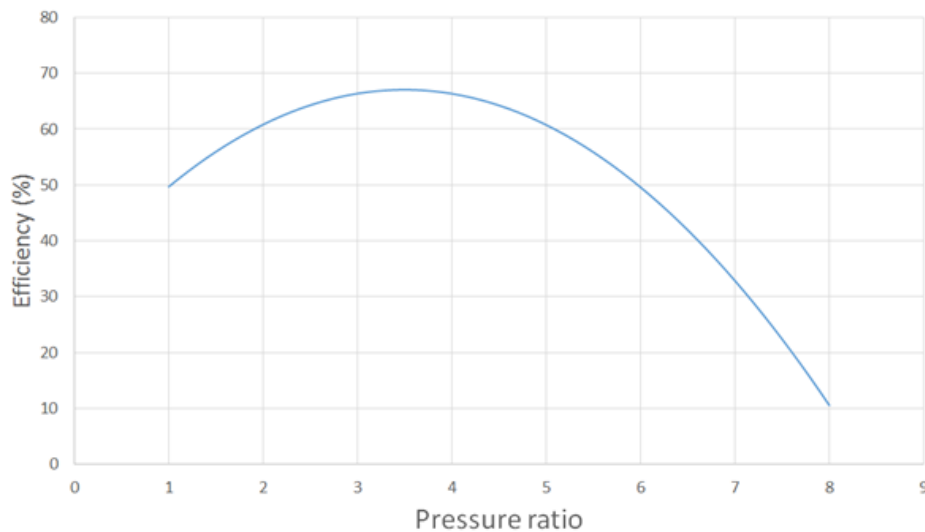


Figure 7: Total efficiency of compressor as a function of pressure ratio

In the refrigeration system presented in this section, the low temperature level does not have a separate condenser/gas cooler, thus all the rejected heat is transferred to the high stage compression. This means that the low stage compression is responsible for part of the electricity needed to run the high stage compressor. This part is given by (Gavarrell, 2011):

$$E_{hs,f} = E_{hs} \frac{Q_{2,f} + E_{ls}}{Q_{2,f} + E_{ls} + Q_{2,m}} \quad (5)$$

In the above equation, E represents the electrical energy needed to run the compressor, Q_2 represents the cooling demand, hs the high stage compression, ls the low stage compression, m the medium temperature level, and f the freezing temperature level.

This factor is reflected in the cooling COP of both medium and freezing temperature levels:

$$COP_m = \frac{Q_{2,m}}{E_{hs} - E_{hs,f}} \quad (6)$$

$$COP_f = \frac{Q_{2,f}}{E_{ls} + E_{hs,f}} \quad (7)$$

The power to run auxiliary fans and pumps, if existing, are not considered in the calculations.

2.4 Heating system

In the simulations, two methods of heating are used in the supermarket. The first is a ground source heat pump designed specifically to cover the heating needs of the supermarket. The second relies on heat recovery to supply the heating demand completely from the refrigeration system. The following sections detail the heating systems.

2.4.1 Heat pump

The first type of heating system simulated in the study is a ground source heat pump with R407C as refrigerant. In order to provide heating at the same temperature level as the heat recovery, the condensation temperature is chosen to be 40 °C. The heat pump has no sub-cooling, 5 K of internal superheating and 5 K of external superheating. The compressor total efficiency is assumed to be 65% while the approach temperature between the borehole brine and the evaporation temperature is assumed to be 10 K. In order to have a fair comparison, the number of boreholes and their design is the same as for the thermal energy storage, described in section 2.2.

2.4.2 Heat recovery

For a refrigeration system running with a refrigerant of high discharge pressure, such as CO₂, the preferred method of heat recovery is with a simple desuperheater after the high stage compressor and before the condenser/gas cooler (Sawalha, 2013). Such a design is suitable for the refrigeration system described in section 2.3.

The desuperheater provides the hot water to the supermarket by exchanging heat with the discharge gas. The hot water return from the supermarket to the desuperheater is assumed to be 30 °C and the desuperheater exit gas temperature to be 35 °C (Sawalha, 2013).

The control strategy detailed by Sawalha (2013) is optimized to have a maximum COP in all conditions while recovering all the needed heat: the discharge pressure is increased just enough to recover the heat demand, while keeping a minimum gas cooler output temperature. When the discharge pressure reaches the maximum, 88.5 bar above which increasing the pressure is inefficient, the gas cooler output is increased just enough to recover all needed heat, from the minimum of 5 °C up to 35 °C. No more heat can be recovered from the system when the discharge pressure reaches 88.5 bar and the gas cooler output temperature is 35 °C.

In order to compare the efficiency of the heat recovery method to the efficiency of the heat pump, the heating COP of the refrigeration system with heat recovery is given by the ratio of heat demand to the power needed to supply the heat (Sawalha, 2013):

$$COP_{1,HR} = \frac{Q_1}{E_{HR} - E_{FC}} \quad (8)$$

In the above equation, Q_1 represents the heat demand, E_{HR} the power needed by the compressors of the refrigeration system with heat recovery, and E_{FC} the power needed by the compressors of the refrigeration system on floating condensing mode, thus without heat recovery. The difference between E_{HR} and E_{FC} results into the power needed to supply the heat demand.

Chapter 3

SIMULATION CASES

In the theoretical study, several case scenarios are devised to show the effect of thermal energy storage on the energy consumption of the supermarket. The cases are combinations of different heating systems and control strategies of the storage. Overall there are 5 scenarios, detailed below.

3.1 Floating condensing and heat pump (FC+HP)

The first case is formed of separate refrigeration and heat pump systems. The refrigeration runs on floating condensing mode and is not connected to the ground source heat pump, as illustrated in Figure 8. The number of boreholes is 20, as explained in section 2.2. For a fair comparison, the same number of boreholes is considered for the remaining cases.

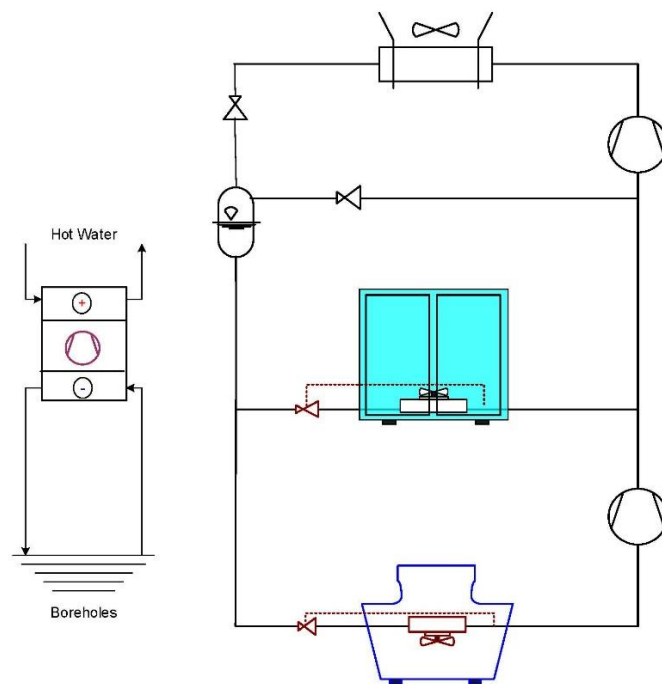


Figure 8: Floating condensing and heat pump (FC+HP)

This case, abbreviated as FC+HP, is the reference scenario and will be compared to all the following cases.

3.2 Floating condensing and connected heat pump (FC+HPC)

In the second case, the refrigeration system running on floating condensing mode is connected to the ground source heat pump as Figure 9 shows. The borehole brine, after being cooled down by the heat pump, sub-cools the gas cooler output in the refrigeration system. This sub-cooling is only done when the heat pump is turned on and is supplying the heating needs.

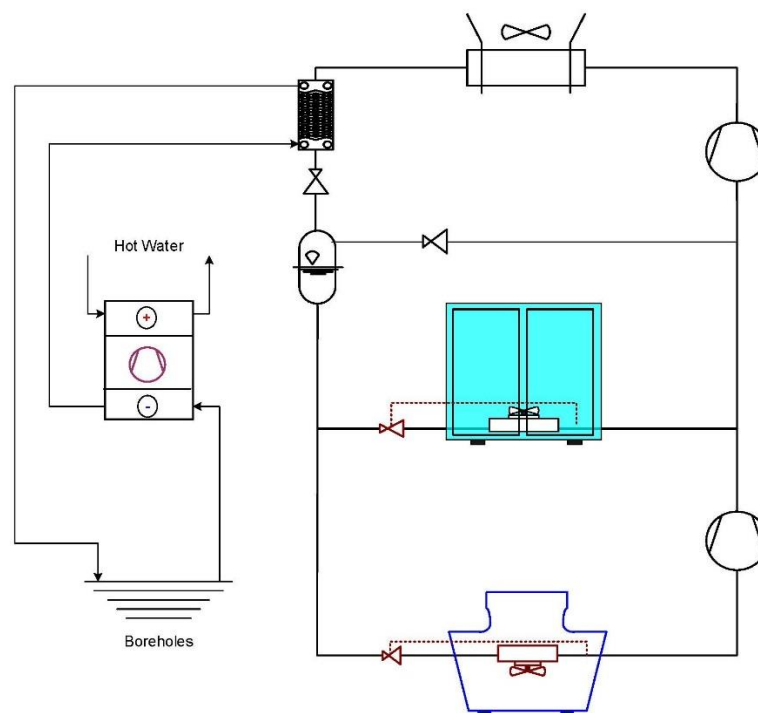


Figure 9: Floating condensing and connected heat pump (FC+HPC)

This case, abbreviated as FC+HPC, is an existing commercial design and is therefore of interest.

3.3 Floating condensing with borehole thermal energy storage (FC+BTES)

In the third case, the boreholes are performing as thermal energy storage. As in the previous case, when the heat pump is running, the brine, after being cooled down, sub-cools the gas cooler output in the refrigeration system. As a modification from the previous case, the borehole brine continues to sub-cool the gas cooler output even when the heat pump is not turned on.

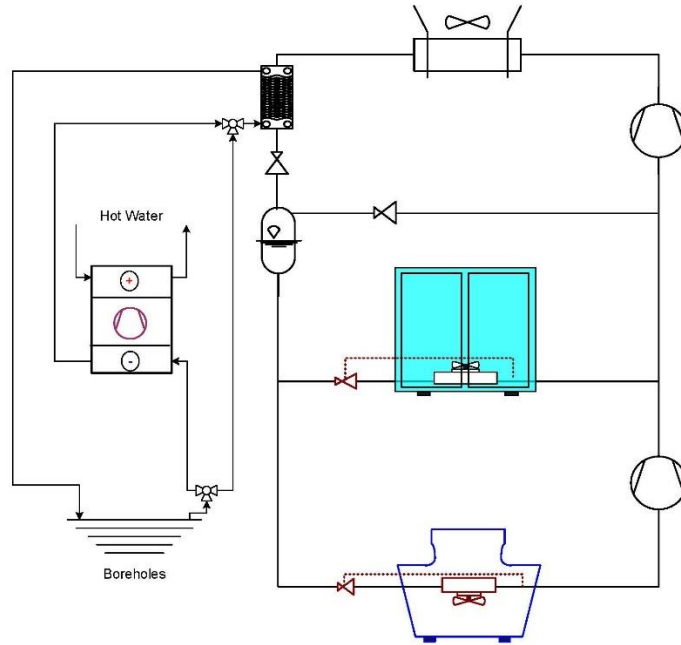


Figure 10: Floating condensing with orehole thermal energy storage (FC+BTES)

The third case, abbreviated as FC+BTES, is shown in Figure 10.

3.4 Complete heat recovery (CHR)

The fourth case consists only of the refrigeration system. The heating is provided by heat recovery, shown in Figure 11.

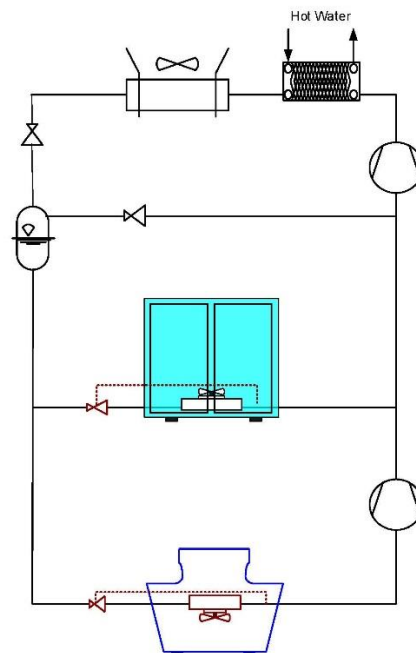


Figure 11: Complete heat recovery (CHR)

With the control strategy described in section 2.4.2, the heat recovery supplies 99.37% of the heating demand. This case, abbreviated as CHR, is the only one without any boreholes and is considered as a reference heat recovery scenario.

3.5 Complete heat recovery with borehole thermal energy storage (CHR+BTES)

The final scenario adds a borehole thermal energy storage to the previous case, where the heating is provided completely by the heat recovery. As modeled in Figure 12, the medium temperature level evaporator cools the ground during the colder days of winter, when the heating demand is higher than 300 kW. The stored cooling energy is then used to sub-cool the gas cooler output during the summer.

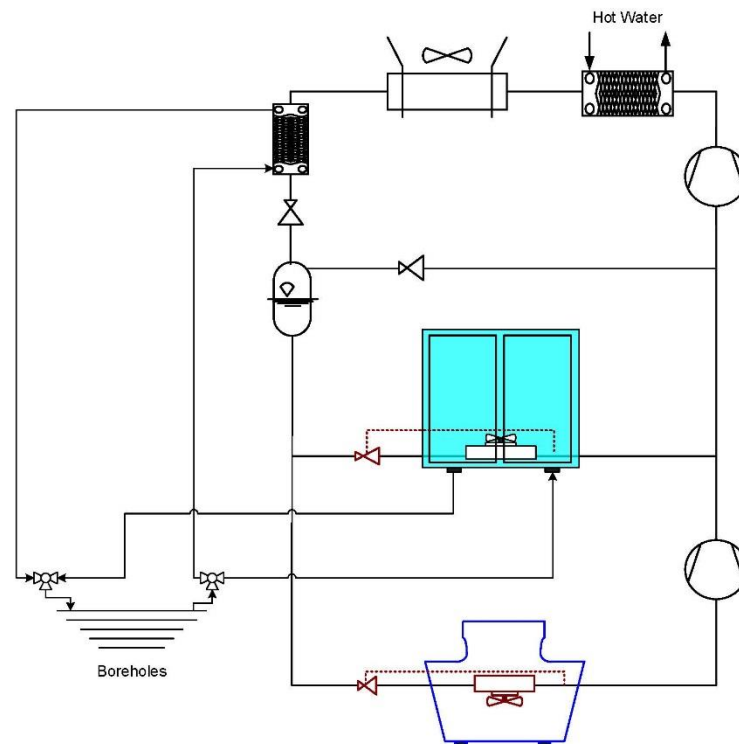


Figure 12: Complete heat recovery with borehole thermal energy storage (CHR+BTES)

This scenario, abbreviated as CHR+BTES, is similar to the system described in section 1.4.2, and if more heat is needed, the system control can be adjusted to take more heat from the ground thus increasing the heat recovery.

Chapter 4

SIMULATION RESULTS

With the input parameters described in Chapter 2, a simulation is carried out on the performance of the scenarios presented in Chapter 3, over a period of 10 years. The simulation results include hourly values of temperature, pressure, and enthalpy of interest points in the system cycles, the power needed to run the compressors of the refrigeration and the heating systems, the cooling COP for the two levels of temperature, the heating COP, and the temperature of the borehole brine.

The following sections compare the results of the simulations of each scenario to the one of the reference case. The graphs presented below show the cooling COP of the medium temperature level as a function of the ambient temperature, the heating COP as a function of the heat demand, and the combined power needed to run the refrigeration and the heating systems as a function of the ambient temperature. The cooling COP of the low temperature level follows the same pattern as the COP of the medium temperature level.

Although the simulation is run over 10 years to take into account the temperature changes in the ground, the results shown in the following sections present only the values of the last year.

4.1 FC+HP vs. FC+HPC

The first section compares the reference case FC+HP, where the refrigeration and the heat pump are separate, to the case FC+HPC where the brine from the heat pump sub-cools the gas cooler output in the refrigeration system. The objective of this comparison is to show the effect of connecting the heat pump to the refrigeration system, only during winter.

The cooling COP of the medium temperature level is shown in Figure 13. It can be seen that the reference case, FC+HP, has a constant COP on cold days, until 0 °C. There is a small drop of COP until 5 °C, above which the COP decreases sharply. This is because the condensation temperature is held at 10 °C when the temperature drops below 5 °C. Further sub-cooling of the refrigerant is possible until a minimum of 5 °C while keeping an approach temperature of 5 K with the ambient air, as detailed in section 2.3.

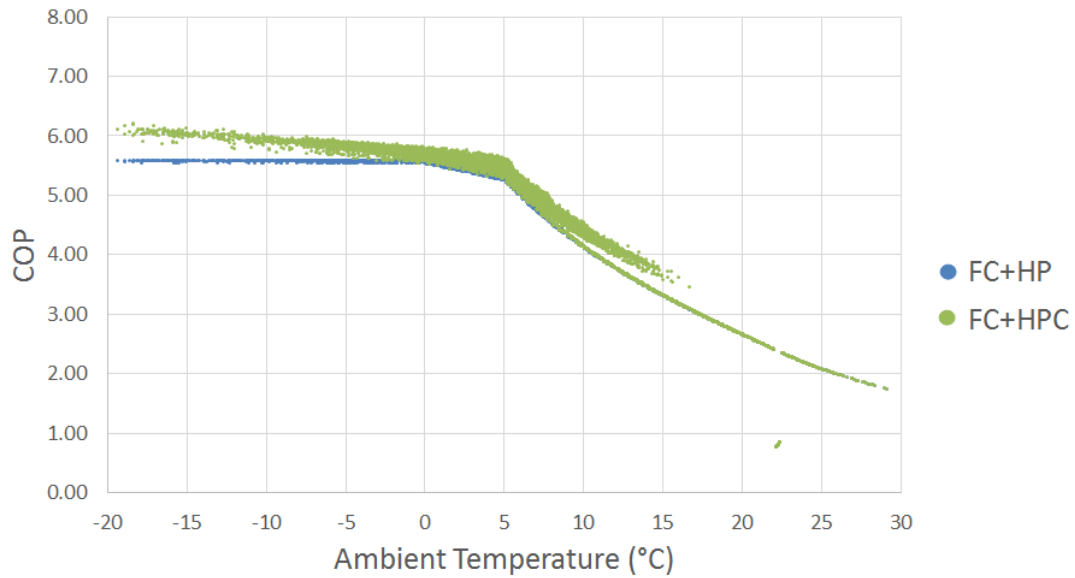


Figure 13: Cooling COP as a function of ambient temperature, FC+HP vs. FC+HPC

Figure 13 shows that FC+HPC has a higher COP than the reference case below ambient temperatures of 15 °C. Above this temperature, the two curves overlap and show the same COP for both cases. The reason for the improvement of the COP is that below 15 °C there is a heat demand and the heat pump starts operating, which leads to sub-cooling of the gas cooler output and a better COP (Granyrd, et al., 2011).

As the ambient temperature decreases, the heat demand increases, and the borehole brine temperature further decreases. This leads to further sub-cooling, and an increase in the cooling COP of the refrigeration system.

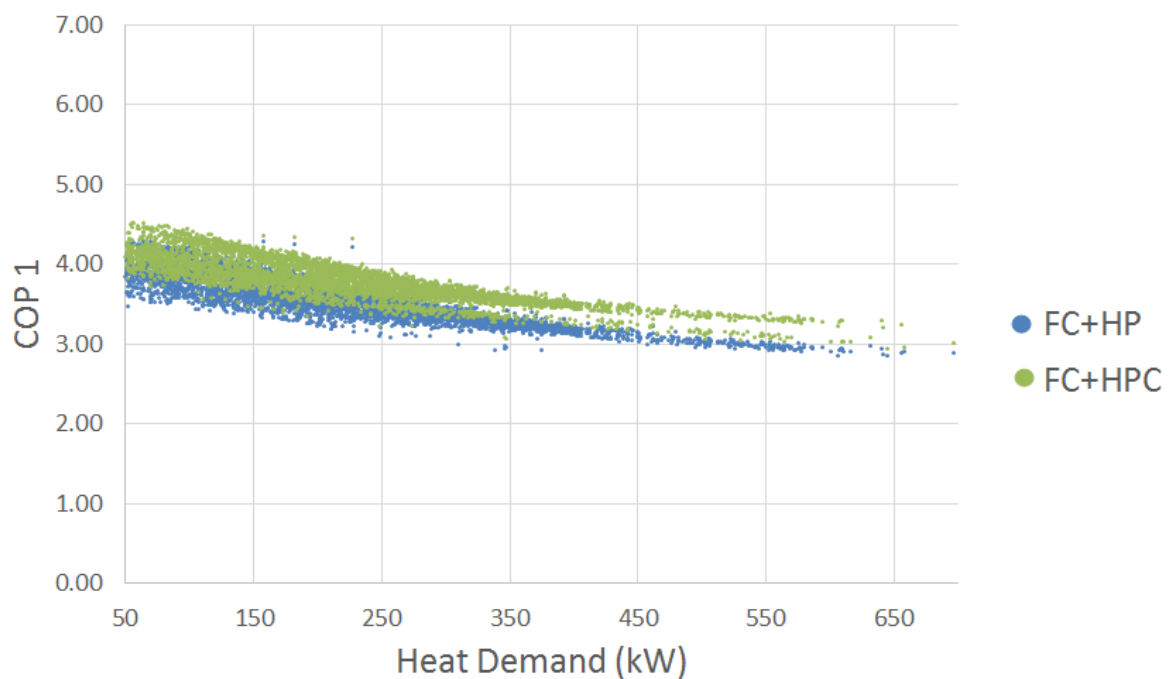


Figure 14: Heating COP as a function of heat demand, FC+HP vs. FC+HPC

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The heating COP as a function of the heat demand is shown in Figure 14 for both systems. As the heating demand increases, the mass flow of the refrigerant increases, which leads to a higher power consumption and a lower heating COP.

Comparing with the reference case, FC+HPC has a higher heating COP, because as the borehole brine sub-cools the gas cooler output in the refrigeration system, the brine itself is heated up, which leads to a higher evaporation temperature in the heat pump and a higher heating COP.

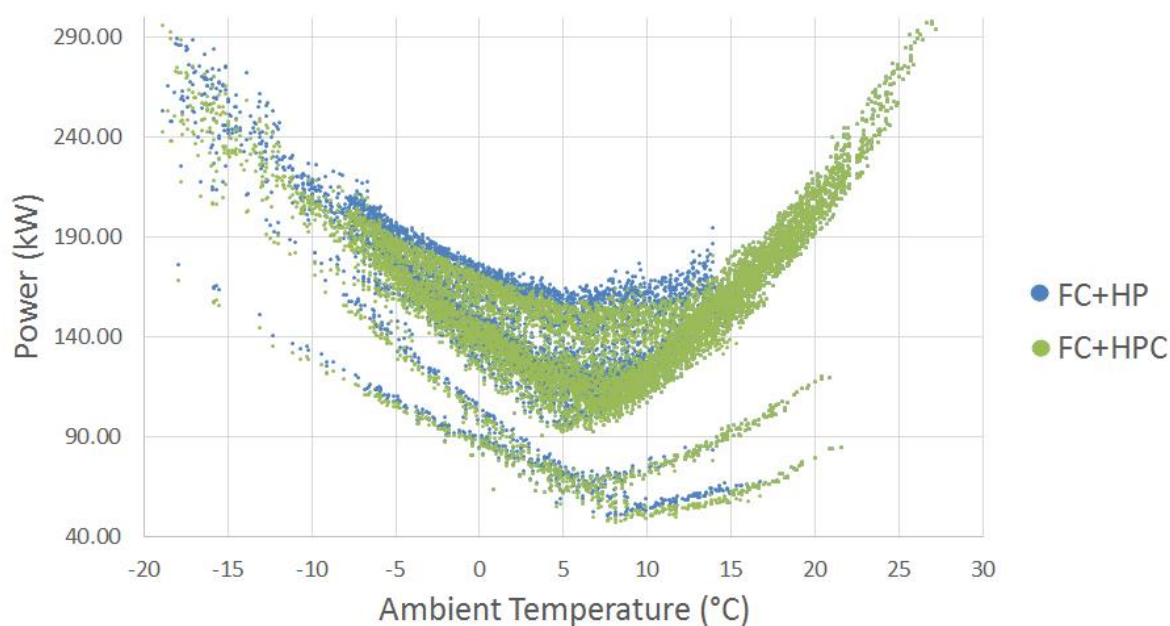


Figure 15: Total power needed as a function of ambient temperature, FC+HP vs. FC+HPC

The total power needed to operate both the refrigeration system and the heat pump is shown in Figure 15. Below ambient temperatures of 0 °C, the power needed to run the refrigeration is almost constant because the cooling loads for both temperature levels are almost constant, as shown in Figure 3, and the COP is constant, as shown in Figure 13. Therefore, it can be deduced from Figure 15 that the power needed to operate the heat pump decreases as the ambient temperature increases and as the heating demand decreases. Above ambient temperatures of 15 °C, the heat pump is turned off and the power shown in the graph represents the power needed to run the refrigeration system in floating condensing mode.

Comparing with the reference case, FC+HPC has a lower power need to run both the refrigeration and the heat pump systems, when the ambient temperatures are lower than 15 °C. This is because the refrigeration system runs more efficiently when there is sub-cooling, as shown in Figure 13, and so does the heat pump because of higher evaporation temperatures, as shown in Figure 14.

Overall, FC+HPC has an annual total energy usage of 1339 MWh, 2% less than the reference case. The refrigeration system uses 1004 MWh/year, 1.8% less than the reference case, and the heat pump uses 335 MWh/year, 2.8% less than the reference scenario.

4.2 FC+HP vs. FC+BTES

The second comparison with the reference case is FC+BTES, where the refrigeration system is connected to the heat pump and the boreholes operate as a thermal energy storage. As a difference from FC+HPC, FC+BTES has sub-cooling of the gas cooler output even in the summer, when the heat pump is not running. The objective of this comparison is to show the effect of thermal energy storage on a refrigeration system operating in floating condensing mode.

The medium temperature cooling COP for the two scenarios is shown in Figure 16. It can be seen that the scenario with the borehole thermal energy storage has a better cooling COP for all ambient temperatures. As explained for the FC+HPC scenario in the previous section, the COP improvement is greater as the ambient temperature decreases. The reason for this is that for low ambient temperatures the heat demand increases, the load on the heat pump increases and the borehole brine temperature decreases. This allows further sub-cooling in the refrigeration system, which improves the cooling COP.

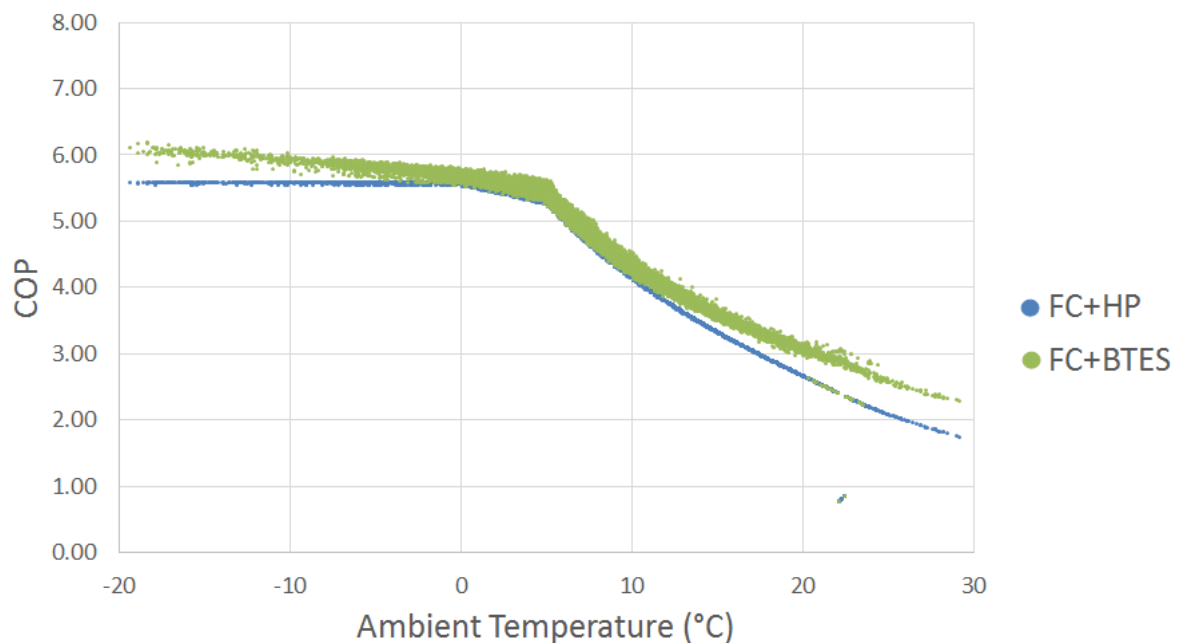


Figure 16: Cooling COP as a function of ambient temperature, FC+HP vs. FC+BTES

As a difference from FC+HPC, FC+BTES has a better COP than the reference scenario even when the ambient temperatures are high. This is because the borehole thermal energy storage provides sub-cooling to the refrigeration system even when the heat pump is not operating.

It can also be noted that as the ambient temperature increases, the improvement in the cooling COP gets larger. The reason for this is that high ambient temperatures force the condensation temperature to be higher and this has a negative effect on the COP. But with

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sub-cooling provided from the thermal energy storage, it is possible to lower the expansion valve inlet temperature and have a better COP.

The heating COP of the two systems as a function of the heating demand is shown in Figure 17. As seen with the previous comparison, FC+BTES has a higher heating COP, because when the borehole brine sub-cools the gas cooler output in the refrigeration system, the brine itself is heated up, which leads to a higher evaporation temperature in the heat pump and a higher heating COP.

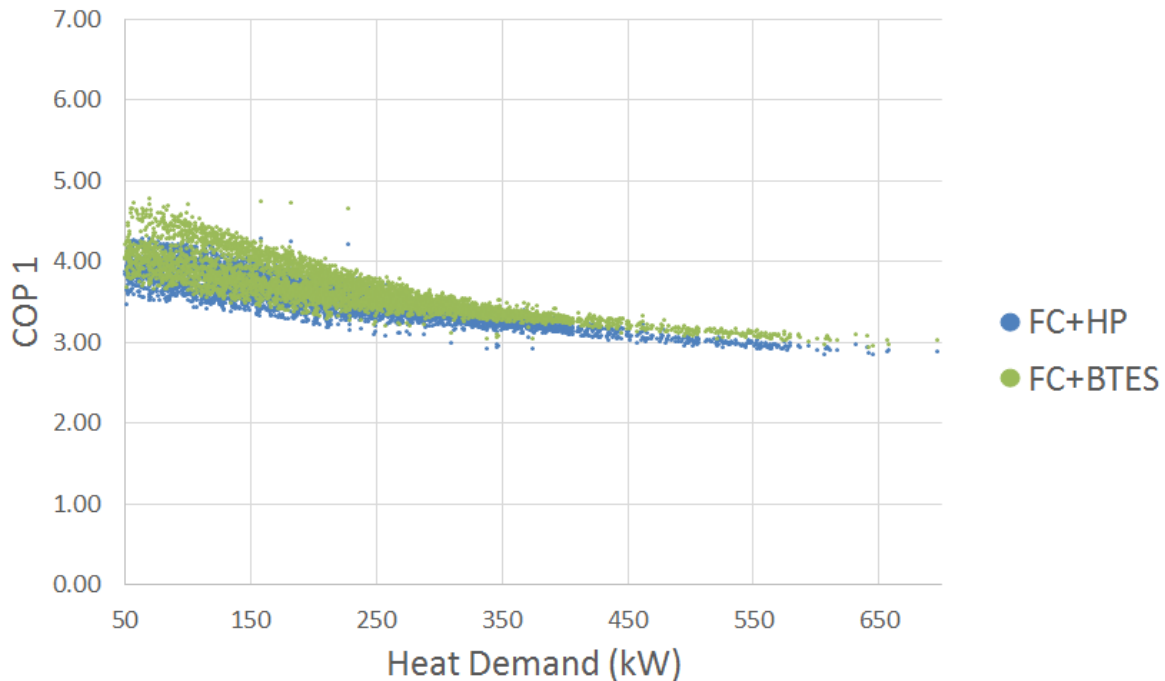


Figure 17: Heating COP as a function of heat demand, FC+HP vs. FC+BTES

The total power needed to operate the refrigeration and the heating systems for both scenarios is presented in Figure 18. It can be seen that FC+BTES has a lower power need for all ambient temperatures. For temperatures below 10 °C, both the heating and the cooling COP's of FC+BTES are higher than the reference case. This infers a lower total power need to operate the heat pump and the refrigeration system.

In the previous comparison, the power needed to run the refrigeration system above 10 °C was the same for both scenarios, but in this comparison, FC+BTES has a lower power need to operate the refrigeration system as the cooling COP is higher. The difference in the power load intensifies as the ambient temperature increases. The effect of the thermal energy storage and its sub-cooling is shown clearly in these temperatures.

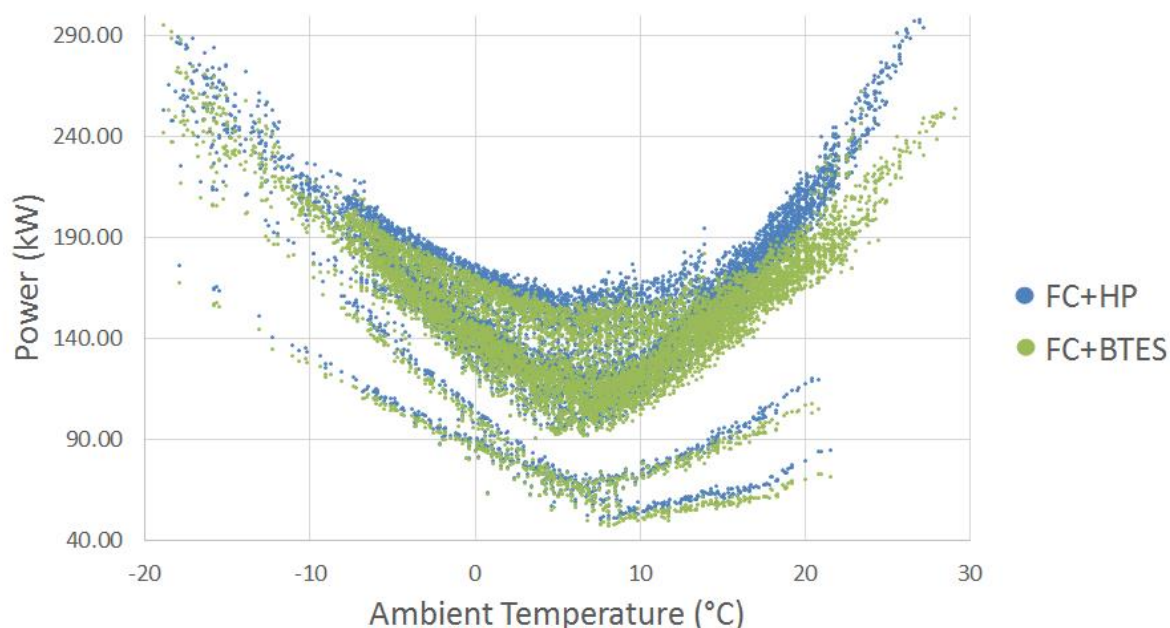


Figure 18: Total power needed as a function of ambient temperature, FC+HP vs. FC+BTES

Overall, FC+BTES has an annual total energy usage of 1281 MWh, 6.3% less than the reference case. The refrigeration system uses 948 MWh/year, 7.3% less than the reference case, and the heat pump uses 333 MWh/year, 3.5% less than the reference scenario.

4.3 FC+HP vs. CHR

In the third section, FC+HP, the reference scenario with separate refrigeration and heat pump systems, is compared to CHR, the scenario with complete heat recovery and no boreholes. As described in section 3.4, CHR does not include any boreholes and supplies the heating needed through a heat recovery system, controlled for highest COP. This comparison shows the effect of having a heat recovery instead of a heat pump, for the heat source.

The medium temperature cooling COP of the two systems as a function of the ambient temperature is shown in Figure 19. It can be seen that the COP of CHR is lower than the COP of the reference case for ambient temperatures below 10 °C. The reason is that, for CHR, the refrigeration system is the heat provider. As the heating demand starts, the refrigeration system no longer operates in floating condensing mode, and it should increase the discharge pressure to meet the heating demand. An increase in the discharge pressure increases the condensation temperature and thus decreases the cooling COP.

As the ambient temperature decreases, the heating demand increases and the refrigeration system increases the discharge pressure even more. Thus the difference in the cooling COP's increases as the ambient temperature decreases.

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For ambient temperatures higher than 10 °C, both curves coincide. The reason for this is that there is no heating demand at these temperatures, and CHR can operate in floating condensing mode, just like FC+HP.

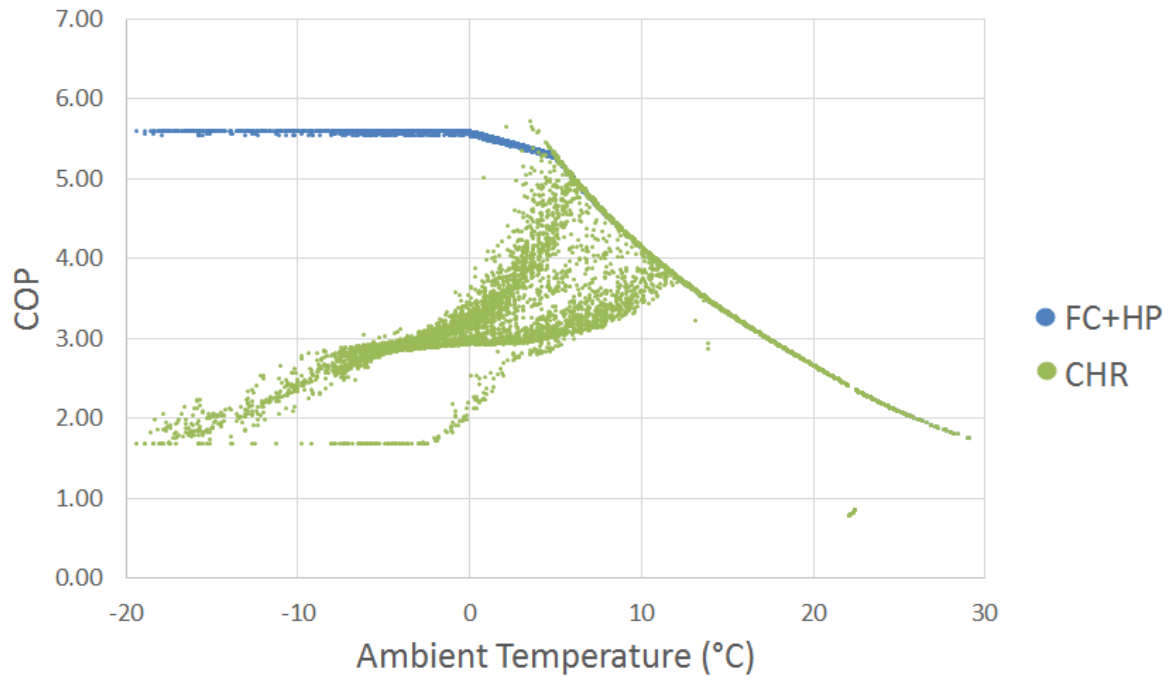


Figure 19: Cooling COP as a function of ambient temperature, FC+HP vs. CHR

The cooling COP includes the total power used by the compressor. For CHR, the compressor is used for providing both cooling and heating therefore the cooling COP does not represent a fair comparison of efficiency between the 2 cases.

The heating COP of the two systems as a function of the heating demand is shown in Figure 20. It should be noted that for CHR, the curve represents COP_{HR} , as given by equation (6). The heating COP of CHR is greater than the one of FC+HP for the whole heating demand range, although it has variations.

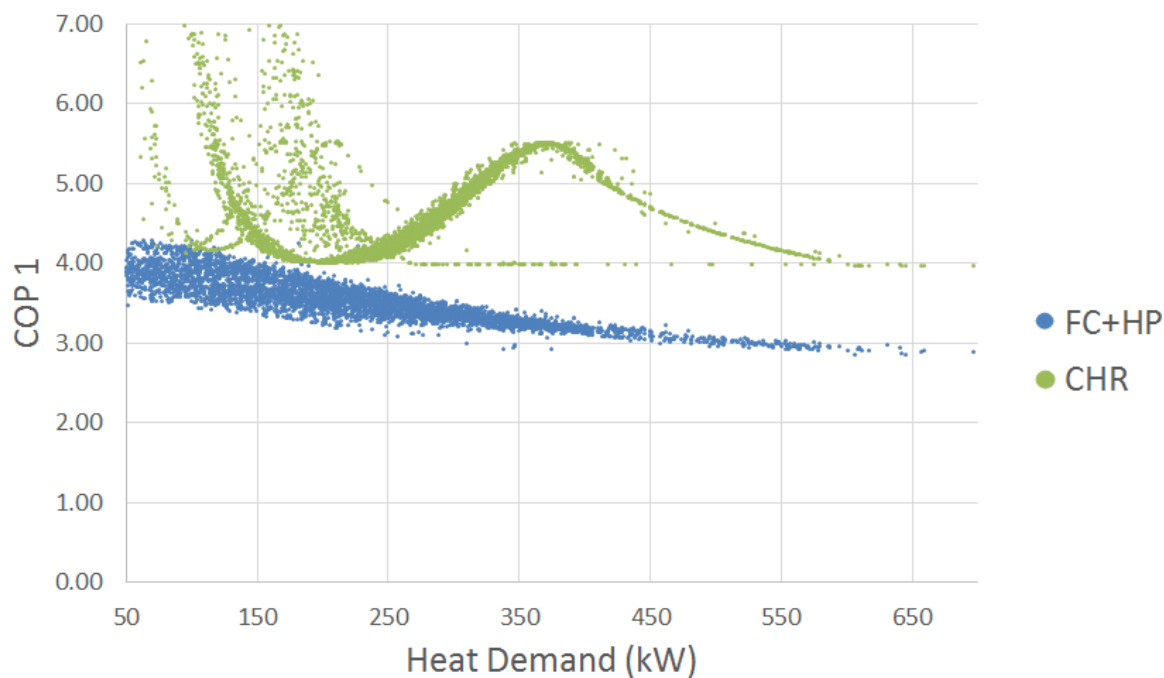


Figure 20: Heating COP as a function of heat demand, FC+HP vs. CHR

The heating COP of CHR is much higher than the reference case's for low heating demand. The reason is that, as the heating demand is low, the refrigeration system can recover all the needed heat with a minimum increase in the discharge pressure, which does not require much power.

As the heat demand increases, the COP has a negative slope because recovering more amount of heat requires much larger discharge pressure increase and therefore much larger compressor power. The slope of the COP becomes positive for heat demand higher than 200 kW. At this point, the refrigeration system starts to run in trans-critical mode and a slight increase in the discharge pressure, provides a much higher heat recovery.

This positive slope continues until a heat demand of 380 kW. At this point, the discharge pressure is at its maximum of 88.5 bar and is not increased anymore. Instead, the gas cooler runs at reduced capacity and its exit temperature increases. This results into a higher mass flow of refrigerant and has a negative effect on the heating COP.

The cooling COP, in Figure 19, does not show how effective is CHR compared to FC+HP because the heating system in CHR is incorporated into the refrigeration system. Figure 21, which shows the total power needed to run the systems for both scenarios, gives a better comparison about the effect of the heat recovery.

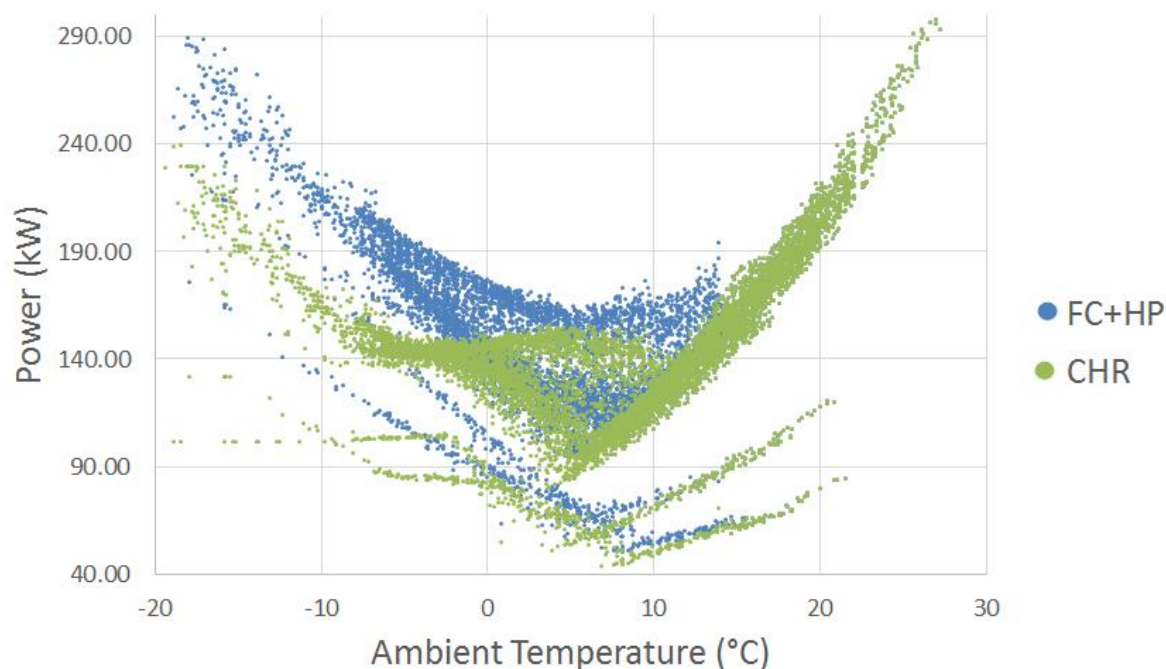


Figure 21: Total power needed as a function of ambient temperature, FC+HP vs. CHR

For ambient temperatures less than 10 °C, it is shown that the power needed to run CHR is much less than the power needed to run FC+HP. This is the region when the heating demand is positive, therefore it can be deduced that although the heat recovery system has higher discharge pressures, its power demand is still lower than the combined power needed to run a separate heat pump and a refrigeration system.

For ambient temperatures higher than 10 °C when there is no heating demand, both refrigeration systems operate on floating condensing mode and therefore they need the same amount of power. This is reflected by the superimposed curves.

Overall, CHR has an annual total energy usage of 1257 MWh, 8.1% less than the reference case.

4.4 FC+HP vs. CHR+BTES

The last scenario to be compared with the reference case is CHR+BTES, where the refrigeration system has heat recovery and is connected to borehole thermal energy storage. This comparison shows the effect that the heat recovery and the thermal energy storage bring to a refrigeration system with a separate heat pump.

The medium temperature cooling COP as a function of the ambient temperature is shown for both scenarios in Figure 22.

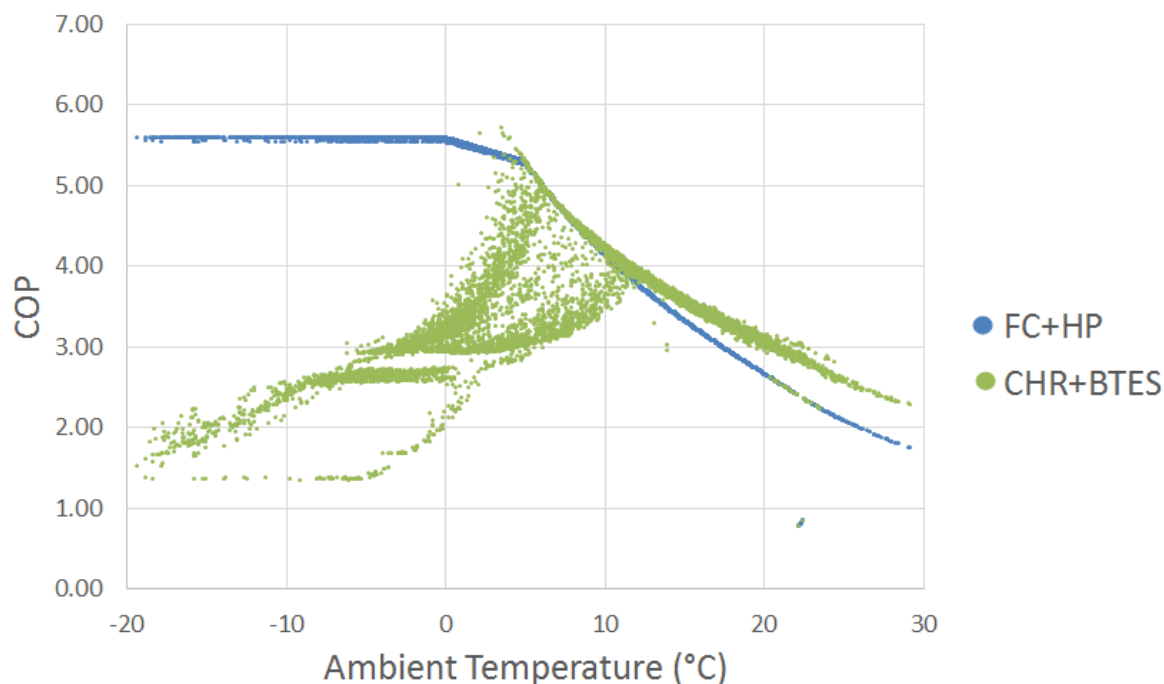


Figure 22: Cooling COP as a function of ambient temperature, FC+HP vs. CHR+BTES

For ambient temperatures lower than 10 °C, the cooling COP of CHR+BTES is lower than the reference case FC+HP. Because of the heating demand, the refrigeration system has to increase the discharge pressure to recover the needed heat, which causes more power to be used by the compressor and therefore lower cooling COP.

As the ambient temperature decreases, the heating demand increases and the refrigeration system increases the discharge pressure even more. Thus the difference in the cooling COP's increases as the ambient temperature decreases.

As a difference from the previous comparison, a jump can be noticed in the cooling COP of CHR+BTES when the COP has a value of 2.8. At this point, the medium temperature evaporator starts to cool the borehole thermal energy storage and therefore adding the cooling of the medium temperature level. This in turn increases the mass flow of the refrigerant, thus more heat can be recovered with lower discharge pressure. The increase in mass flow increases the power consumption of the compressor and thus lowers the cooling COP.

For ambient temperatures higher than 10 °C, there is no heat demand and both refrigeration systems operate in floating condensing mode. As a difference from the previous comparison, the gas cooler output in CHR+BTES is sub-cooled by the thermal energy storage. This provides a better cooling COP than a refrigeration system without sub-cooling, as in the reference case FC+HP.

As the ambient temperature increases, the effect of the sub-cooling becomes more visible.

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The heating COP of the two systems as a function of the heating demand is shown in Figure 23. It should be noted that for CHR+BTES, the curve represents COP_{HR} , as given by equation (6). The heating COP of CHR+BTES is greater than the one of FC+HP for the whole heating demand range, although it has variations.

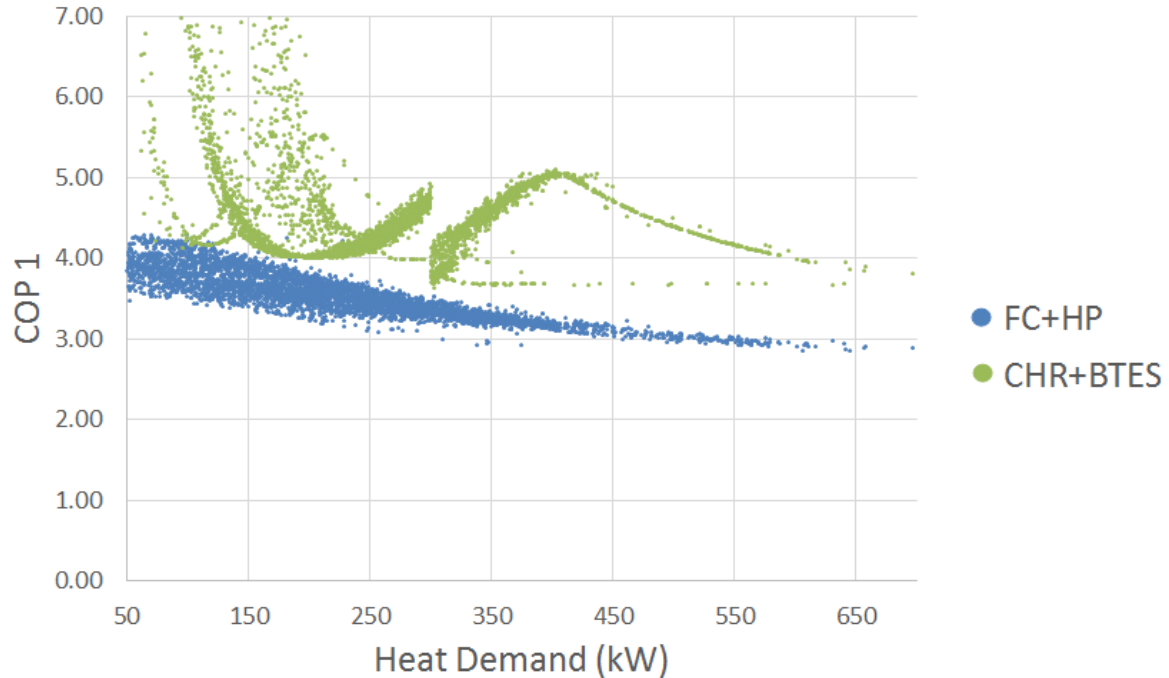


Figure 23: Heating COP as a function of heat demand, FC+HP vs. CHR+BTES

The variations in the heating COP of CHR+BTES, except the jump at heat demand of 300 kW, can be explained by the same reasons as for CHR, detailed in the previous comparison. As for the jump happens at a heat demand of 300 kW, at which point heat is taken from the thermal energy storage. As explained previously, this helps in storing cooling energy in the boreholes and provide heat recovery at lower discharge pressures.

CHR+BTES is a refrigeration system with heat recovery and has no heat pump. Therefore the cooling COP shown in Figure 22 does not fully compare the scenario to the reference case. A better picture is given when comparing the total power needed to run the systems as a function of the ambient temperature, shown in Figure 24.

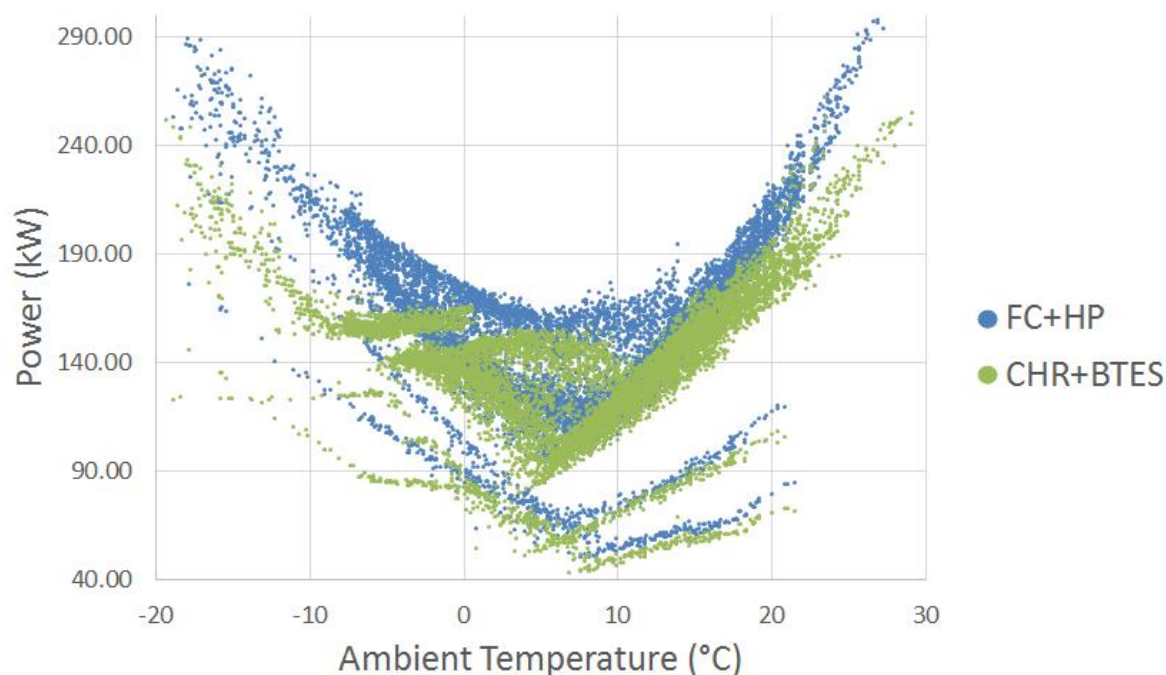


Figure 24: Total power needed as a function of ambient temperature, FC+HP vs. CHR+BTES

It can be seen that CHR+BTES consumes less power than the reference case FC+HP, for all ambient temperatures. It can be inferred that, as it was shown in the previous comparison, it is much profitable to provide heating with heat recovery rather than with a heat pump.

The graph shows also the effect of the thermal energy storage on the power consumption. Although there is a rise in power needed at the coldest temperatures, when the borehole is being charged by cooling energy, but the power needed during warmer temperatures is much lower than the reference case because of the sub-cooling provided by the storage.

Overall, CHR+BTES has an annual total energy usage of 1210 MWh, 11.5% less than the reference case.

4.5 CHR vs. CHR+BTES

The previous section compares CHR+BTES to the reference case and shows the effect of having thermal energy storage and heat recovery instead of a heat pump. In order to show the effect of adding thermal energy storage to a system that already provides the heating by heat recovery, CHR+BTES is compared to CHR.

The medium temperature cooling COP as a function of the ambient temperature is shown in Figure 25. For ambient temperatures between -5 °C and 10 °C, the two curves coincide and both scenarios have the same cooling COP. During this period, the thermal

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energy storage is not being utilized. There is no cooling energy that is being stored, and there is no opportunity to further sub-cool the gas cooler output in the refrigeration system.

For ambient temperatures below $-5\text{ }^{\circ}\text{C}$, the thermal energy storage is being charged by cooling energy. Heat is being taken from the boreholes by the medium temperature evaporator. This increases the refrigerant mass flow and thus the power consumed from the compressor. Therefore during this period, the cooling COP of CHR+BTES is lower than the COP of CHR.

For ambient temperatures above $10\text{ }^{\circ}\text{C}$, sub-cooling of the gas cooler output is possible by the thermal energy storage. Thus, the cooling energy stored during winter is utilized for sub-cooling and the effect is an improvement of the cooling COP.

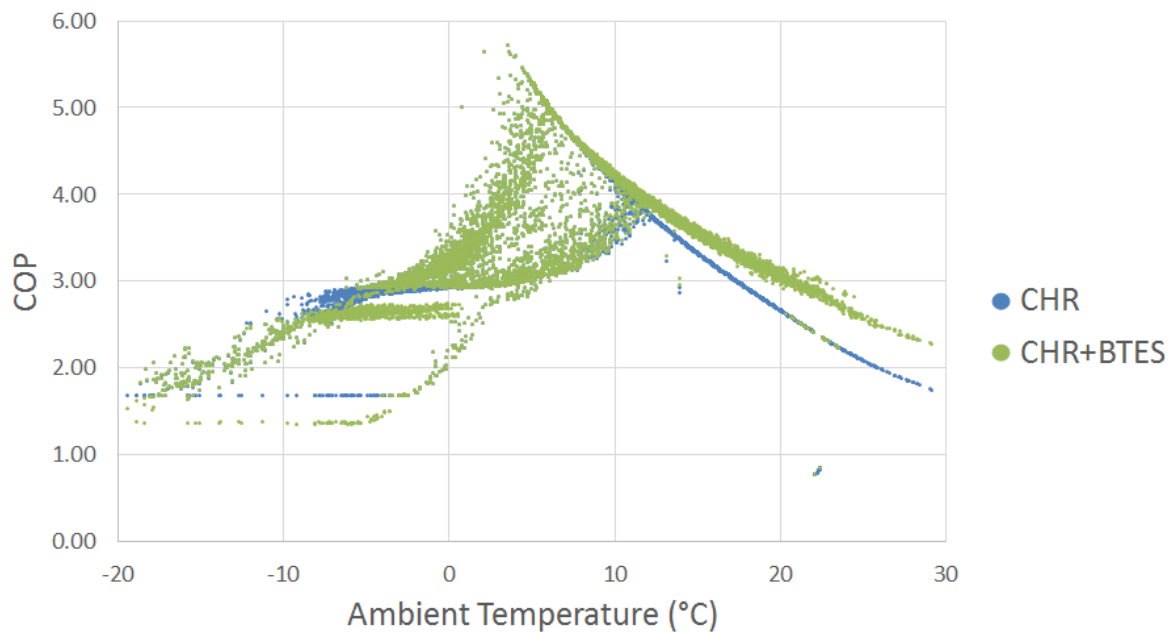


Figure 25: Cooling COP as a function of ambient temperature, CHR vs. CHR+BTES

The comparison of the heating COP as a function of the heat demand is presented in Figure 26. It should be noted that both curves show the COP_{HR} , as given by equation (6).

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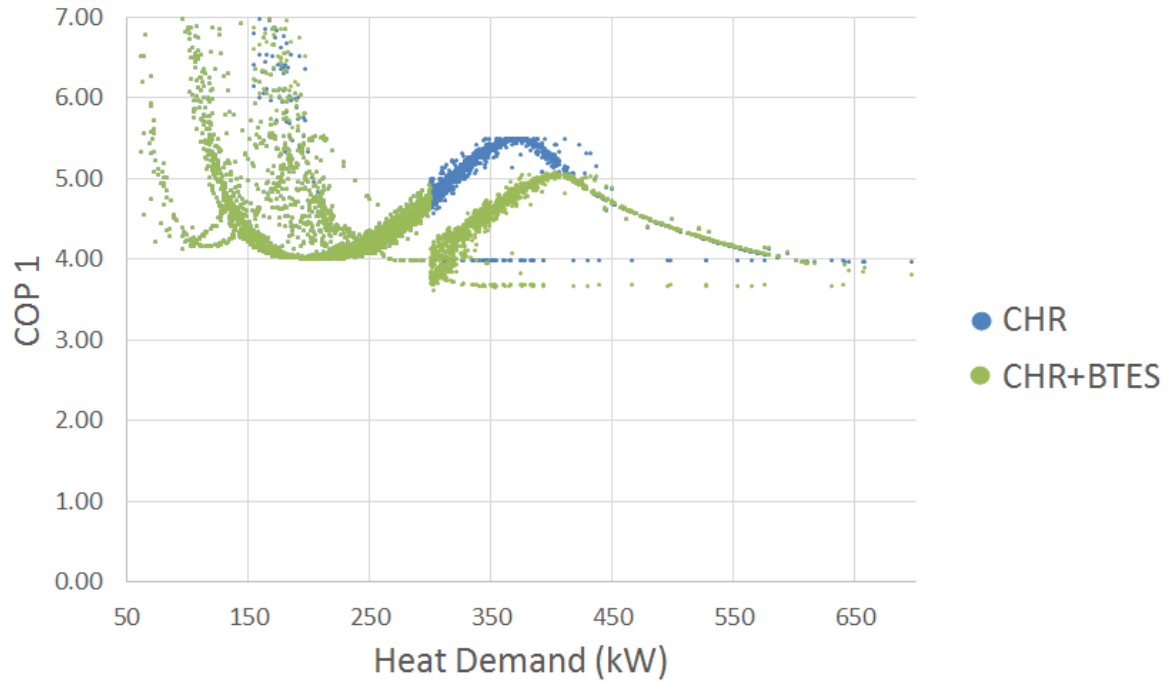


Figure 26: Heating COP as a function of heat demand, CHR vs. CHR+BTES

For heat demands lower than 300 kW, the heating COP is the same for both systems and the 2 curves are superimposed. For heat demands above 300 kW, CHR+BTES starts charging the boreholes with cooling energy, thus it increases the refrigerant mass flow. Although the increase in mass flow allows heat recovery with lower discharge pressures, its effect on the compressor power is much greater and therefore the heating COP of CHR+BTES is less than the COP of CHR.

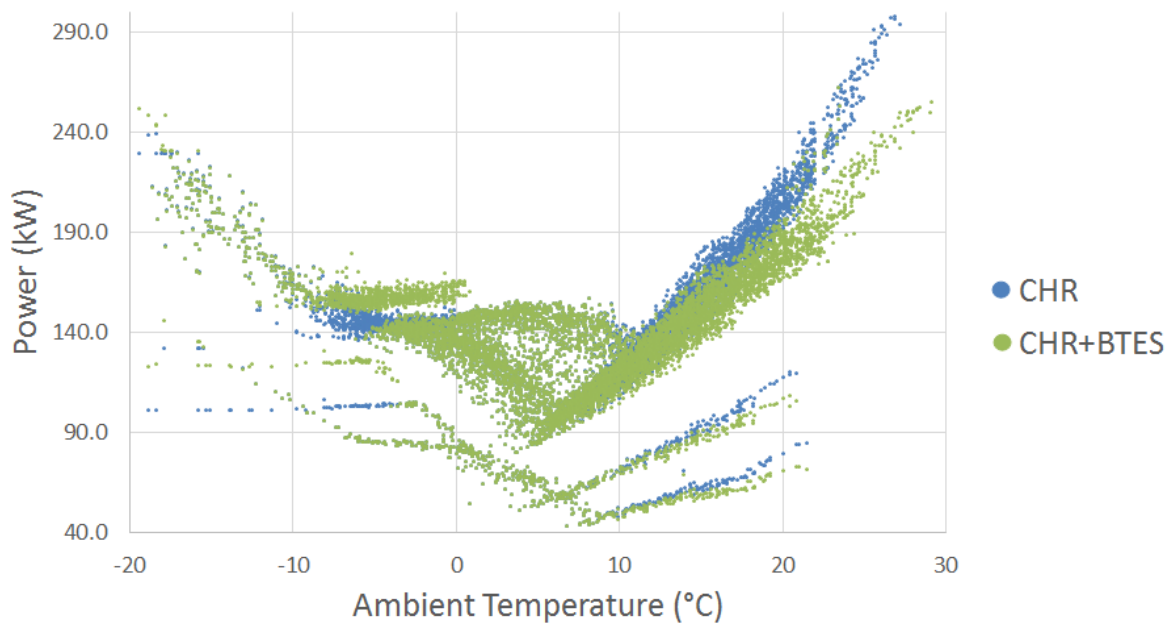


Figure 27: Total power needed as a function of ambient temperature, CHR vs. CHR+BTES

SIMULATION RESULTS

The comparison of the power consumption as a function of the ambient temperature is shown in Figure 27. For ambient temperatures between $-5\text{ }^{\circ}\text{C}$ and $10\text{ }^{\circ}\text{C}$ the curves coincide and both scenarios need the same amount of power to operate the refrigeration system. As explained earlier, during this period CHR+BTES is not utilizing the thermal energy storage.

For ambient temperatures below $-5\text{ }^{\circ}\text{C}$, CHR+BTES has a higher power demand. This is caused by the higher refrigerant mass flow, as the thermal energy storage is being charged with cooling energy by the medium temperature level.

Although CHR+BTES consumes more power during winter, it needs significantly less power for ambient temperatures higher than $10\text{ }^{\circ}\text{C}$. The cooling energy that was stored in the ground during winter time, is being used to provide sub-cooling to the gas cooler output. The sub-cooling improves the cooling COP and allows less power usage.

Overall, CHR+BTES has an annual total energy usage of 1210 MWh, 3.7% less than CHR.

Chapter 5

ECONOMIC ANALYSIS

The final step in evaluating the potential of thermal energy storage in supermarkets is to analyze the cost of incorporating the discussed storage techniques into the existing systems.

The reference case, FC+HP, has a refrigeration system and a separate ground source heat pump. The second scenario, FC+HPC, uses the brine from the heat pump to provide sub-cooling to the refrigeration system. Both scenarios have the same heat pump with the same number of boreholes, therefore the cost of upgrading FC+HP to FC+HPC is negligible relative to the savings the upgrade provides.

The third scenario, FC+BTES, uses the borehole brine, cooled by the heat pump during winter, to provide sub-cooling to the refrigeration system. This scenario also uses the same heat pump with the same number of boreholes. Whether upgrading from FC+HP or FC+HPC, implementing the solution of FC+BTES has negligible costs compared to its savings. However, it is interesting to evaluate the cost of adding additional boreholes to the existing ones in the third scenario FC+BTES.

The fourth scenario, CHR, is a heat recovery system with no heat pump and no boreholes. It is considered the reference case for the scenario with a heat recovery system and borehole thermal energy storage, CHR+BTES. The costs and savings of these two scenarios, CHR and CHR+BTES, will be compared and analyzed.

The economic analysis tool used in this chapter is the life cycle cost analysis. The LCC tool computes the present cost of a system over its lifetime. It takes into account the investment cost, the energy cost and the maintenance cost to evaluate the most cost effective scenario. The following chapter will analyze the life cycle cost of adding additional boreholes to the FC+BTES case, and the cost of adding a thermal energy storage to the CHR case in order to upgrade it to CHR+BTES.

5.1 Life cycle cost analysis parameters

The life cycle cost adds the present value of 3 factors, the investment cost, Inv , the energy cost LCC_E , and the maintenance cost, LCC_M :

$$\text{LCC} = \text{Inv} + \text{LCC}_E + \text{LCC}_M \quad (9)$$

The investment costs includes the cost of adding the borehole thermal energy storage. It is assumed to be 250 SEK/m of borehole depth (Byggmentor, 2013).

The annual electrical energy of each scenario, AE, is calculated by the theoretical simulation, detailed in the previous chapters. The present value of the annual energy costs is given by:

$$\text{LCC}_E = C_{AE} \times \text{AE} \times \frac{1 - [1 + (i - p)]^{-n}}{i - p} \quad (10)$$

The cost of electricity, C_{AE} , is assumed to be 1 SEK/kWh, and because the future price variations cannot be accurately forecasted, the cost of electricity is assumed to be increasing with the inflation rate (Statistika centralbyrån, 2013). The interest rate, i , and the inflation rate, p , are assumed to be 6% and 2% respectively. The life cycle cost analysis is over a period of 30 years, indicated by n in the above equation.

For simplicity, the maintenance cost of the scenarios are neglected.

5.2 Life cycle cost analysis

First, the life cycle cost of FC+BTES is analyzed when additional boreholes are used in the system. Although it has a significant investment costs, adding boreholes has several benefits. The borehole brine temperature variations over the years will be less sensitive to the heating and cooling loads. With a less varying brine temperature, the heat pump will work more efficiently and the thermal energy storage will provide more sub-cooling to the refrigeration system.

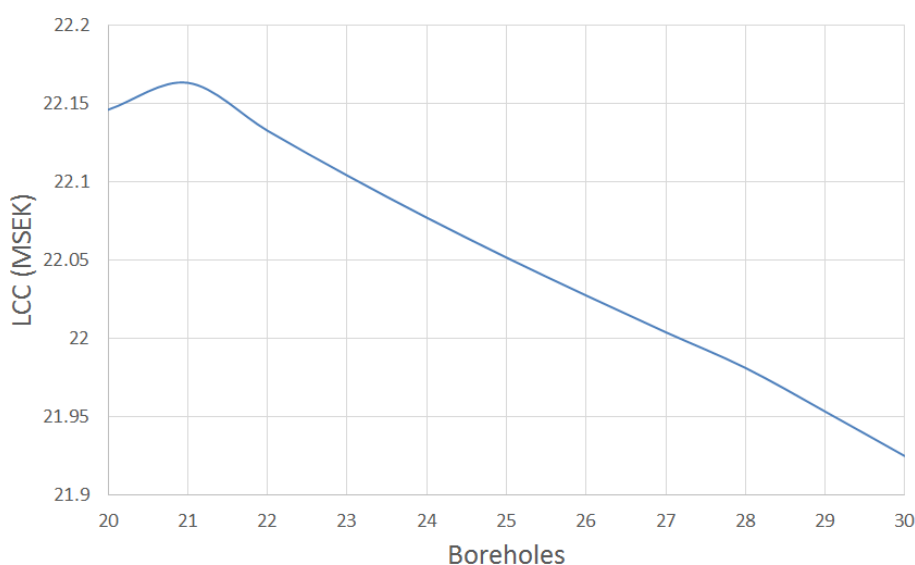


Figure 28: Life cycle cost of FC+BTES for each additional borehole

The life cycle cost of FC+BTES for each additional borehole is presented in Figure 28 in millions of SEK. It can be seen that as the number of boreholes increases the life cycle cost of FC+BTES decreases. It is therefore more cost-effective to have a high number of boreholes, although the space area limitation should be taken into consideration.

Second, an analysis of life cycle cost is performed for the simulation cases CHR and CHR+BTES. As boreholes are not included in the scenario CHR, investing in 20 boreholes and thus upgrading CHR into CHR+BTES is costly. The objective of this life cycle cost analysis is to evaluate the cost-effective scenario for a refrigeration system that has heat recovery. Using the equations (9) and (10) and the input parameters described in the previous section, the life cycle cost of CHR is calculated to be 21.7 million SEK, whereas the one of CHR+BTES 21.9 million SEK. Over the calculated period of 30 years, the difference of the two life cycle costs is not large. Thus, CHR+BTES can be considered beneficial for longer periods or when the electricity price increase is higher than the inflation.

Chapter 6

DISCUSSIONS

The performance of the simulation cases presented in Chapter 3 have been theoretically analyzes. As a summary, Figure 29 presents the annual energy consumption of the refrigeration and the heat pump system of each scenario. Each simulation case is compared with the reference case, FC+HP, and the difference in total energy use is presented by the percentages at the top of each column. The difference in the energy consumption of the heat pump system is shown at the top part of the FC+HP and FC+BTES columns, whereas the difference in the energy consumption of the refrigeration system at the top part of the columns.

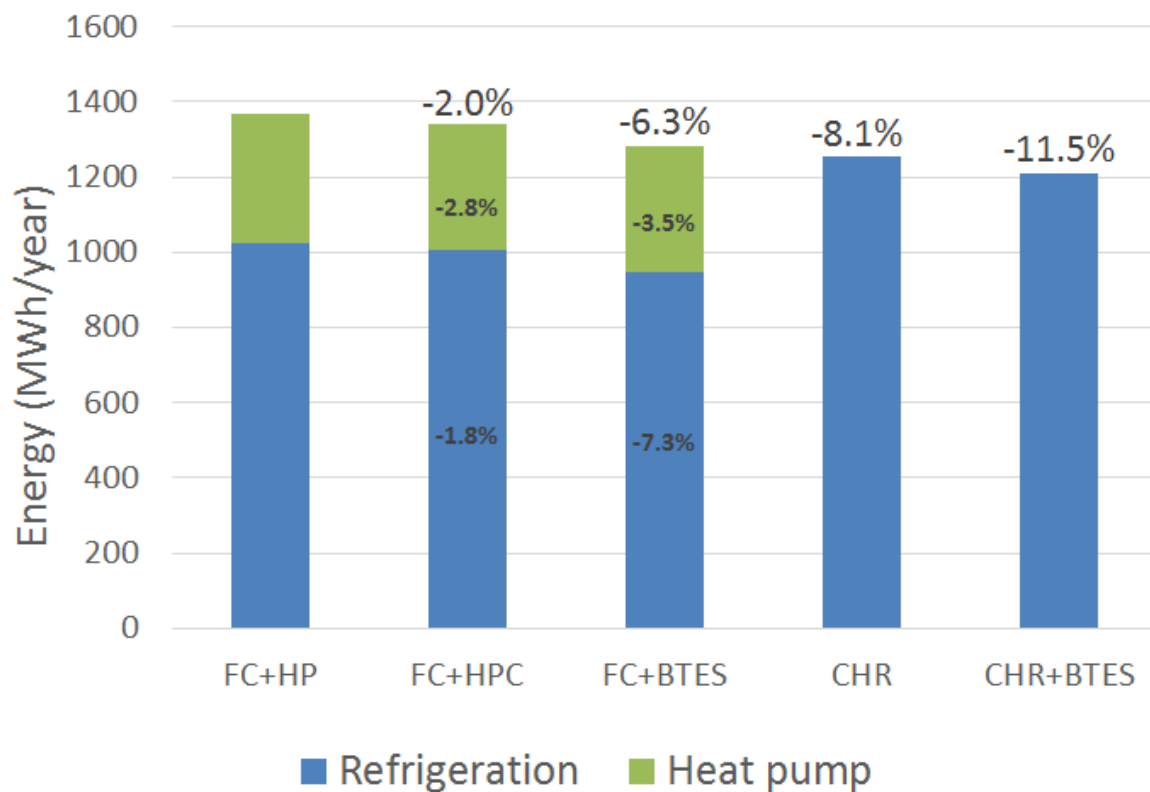


Figure 29: Summary of annual energy usage

DISCUSSIONS

The reference case has the highest energy consumption among the simulation cases. It uses a heat pump to provide the heating needs and does not utilize the thermal energy storage capabilities of the boreholes.

Connecting the heat pump to the refrigeration system, as in FC+HPC, is beneficial for both the heat pump and the refrigeration system. The brine that leaves the heat pump sub-cools the refrigeration system and therefore heats up itself. The sub-cooling improves the refrigeration system, which uses 1.8% less energy than the reference case, and the increased brine temperature increases the heat pump evaporation temperature and thus improves the heat pump, which uses 2.8% less energy. Upgrading the reference case to FC+HPC does not require a large investment, nevertheless provides savings in the annual electricity usage.

Utilizing the boreholes as thermal energy storage, as in FC+BTES, offers the best option for refrigeration systems operating in floating condensing mode. The heat pump is connected to the refrigeration system, similar to FC+HPC, therefore it provides all the benefits stated for the previous case. In addition, the borehole is connected to the refrigeration system. This connection provides sub-cooling even when the heat pump is not operating. The amount and the potential of sub-cooling is much greater in the summer, as it makes the refrigeration system more efficient and raises the brine temperature considerably, thus increasing the efficiency of the heat pump during winter. Unlike the previous case, FC+BTES helps balance the cooling and heating loads during the whole year. Compared to the reference case, the annual electricity usage by the heat pump system is reduced by 3.5% and the annual electricity usage by the refrigeration system is reduced by 7.3%, thus resulting into a 6.3% decrease in the overall electricity usage. Upgrading the reference case or the FC+HPC case to FC+BTES requires minimal investment costs and provide substantial amount of savings. Moreover, investing in additional boreholes is cost-effective if the space is available.

Providing heat through heat recovery rather than a heat pump proves to be more efficient for a supermarket. The CHR case provides 99.37% of the heating demand while using 8.1% less energy than the combined heat pump and refrigeration system of the reference case. The heat recovery cases perform even better than FC+BTES, which uses borehole thermal energy storage to improve the efficiency of the system. Moreover, the investment cost of CHR are much lower than the investment cost of the reference case, as the heat pump is replaced by a desuperheater heat exchanger.

Finally, the case with the lowest annual energy consumption is CHR+BTES. The case, which uses borehole thermal energy storage coupled with a heat recovery system, has an annual electricity usage 11.5% less than the reference case, and 3.7% less than CHR, the case with heat recovery system but without the thermal energy storage. The upgrading cost of CHR into CHR+BTES includes the investment of the boreholes, which may seem costly, but the 2 cases have similar life cycle costs and CHR+BTES may prove to be more cost-effective with higher electricity prices. Unlike CHR, CHR+BTES can adjust its limit of heat recovery by taking more heat from the ground during winter time thus increasing the cooling load of the medium temperature level.

Chapter 7

CONCLUSIONS

The potential of thermal energy storage in CO₂ supermarkets have been investigated. After presenting daily and seasonal energy storage technologies suitable for supermarkets, the research focused on the seasonal storage alternative, choosing boreholes as the solution.

Five cases of supermarket systems have been presented for simulation. The first is the reference case, FC+HP, which provides the heating with a ground source heat pump, and does not have a thermal energy storage. The second case, FC+HPC, is based on the reference case, except the heat pump is connected to the refrigeration system, providing sub-cooling during the operation time of the heat pump. The third case, FC+BTES, is also based on the reference case, but the stored cooling energy in the ground by the heat pump is used to provide sub-cooling to the refrigeration system during the warm periods of the year, thus utilizing the boreholes as a thermal energy storage. The fourth case, CHR, provides the heating by means of heat recovery with a desuperheater and does not have a thermal energy storage. It is considered to be the reference case of the fifth case, CHR+BTES, where boreholes are used to store cooling energy during the coldest days of the winter, and use the stored heat to provide sub-cooling to the refrigeration during the warmer days.

First, the simulation results show that FC+HPC, which is commercially applied, performs better than the reference case, FC+HP, but with a slight change in the design and the control system, it can use the boreholes as a thermal energy storage and perform more efficiently, as in case FC+BTES. Second, it has been proved that the supermarket systems with heat recovery, CHR and CHR+BTES, perform better than the ones using a ground source heat pump. Third, although adding a thermal energy storage to the heat recovery system can reduce the electricity usage, as in CHR+BTES, it proves to have a slightly higher life cycle cost.

Therefore, it can be concluded that existing supermarket systems with ground source heat pump, can improve their performance drastically by small changes in the design and the control systems, by using the boreholes as thermal energy storage. As for new supermarket systems, the case with heat recovery is recommended, CHR, but because of

CONCLUSIONS

its limit in maximum recoverable heat, adding borehole thermal energy storage provides more flexibility by taking more heat from the ground.

For future work, the daily thermal energy storage technologies can be evaluated to analyze their potential in supermarkets and compare them with seasonal thermal energy storage techniques, thermodynamically and economically.

The simulation cases can be more refined to include any auxiliary power such as borehole pumps and condenser fans.

Various other simulation cases with different system designs and control strategies can be investigated.

The life cycle cost analysis can be more detailed by including different scenarios for electricity price forecast and by adding the maintenance and other costs associated with the investment of boreholes.

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