Resurseffektiva kyl- och värmepumpssystem

Thermal Storage in Supermarket Refrigeration System

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Energimyndigheten

Sammanfattning

Denna studie utvärderar potentialen för termisk lagring i livsmedelsbutikers energisystem. Både kort och långtidslagring har undersökts och besparingen i energianvändning från kylsystem har beräknats.

I korttidslagrings scenarier har både sensibla och latenta typer av termisk lagring studerats. Olika scenarier som utnyttjar variationer av utomhustemperaturer har analyserats. I långtidslagring har borrhål för säsongslagring använts. Olika scenarier som kopplar borrhålet till kylsystemet för att täcka kyl-och värmebehov har föreslagits och utvärderats.

I korttidslagringen, visade sig att lagringsvolymen ska vara stor för att nå tillräckliga energibesparingar. Det behövs 10 och 6.3 m3 lagringsvolym av köldbärare och PCM för att uppnå 2.6 och 8% daglig energibesparing. Eftersom det är svårt att hitta plats till stora volymer i maskinrummet, kan de korttidslagringar föreslagna i denna studie ha begränsade möjligheter för livsmedelsbutiker.

I långtidslagring, där borrhål har kopplats till kylsystemet för att leverera ytterligare underkylning, kan energibesparingar bli cirka 6,3% av den årliga energianvändningen. På uppvärmningssidan, är det bevisat att livsmedelsbutiker med värmeåtervinningssystem är mer effektiva än butiker utan värmeåtervinning. Kylsystemet kan täcka nästan allt värmebehov med högre effektivitet (COP 4-5,5) i jämförelse med en bergvärmepump (COP på 3-4). Om värmebehovet är högre än värmen som kan återvinnas från kylsystemet då kan bergvärmepump användas för att täcka överskottsvärmen. Borrhålet kan användas för underkylning av kylsystemet under sommarperioden.

Summary

This study evaluates the potential of thermal storage in supermarket energy system. Both short and long term storage are investigated and the saving in energy use of refrigeration system is calculated.

In short term storage scenarios, sensible and latent types of thermal storage are studied. Different scenarios to exploit the ambient temperature difference are analyzed. In long term storage, borehole as seasonal storage unit is used. Different scenarios of connecting the borehole to the refrigeration system to cover the cooling and heating demands were suggested and evaluated.

In short term storage, the needed storage volume for the refrigeration system of a real supermarket was found to be rather large to have acceptable saving in energy consumption. About 10 and 6.3m^3 of storage unit volumes are required for brine and PCM respectively to achieve daily energy saving of 2.6 and 8%. Since it is difficult to fit large volume in machinery room, short term storage with the suggested storage in this study may have limited possibilities for application.

In long term storage, connecting borehole to the refrigeration system to provide additional sub-cooling can have about 6.3% savings in annual energy use. On the heating side, it is proved that the supermarket systems with heat recovery can perform more effectively than the reference case. The refrigeration system can cover almost all the heat load at higher efficiency (heating COP of 4-5.5) compared to a ground source heat pump (Heating COP of 3-4). If the heating demand is higher than what can be recovered from the refrigeration system then ground source heat pump can be used to cover the excess heat and the borehole can sub-cool the refrigeration system in the summer period.

Acknowledgement

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Content

1 INTRODUCTION

1.1 Background

Supermarkets are intensive consumers of energy using electricity and heating energy to cover their cooling and heating demand. In Sweden, supermarkets are responsible for 3% of the total national electricity use (Kullheim, 2011). About 50% of the electricity in supermarkets is used by the refrigeration system, other sections such as lighting, kitchen and fans are responsible for the rest (Furberg, et al. 2000). Beside the intensive electricity usage heating needs can account to 15% of the total annual energy usage of a medium-sized supermarket (Arias, 2005). This broad energy use encourages making improvements in the efficiency of supermarket refrigeration and heating systems.

The refrigeration systems performance is highly dependent on the boundary conditions. In floating condensing mode where the refrigeration system is supposed to cover only the cooling demand the condensation temperature is connected to the ambient temperature. The lower ambient temperature the higher cooling COP of the system can be achieved due to lower condensation temperature.

The operating boundary conditions could change by using thermal energy storage and higher efficiency can be achieved in severe conditions. In such case, the system will be controlled to run for energy storage when the efficiency is high and the demand is low. Then the storage system will be discharged when the load is high and the efficiency is low.

The short term thermal storage concept is explained in the following simple plot in Figure 1 for three consecutive days. Cooling load shown by Figure 1 varies considerably during day and time, increasing in day time as the ambient temperature increases. Due to high ambient temperature, the refrigeration system is operated at high condensation temperature leading to relatively low efficiency; shown by Figure 2. The concept is to charge the storage device by adding extra load on the system while the system efficiency is high and then use the stored energy in the peak time to improve the COP and reduce the energy consumption.

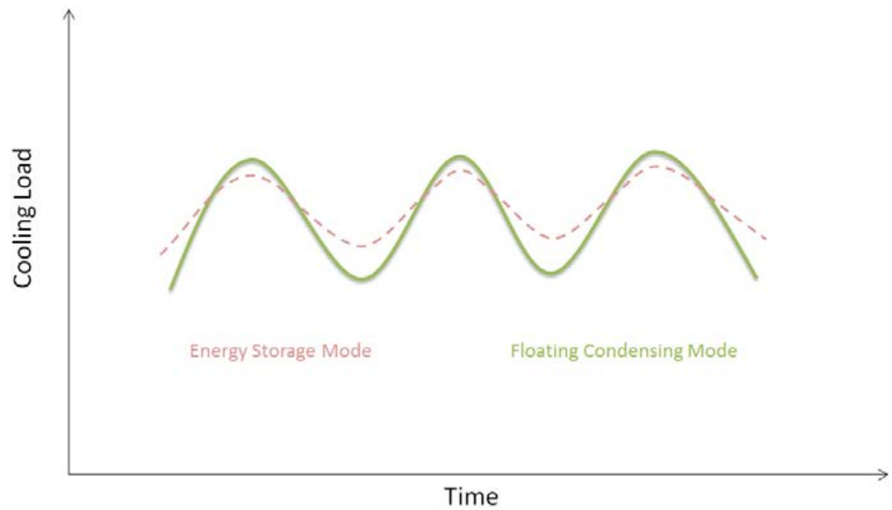

Figure 1 – Explanatory profile of cooling load with and without thermal energy storage

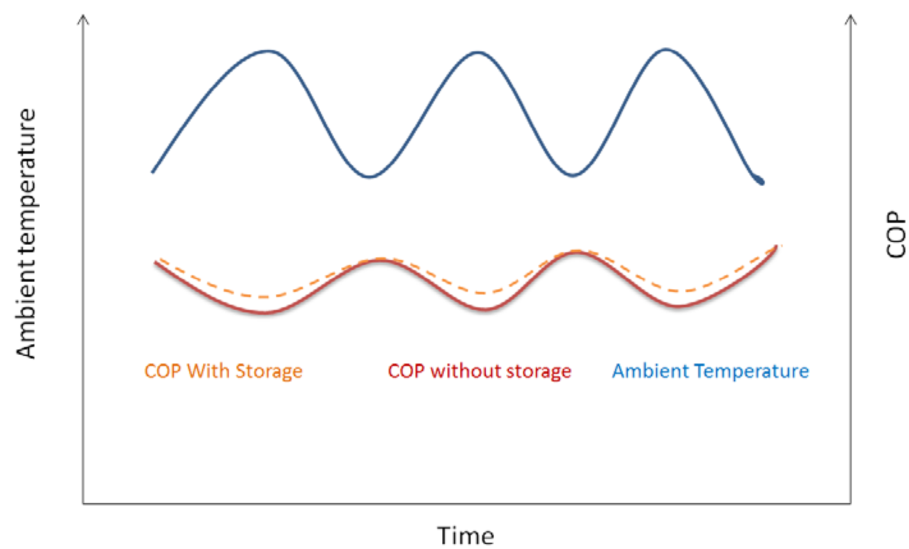

Figure 2 - Explanatory profile of ambient temperature and cooling COP with and without thermal energy storage

Similar concept can be applied in long term thermal storage. Charging the thermal storage device/system is done in the cold period, for instance taking heat from the ground for space heating. In the summer, the cold underground can be used to cool the refrigeration system and reduce its energy use; i.e. increasing cooling COP. This will lead to the underground warming up during the summer, which will help to cover the winter heating load more effectively.

1.2 Aim of study

The objective of this study is to investigate the potential of short and long term thermal energy storage in supermarket refrigeration systems, and to evaluate its influence on the efficiency (or energy use) of the refrigeration system providing simultaneous cooling and heating.

Several short and long term thermal energy storage technologies for supermarket will be investigated. The influence of different control strategies on system performance will be studied and economical evaluation of long term thermal storage will be taken into consideration.

1.3 Methodology

Literature study is carried out to summarize the possible thermal energy storage applications in supermarket refrigeration system. Different methods and technologies for both short and long term thermal storage are reviewed. Additionally the results of thermal energy storage usage from literature are expressed briefly.

Then a refrigeration system solution for simultaneous covering of cooling and heating demands (including the control strategy for heat recovery) in average size is described. Also the method of refrigerant mass flow rate calculation and the dentitions for cooling and heating COPs are expressed. This refrigeration system is considered as reference case.

The study is divided into two parts of short and long term thermal energy storage analyses. For both types of storage, the chosen storage method and the corresponding governing equations to model the storage system are explained. Computer simulation model has been built to simulate the performance of the refrigeration system in connection with thermal storage device.

In short term, the computer model is focused on cooling side for relatively short period of time. Different scenarios of thermal energy storage to improve the efficiency and reduce the energy use are investigated. The result for each scenario is compared to the reference case and the savings in daily energy use is calculated.

In long term, the computer model is used to investigate the system performance on both cooling and heating sides in seasonal scale. Different scenarios of using seasonal thermal storage in connection with the refrigeration system are devised and investigated. Similar to short term case, the results are compared to the reference case and the long term savings in energy consumption is calculated. Additionally, economic analysis is done for long term thermal storage system.

Finally the best thermal storage type and the best scenario for an average size supermarket in Sweden are suggested.

1.4 Literature survey

Different thermal storage technologies are investigated in this project. In this section possible methods presented in scientific literature for short and long term thermal storage are reviewed. This helped in defining the possible system solutions that are modelled in this study.

1.4.1 Daily storage

The principle of short term storage is based on the load shifting between day and night. During night time, system efficiency is higher due to lower ambient temperature. At this time storage unit is charged (cooled down) by the refrigeration system and later as the ambient temperature increases the stored cold medium will be used to cool down the system and increase its efficiency.

Several concepts for short term thermal storage have been found in literature, for instance Ure and Beggs (1997) suggested an ice tank to act as storage unit incorporating with the heating and refrigeration systems in a supermarket. The ice tank is charged by refrigeration system during night and the stored energy is used for cooling purposes in the HVAC system in the next day (Ure & Beggs, 1997).

American company of Calmac investigated the use of ice tank and PCM (Phase Change Material) as storage unit in supermarkets. The ice tank was used to make extra sub-cooling in the day resulting in large volume of ice. The PCM was charged during off-peak hours and used to cover the cooling demand in peak hours. The PCM storage unit could cover all the cooling demand in medium temperature cycle (Calmac Manufacturing Corporation, 2002).

Using PCM as storage unit has been the matter of interest in other refrigeration system not specifically supermarkets. Wang, et al. (2007) analyzed integration of PCM units in a simple refrigeration plant experimentally. Result showed that by making extra sub-cooling in the system, the efficiency can be improved up to 8 % in UK climate (Wang, et al. 2007). Liu (2010) worked on thermal storage in truck refrigeration system as his PhD thesis. Liu studied the usage of PCM in plate heat exchangers mathematically and experimentally. He concluded that there is potential to reach about 73 % running cost reduction if the charging process would be carried out during off-peak tariff periods (Liu, 2010).

Function of attaching PCM slabs to the freezer cabinets was also studied as a thermal storage unit to reduce the energy consumption and maintain the products quality in case of power failure. Raeisi, et al. (2013) studied the PCM usage attached to the cabinet walls inside the freezers. Two shapes of PCM were investigated, pouches and honeycomb shaped. The PCM pouches have 5 % increase in energy consumption because of extra heat resistance made by pouches but usage of honeycomb shaped PCM reduced the energy consumption by 2 % (Raeisi, et al., 2013).

Azzouz, et al. (2008) analyzed the dynamic model of PCM slab attached on the outside wall of refrigerator evaporator. The model was validated by experimental tests as well. The results showed that with additional thermal inertia higher evaporating temperature could be achieved and the existence of PCM lets the system to maintain the product quality at desired level and acceptable temperature in case of power failure for a period of 4 to hours depending on the load. Oro, et al. (2012) also evaluated the effect of PCM usage in the freezers in case of power failure experimentally. The conclusion was that, during 3 hours of power failure, temperature of products could be kept 4-6 degree lower than reference case without PCM usage Oro, et al. (2012).

1.4.2 Seasonal storage

The common way of long term storage is using borehole as storage unit. In long term thermal energy storage, the borehole in the winter is used in connection with heat pump to cover the heating demand. Then the cold ground is used to sub-cool the refrigeration system in summer time. The heat rejected in the ground in the summer time leaves the ground at relatively high temperature for the winter operation which is also good for the heat pump operation.

Titze, et al. (2012) investigated the integration of borehole storage unit with refrigeration and HVAC system in a supermarket, a schematic is shown in Figure 3. In such case the brine is cooled down by refrigeration system before entering the borehole. This extra load fed to the refrigeration system, helped the system to recover heat on the high pressure side and cover the total heating demand. The stored cold brine was used to provide extra sub-cooling for the refrigeration system in summer time.

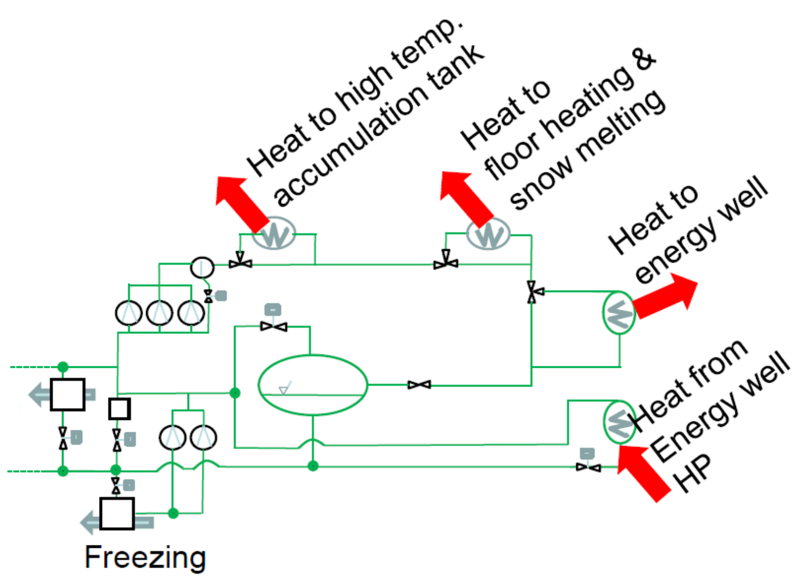

Figure 3 - Refrigeration system integrated with borehole thermal storage (Titze, et al., 2012)

2 SYSTEM DESCRIPTION

2.1 Refrigeration system

CO_2 booster system shown in is common supermarket refrigeration system in Sweden in new installation (Sawalha, 2013). The input parameters to the simulation program for modelling the refrigeration system are supported by field measurements (Gavarrell, 2011).

In such system two cooling temperature levels are required to maintain the chilled and frozen products around 3 °C and -18°C respectively. The evaporation temperature in low temperature level is set to -30 °C. The low level evaporator is assumed to have 10K internal and 10K external superheating. The evaporation temperature in medium temperature level is -10 °C with the same degree of superheating as low temperature level; i.e. 10K each.

The refrigerant in low temperature level is compressed by low stage compressor, transferred to medium temperature level and mixed with the refrigerant coming out of medium temperature evaporators. After mixing, the refrigerant is compressed by high stage compressor. The total heat is rejected by gas cooler to the ambient. At the gas cooler outlet 5K approach difference is considered between the refrigerant and ambient air.

In order to improve the system efficiency flash gas bypass is used. It reduces the enthalpy of refrigerant entering the low temperature evaporators. The vapor is separated in an intermediate pressure receiver then expanded after the medium temperature evaporators, as can be seen in Figure 4. The pressure in flash gas receiver is about 3bar higher than the pressure of medium level.

CO_2 Booster system has the capability to operate in trans-critical region. In this zone, the pressure and temperature are independent. In this case, the discharge pressure is controlled so the system will have the highest COP, the optimum pressure is calculated according to the correlation shown by equation 1 (Sawalha, 2008).

$$\mathbf{P_{Discharge,opt} = 2.7 \times T_{gc,exit} - 6} \qquad \textbf{(1)}$$

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

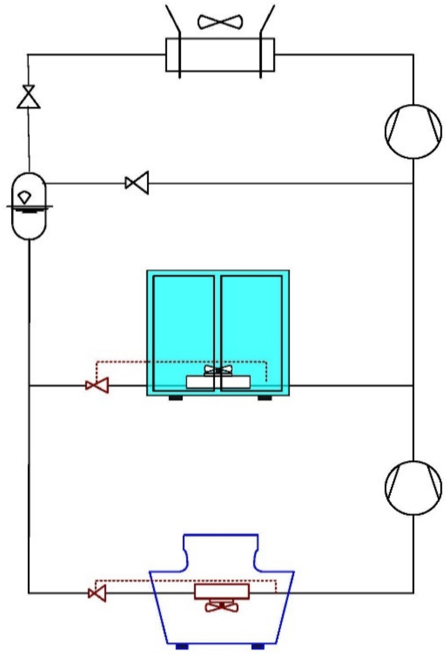

Figure 4 – CO2 booster refrigeration system

The compressor work in low and high stage is calculated using total efficiency method; expressed by equation 2. The total efficiency is calculated from the compressor manufacturer data (Gavarrell, 2011). The mass flow rate is obtained from cooling demand which is an input parameter, and by calculating the enthalpies around the evaporators.

$$\dot{\mathbf{E}}_{\mathbf{Total}} = \frac{\dot{\mathbf{m}}\ \Delta \mathbf{h}_{\mathbf{is}}}{\boldsymbol{\eta}_{\mathbf{Total}}} \tag{2}$$

In the high stage compressor there is a share related to the refrigerant coming from low stage compressor calculated by equation 3, calculation details can be found in the thesis of Gavarrell (2011) (Gavarrell, 2011).

$$\dot{\mathbf{E}}_{\mathbf{hs,f}} = \dot{\mathbf{E}}_{\mathbf{hs}} \frac{\dot{\mathbf{Q}}_{\mathbf{2,f}} + \dot{\mathbf{E}}_{\mathbf{ls}}}{\dot{\mathbf{Q}}_{\mathbf{2,f}} + \dot{\mathbf{E}}_{\mathbf{ls}} + \dot{\mathbf{Q}}_{\mathbf{2,m}}} \tag{3}$$

Where $\mathrm{E_{hs}}$ and $\mathrm{E_{ls}}$ represents the high stage and low stage compressor work respectively and $\mathrm{E_{hs,f}}$ is the share of low stage cooling load in high stage work.

The cooling COP for medium and low temperature levels are calculated by equations 4 and 5 respectively. For medium level COP the share of low stage compressor is withdrawn from the total high stage work. On the other hand for low level COP this share is added to the compressor work of low stage.

$$\mathbf{COP_m} = \frac{\dot{\mathbf{Q}}_{\mathbf{2,m}}}{\dot{\mathbf{E}}_{\mathbf{hs}} - \dot{\mathbf{E}}_{\mathbf{hs,f}}} \quad \mathbf{(4)}$$

$$\mathbf{COP_f} = \frac{\dot{\mathbf{Q}}_{\mathbf{2,f}}}{\dot{\mathbf{E}}_{\mathbf{ls}} + \dot{\mathbf{E}}_{\mathbf{hs,f}}} \quad \mathbf{(5)}$$

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

2.2 Heat recovery

Refrigeration system in supermarket is used not only to provide cooling but also to cover the heating demand. In this section, control strategy for heat recovery is explained since the heat recovery will be used for comparison with ground source heat pump system in the seasonal thermal storage scenarios.

In CO2 booster heat is recovered by using a heat exchanger as desuperheater after the high stage compressor and before gas cooler. Heat is transferred by desuperheater to water loop which is connected to heating system (Sawalha, 2013). The amount of recovered heat is highly dependent on the temperature of water returning from heating system. In reality return temperatures varies significantly as the ambient temperature and consequently heating demand changes.

The control strategy of heat recovery is based on Sawalha (2013) strategy. The control method aims to achieve the highest cooling and heating COP for the given conditions. The suggested optimum discharge pressure is given by equation 6, it is the maximum discharge pressure which refrigeration system should be operated at.

$$\mathbf{P_{Discharge,max}} = \mathbf{2.7} \times \mathbf{T_{desuperheater,exit}} - \mathbf{6} \quad \mathbf{(6)}$$

Discharge pressure is allowed to be raised up to maximum discharge pressure to cover the demand while gas cooler should work with full capacity to provide maximum level of sub-cooling. For recovering more heat, instead of raising the discharge pressure gas cooler should be regulated to get higher gas cooler exit temperature resulting in higher mass flow rate of refrigerant while discharge pressure is fixed at maximum level. Higher mass flow rate enables the system to recover more heat at fixed discharge pressure.

The heating efficiency for refrigeration system can be expressed in equation 7, it is a way to compare the heat recovery of the system to a conventional heating system such as heat pump (Sawalha, 2013).

$$\mathbf{COP_{1,HR}} = \frac{\dot{\mathbf{Q}}_{\mathbf{1}}}{\dot{\mathbf{E}}_{\mathbf{HR}} - \dot{\mathbf{E}}_{\mathbf{FC}}} \quad \mathbf{(7)}$$

In the above equation, Q_1 represents the heat demand, E_{HR} the compressor power in heat recovery mode, and E_{FC} the compressor power in floating condensing mode. In another word, the difference between E_{HR} and E_{FC} is the net power given to the refrigeration system to cover the heating demand.

3 SHORT TERM THERMAL STORAGE

The objective of this section is to investigate the potential of short term thermal energy storage in supermarkets.

The refrigeration system is modelled according to correlations expressed in section 2.1. The cooling load profiles of real supermarket focusing on summer period are obtained from field measurement and used in the simulation model. Different methods of short term thermal energy storage used in this study and the corresponding governing equations to model each method are explained as well. The storage system is modelled in connection with refrigeration system.

Different scenarios and system solutions of short term thermal storage to improve the efficiency of the refrigeration system and reduce its energy use are described. Finally the influence of short term thermal storage is investigated and the results for each scenario are discussed.

3.1 Daily cooling & heating demands profiles

The refrigeration system is simulated by computer model using the energy profiles of real supermarket as input parameters. In this section the focus is on short term thermal storage in summer period and the main input parameters to the computer model are ambient temperature and cooling demands at low and medium level of evaporation.

Ambient temperature and cooling demands of a real supermarket for one week in July are shown by Figures 5-6. The supermarket used in this study is located in north of Sweden in Piteå town. The measured data are obtained by the field measurement from online interface of IWMAC (IWMAC, 2014) for a week in July 2012. The data is synchronized and averaged over each five minutes.

The focus is on summer period since there is relatively large difference between day and night ambient temperatures. The variation of ambient temperature can reach to 12K (16th of July), as can be seen in Figure 5. Such difference provides great opportunity for the thermal storage system to be charged effectively during night by refrigeration system with rather high efficiency and discharged during day. The stored cold in the storage system can provide extra sub-cooling for the refrigeration system in day improving the efficiency of the refrigeration system. The larger the ambient temperature difference, the higher efficiency of thermal storage system is expected.

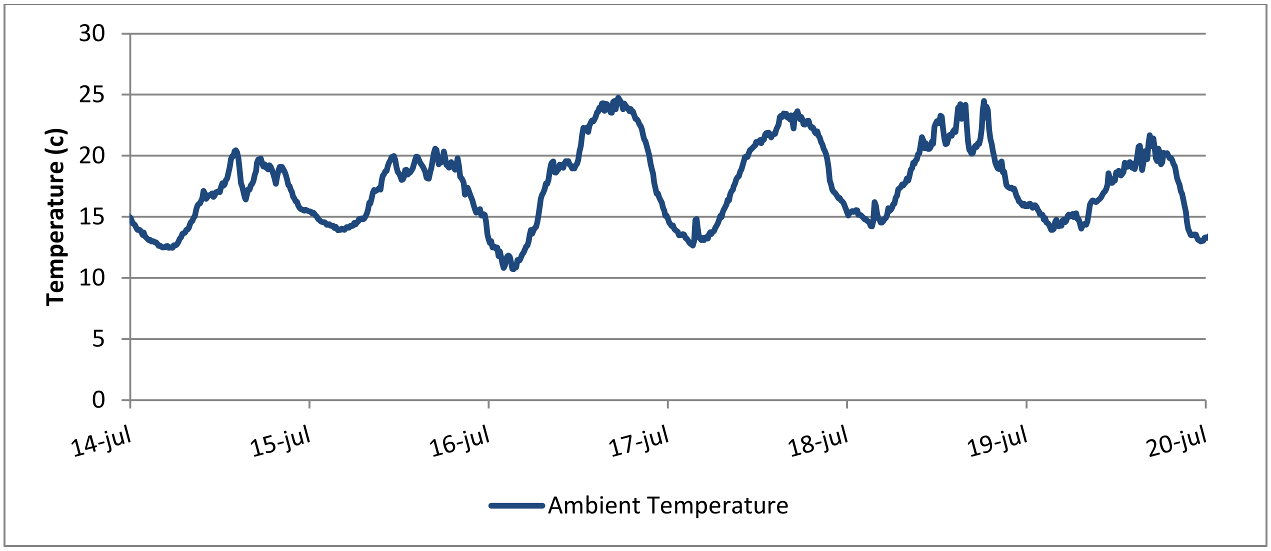

Figure 5 – Ambient temperature profile for a week in July 2012 for a real supermarket in Piteå

Cooling demand for low and medium temperature evaporators are shown in Figure 6. As can be seen the cooling load at low temperature level remains relatively constant about 20kW during 24 hours, but the cooling load at medium temperature varies significantly from 40-50kW during night to 60-90kW during day. Higher cooling load and lower efficiency due to higher ambient temperature during day mean the refrigeration system uses more energy to cover the cooling demand. The concept of short term thermal storage is to charge the storage system in night period where the cooling load on the refrigeration system is lower and the system operates more effectively, and use the stored cold in the day period.

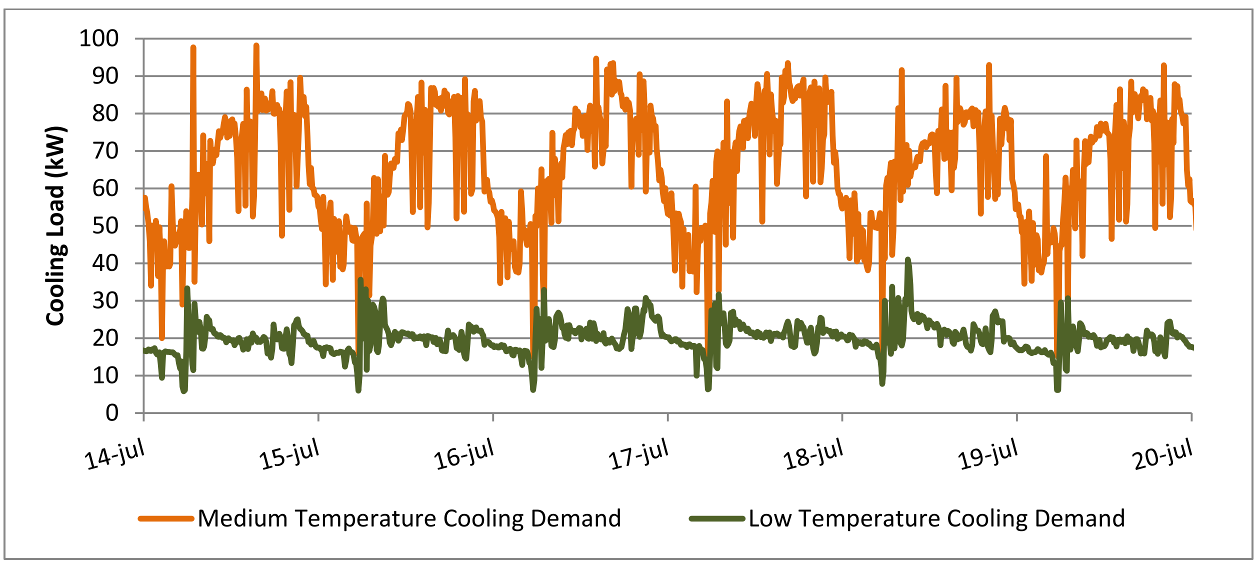

Figure 6 – Cooling demands profile for a week in July 2012 for a real supermarket in Piteå

3.2 How to store energy in short term: Methods & Technologies

Physical thermal storage is divided into two forms; sensible and latent. In sensible type, no phase change occurs and by transferring heat from or to the storage unit, its temperature increases or decreases. The major disadvantage of sensible thermal storage is that it requires rather large volume of storage material. By far, sensible storage is the most common way of heat storage (Mehling, et al. 2008).

In latent type, phase change occurs and large amount of energy can be stored by solidification and melting processes and heat transfer is done with constant temperature of storage material. The advantage of this method is that volume of the storage unit is much lower compared to sensible type. However through the phase change, volume of the storage material might change and this should be considered in design of the storage unit.

In this research both sensible and latent forms of thermal storage are studied. For sensible storage, brine tank is used and for latent storage ice slurry tank and PCM heat exchanger are utilized. The storage system is modelled in connection with the refrigeration system according to the governing equations explained in the following sections.

3.2.1 Brine Tank Modelling

In this type of storage only sensible heat transfer occurs. The storage unit is charged and discharged during night and day by the brine flow returning from the refrigeration system. The tank temperature and stored thermal energy can be calculated by equation 8-9.

$$\frac{dE}{dt} = \dot{m}(Cp_{in}T_{in} - Cp_{out}T_{out}) \quad \textbf{(8)}$$

$$E_{new} = E_{old} - dE \quad \textbf{(9)}$$

Where E represents the energy stored in storage unit and dE is the variation of energy in storage tank per each time step. In this analysis Ethyl alcohol-Water with weight concentration of 29.7% and freezing temperature of -20℃, is used as storage material. The thermo-physical properties of Ethyl alcohol is obtained from Melinder (2007).

3.2.2 Ice Slurry Tank Modelling

Ice slurry is the mixture of water solutions and small ice crystals (Kauffeld, et al., 2005). Usually the solution is consisted of water and freezing points depressants. The depressants are normally the same as additives in secondary fluids like salts and alcohols. The ice crystals are generated by some ice generator continuously which is a very difficult and complicated process (Kauffeld, et al., 2005). The schematic layout of ice generator connected to refrigeration system to produce ice particles is shown by Figure 7.

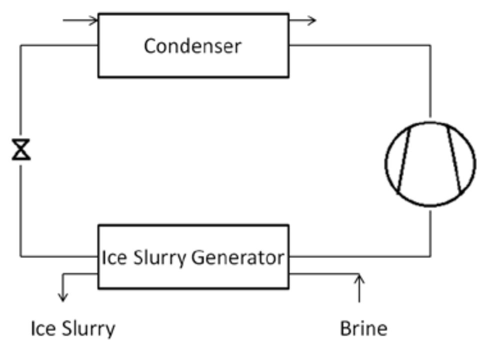

Figure 7 – Schematic layout of Ice Slurry generator in connection with refrigeration system

As can be seen in Figure 7, brine or ice slurry with low concentration of ice crystal enters the ice generator and act as heat source in the charging process. In the ice generator the brine or ice slurry passes through the evaporator, heat is extracted from the solution and concentration of ice increases. In discharging process without ice generator ice slurry is being circulated in contact with the refrigeration system exposed to heat flux. The melting of ice crystals makes large enthalpy change while the temperature of carrier fluid varies few degrees. Since the latent and sensible heat transfer occurs, it is expected to have lower volume of storage compared to brine tank.

The governing correlations to model the ice slurry tank are given by equations 10-11:

$$\frac{\mathbf{dE}}{\mathbf{dt}} = \dot{\mathbf{m}}(\mathbf{h_{in}} - \mathbf{h_{out}}) \tag{10}$$

$$\mathbf{E_{new}} = \mathbf{E_{old}} - \mathbf{dE} \tag{11}$$

In this study, it is aimed to use solution of Ethyl alcohol with additive mass fraction of 0.04 as the material for ice slurry storage tank. It is assumed that the additive mass fraction of the solution is maintained constant in both charging and discharging processes.

3.2.3 Phase change material (PCM)

Using PCM as storage material requires large thermal energy due to phase change in the material from solid to liquid form and vice versa. Large number of PCM materials are investigated and studied in the literature. Hasnain(1998), Farid, et al. (2004) and Agyenim, et al. (2010), reviewed the applicable materials including inorganic and organic materials thoroughly. Inorganic materials are mostly salts and salt hydrates while organic materials are compounds of paraffins and non-parraffins (Hasnain, 1998). Both organic and inorganic materials have advantages and disadvantages. A good PCM should have specific characteristics:

- Melting point of PCM should be in the desired range, so the phase change occurs.
- High latent heat of fusion per unit of volume to store high amount of energy with phase change transition
- High thermal conductivity to facilitate the heat transfer from the fluid to PCM and vice versa.
- High density to use smaller volume
- Small volume change within the phase change otherwise the volume change due to phase transformation should be considered in the design of the container
- Chemical stability, it should not degrade after large number of cycles
- No flammability or corrosively

Some of the PCMs suffer from incongruent melting and supercoiling. Incongruent melting which happens mostly for salt hydrates is phenomena that the material melts with decomposition leading to water forming and lower hydrated salt. Water with lower density remains at top and lower hydrated salts lies in the bottom. Incongruent melting results in lower storage efficiency (Farid, et al., 2004). Supercooling which is another major problem is that the solidification does not start sharply at the freezing point. The material is cooled down to few degrees below the freezing point and then the phase change starts. To avoid incongruent melting and supercoiling some agent can be added. Nowadays PCMs are being produced commercially by several PCM manufacturing companies (RUBITHERM, 2014).

PCM could be used in different container shapes. Agyenim, et al. (2010) reported about slab containers and three types of cylindrical storage unit shown by Figure 8. In pipe model, Figure 8(a), the PCM is located in the shell and heat transfer fluid flows in the tube. In the Cylinder model, Figure 8(b), it is on the way around. The PCM is placed in the shell and the fluid flows in the shell. In the shell and tube model which is a complex of pipe model with PCM in the shell and fluid in the tubes. Also large number of PCM slabs shown by Figure 8(d), can be put aside each other to form rectangular type of container.

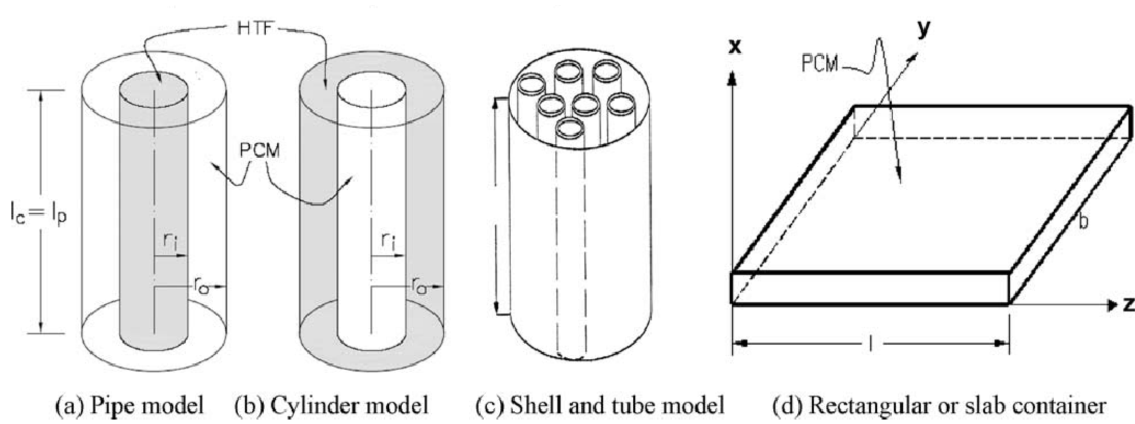

Figure 8 – PCM containers (Agyenim, et al. 2010)

PCM thermal storage modeling

Many researchers have carried out investigations about modelling of heat transfer in PCM containers. In this study, slab container is chosen as PCM storage unit and to do the simulation, Liu (2010) model is used to simulate the thermal storage unit. The model is described briefly in the following section.

The commercial PCM of RT4 which its thermo physical properties are expressed by Table 1 is used. The dimensions of PCM slab is set to be 0.8, 0.4 and 0.05m for length, width and height respectively and the gap between the slabs is considered to be 0.04m.

Ethyl alcohol-Water with weight concentration of 29.7% and freezing temperature of -20℃, is used as heat transfer fluid through the PCM container.

Table 1 (RUBITHERM, 2014)

RT4 Thermo physical properties	
Melting-Freezing point [℃]	4
Latent heat of fusion [kJ/kg]	182
Specific heat [kJ/(kg.K)]	2
Density [kg/m^3]	770-880
Heat conductivity [W/(m.K)]	0.2

The rectangular storage unit shown in Figure 9 is comprised of several PCM slab placed horizontally and in parallel. The heat transfer fluid enters the storage unit, flows among the slabs melting or solidifying the PCM and exits the storage unit. Each slab is divided into several elements in x direction and governing equation is solved for each elements at each time step.

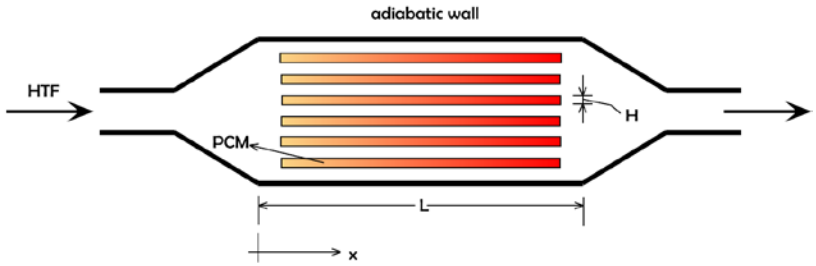

Figure 9 – Rectangular PCM container (Liu, 2010)

In the simulation model following assumptions are taken into account (Liu, 2010):

- The thermo-physical properties of PCM remains constant within single phase
- Axial thermal conduction is neglected
- There is no temperature gradient of heat transfer fluids normal to the direction of flow
- The container is well insulated and there is no heat loss to surroundings
- Thermal resistance between PCM and container wall is ignored.
- Supercooling of PCM is neglected.

Equation 12 represents the governing correlation of energy balance for the heat transfer fluid:

$$\frac{\partial T_f}{\partial t} + \frac{\dot{m}}{\rho_f A_f}\frac{\partial T_f}{\partial x} = \frac{UP}{\rho_f A_f C_f}(T_f - T_w) \quad \textbf{(12)}$$

Where T_f and T_w are fluid and wall temperature respectively (℃), C_f, A_f, ρ_f are specific heat ($\frac{J}{kg.K}$), crossing area (m^2) and density of fluid ($\frac{kg}{m^3}$). The 1st term on the left side of equation represents the fluid temperature variation in time and 2nd term expresses the thermal capacitance of the fluid in the element in x direction. The term on the right side of equation states the convection heat transfer from fluid to the wall or vice versa. Equation 12 is converted from partial differential form to algebraic equation by using backward time and upwind scheme approximation expressed by equations 13-14 where i and j are the symbols of element location and time respectively:

$$\frac{\partial T_f}{\partial t} = \frac{T_f{}^j(i) - T_f{}^{j-1}(i)}{\Delta t} \quad \textbf{(13)}$$

$$\frac{\partial T_f}{\partial x} = \frac{T_f{}^j(i) - T_f{}^j(i-1)}{\Delta x} \quad \textbf{(14)}$$

Equations 15-16 expresses the convection heat transfer from the fluid to the wall and conduction in the slab wall.

$$\mathbf{q_{Convection}}^{j}(\mathbf{i}) = \mathbf{UA}(\mathbf{T_f}^{j}(\mathbf{i}) - \mathbf{T_w}^{j}(\mathbf{i})) \quad \textbf{(15)}$$

$$\mathbf{q_{Condution}}^{j}(\mathbf{i}) = \frac{\mathbf{k_W A}(\mathbf{T_f}^{j}(\mathbf{i}) - \mathbf{T_w}^{j}(\mathbf{i}))}{\mathbf{dX_W}} \quad \textbf{(16)}$$

Where U is the convective heat transfer coefficient $(\frac{\mathrm{W}}{\mathrm{m^2K}})$, A the wall area which is in contact with heat transfer fluid $(\mathrm{m^2})$, $\mathrm{k_W}$ the thermal conductivity of metal between the fluid and PCM and $\mathrm{dX_W}$ is length of the element (m). The convective heat transfer coefficient is calculated according to equation 17-19 (Liu, 2010):

$$\mathbf{U} = \frac{(\mathbf{Nu\ k_f})}{\mathbf{D_e}} \quad \textbf{(17)}$$

$$\bar{\mathbf{x}} = \frac{(\frac{\mathbf{x}}{\mathbf{D_e}})}{(\mathbf{Re.Pr})} \quad \textbf{(18)}$$

$$\mathbf{Nu} = \begin{cases} \mathbf{1.223(\bar{x})^{-(\frac{1}{3})} + 0.4} & \bar{\mathbf{x}} \le \mathbf{0.001} \\ \mathbf{7.541 + 6.874(10^3\ \bar{x})^{-0.488}\ e^{-245\bar{x}}} & \bar{\mathbf{x}} > \mathbf{0.001} \end{cases} \quad \textbf{(19)}$$

Where Nu, $\mathrm{D_e}$, Re and Pr are Nusselt number, equivalent diameter (m), Reynolds and Prandtl number. The initial and boundary conditions are expressed by equation 20-22 (Liu, 2010):

$$\mathbf{T_f}^{\mathbf{j=0}}(\mathrm{i}) = \boldsymbol{T_{initial}} \quad \textbf{(20)}$$

$$\frac{\partial \mathbf{T_f}}{\partial \mathbf{x}}\ (\boldsymbol{i} = \mathbf{0}\ ) = \ \mathbf{0} \quad \textbf{(21)}$$

$$\frac{\partial \mathbf{T_f}}{\partial \mathbf{x}}\ (\boldsymbol{i} = \boldsymbol{L}\ ) = \ \mathbf{0} \quad \textbf{(22)}$$

As the heat is transferred from fluid to PCM node, PCM can be in either solid or liquid or in transition form. So depending on the state of PCM, heat can be used to increase or decrease the element temperature or change the liquid fraction. The melting and freezing process are shown by Figure 10.

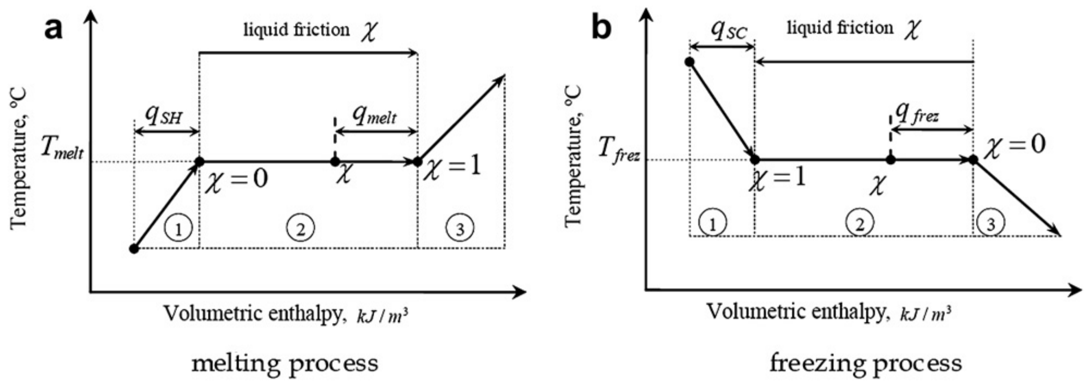

Figure 10 – Schematic profile of melting and solidification processes (Liu, 2010)

In order to solve the problem in the PCM node four new terms are introduced and expressed by equation 23-26: q_{SH}, q_{melt}, q_{SC}, q_{frez} where q_{SH} and q_{melt} are the heat required to increase the node temperature to melting temperature and the heat needed to melt the entire node respectively. On the other side, q_{SC} and q_{frez} are the heat needed to decrease the node temperature to freezing temperature and the required heat to freeze the entire element.

$$\mathbf{q_{SH}}^{\mathbf{j-1}}(\mathbf{i}) = \boldsymbol{\rho}_{\mathbf{s}}\left(\frac{\Delta \mathbf{V}}{\Delta \mathbf{t}}\right)\mathbf{C_s}[\mathbf{T_{melt}} - \mathbf{T_{node}}^{\mathbf{j-1}}(\mathbf{i})] \tag{23}$$

$$\mathbf{q_{melt}}^{\mathbf{j-1}}(\mathbf{i}) = [\mathbf{1} - \mathbf{x_{node}}^{\mathbf{j-1}}(\mathbf{i})]\boldsymbol{\rho}_{\mathbf{l}}\left(\frac{\Delta \mathbf{V}}{\Delta \mathbf{t}}\right)\boldsymbol{\lambda} \tag{24}$$

$$\mathbf{q_{SC}}^{\mathbf{j-1}}(\mathbf{i}) = \boldsymbol{\rho}_{\mathbf{l}}\left(\frac{\Delta \mathbf{V}}{\Delta \mathbf{t}}\right)\mathbf{C_l}[\mathbf{T_{frez}} - \mathbf{T_{node}}^{\mathbf{j-1}}(\mathbf{i}) \tag{25}$$

$$\mathbf{q_{frez}}^{\mathbf{j-1}}(\mathbf{i}) = \mathbf{x_{node}}^{\mathbf{j-1}}(\mathbf{i})\boldsymbol{\rho}_{\mathbf{l}}\left(\frac{\Delta \mathbf{V}}{\Delta \mathbf{t}}\right)\boldsymbol{\lambda} \tag{26}$$

The governing equations to calculate the node temperature and liquid fraction of the element depend on its state. The correlations for melting and freezing process are expressed by Table 2 - 3 respectively. For instance in solid state where the node temperature is lower than melting temperature, the governing equation depends on the amount of transferred heat. If the transferred heat (q_{Conv}) is less than the heat required for the PCM to reach melting temperature (q_{SH}), PCM node remains in solid state (x_{node}=0) and the node temperature will be raised. If the transferred heat (q_{Conv}) is more than the heat required for the PCM to reach melting temperature (q_{SH}), PCM node enters the two phase state and the node temperature reaches the melting temperature.

The liquid fraction expresses the fraction of liquid in PCM node when it is in two phase state. The density ($\frac{kg}{m^3}$) and specific heat of PCM ($\frac{J}{kg.K}$) in two phase state are calculated by equation 27-28:

$$\rho_k = \rho_l x + \rho_s(1 - x) \quad (27)$$

$$C_k = C_l x + C_s(1 - x) \quad (28)$$

Table 2 - (Liu, 2010)

Solid state to two phase state $T_{node}^{j-1}(i) < T_{melt}$	
$q_{Conv}^{j}(i) < q_{SH}^{j-1}(i)$	$q_{Conv}^{j}(i) = \rho_s \left(\frac{\Delta V}{\Delta t}\right) C_s [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 0$
$q_{Conv}^{j}(i) = q_{SH}^{j-1}(i)$	$T_{node}^{j}(i) = T_{melt}$ $x_{node}^{j}(i) = 0$
$q_{Conv}^{j}(i) > q_{SH}^{j-1}(i)$	$T_{node}^{j}(i) = T_{melt}$ $q_{Conv}^{j}(i) - q_{SH}^{j-1}(i) = [x_{node}^{j}(i) - x_{node}^{j-1}(i)]\, \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda$
Two phase state to liquid state $T_{node}^{j-1}(i) = T_{melt}$	
$q_{Conv}^{j}(i) < q_{melt}^{j-1}(i)$	$T_{node}^{j}(i) = T_{melt}$ $q_{Conv}^{j}(i) = [x_{node}^{j}(i) - x_{node}^{j-1}(i)]\, \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda$
$q_{Conv}^{j}(i) = q_{melt}^{j-1}(i)$	$T_{node}^{j}(i) = T_{melt}$ $x_{node}^{j}(i) = 1$
$q_{Conv}^{j}(i) > q_{melt}^{j-1}(i)$	$q_{Conv}^{j}(i) - q_{melt}^{j-1}(i) = \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 1$
liquid state $T_{node}^{j-1}(i) > T_{melt}$	
$q_{Conv}^{j}(i) = \rho_l \left(\frac{\Delta V}{\Delta t}\right) C_l [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 1$	

The same sets of equations are expressed for freezing process in Table 3.

Table 3 - (Liu, 2010)

Liquid state to two phase state $T_{node}^{j-1}(i) > T_{frez}$	
$\lvert q_{Conv}^{j}(i)\rvert < \lvert q_{SC}^{j-1}(i)\rvert$	$q_{Conv}^{j}(i) = \rho_l \left(\frac{\Delta V}{\Delta t}\right) C_l [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 1$
$\lvert q_{Conv}^{j}(i)\rvert = \lvert q_{SC}^{j-1}(i)\rvert$	$T_{node}^{j}(i) = T_{frez}$ $x_{node}^{j}(i) = 1$
$\lvert q_{Conv}^{j}(i)\rvert < \lvert q_{SC}^{j-1}(i)\rvert$	$T_{node}^{j}(i) = T_{frez}$ $q_{Conv}^{j}(i) - q_{SC}^{j-1}(i) = [x_{node}^{j}(i) - x_{node}^{j-1}(i)]\, \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda$
Two phase state to solid state $T_{node}^{j-1}(i) = T_{frez}$	
$\lvert q_{Conv}^{j}(i)\rvert < \lvert q_{frez}^{j-1}(i)\rvert$	$T_{node}^{j}(i) = T_{frez}$ $q_{Conv}^{j}(i) = [x_{node}^{j}(i) - x_{node}^{j-1}(i)]\, \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda$
$\lvert q_{Conv}^{j}(i)\rvert = \lvert q_{frez}^{j-1}(i)\rvert$	$T_{node}^{j}(i) = T_{frez}$ $x_{node}^{j}(i) = 0$
$\lvert q_{Conv}^{j}(i)\rvert > \lvert q_{frez}^{j-1}(i)\rvert$	$q_{Conv}^{j}(i) - q_{frez}^{j-1}(i) = \rho_k \left(\frac{\Delta V}{\Delta t}\right) \lambda [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 0$
Solid state $T_{node}^{j-1}(i) < T_{frez}$	
$q_{Conv}^{j}(i) = \rho_s \left(\frac{\Delta V}{\Delta t}\right) C_s [T_{node}^{j}(i) - T_{node}^{j-1}(i)]$ $x_{node}^{j}(i) = 0$	

3.3 Short term thermal storage case studies

Different possible system solutions with short term thermal storage are evaluated in this section. The aim is to see how much daily energy savings can be gained by each scenario at different conditions.

3.3.1 Thermal storage at medium temperature (TSMT)

The idea in the first case illustrated in Figure 11 is to use the stored cold fluid in the thermal storage unit for sub-cooling of the refrigeration system. The storage unit is charged by a separate evaporator in medium temperature level during night time. During day time, the storage unit with low temperature provides sub-cooling for the system where the ambient temperature increases.

In this case, the evaporation in separate evaporator is done at the same temperature of medium temperature cabinets; which is -10℃. The refrigerant exiting from this evaporator is compressed by the high stage compressor after mixing with other flows. Internal superheat for the separate evaporator is considered to be 5K. Assuming an approach temperature difference of 2K at the evaporator inlet, the temperature of storage unit cannot be reduced lower than -3℃. In the evaporator 5K of temperature drop is assumed for the working fluid.

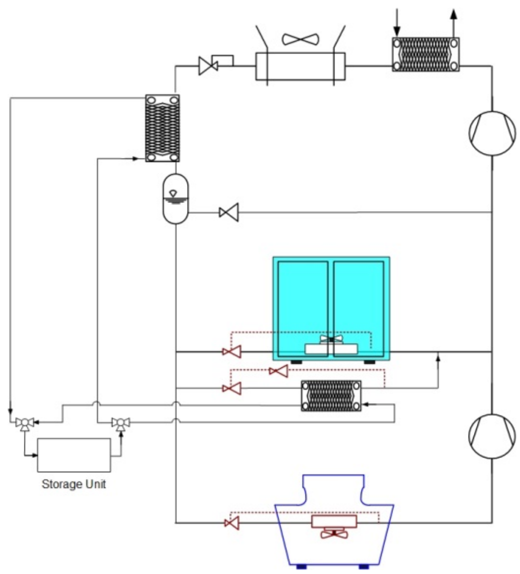

Figure 11 – Schematic layout of TSMT scenario, thermal storage at medium temperature

In this scenario brine tank and PCM container can be used as the storage units. There is limitation to use ice slurry in this case since ice slurry should be used in the temperature range lower than the freezing point. The freezing temperature for Ethyl Alcohol-Water with additive concentration of 0.04 which was presented in previous section is about -1.7℃ (Melinder, 2007). It means that if this solution would be used as the working fluid for ice slurry tank, the range of temperature change will be from -3 to -1.7℃ which this small temperature change results in low enthalpy difference and consequently large volume. Regarding this limitation, only brine and PCM are used for the storage unit.

Discharging the storage unit is done by the sub-cooler located after the gas cooler. The working fluid (Ethyl Alcohol-Water) coming from storage unit provides extra sub-cooling for the refrigeration system. The working fluid with high temperature returns to the storage unit and increases the storage unit temperature. The approach difference at the inlet and outlet is assumed to be 8℃, this assumption results in effectiveness of sub-cooler between 0.4-0.6.

3.3.2 Thermal storage at higher temperature (TSHT)

The 2nd scenario shown by Figure 12 has similar thermal storage concept to the 1st scenario, but the process of charging the storage tank is done at higher evaporation temperature. The charging evaporator is connected to a separate compressor which enables the evaporator to cool down the working fluid at higher evaporation temperature. Working at higher evaporation temperature requires lower charging power which can lead to higher energy savings. But on the other side, the minimum temperature which the storage unit can reach is higher compared to 1st scenario and higher storage tank volume is needed for the same storage capacity.

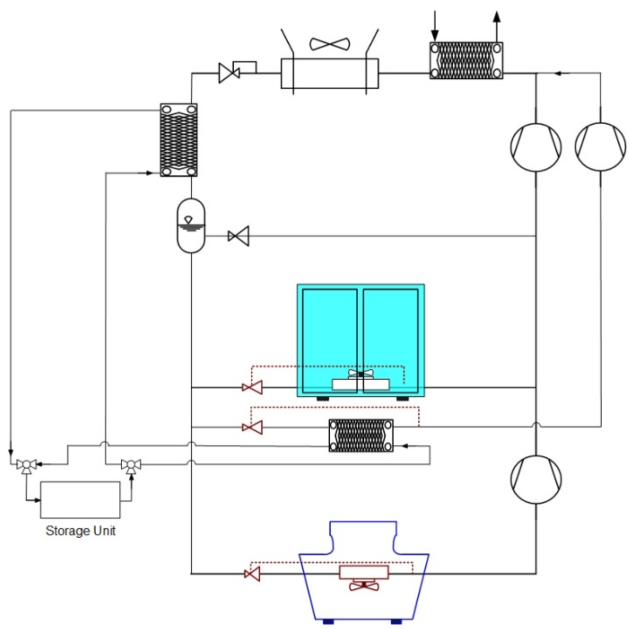

Figure 12 - Schematic layout of TSHT scenario, thermal storage at higher temperature

3.3.3 Storage in the frozen food (SFF)

This scenario focuses on the thermal storage by the mass of products in the low temperature cabinets. This case is done specifically for freezing cabinets. The idea is that energy can be stored by the product itself during night time when the system has rather good efficiency. The temperature of product can be reduced lower than the required temperature. During day time as the system works with lower efficiency, lower power is consumed and temperature of products is raised to the maximum allowable level. The major advantage of this system is that no extra cost is required. This scenario is merely a matter of control strategy to exploit the advantage of the temperature difference between day and night.

It is assumed that lowering the temperature of freezing products has no effect on the evaporation temperature. The fans in the cabinets have constant speed but changing the set point to lower air temperature will results in longer running time of the compressors when the load is low at night. The average allowable temperature at which the products should be maintained is considered to be -18℃. The product temperature is lowered down to -22℃ during night and then raised again during day time.

In this section for the input parameters, the energy profiles described in section 3.1 is not used. For this part, the simulated cooling load for a real super market from CyberMart software is used and the load for one freezing cabinet is calculated (Arias, 2005).The cabinet type is Electrolux-UB 250-2 with dimensions of 2.5, 0.82 and 0.3 m. 80% of the product in the cabinet is considered to be water and analysis is done for this amount of water.

3.4 Simulation results for short term thermal storage scenarios

Each scenario is simulated using the energy profiles and outdoor temperature as input parameters for 24 hours. The result for each scenario is compared to the reference case described in section 2.1 and savings in energy consumption of the refrigeration system is calculated. Daily energy savings are calculated by comparing the daily energy use of this case study to the reference case, expressed in section 2.1.

3.4.1 Results for TSMT scenario (Thermal storage at medium temperature)

The calculations for this scenario are done for the period of 18th of July, 2012 for the refrigeration system of the supermarket specified in section3.1. The storage volume is 10 and 2.5 m^3 for brine and PCM respectively. The results are shown in Figure 13-14. Figure 13 shows the temperature profile of brine tank and the average temperature of fluid in PCM container. The charging process starts in night time and stops at 6:00 A.M. Then the discharging process starts from 9:00 A.M until the temperature of storage material reaches its initial temperature. Brine tank reaches the minimum temperature by the end of charging process but fluid average temperature in PCM container reaches temperature of 4℃.

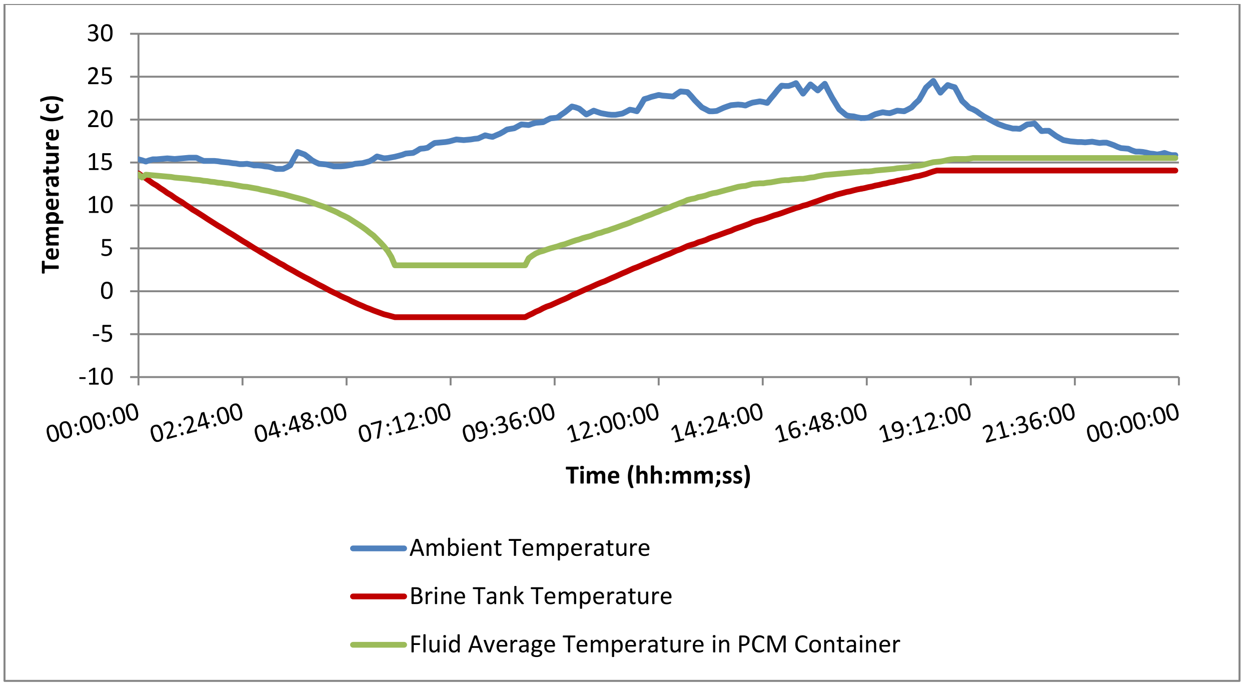

Figure 13 – Temperature profile for storage unit and outdoors for 18th of July, 2012

Cooling COP of the system for both brine and PCM is illustrated in Figure 14. During the charging process the COP of the system with storage is the same as the reference case. When the discharging process starts, the COP of the system with storage is higher compared to reference case due to additional sub-cooling provided by storage unit.

At the start of the discharging process, brine case gives higher COP since the brine tank temperature is lower than the fluid temperature exiting the PCM container. But at the end of the discharging process, the PCM case gives higher COP since the outlet temperature of fluid from PCM container is lower than the brine tank temperature which cannot be seen in Figure 13 since the average temperature of PCM is plotted.

The daily savings of energy use for brine and PCM is 2.6 and 6% respectively which seems to be low considering the quite large volume of storage units. In brine case the storage unit is $10\mathrm{m}^3$ and in Case of PCM the container volume is about 1.8 times the PCM volume. High power consumption of the refrigeration system during charging process of thermal storage unit results in low saving in daily energy use. If the temperature difference between day and night and consequently the difference in COP of reference refrigeration system would be more prominent, higher energy savings will be obtained.

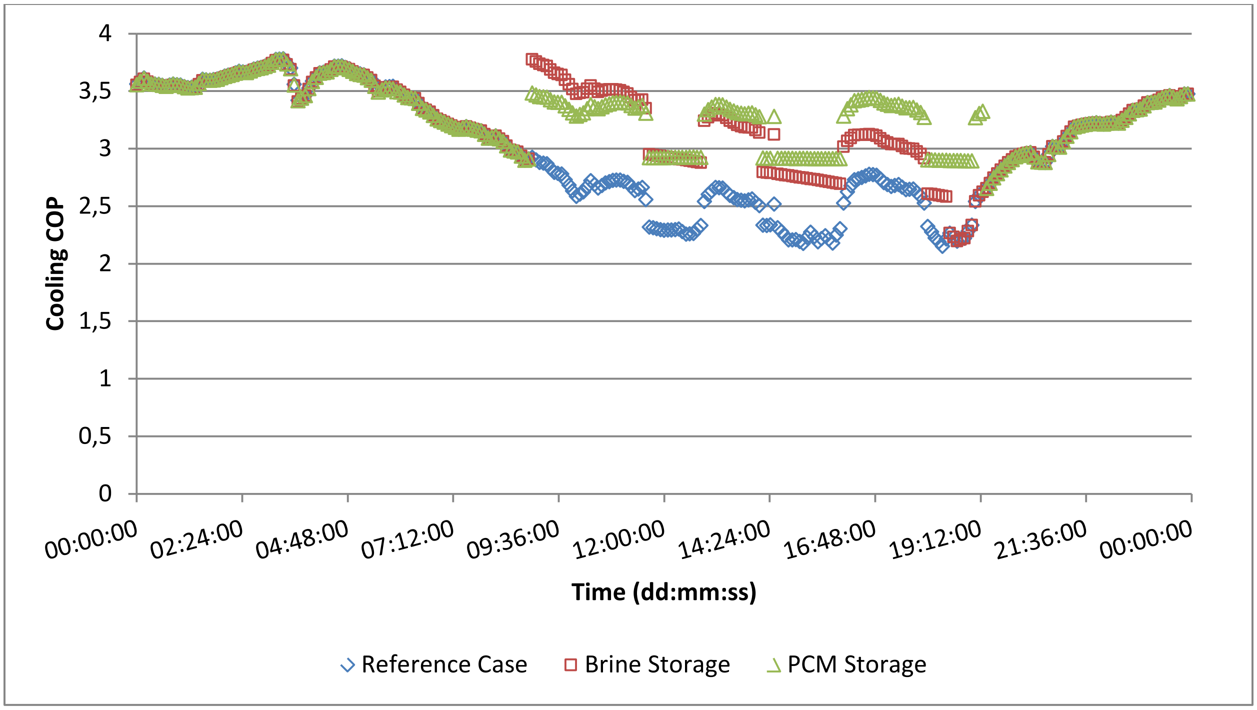

Figure 14 – Cooling COP of the system with and without thermal storage (brine and PCM for one day)

To have better picture of daily energy saving numbers for given storage volume the simulation is done for different volumes for 3 days of 16th, 17th and 18th of July. The energy use savings against the volume is illustrated in Figure 15-16 for brine and PCM respectively.

The savings curve for each day is different due to different ambient temperature profile. For instance 16th of July where the gap between maximum and minimum temperature is larger, the storage system gives generally higher energy savings; however, the difference is insignificant. The days of 17th and 18th of July have lower energy use savings since the ambient temperature varies to lower extent from night to day.

There is saving peak for each day. The results show the larger volume makes higher energy use savings but after the peak point, the energy use saving numbers reduces as the volume increases. It is because there is not enough load on refrigeration system to use the stored energy and storage unit temperature does not reach the initial temperature before charging process. In this study, calculation is done specifically for one day. If the calculations proceed to the following day then the energy needed to charge the system will be lower and higher energy use savings can be achieved. The peak shown in Figure 15 is due to certain calculations for one day.

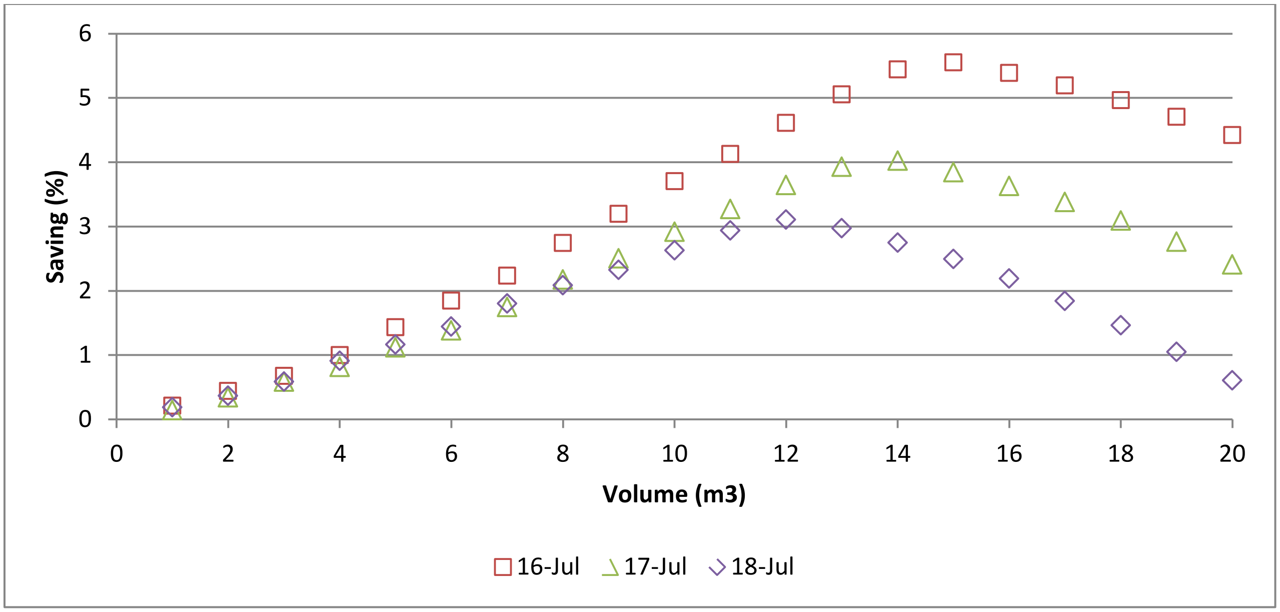

Figure 15 – Daily energy use savings vs. volume of brine tank storage for three different days

For the 18th of July the maximum energy use saving with brine tank is about 3% with tank volume of 12m^3. The highest amount of energy use saving for PCM at the same day is about 10% with PCM volume of 4.5m^3, such PCM volume means that the volume of PCM container will be about 8m^3. Higher maximum energy use savings are obtained for other days but at larger volume.

The results show that quite large storage volume is needed to have reasonable energy savings; to save 3-4 % of energy use more than 10m3 if brine will be needed, if PCM is used to save 4% about 3m3 of PCM and 5,4m3 of PCM container will be needed. The calculations in following scenario is done to see how the energy saving figures change if storage units are operated at higher temperature.

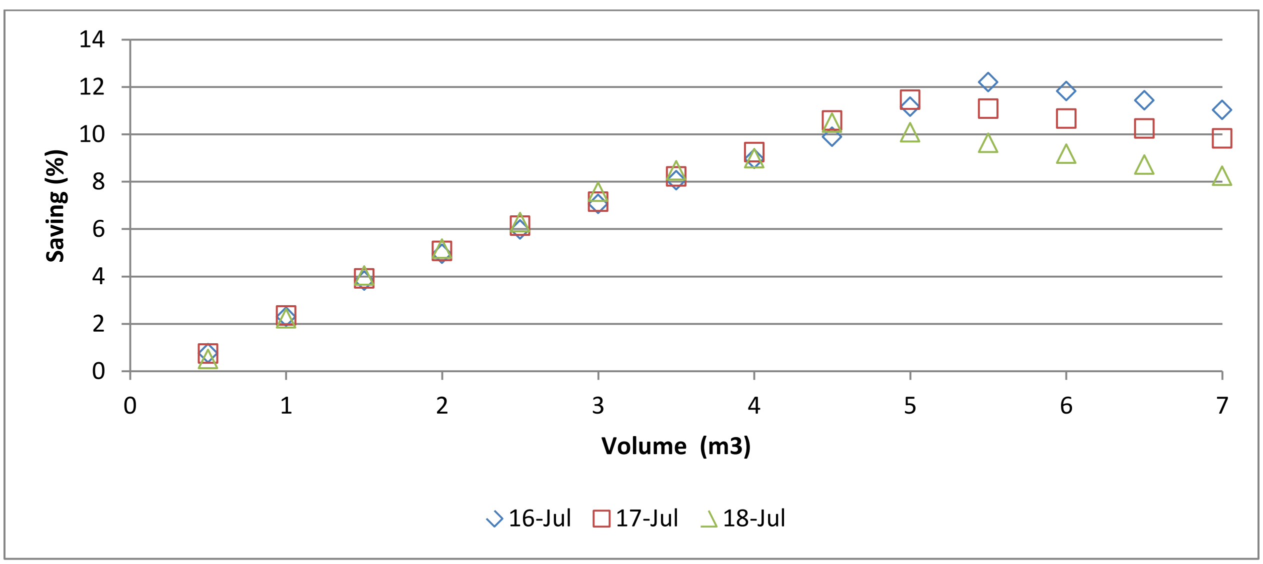

Figure 16 - Daily energy use savings vs. volume of PCM storage unit for three different days

3.4.2 Results for TSHT scenario (Thermal storage at higher temperature)

In this scenario charging the storage system at higher evaporation temperature is investigated. The results are shown in Figure 17-18. Daily energy use savings are calculated by comparing the daily energy use to the reference case, expressed in section 2.1.

Figure 17 shows the daily energy use savings against the evaporation temperature during charging time for tank volume of 8, 10 and 12m^3. The maximum energy use saving is achieved at evaporation temperature of -6 to -8℃. The increase in energy use savings is insignificant and for evaporation temperatures higher than -6℃ lower savings can be observed. The comparison between different volumes shows that at low evaporation temperatures, the increase in energy use savings is higher for larger volumes.

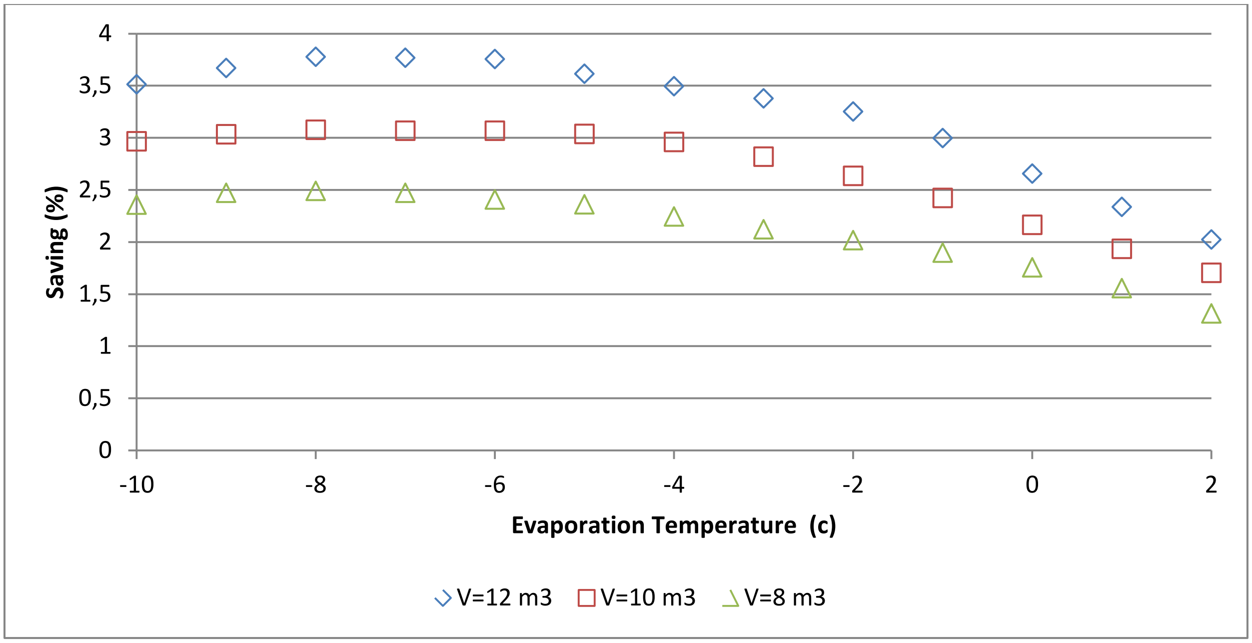

Figure 17 - Daily energy use savings VS. evaporation temperature of charging for brine

The same results for PCM can be observed in Figure 18. The analysis is done for PCM volumes of 1, 2, 3 and 4m^3. For low volumes there is no increase in saving but for higher volumes the maximum energy use savings is at evaporation temperature of about -7 to -9℃. After the peak point the amount of energy use savings reduces steeply.

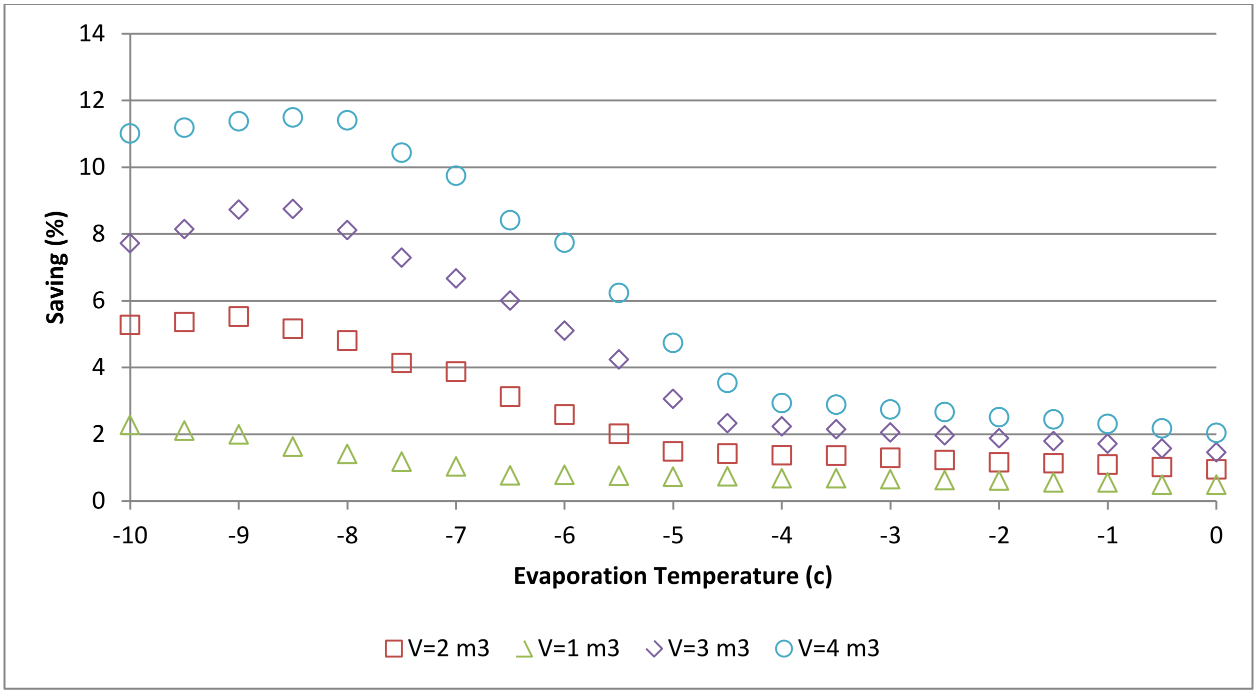

Figure 18 - Daily energy use savings vs. evaporation temperature of charging for PCM

Generally, this scenario has little improvement compared to the charging the storage system with evaporation temperature of -10℃. As the evaporation temperature increases lower energy use saving is achieved except the thermal storage at evaporation temperatures close to reference temperature (-10). There is slight increase in savings for large volumes at evaporation temperature slightly higher than the reference case temperature.

3.4.3 Results for storage in the frozen food (SFF)

In this part, the storage by frozen food in the low temperature cabinets is evaluated. As explained in section 3.3.3, 80% of the freezer volume (2.5x0.92x0.3 m^3) is assumed to be water and calculations are done for this amount of water for one freezer. The cooling load for one freezer cabinet is simulated and calculated by CyberMart software for a real supermarket (Arias, 2005).

The calculation result about the storage by frozen food in freezer is shown in Figure 19. Analysis is done for 24 hours, 8th of July. The charging is done during night from 0:00 to 7:00 A.M and discharging is carried out from 10:00 to 8:00 P.M. The saving in power consumption of compressors corresponding to one freezer is insignificant about 0.4% of the reference case. The reason for low saving is that the stored energy in the products corresponding to 4K temperature change, is insignificant respect to the cooling load of the cabinets. To achieve higher saving, higher thermal mass is needed to be stored.

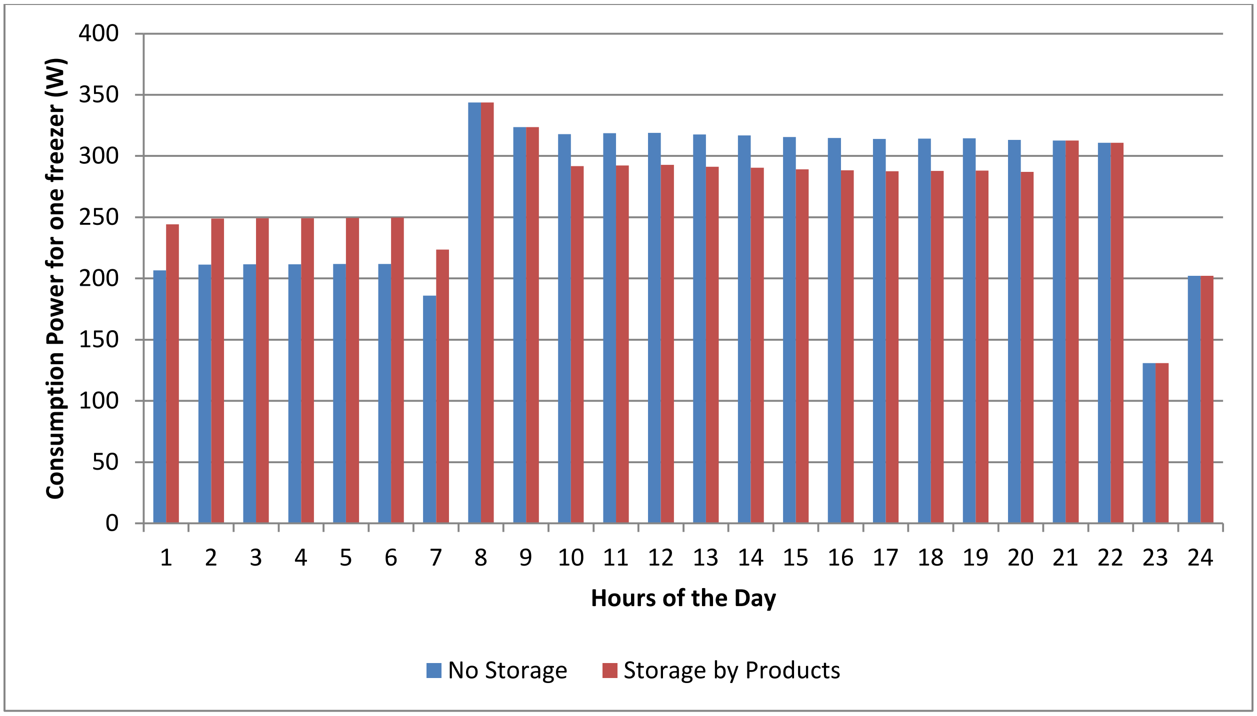

Figure 19 – Power consumption for a freezer for the 8th of July, SFF case

For increasing the thermal mass in the freezers, PCM can be used and attached to the wall inside the freezers. Due to phase change in PCM higher thermal mass can be stored during night when the efficiency is relatively high. Some researchers such as Azzouz et al. (2008) and Raeisi et al. (2013) have studied concept of using PCM in cabinets experimentally. This can be matter of further investigations for the future. Thermal storage by the help of both frozen products and PCMs in freezers can be modeled and studied.

4 LONG TERM (SEASONAL) THERMAL STORAGE

4.1 Simulation parameters

The aim of this chapter is to evaluate the potential of long term thermal energy storage in supermarkets. The simulation needs several input parameters such as seasonal cooling and heating demands of the supermarket. This chapter describes the input parameters to be used in the long term simulation and the design of the thermal energy storage. Different simulation cases are expressed and the simulation results are discussed as well.

4.2 Seasonal cooling & heating demands profiles

The first input parameter for the simulation is the energy profiles of the supermarket. This includes the medium and low temperature cooling demands 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 (Arias, 2005). The chosen supermarket is located in Stockholm, Sweden and is quite large one with an area of 14000 m2.

Figure 20 shows the hourly cooling demand of the supermarket for both medium and low temperature levels as a function of the outdoor 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 at around 65 kW.

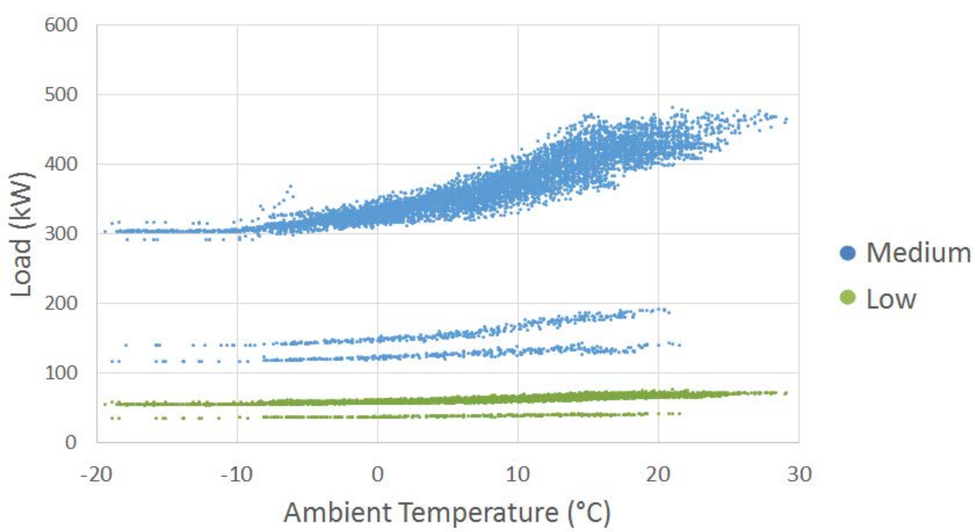

Figure 20 - Medium and low temperature cooling demand as a function of ambient temperature

Figure 21 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.

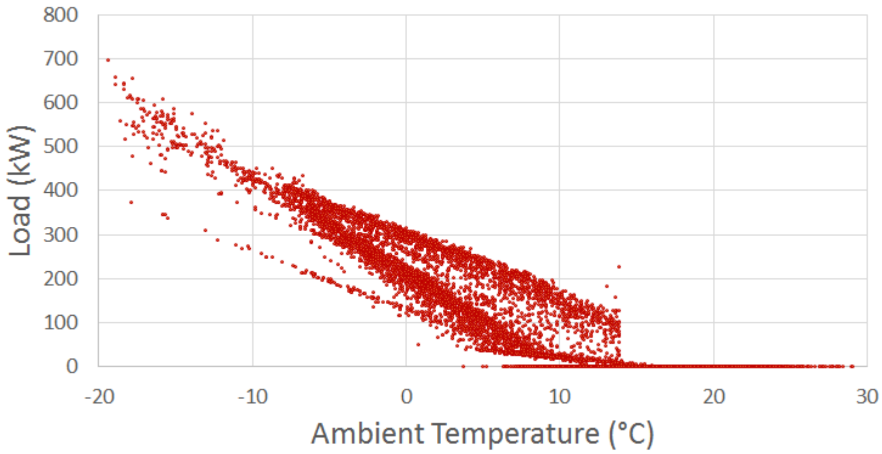

Figure 21 - Heating demand as a function of temperature

4.2.1 Long term thermal energy storage technology

This study focuses on seasonal thermal storage, thus this section describes the design of borehole thermal energy storage, which includes the rock, the brine, 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 ($\frac{W}{m.K}$), rock density of 2700 kg/m3, rock specific heat capacity of 830 ($\frac{J}{kg.K}$), 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 ($\frac{W}{m.K}$), 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 ($\frac{W}{m.K}$). As Figure 22 shows, the borehole has a diameter of 110 mm and a shank spacing of 70 mm, the distance between the centers of the pipes.

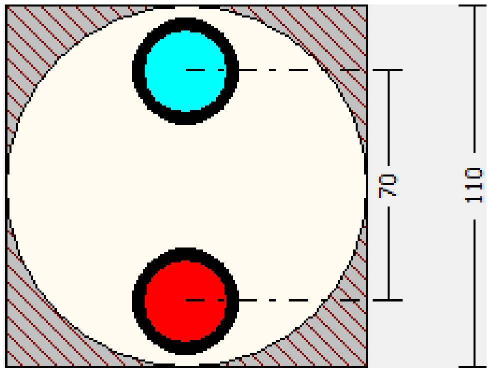

Figure 22 - Cross sectional view of a borehole

The Earth Energy Designer software uses the above mentioned properties of the borehole to calculate the effective thermal resistance as 0.148 ($\frac{\mathrm{m.K}}{\mathrm{W}}$). 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):

$$T_{br}(t) = \frac{q}{4\pi\lambda_{rock}} \times \left(ln\left(\frac{4\alpha_{rock}t}{r_{bh}^2}\right) - \gamma \right) + q \times R_{bh} + T_0 \tag{29}$$

In the above equation, q is the heat extraction rate (W), λ_{rock} is the rock thermal conductivity ($\frac{\mathrm{W}}{\mathrm{mK}}$), α_{rock} is the rock thermal diffusivity ($\frac{\mathrm{m}^2}{\mathrm{s}}$), r_{bh} is the borehole radius (m), γ is Euler's constant, R_{bh} is the effective borehole thermal resistance as described above, and T0 is the undisturbed ground temperature. The heat extraction rate, q, is not constant and is given by:

$$q(t) = \begin{cases} q_1, & t_1 < t < t_2 \\ q_2, & t_2 < t < t_3 \\ & \quad \dots \\ q_n, & t_{n-1} < t < t_n \end{cases} \tag{30}$$

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

$$T_{br}(t) = \sum_{n=1}^{N} \left(\frac{q_n - q_{n-1}}{4\pi\lambda_{rock}} \times \ln(t - t_n) \right) + \frac{q_N}{4\pi\lambda_{rock}} \times \left(ln\left(\frac{4\alpha_{rock}}{r_{bh}^2}\right) - \gamma \right) + q_N \times R_{bh} + T_0 \tag{31}$$

4.3 Seasonal thermal storage case studies

Several case scenarios are devised to show the effect of long term thermal energy storage on the energy use in the supermarket. The cases are combinations of different heating systems and control strategies of the storage. Overall there are four cases, detailed below.

4.3.1 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 23. The number of boreholes is 20. For a fair comparison, the same number of boreholes is considered for the other scenarios.

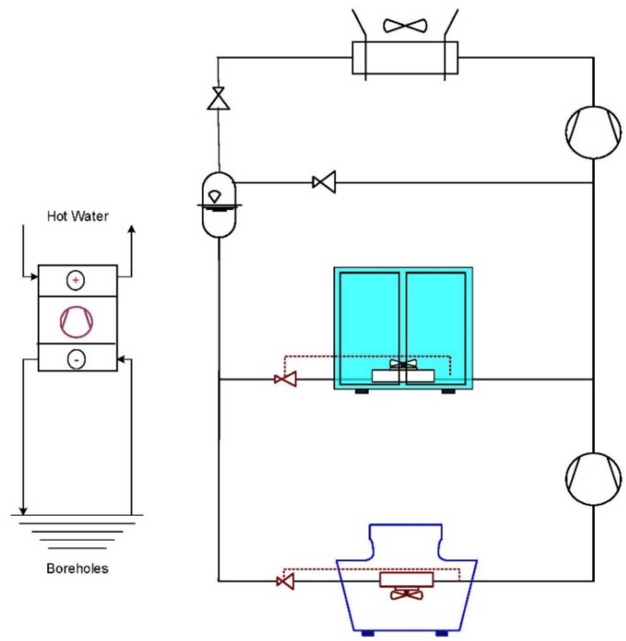

Figure 23 - Floating condensing and heat pump (FC+HP)

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

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

In the case, the boreholes are performing as thermal energy storage system. The brine is used for both heat pump and refrigeration system. The borehole brine is supposed to sub-cool the gas cooler output even when the heat pump is not working. This case, abbreviated as FC+BTES, is shown in Figure 16.

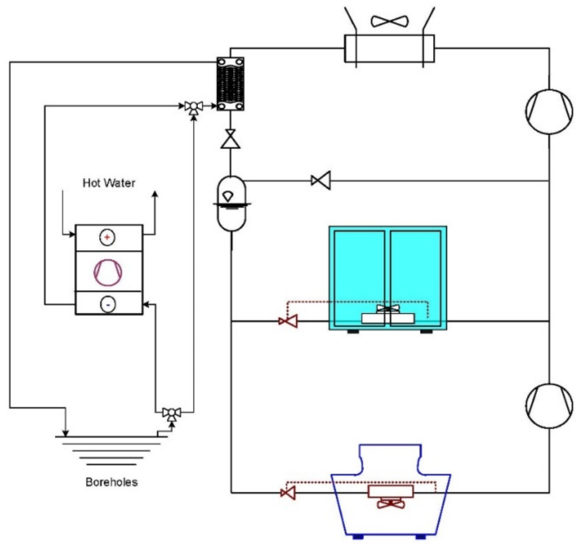

Figure 24 - Floating condensing with borehole thermal energy storage (FC+BTES)

4.3.3 Complete heat recovery (CHR)

The third case consists only of the refrigeration system. The heating is provided by heat recovery, shown in Figure 25. With the control strategy described in section 2.2, the heat recovery supplies almost all 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.

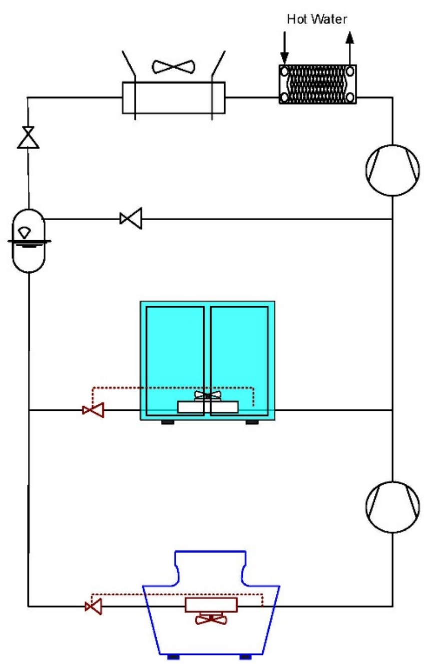

Figure 25 - Complete heat recovery (CHR)

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

The final scenario adds borehole thermal energy storage to the previous case, where the heating is provided completely by the heat recovery. As modeled in Figure 26, the medium temperature level evaporator cools the ground during the colder days of winter, when the heating demand is higher than what the refrigeration system can provide. Then the stored cooling energy in borehole is used to sub-cool the gas cooler output during the summer. This scenario, abbreviated as CHR+BTES.

Since the modelled refrigeration system in this study can cover almost all the heating demand needed then this scenario was not calculated. However it is an interesting possibility for installation with quite large heating demands.

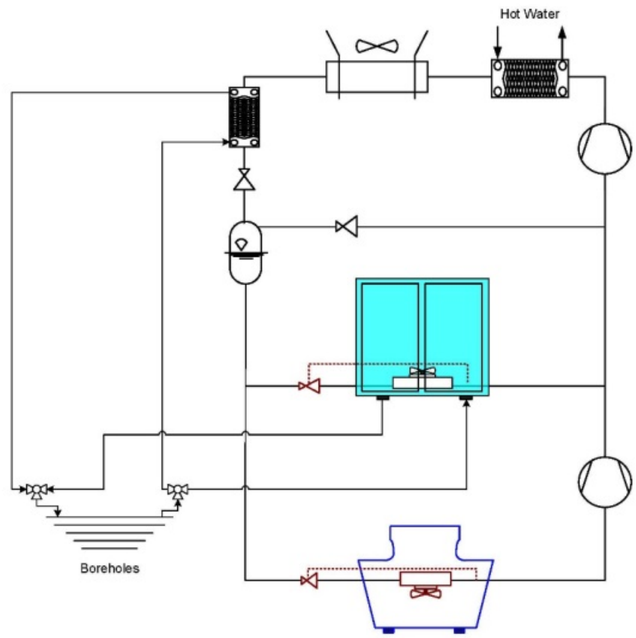

Figure 26 - Complete heat recovery with borehole thermal energy storage (CHR+BTES)

4.4 Simulation results of long term thermal storage scenarios

With the input parameters described in section 4.1.1, a simulation is carried out on the performance of the scenarios, over a period of 10 years. The simulation results include cooling and heating COP, the temperature of the borehole brine and the power needed to run the refrigeration and the heating systems.

The following sections compare the results of the simulations of each case study to the one of the reference case. The results shown in the following sections present only the values of the last year of the simulation period.

4.4.1 FC+HP vs. FC+BTES

This is a comparison of FC+BTES with the reference case. The objective of this comparison is to show the effect of thermal energy storage on a refrigeration system operating in floating condensing mode.

The cooling COP at medium temperature for the two scenarios is shown in Figure 27. It can be seen that the scenario with the borehole thermal energy storage has a higher cooling COP for all ambient temperatures. As can be seen in Figure 27 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.

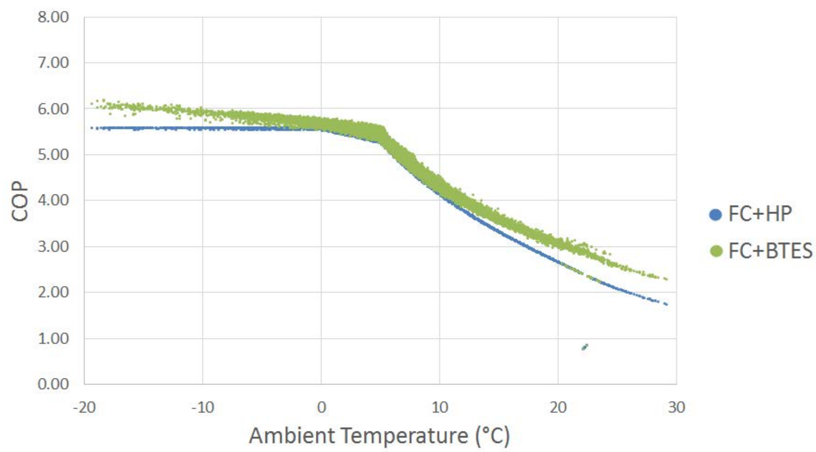

Figure 27 - Cooling COP as a function of ambient temperature, FC+HP vs. FC+BTES

It can be noted that in floating condensing mode in high ambient temperatures, the cooling COP drops. The FC+BTES has higher COP at high ambient temperatures compared to reference case which is due to extra sub-cooling made by borehole.

The heating COP (COP1) of the two systems as a function of the heating demand is shown in Figure 28. 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.

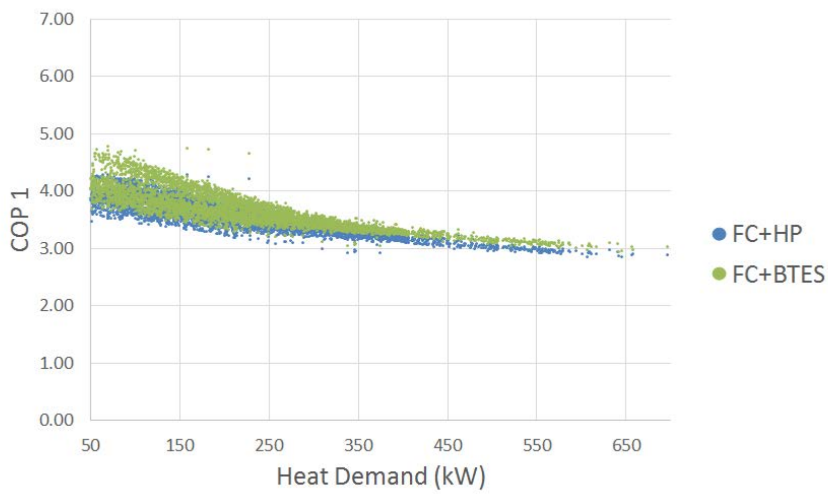

Figure 28 - Heating COP as a function of heat demand, FC+HP vs. FC+BTES

The total power needed to operate the refrigeration and the heating systems for both scenarios is presented in Figure 29. It can be seen that FC+BTES has a lower power need for all ambient temperatures. 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.

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.

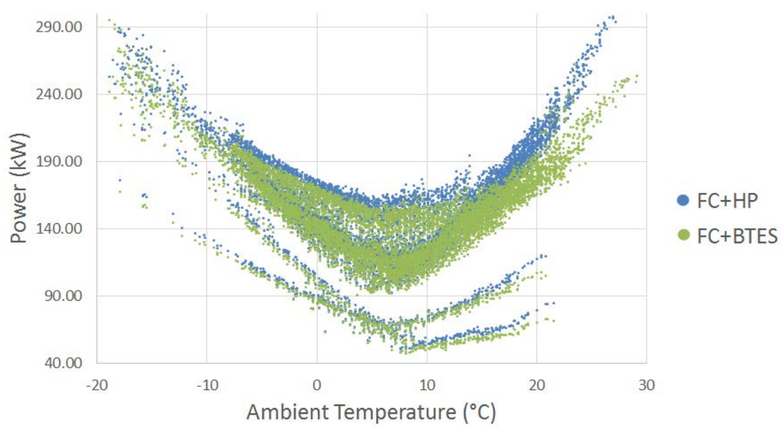

Figure 29 - Total power needed as a function of ambient temperature, FC+HP vs. FC+BTES

4.4.2 FC+HP vs. CHR

In this section, FC+HP, the reference scenario is compared to CHR, the scenario with complete heat recovery and no boreholes. As described in section 4.3.2, 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 30. It can be seen that the cooling COP of CHR is lower than 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.

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.

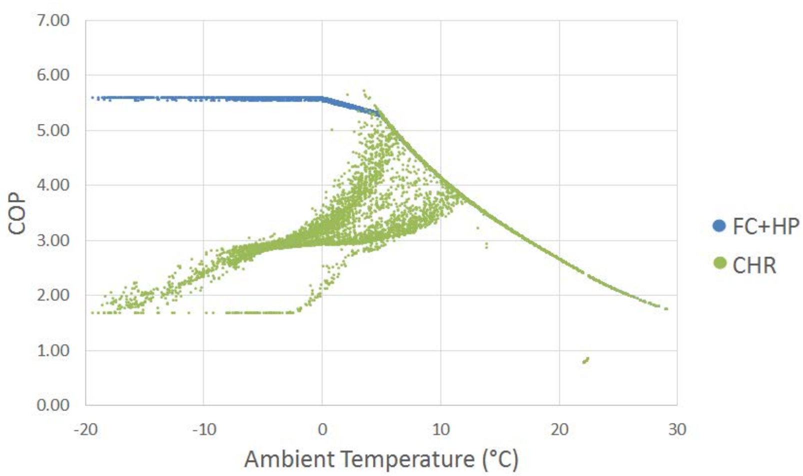

Figure 30 - Cooling COP as a function of ambient temperature, FC+HP vs. CHR

The heating COP of the two systems as a function of the heating demand is shown in Figure 31. It should be noted that for CHR, the curve represents COPHR, as given by equation 7. The heating COP of CHR is greater than the one of FC+HP for the whole heating demand range, although it has variations.

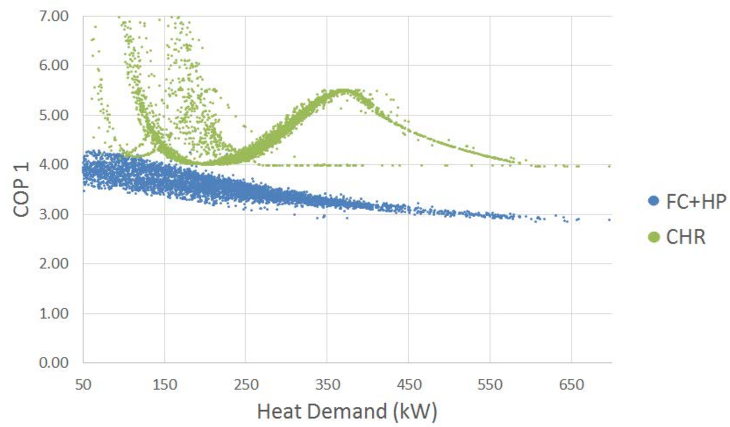

Figure 31 - Heating COP as a function of heat demand, FC+HP vs. CHR

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 88.5bar 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.

Figure 32, which shows the total power needed to run the systems for both scenarios, gives a better comparison about the effect of the heat recovery. 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.

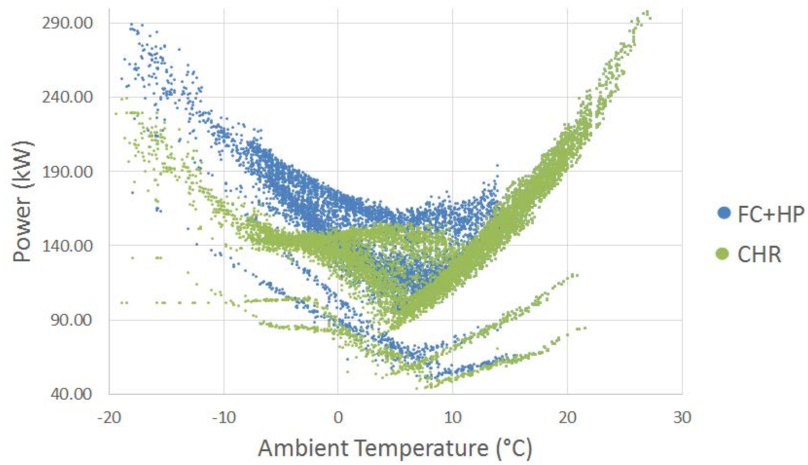

Figure 32 - Total power needed as a function of ambient temperature, FC+HP vs. CHR

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

4.5 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 costs and savings of two scenarios, CHR and CHR+BTES, are compared and analyzed.

The economic analysis tool used in this section is the life cycle cost analysis. 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 are analyzed.

4.5.1 Life cycle cost analysis parameters

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

$$\mathrm{LCC} = \mathrm{Inv} + \mathrm{LCC_E} + \mathrm{LCC_M} \tag{32}$$

The investment costs includes the cost of adding the borehole thermal energy storage. It is considered 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:

$$\mathrm{LCC_E} = \mathrm{C_{AE}} \times \mathrm{AE} \times \frac{1 - [1 + (i - p)]^{-n}}{i - p} \tag{33}$$

The cost of electricity, CAE, 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.

4.5.2 Life cycle cost analysis

First, the life cycle cost of FC+BTES is analyzed for supermarket with an area of 14000 m^2 described in section 4.2. 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.

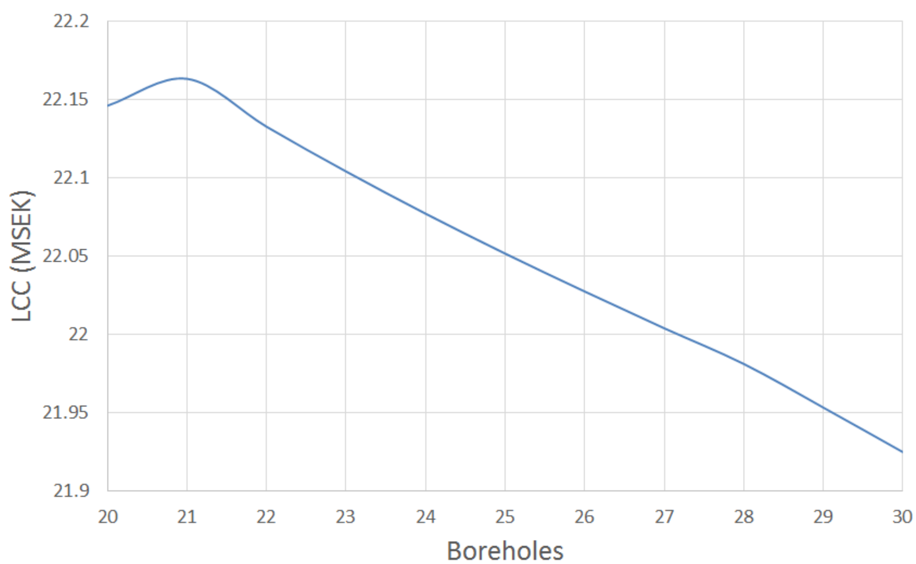

Figure 33 - Life cycle cost of FC+BTES for each additional borehole

The life cycle cost of FC+BTES for each additional borehole is presented in Figure 33 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.

5 CONCLUSION

The potential of thermal energy storage in CO_2 supermarkets is investigated in both short and long term.

In short term thermal storage, brine and PCM usage is evaluated. In brine case, the analysis shows that rather large volume of storage tank is needed; while the savings in power consumption are limited; $10m^3$ of storage tank volume for 2.6-3% savings in daily energy use. PCM usage proves to have higher savings with smaller volume; $6.3m^3$ of PCM container for 8% savings in daily energy use. Charging the storage system at evaporation temperatures 2-3K higher than the reference case (-10℃) shows little improvement in savings. Additionally storage by frozen products in the cabinets shows very little savings in energy consumption.

In long term thermal storage, the usage of borehole connected to the refrigeration system to make additional sub-cooling shows higher efficiency compared to reference case leading to 6.3% savings in annual energy use. On the heating side, it is proved that the supermarket systems with heat recovery can perform more effectively than the reference case (using heat pump). The refrigeration system can cover almost all the heat load at higher efficiency (heating COP of4-5.5) compared to a ground source heat pump (Heating COP of 3-4). If the heating demand is higher than what can be recovered from the refrigeration system then ground source heat pump can be used to cover the excess heat and the borehole can sub-cool the refrigeration system in the summer period.

6 FUTURE WORK

For future work, the further potential of heat storage in short and long term storage can be evaluated.

As expressed previously, the storage by frozen products in the freezers can be improved by using additional thermal mass. PCM in different forms such as slab can be attached to the wall inside the freezers.

Also, the potential of free cooling to make sub-cooling can be investigated. Free cooling can be used in the same concept as short term case studies providing additional sub-cooling analyzed in this investigation. Storage unit can be charged by outdoor air during night. Discharging is done by the sub-cooler during day providing additional sub-cooling for the refrigeration system. The advantage of this concept is that there is no extra load on the refrigeration system during charging, but on the other hand the temperature of storage unit is limited to difference in ambient temperature of day and night. Thermal storage by free cooling can be matter of further investigation.

Short term thermal storage cannot be only used for cooling purposes but also for heating purposes specifically for the cases where the heat load is high and the refrigeration system cannot cover all the heating demand. The heat which is normally rejected by gas cooler to the ambient can be stored and the stored energy in the storage unit can be used as additional heat source for the refrigeration system.

The same concept of heat storage can be done for long term case. Borehole in connection with refrigeration system can be used for both cooling and heating purposes specifically in cases where the refrigeration system is unable to cover the heat load totally. In such case, in the cold days of winter the medium temperature level evaporator cools the ground, when the heating demand is higher than what the refrigeration system can provide. This extra heat source on the refrigeration system enables the system to recover more heat. Then the stored cooling energy in borehole can be used to sub-cool the gas cooler output during the summer provide additional sub-cooling for the system. This can be an interesting matter of future investigation for quite large hypermarkets where the heating demand is much higher than what the refrigeration system can cover.

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