CHALMERS

Competitiveness of District Cooling in Energy Efficient Supermarkets

Master of Science Thesis within the Sustainable Energy Systems programme

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CHALMERS UNIVERSITY OF TECHNOLOGY
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ABSTRACT

This thesis concerns utilization of district cooling in supermarkets. Today, district cooling is primarily used in hotels, hospitals, offices and computer centrals. This study investigates the possibility to expand the range of customers to also include supermarkets. The studied use includes comfort cooling as well as condenser cooling. Condenser cooling implies increasing the COP of a chiller by decreasing its condensation temperature. Chillers are inevitable since the temperature levels of chilled and frozen food are too low to directly use the district cooling.

The aim of the thesis is to determine the highest possible price of district cooling the supermarket owner is willing to pay in order to connect the supermarket to the district cooling network.

Two different reference supermarkets are analysed, one representative for a supermarket today and one representative for a future supermarket. The major difference is the energy efficiency of the display cabinets.

Four different system designs are investigated, using district cooling for both comfort cooling and condenser cooling, not using district cooling at all, using district cooling only for comfort cooling and finally, using district cooling only during the warm part of the year.

The results show that the prices charged for district cooling are in general too high to make district cooling competitive in a typical supermarket today. In the typical supermarket in the future it is more favourable. Using district cooling in the future reference supermarket for both comfort cooling and condenser cooling implies an annual demand of 331 MWh and a competitive price of 216 SEK/MWh. Using district cooling only when the outdoor temperature exceeds 10 °C implies a demand of 210 MWh/year and a competitive price of 290 SEK/MWh and using district cooling only for comfort cooling implies a demand of 80 MWh/year and a competitive price of 599 SEK/MWh.

The typical future supermarket is fully possible to design at the present day, this would decrease the energy demand substantially and make district cooling competitive.

Keywords: Condenser cooling, district cooling, supermarkets

Preface

This work is the concluding part of my master studies in Sustainable Energy Systems at Chalmers University of Technology. The work was carried out at the department of district cooling at Göteborg Energi during April – October 2010. The study is made in collaboration with a doctoral programme about energy efficient systems and suitable indoor climate in supermarkets at SP Technical Research Institute of Sweden.

I would like to express my gratitude to everyone who has supported me or in any other way contributed to this thesis. Thank you Anna Svernlöv, manager at the department of district cooling at Göteborg Energi, for giving me this opportunity and providing me an ideal working environment. I would also like to thank all employees at the department of district cooling at Göteborg Energi for always being helpful and supportive. Thank you Ulla Lindberg, Lic.Eng. at SP Technical Research Institute of Sweden, for great guidance and for sharing precious knowledge about energy systems in supermarkets. Finally, I would like to thank my examiner Professor Per Fahlén for precious input and support.

Göteborg, October 2010
Peter Filipsson

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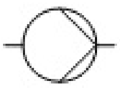
Designations

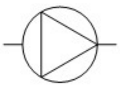
Abbreviations

ACH	Air changes per hour
CAV	Constant air volume
COP	Coefficient of performance
HVAC	Heating, ventilation and air conditioning
HX	Heat exchanger
SFP	Specific fan power
SPF	Seasonal performance factor
VAV	Variable air volume

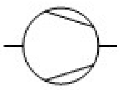
Symbols



Pump

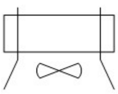


Fan



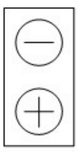
Compressor

Expansion device

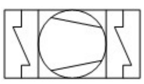


Dry cooler

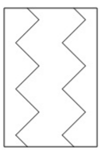
Heating and cooling coil



Recuperative heat exchanger



Chiller



District cooling substation

Nomenclature

A	Area	[m^2]
A	Investment cost	[SEK]
C	Energy price	[SEK/MWh]
c_p	Specific heat capacity	[J/kg·K]
I	Net present value factor	[-]
i	Discount rate	[-]
K	Heat transfer coefficient	[W/K]
L	Length	[m]
n	Economic life	[years]
Δp	Pressure drop	[Pa]
Q	Energy	[J]
$\dot{Q}$	Power	[W]
q	Annual increase of energy price	[-]
$\dot{q}$	Specific power	[W/m]
r	Heat of vaporization	[J/kg]
T	Temperature	[K]
t	Temperature	[°C]
$\dot{V}$	Volumetric flow	[m^3/s]
$\dot{W}$	Electric power	[W]
Δx	Change of humidity ratio	[kg_{water}/kg_{air}]
γ	Cooling load correction factor due to indoor humidity	[-]
η	Efficiency	[-]
η_c	Carnot efficiency	[-]
ρ	Density	[kg/m^3]
φ	Relative humidity	[-]

Subscripts

a	air
cc	comfort cooling
DC	district cooling
DH	district heating
el	electricity
h	heat
HR	heat recovery
in	indoor
inf	infiltration
int	internal
max	maximum
out	outdoor
rec	recovery
s	supply
tr	transmission
v	vapour
vent	ventilation
1	Hot side of a chiller (e.g. T_1 denotes the condensation temperature)
2	Cold side of a chiller (e.g. COP_2 denotes the coefficient of performance in cooling mode)

Additional subscripts are used, but they are considered self explanatory.

Abbreviations used to denote streams in Figure 15, Figure 18, Figure 20 and Figure 22

Cond.	Refrigerant in the condensers
Cond. (C)	Refrigerant specifically in the condensers of the chilled-food chillers
Cond. (F)	Refrigerant specifically in the condensers of the frozen-food chillers
CC	Condenser coolant (a secondary coolant with condensers as only heat source)
DC	District cooling water
OA	Outdoor air passing the dry cooler
SC	Secondary coolant
VA	Ventilation air

1 Introduction

1.1 Background

The use of district cooling in Sweden has grown steadily since the first commercial plant was taken into operation in 1992. District cooling is primarily used for air conditioning and cooling of industrial processes and computer centrals.

Estimations show that supermarkets stand for 3 - 5 % of the total use of electricity in industrialized countries (Arias et al. 2010[4]). In Sweden in 2009, that corresponds to 4.2 – 6.9 TWh (The Swedish Energy Agency, 2010b[25]). Around 45 % of the electricity use in a typical Swedish supermarket is due to food refrigeration. A typical open vertical display cabinet of 5 m consumes as much electricity as a Swedish electrically heated detached family house annually. The use of electricity in a vapour compression chiller is directly related to the difference between the condensation temperature and the evaporation temperature. The common practice today is to reject the condenser heat in dry coolers located outside the supermarket, or to recover the condenser heat in a heat recovery system connected to the air conditioning system. By rejecting the condenser heat to district cooling water, a low and steady condensation temperature is achieved which may lead to a lower consumption of electricity.

Different ways to reject heat from the condensers of chillers in supermarkets have been investigated by Haglund Stignor (2003)[15]. The investigation was done with respect to costs, use of electricity and environmental impact. The conclusion was that the district cooling must be very cheap in order to compete with the other alternatives.

A technical procurement competition carried out in 1997 showed that there is a great potential in making display cabinets more energy efficient. The present development in energy use in supermarkets may make district cooling a more beneficial alternative. One big drawback with the district cooling alternative is that the possibility to recover the condenser heat is lost. But more energy efficient cooling cabinets will leak less cold air, and therefore reduce the need for heat and increase the need for comfort cooling.

1.2 Purpose and Aim of the Study

The purpose of this Master thesis is to study how the profitability of connecting a supermarket to the district cooling network will evolve the coming decades. In addition to study the present situation a future scenario is studied. The most important difference between the cases is the energy efficiency of the display cabinets.

The aim of this Master thesis is to determine the highest possible price the supermarket owner is willing to pay for district cooling in order to connect the supermarket to the district cooling network, today and in the future. This price will henceforth be referred to as the competitive price of district cooling. It shall also be determined to what purpose the district cooling should be used and how much.

1.3 Method and Limitations

By reviewing previous work on energy use in supermarkets a reference supermarket is defined. By studying the present trends in this area a prediction about a typical supermarket in the year 2030 is made. The defined parameters make it possible to calculate the demand for energy.

The next step is to elaborate a number of alternative solutions to meet the demand for energy in the reference supermarket. Calculations are made in order to decide the need for externally added energy for each alternative. All calculations are made using the numerical calculation programme Matlab. With information about costs for all required components a fair comparison can be made. This will end up with a highest possible price the supermarket owner can be willing to pay in order to connect the supermarket to the district cooling network. Since this study deals with many uncertain parameters, a thorough sensitivity analysis will be carried out.

The study does not include any measurements made in real supermarkets. Nor will it end up with any detailed technical solutions regarding control systems etc. Geographically, the study is limited to comprise a supermarket in Göteborg. Göteborg is located on the southwest coast of Sweden and has a maritime climate. The weather data used in this study implies an annual mean temperature of 9.7 °C. Climatic variations make it possible to achieve different results in other regions. Regarding the production of district cooling only Göteborg Energi´s production is studied. Only the sales area of the supermarket is studied. Regarding the refrigeration system, a completely indirect system is studied.

1.4 Outline of the Report

Section 2 of this report gives an introduction to energy use in supermarkets. Topics important later in the report are dealt with more thoroughly.

Important parameters and energy demand of the studied supermarkets are defined in section 3.

Alternative designs to meet the energy demand are presented in section 4.

The results, in terms of energy demand, costs and profitability of district cooling are presented in section 5.

A discussion is carried out in section 6 and conclusions are finally drawn in section 7.

2 Energy Use in Supermarkets

The energy system of a supermarket can be observed as several subsystems interacting with each other, e.g. refrigeration system, cabinet system and HVAC system (Arias 2005, p. 104[2]). Figure 1 illustrates how the subsystems are interrelated with each other. To exemplify the interrelations between the subsystems one can assume that the outdoor temperature decreases. A decrease of outdoor temperature will very likely cause a decrease of indoor relative humidity. Dryer indoor air decreases the losses and consequently the cooling load in the display cabinets. If the supermarket uses a condenser heat recovery system there will be less heat to recover. The need for external heat will consequently increase both directly due to higher transmission and infiltration losses through the building envelope and indirectly due to less heat to recover in the condenser heat recovery system.

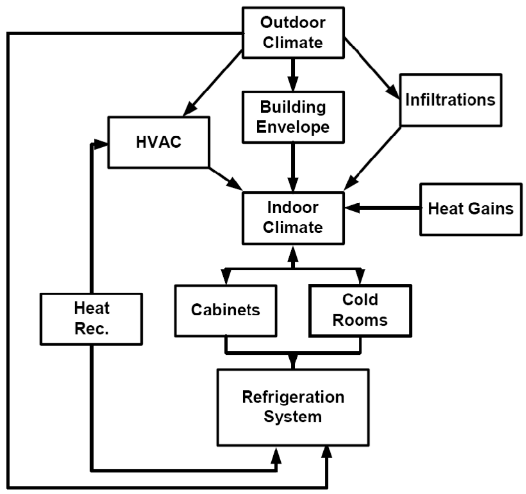


Figure 1 Model of the different subsystems in a supermarket (Arias 2005, p. 105[2])

The performance of the display cabinets influences the indoor climate to a large extent. Display cabinets are major emitters of cold air. This has an impact on the HVAC system and causes an increased demand for heating. The leakage of cold air also causes large vertical temperature gradients in front of the cabinets, which impairs the thermal comfort (Lindberg 2009[18]). On the other hand, more energy efficient display cabinets will cause a higher demand for comfort cooling, especially during summer.

Supermarkets are major consumers of electricity. An average Swedish supermarket uses around 320 kWh_{el}/m^2 annually (The Swedish Energy Agency 2010a, p. 29[24]). However, the cost of energy is only one percent of the overall cost for a supermarket, but halving the cost of energy results in an increase of profits by 17 % (Arias et al. 2004, p. 7[3]).The refrigeration system and lighting are by far the largest causes for the use of electricity. This is illustrated in Figure 2.

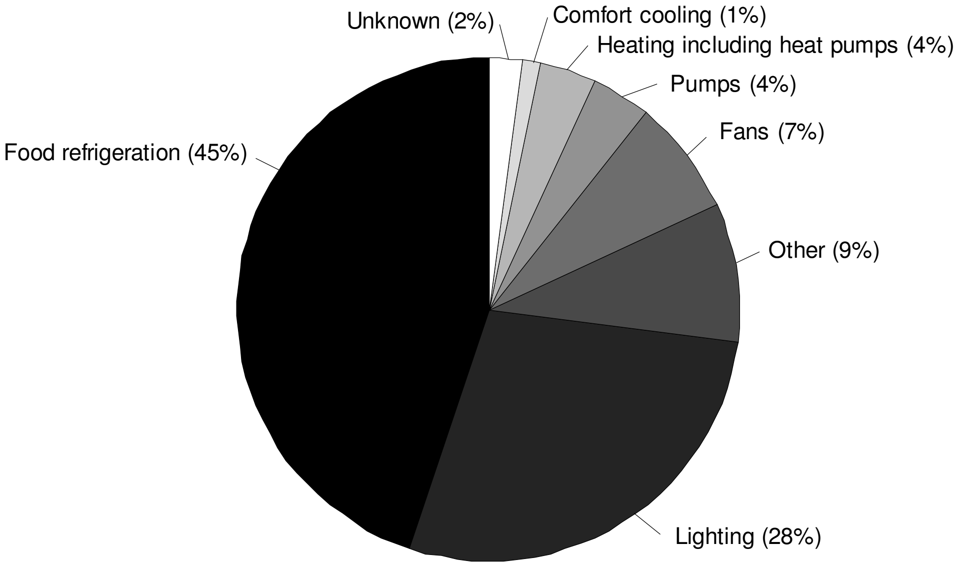


Figure 2 Distribution of electricity use in an average Swedish supermarket. (The Swedish Energy Agency 2010a, p. 29[24])

2.1 The Heat Balance of a Building

The heat balance of a building is influenced by three groups of factors:

- The building envelope
- The outdoor climate
- The activities in the building

The building envelope is defined by its area, the heat transmission coefficient, the air infiltration rate and the heat storage capacity of the building structure.

The most important factors of the outdoor climate are for a supermarket the temperature and the humidity, but the solar radiation and wind speed does also influence the heat balance.

The activities in the building determine how much internally generated heat there is. This includes heat from people, lighting, equipment, heat emitted or absorbed by the building structure and heat from solar irradiation. In contrast to other commercial buildings, supermarkets often have a very low net amount of internally generated heat. The reason is cold display cabinets which absorb a lot of heat. As in all commercial buildings, the internal generation of heat differs a lot between daytime and nighttime. It is important to take this into consideration when making calculations.

To be able to keep the indoor climate within some comfort requirements, the building needs a HVAC system. The main tasks of the HVAC system are to control the temperature, humidity and cleanliness in the building. This is achieved by conditioning the air supplied into the building. The HVAC system may include a heat exchanger located between the supply air and the exhaust air. To further decrease the need for heat, the system can recover heat from the condensers in the refrigeration system.

2.2 Display Cabinets

Display cabinets are an important part of the energy system of a supermarket. They can differ quite a lot regarding temperature levels, accessibility, loading of goods, type of opening, cooling distribution etc (Fahlén 2000[11]). In this study, vertical chilled-food cabinets and horizontal frozen-food gondolas are included. Vertical cabinets are very common due to the possibility to expose a large amount of goods in a small floor area. In cabinets without doors, the cold air in the cabinet is separated from the ambient air by an air curtain. Studies have been done to understand and improve the air curtains, but it is very hard to design an air curtain which is not disturbed by moving people, ventilation air, differences in cabinet loading etc. The cooling load in an open vertical display cabinet is linearly proportional to the difference in specific enthalpy between the air in the cabinet and the surrounding air (Axell 2002, p. 106[5]).

There are many methods to decrease the energy demand in a display cabinet. One efficient measure is night covering. Lindberg et al. (2010)[19] has shown that a covered vertical chilled-food cabinet has 67 % lower cooling load. Mounting doors or lids on cabinets and gondolas is another efficient measure to decrease the infiltration losses. Lindberg et al. (2010)[19] has investigated how the direct electric input, the heat extraction rate and the food temperatures are influenced by mounting doors on vertical display cabinets. The results show that the heat extraction rate decreases by 66 % at daytime operation and by 53 % at nighttime operation when mounting doors on the cabinet. These percentages are valid for a door opening frequency of 10 openings per hour. If the frequency is increased to 30 openings per hour the reduction in heat extraction rate is 61 % during daytime and 46 % during nighttime. In addition to lower heat extraction rate, the use of doors made the temperature distribution in the cabinet much more uniform and made it possible to increase the brine inlet temperature from -8 ˚C to +2 ˚C.

The cooling load in a display cabinet is both due to sensible heat and latent heat. The sensible heat load is due to the necessary change in temperature while the latent heat load is due to the fact that some moisture condenses when cooling the air. This phenomenon makes the cooling load higher in the summer, even if the indoor temperature is kept constant all year around, since indoor air is almost always more humid during the summer. The magnitude of this is presented in Figure 3.

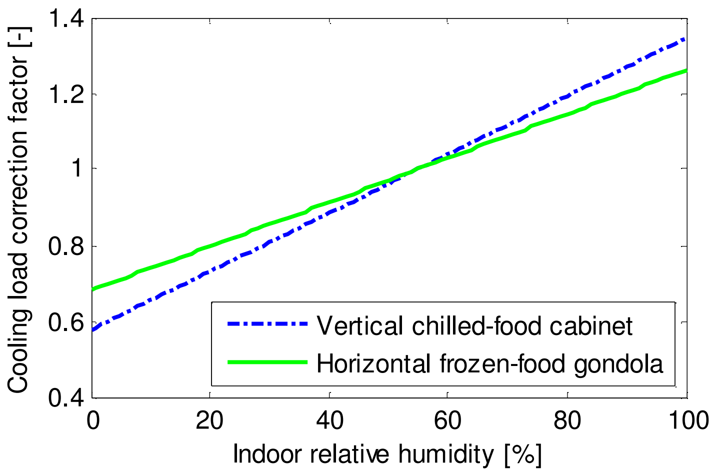


Figure 3 Cooling load as a function of the ambient humidity (Arias 2005, p. 124[2])

The cooling load in a display cabinet is on one hand due to difference in enthalpy between the air in the cabinet and the ambient air and on the other hand due to internal loads in the cabinet. The magnitude of the different contributions to the total load differs quite a lot in a vertical chilled-food cabinet and a horizontal frozen-food gondola. The contributions to the total cooling load, $\dot{Q}_C$, at an ambient temperature of 25 °C are presented in Table 1, and illustrated in Figure 4.

Table 1 The heat balance in display cabinets (Arias 2005[2] & Fahlén 2000[11])

		Vertical chilled-food cabinet	Horizontal frozen-food gondola
Heat transfer through conduction	$\dot{Q}_1$	7 %	11 %
Heat transfer through infiltration	$\dot{Q}_2$	64 %	23 %
Heat transfer through radiation	$\dot{Q}_3$	11 %	45 %
Heat from internal lighting	$\dot{Q}_4$	11 %	0 %
Heat from fans and heating wires	$\dot{Q}_5$	7 %	17 %
Heat from defrost equipment	$\dot{Q}_6$	0 %	4 %

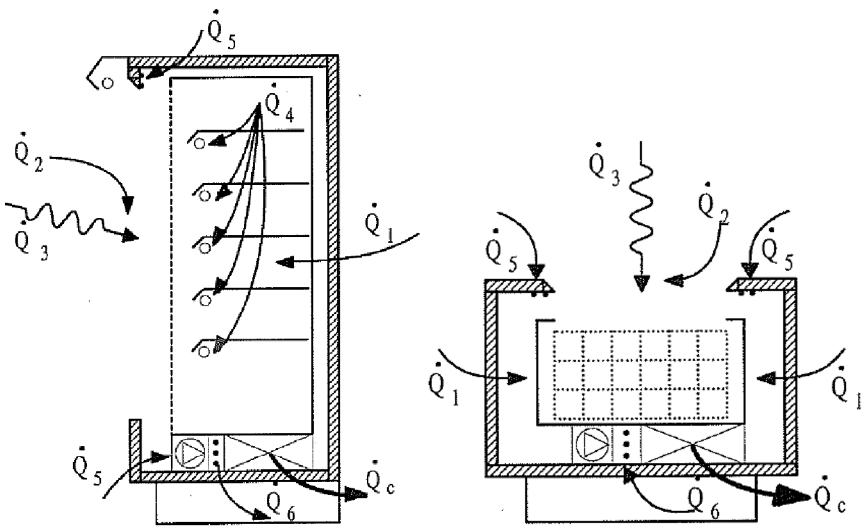


Figure 4 Energy balances of a vertical chilled-food cabinet and a horizontal frozen-food gondola (Fahlén 2000[11])

The energy demand in a horizontal gondola is much less than in a vertical cabinet. The reason is that the cold and heavy air in the gondola is protected from pouring out by the walls of the gondola. In addition to that, it is rather common to mount lids on horizontal gondolas.

2.3 The Vapour Compression Process

Since the supply temperature of the brine in display cabinets is kept well below 6 °C, district cooling cannot directly be used to cool the food. It is therefore inevitable to use a chiller. This can be done in a vast amount of configurations but almost all include a vapour compression process. The vapour compression chiller makes use of the fact that the saturation temperature of a fluid is dependent on its pressure. With a compressor and an expansion device the chiller can absorb heat at a low temperature (in the evaporator) and reject the heat at a higher temperature (in the condenser). The COP represents the efficiency of the chiller and is defined as the relation between achieved cooling and electrical work required (it is common to use the term COP_2 to refer to this definition. All COP presented in this study refer to this definition). The COP increases when the difference between condensation and evaporation temperature decreases. The relation is presented in Figure 5. In modern supermarkets, the chillers are operated with a floating condensation temperature. It means that the condensation temperature is adjusted according to the outdoor temperature. This yields a high COP when it is cold outside and a lower COP when it is warm outside.

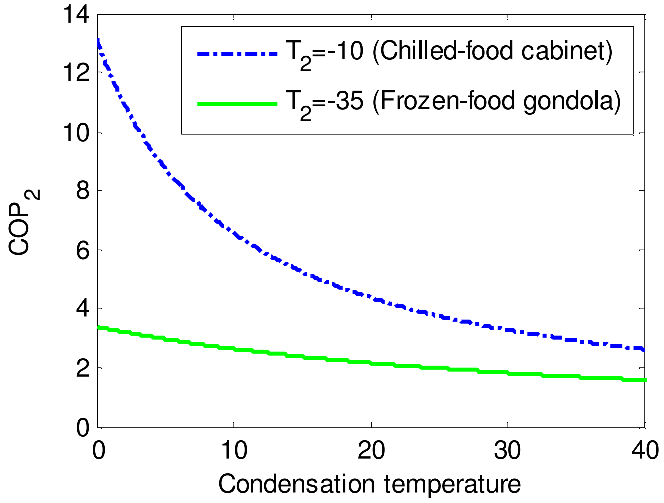


Figure 5 COP_2 as a function of the condensation temperature (at a Carnot efficiency of 0,5)

2.4 Supermarket Refrigeration System

Refrigeration systems in supermarkets can be designed in several different ways. Some examples are:

- Direct systems (distributed or not)
- Completely indirect systems
- Partially indirect systems
- Different kinds of cascade systems

This study involves completely indirect systems. A schematic of such system is presented in Figure 6.

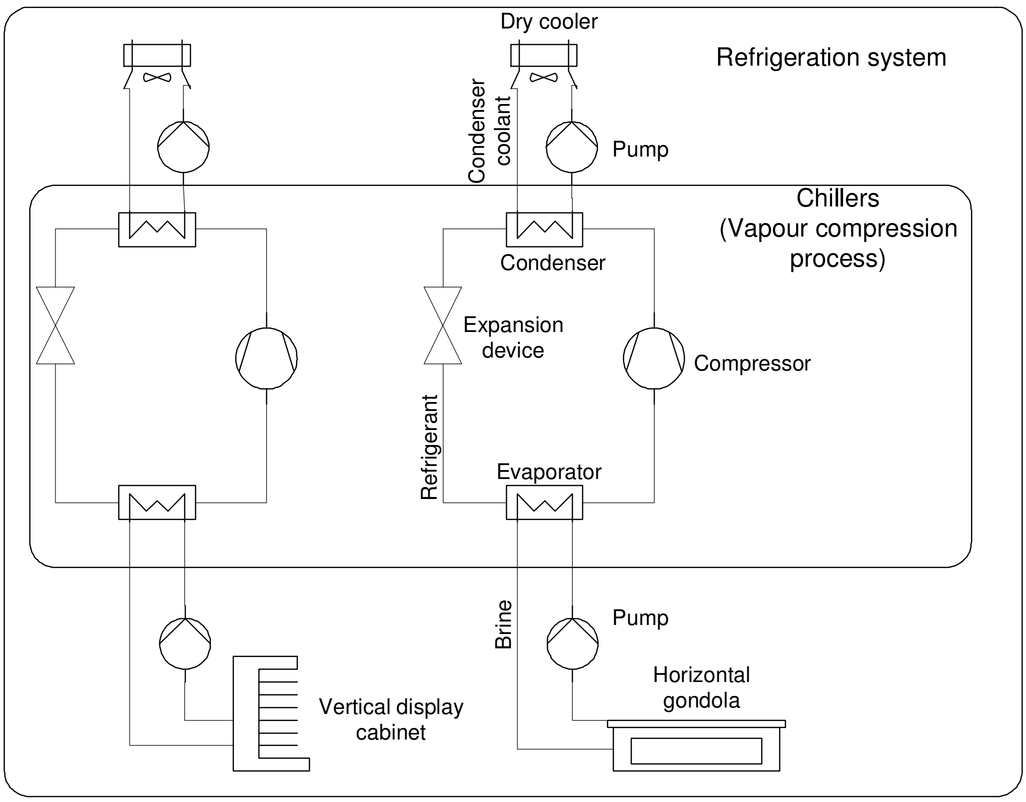


Figure 6 Schematic of a completely indirect refrigeration system

In a direct system, the condensation occurs in roof-mounted condensers and the evaporation occurs in the display cabinets. The advantages with an indirect system are less amount (and leakage) of refrigerant, more efficient defrosting, easier load-shedding and better possibilities to recover condenser heat. The disadvantages with indirect systems are more pump work, higher investment cost and lower efficiency. In addition to the cabinets and gondolas, cooling is also needed in cold storerooms.

2.4 District Cooling

In a district cooling system, water is chilled in a central plant and distributed to customers through a network of pipes. A big advantage is that the cooling is produced in a large plant with high efficiency. In Göteborg, the river Göta älv is used to cool the district cooling water during the cold part of the year, this is known as free cooling. During the warm part of the year, absorption heat pump technology is used. It makes use of waste heat to produce cooling, which is very beneficial since the demand for district heating is usually very low in the summer. Vapour compression technology is used when the capacity of the free cooling and absorption technology is insufficient. The temperature of the delivered cold water is 6 °C and the return temperature should be 16 °C.

2.5 Condenser Cooling

As presented in Figure 5, a low condensation temperature generates a high COP. A simple analysis can be made in order to determine the value of the cooling used to increase the COP of a chiller. A cooling load, $\dot{Q}_2$, is supplied by a chiller cooled by free outdoor air at a certain efficiency, COP_{AIR}. By cooling the condenser with district cooling water, the same cooling load can be supplied at another efficiency, COP_{DC}. With a given price for electricity, C_{el}, it is possible to determine which price for district cooling, C_{DC}, that equalizes the cost of energy of the two cases. The cost of energy needed in the case with air-cooled condenser equals

$$C_{el} \cdot \frac{\dot{Q}_2}{COP_{AIR}} \qquad \text{(eq. 2.1)}$$

The cost of energy needed for the case with a condenser cooled by district cooling is expressed as

$$C_{el} \cdot \frac{\dot{Q}_2}{COP_{DC}} + C_{DC}\left(\dot{Q}_2 + \frac{\dot{Q}_2}{COP_{DC}}\right) \qquad \text{(eq. 2.2)}$$

By setting the cost of energy for the two cases equal the price of district cooling is obtained as

$$C_{DC} = C_{el}\left(\frac{\frac{COP_{DC}}{COP_{AIR}} - 1}{COP_{DC} + 1}\right) \qquad \text{(eq. 2.3)}$$

This relation is illustrated in Figure 7.

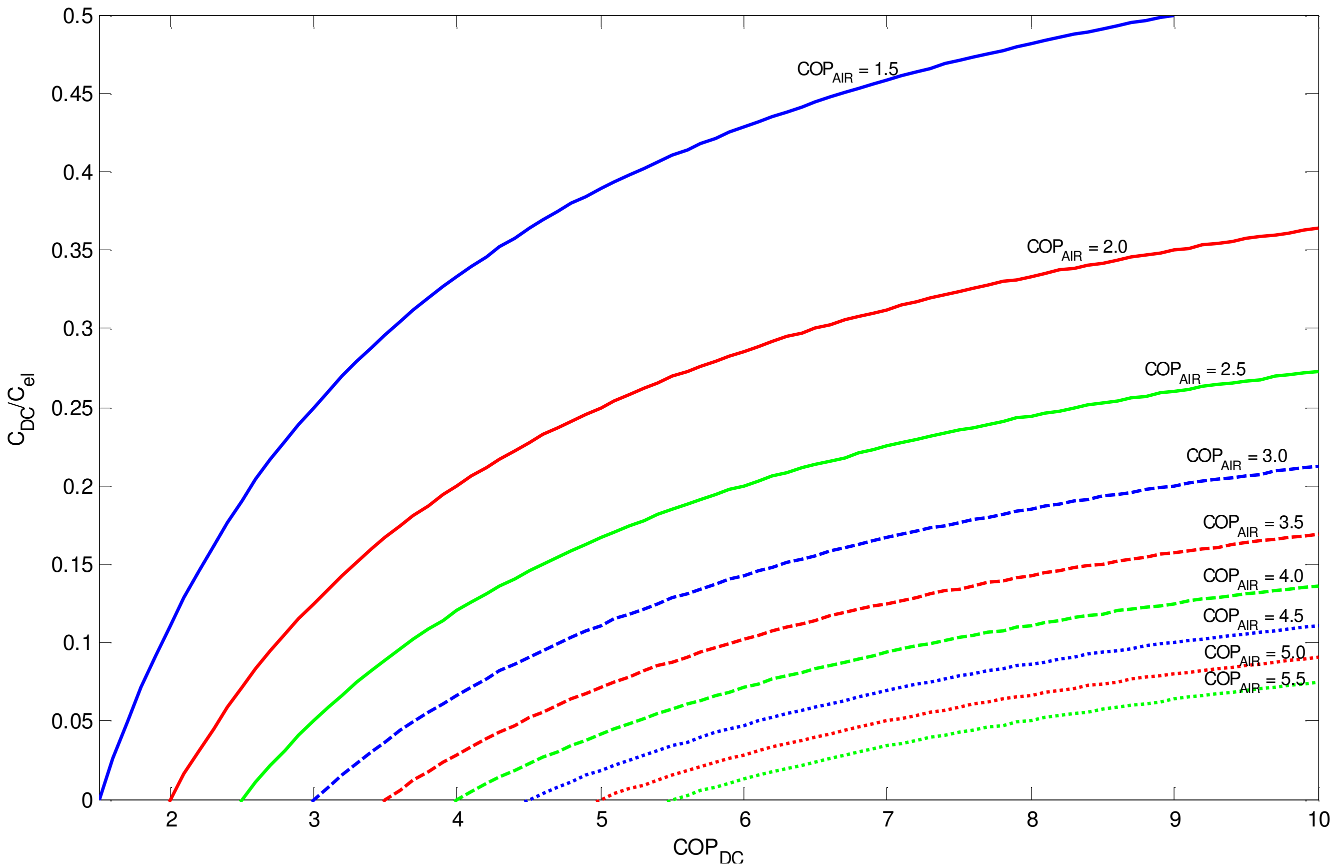


Figure 7 The relative price of district cooling as a function of COP with and without a district cooling cooled condenser

This simple analysis may give an indication of the potential of district cooling cooled condensers. But in reality, a lot more parameters must be taken into consideration before making any conclusions. Some of the parameters are:

- Costs of required components (dry coolers, heat exchangers etc.)
- Auxiliary energy needed (fan power, pump power etc.)
- Demand for heating and comfort cooling

2.6 Condenser Heat Recovery

Recovering heat from the condensers is often considered an efficient measure to improve the energy efficiency of a supermarket. Of course, it seems wasteful to reject heat in roof-mounted dry coolers when there is a heat demand in the supermarket. But according to Johnsson[27] it is very rare that these systems operate optimally. The reasons are claimed to be the complexity and the large number of actors involved in the energy system of a supermarket (retailer, property owner, consultants and different contractors dealing with refrigeration, piping, control systems, ventilation, automation etc.). As previously indicated, one problem with condenser heat recovery is that there is less heat to recover when the heat demand is high and vice versa. Another big drawback is the fact that the condensation temperature must be increased in order to make the condenser heat hot enough. This will affect the COP of the chiller. If the heat demand is small compared to the cooling load supplied by the chiller, the increase of condensation temperature will cause more harm than good.

In a system with condenser heat recovery, the condensation temperature is typically 10 °C above the indoor supply air temperature when there is a heat demand (otherwise 10 °C above the outdoor air temperature) (Johnsson[27]). Assume a cooling load, $\dot{Q}_2$, and a heat demand, $\dot{Q}_H$. When utilizing condenser heat recovery, an efficiency, COP_{HR}, is obtained and the electricity demand equals

$$\frac{\dot{Q}_2}{COP_{HR}} \qquad \text{(eq. 2.4)}$$

A system rejecting all heat in roof-mounted dry coolers, having an efficiency of COP_{AIR} and an electric heater with an efficiency of 100%, the total electricity demand equals

$$\frac{\dot{Q}_2}{COP_{AIR}} + \dot{Q}_H \qquad \text{(eq. 2.5)}$$

The following inequality must hold in order to have lower electricity consumption in the system with condenser heat recovery.

$$\frac{\dot{Q}_2}{COP_{AIR}} + \dot{Q}_H > \frac{\dot{Q}_2}{COP_{HR}} \qquad \text{(eq. 2.6)}$$

By using the concept of Carnot coefficient of performance, equation 2.6 can be rewritten as

$$\frac{\dot{Q}_H}{\dot{Q}_2} > \frac{T_s - T_{out}}{\eta_c \cdot T_2} \qquad \text{(eq. 2.7)}$$

η_c is the Carnot efficiency, T_s the indoor supply air temperature, T_{out} the outdoor air temperature and T_2 the evaporation temperature expressed in kelvin. This implies a lower limit of the heating demand in order to make heat recovery a better alternative than an electric heater. The higher the heat demand is, the more beneficial it is to recover heat. But in addition to the lower limit of equation 2.7, there is also an upper limit depending on the amount of available heat in the condenser. Altogether, the constraints are expressed as

$$\frac{T_s - T_{out}}{\eta_c \cdot T_2} < \frac{\dot{Q}_H}{\dot{Q}_2} < 1 + \frac{T_s + 10 - T_2}{\eta_c \cdot T_2} \qquad \text{(eq. 2.8)}$$

In theory, it is easy to have many chillers and be able to decide how many of them to recover heat from and thereby controlling $\dot{Q}_2$. This is rarely done efficiently in reality.

2.7 Trends in Energy Use in Supermarkets

There is a clear ambition among supermarket owners to use energy more efficiently. ICA aims at cutting their direct and known emissions by 30 % to 2020, compared to 2006 (ICA AB 2010, p. 8[16]). Axfood is planning to reduce the use of energy per square meter by 30 % until 2015 (Axfood 2010, p. 4[7]) and the Swedish Co-operative Union KF intends to decrease the use of electricity per sales area by 30 % to 2020, compared to 2008 (KF 2010, p. 37[17]). These three actors cover 44 % of the supermarkets in Sweden and 78 % of the total turnover (Ågren 2008[26]).

At the same time, more and more of the range of products are chilled and frozen food. The consumption of deep frozen food in Sweden was 50.6 kg/capita in 2007 compared to 35.5 kg/capita in 1995 and 26.2 kg/capita in 1985 (Börjesson red. 2008, p. 1[9]). There is also a trend towards more use of electrical devices in supermarkets, e.g. ovens and roasters.

The specific use of electricity in Swedish supermarkets was increased by 8 % between 1990 and 2009 (The Swedish Energy Agency 2010a, p. 45[24]). This is presented in Figure 8.

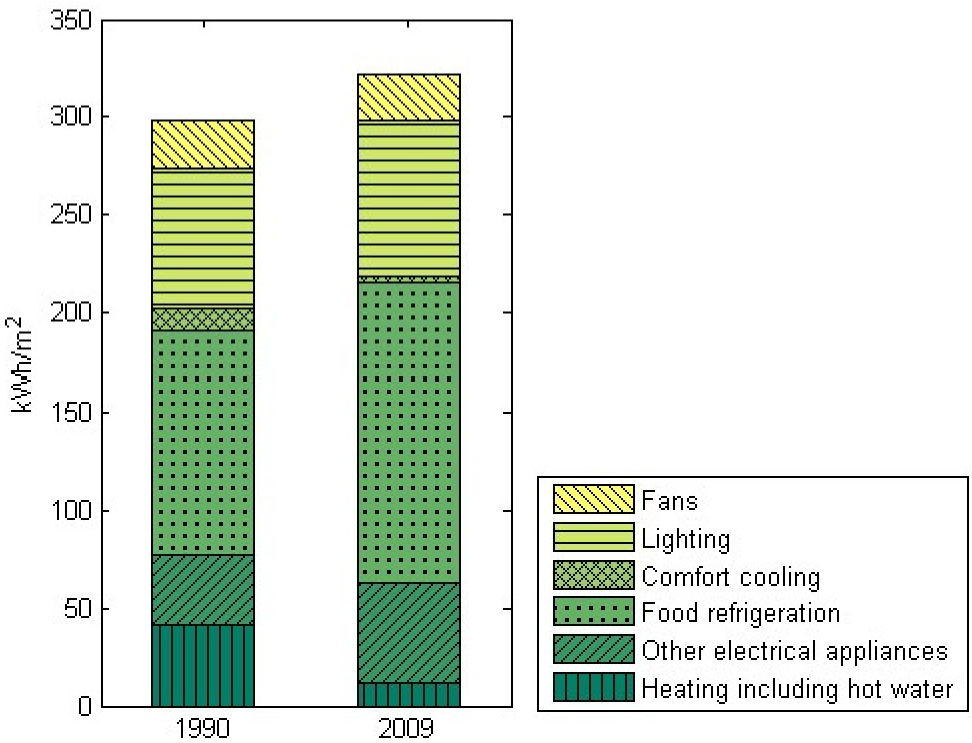


Figure 8 Use of electricity in Swedish supermarkets in 1990 and 2009 (The Swedish Energy Agency 2010a[24])

The electricity consumed by lighting was increased from 72 kWh/m^2 in 1990 to 79 kWh/m^2 in 2009 despite the fact that new energy efficient lighting techniques emerged during this period. The explanation is more illumination (more light in the supermarket) and extended opening hours. The use of CAV ventilation systems in Swedish commercial premises decreased slightly from 81 % in 1990 to 76 % in 2009 at the expense of more VAV ventilation systems (The Swedish Energy Agency 2010a, p. 48[24]).

A barrier for energy efficiency measures in supermarkets is the relation between the retailer and the property owner. The heating cost for the supermarket can be fixed and included in the rent. Hence, the retailer will not benefit from lower heating demand when installing energy efficient display cabinets that spill less cold air. This phenomenon is called misplaced incentives and is a well known barrier for energy efficiency in the real estate industry (Golove, W. & Eto, J. 1996, p. 9[14]).

3 The Reference Supermarkets

To be able to calculate the demand for energy and components two reference supermarkets are defined; one model representative for a supermarket today and one representative for a supermarket in 2030. The area, flow of customers and amount of display cabinets are based on "Butik C" in Axell, Lindberg and Lidbom (2004)[6].

To make it easier to draw any conclusions about causes and effects some basic parameters are assumed to be unchanged between 2010 and 2030, e.g. the dimensions of the supermarket, the flow of customers and the outdoor climate. The data of outdoor climate originates from measurements made in Göteborg by The Swedish Meteorological and Hydrological Institute in 2008. Note that 2008 is a leap year. The annual mean temperature in Göteborg in 2008 was 9.7 ˚C. The weather data, and consequently all simulations made in this study, are made with a resolution of 3 hours.

The indoor temperature is always 20 ˚C. This simplification disregards the possibility to have differentiated temperatures during daytime/nighttime and summertime/wintertime. The indoor absolute humidity is assumed to equal the outdoor absolute humidity in the winter and to be 2 $g_{water}/kg_{dry\ air}$ less humid than the outdoor air in the summer. The difference is assumed to increase linearly between winter and summer and the relation is obtained by studying figure 7.2 in Axell et al. (2004, p. 50)[6]. Details regarding calculations of indoor relative humidity are available in Appendix A.

The average customer is assumed to stay 20 minutes, and five persons are working in the supermarket. Every person is assumed to emit 144 W of sensible heat. The contribution of the sun to the internal heat generation is neglected since supermarkets seldom have windows. Storage of heat in the building structure is also neglected, since the mass of the building structure is small compared to the volume of the supermarket.

Only the sales area is studied, it is 670 m^2 and the flow of customers is 2744 people/day. The opening hours are set to 07:30 – 22:30. These will henceforth be referred to as daytime, and 22:30 – 07:30 will be referred to as nighttime. The ventilation is achieved by a CAV system with different flow rates during daytime and nighttime. The HVAC system includes a recuperative heat exchanger recovering heat from the exhaust air.

It is assumed that 61.5 % of the cooling load in the cabinets and gondolas is sensible (Svensson 2006, p. 13[22]). The dimensions of the supermarket are 34.7 m x 19.3 m x 3.15 m, but only the short side is exposed to the outside air. However, the results are claimed to be applicable to a detached supermarket as well since:

- The energy demands in a supermarket are very much influenced by the design of the HVAC system and the activities in the building (i.e. internal generation of heat), and to a much smaller extent by the properties of the building envelope.
- Detached supermarkets have better insulated outer walls than supermarkets integrated in bigger buildings (The Swedish Energy Agency 2010a[24]).

A conceptual model of the reference supermarket is presented in Figure 9.

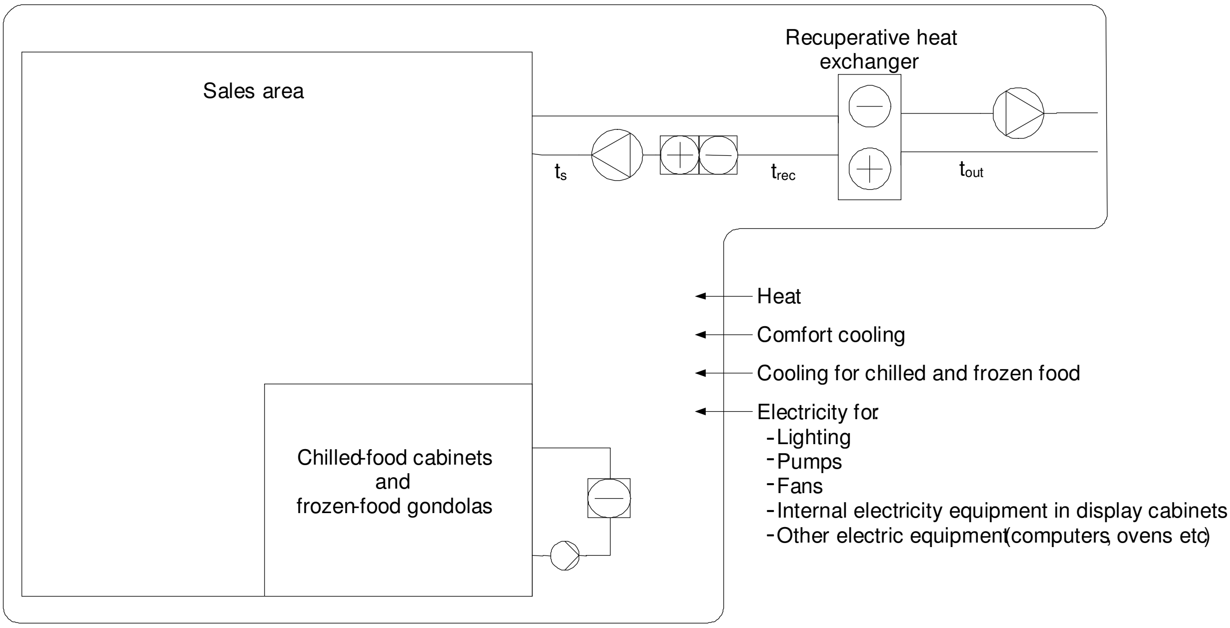


Figure 9 Schematic of the reference supermarket

3.1 The Supermarket of 2010

The values of building envelope transmission coefficients are obtained as an average of the supermarkets studied by the Swedish Energy Agency (2010a) [24]. The infiltration rate is set to 0.2 ACH during daytime and only 0.06 ACH during nighttime since the major part of the infiltration is assumed to occur through gateways and open doors in the entrance.

The recuperative heat exchanger has a maximum temperature efficiency of 70 % and the ventilation system operates at a specific fan power (SFP) of 2 kWs/m^3 at daytime flow rate (SFP is proportional to the square of the flow rate). The ventilation rate during daytime is set to 1.27 m^3/s, this equals the average maximum ventilation rate in the supermarkets studied by the Swedish Energy Agency (2010a) [24] (1.9 l/s and m^2). The ventilation rate is decreased by 50 % during nighttime.

The supermarket has a length of 44 m vertical chilled-food cabinet and 27 m horizontal frozen-food gondola. The performance of the vertical cabinets is specified according to the laboratory measurements made by Lindberg et al. (2010)[19]. The heat extraction rate equals 1804 W/m at daytime operation and 592 W/m at nighttime operation (with night-curtains). The horizontal gondola is assumed to have an average heat extraction rate of 321 W/m during daytime operation, and 224 W/m during nighttime operation. These values are produced by assuming a dimensioning cooling load of 450 W/m at an ambient temperature of 25 °C (Fahlén, 2000[11]), a defrost rate of 5 hours per day, a reduction of cooling load of 10 % due to the actual ambient temperature (Arias, 2005, p. 124[2]) and a cooling load reduction of 30 % due to night covering.

The heat generated within the horizontal frozen-food gondolas (lighting, fans, anti-sweat heaters and defrost equipment) is assumed to be 23 % of the heat extraction rate (Arias 2005, p. 124[2]). In the vertical chilled-food cabinets the internal generation of heat is assumed to be 12 % of the heat extraction rate. This is quite far below the percentage stated in Table 1 even though that was specified at higher ambient temperature. However, this assumption is more in accordance with display cabinets studied in supermarkets in Gothenburg during September 2010 (12 % in this case may imply a load of lighting of 145 W/m and fans/anti-sweat heaters of 70 W/m).

The required brine inlet temperature is set to -8 °C for the vertical cabinet and -33 °C for the horizontal gondola. This is 2 °C above the evaporation temperatures stated in Fahlén (2000, pp. 32-33)[11].

The amount of lighting equals 18.8 W/m^2 during daytime and 3.4 W/m^2 during nighttime. According to the Swedish Energy Agency (2010a, p. 33 & p. 52)[24] the average installed amount of lighting in the sales area of a supermarket is 18.8 W/m^2 and 18 % of the lighting is switched on beyond the opening hours. The supermarket comprises a bake-off oven of 7.6 kW used 3 hours per day and a heated merchandiser of 1.8 kW used during daytime. There is 20 m plug-in cabinet emitting a net amount of heat of 1041 W/m during daytime and 650 W/m during nighttime. This is the average power input of all plug-in cabinets manufactured by Norpe AB. A plug-in cabinet is a display cabinet with both condenser and evaporator enclosed in the cabinet frame. Since the amount of heat emitted in the condenser equals the heat absorbed in the evaporator plus the electricity supplied to the cabinet it can be treated as any other electric equipment (generation of heat equals supply of electricity).

3.2 The Supermarket of 2030

Of course it is not possible to know today what a supermarket in 2030 will look like. But by looking at recent trends a possible scenario can be defined.

The results from Lindberg et al. (2010)[19] are used to predict the performance of the vertical cabinets. The heat extraction rate in the vertical cabinets is 616 W/m during daytime and 280 W/m during nighttime. The heat extraction rate of the horizontal gondolas is assumed to be decreased by 30 % compared to 2010. The required brine inlet temperature is set to +2 °C for the vertical cabinet and -33 °C for the horizontal gondola.

The heat generated within the vertical chilled-food cabinets is assumed to be 17 % of the heat extraction, in accordance with the cabinet studied by Lindberg et al. (2010)[19]. Corresponding percentage is 23 % in the horizontal frozen-food gondolas (Arias 2005, p. 124[2]).

Compared to 2010, the supermarket of 2030 is assumed to have 10 % less infiltration, 5 % less transmission losses and 5 % more display cabinets. The ventilation system is still CAV but the SFP is decreased to 1.25 kWs/m^3 during daytime flow rate. In order to be able to remove the surplus of heat without too low supply air temperatures, the ventilation rate during daytime is doubled compared to 2010. The maximum efficiency of the recuperative heat exchanger is increased to 75 %.

The Swedish Energy Agency (2010a, p. 56)[24] anticipates the potential energy savings of lighting in commercial premises to 11 kWh/m^2 annually at an average uptime of 4000 hours per year. It is assumed that the potential savings are proportional to the uptime which implies an amount of lighting of 15.9 W/m^2 during daytime and 2.9 W/m^2 during nighttime (the lighting uptime is 6460 hours per year in this study). Any increase of the amount of plug-in cabinets is assumed to be compensated by their energy efficiency improvements. Bigger bake-off oven and heated merchandiser are used, the power of these are 9.4 kW and 2.6 kW respectively.

The parameters defining the reference supermarkets are presented in Table 2.

Table 2 Parameters of the reference supermarkets

	2010		2030	
	Daytime (Open)	Nighttime (Closed)	Daytime (Open)	Nighttime (Closed)
Supermarket area [m²]	670		670	
Flow of customers [people/day]	2744	0	2744	0
Vertical chilled-food cabinet [m]	44		46	
Average heat extraction rate in the chilled-food cabinets [W/m]	1804	592	616	280
Horizontal frozen-food gondola [m]	27		28	
Average heat extraction rate in the frozen-food gondolas at [W/m]	321	224	224	157
Ventilation system	CAV and direct recuperative heat exchanger		CAV and direct recuperative heat exchanger	
Temperature efficiency of HX [%]	70		75	
Ventilation rate [m³/s]	1.27	0.64	2.55	0.64
SFP at daytime air flow rate [kWs/m³]	2.0		1.25	
Infiltration rate [m³/s]	0.12	0.04	0.11	0.03
Heat transmission coefficient, outer wall [W/m²K]	0.56		0.44	
Heat transmission coefficient, floor [W/m²K]	0.16		0.16	
Heat generated by lighting, plug-in cabinets, ovens and heated merchandisers[W/m²]	54.9	22.8	53.6	22.3

3.3 Calculations

With information about the reference supermarkets, it is possible to calculate the demands for the four categories of energy presented in Figure 9. All calculations are made in the numerical calculation programme Matlab.

Comfort cooling and heating

The total internal generation of sensible heat in the supermarket can be expressed as

$$\dot{Q}_{int} = \dot{Q}_{people} + \dot{Q}_{lights/equipment} + \dot{Q}_{heat,cabinets} - \dot{Q}_{cool,cabinets} \qquad \text{(eq. 3.1)}$$

$\dot{Q}_{people}$ is the sensible heat emitted from people, $\dot{Q}_{lights/equipment}$ is sensible heat emitted from lighting, plug-in cabinets, ovens etc. (this also includes 50 % of the heat emitted by the ventilation fans). $\dot{Q}_{heat,cabinets}$ is heat from internal heat sources in the conventional cabinets and gondolas and $\dot{Q}_{cool,cabinets}$ is the sensible heat absorbed by the display cabinets and gondolas. The internal generation of heat is presented in Figure 10.

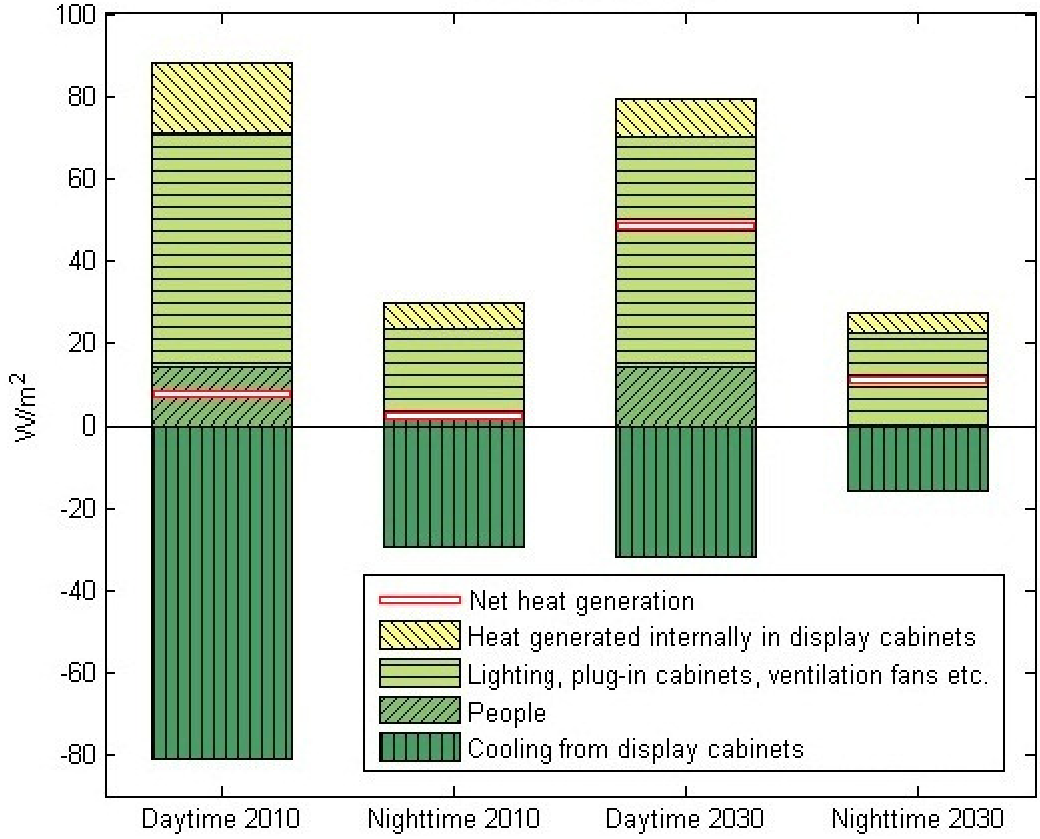


Figure 10 Internal generation of heat

Heat is transferred through the building envelope by infiltration and by transmission. According to Nilsson (2003)[20], these losses can be expressed as

$$\dot{Q}_{inf} = \dot{V}_{inf} \cdot \rho \cdot c_p \cdot (t_{in} - t_{out}) = K_{inf} \cdot (t_{in} - t_{out}) \quad \text{(eq. 3.2)}$$

$$\dot{Q}_{tr} = \sum U_j \cdot A_j \cdot (t_{in} - t_{out}) = K_{tr} \cdot (t_{in} - t_{out}) \quad \text{(eq. 3.3)}$$

U is the heat transmission coefficient of the building envelope, $\dot{V}_{inf}$ is the rate of infiltration, A is the area exposed to the outdoor, t_{in} and t_{out} are the indoor and outdoor temperature respectively and ρ and c_p are the density and specific heat capacity of air.

When the internal generation of heat and the heat losses are known, the required supply air temperature is calculated with an energy balance according to

$$t_s = t_{in} - \frac{\dot{Q}_{int} - (K_{tr} + K_{inf}) \cdot (t_{in} - t_{out})}{\dot{V}_{vent} \cdot \rho \cdot c_p} \quad \text{(eq. 3.4)}$$

$\dot{V}_{vent}$ is the rate of ventilation. The recuperative heat exchanger located between the supply and exhaust air streams recovers heat from the exhaust air and changes the temperature of the supply air from t_{out} to t_{rec}, which is expressed as

$$t_{rec} = t_{out} + \eta_T \cdot (t_{in} - t_{out}) \quad \text{(eq. 3.5)}$$

The temperature efficiency of the heat exchanger, η_T, is controlled between zero and its maximum value in order to minimize the energy demand.

With t_s and t_{rec} known, the demand for heating is calculated as

$$\dot{Q}_h = K_{vent} \cdot (t_s - t_{rec}) \qquad \text{when } t_{rec} < t_s \qquad \text{(eq. 3.6)}$$

K_{vent} equals $\dot{V}_{vent} \cdot \rho \cdot c_p$. When t_{rec} exceeds t_s, there is a demand for comfort cooling. Since condensation will occur when the temperature of the surface of the cooling coil is below the dewpoint of the air, this latent load must be taken into account. In accordance with Nilsson (2003)[20], the demand for comfort cooling is calculated as

$$\dot{Q}_{cc} = K_{vent} \cdot (t_{rec} - t_s) + \rho \cdot \dot{V}_{vent} \cdot \left(r_a \cdot \Delta x_a + c_{p,v} \cdot (t_{rec} - t_s) \cdot \Delta x_a\right) \qquad \text{when } t_{rec} < t_s \qquad \text{(eq. 3.7)}$$

r_a = enthalpy of vaporization (= 2500000 J/kg)
Δx_a = change of airflow humidity ratio (kg/kg)
$c_{p,v}$ = specific heat capacity of water vapour (=1900 J/kgK)

The first term of equation 3.7 represents the sensible cooling load while the second term represents the latent cooling load, i.e. the heat that is released when moisture in the air condenses. When calculating Δx_a, it is assumed that the temperature of the surface of the cooling coil is 8 °C. Calculations regarding Δx_a are presented in Appendix B.

When t_{rec} equals t_s, there is no need for additional heating or cooling. By plotting t_s, t_{rec}, t_{out} and t_{in} in a duration diagram, the demand for heating and sensible cooling is efficiently illustrated (Nilsson 2003[20]). This is presented in Figure 11 and Figure 12. The area between t_s and t_{rec} is proportional to the amount of energy needed and the vertical distance between the lines is proportional to the power. However, it is important to remember that the ventilation flow rate differs in the four cases (see Table 2) and the required heat to change the supply air temperature is proportional to the flow rate. Note that these diagrams do not say anything about the latent cooling load.

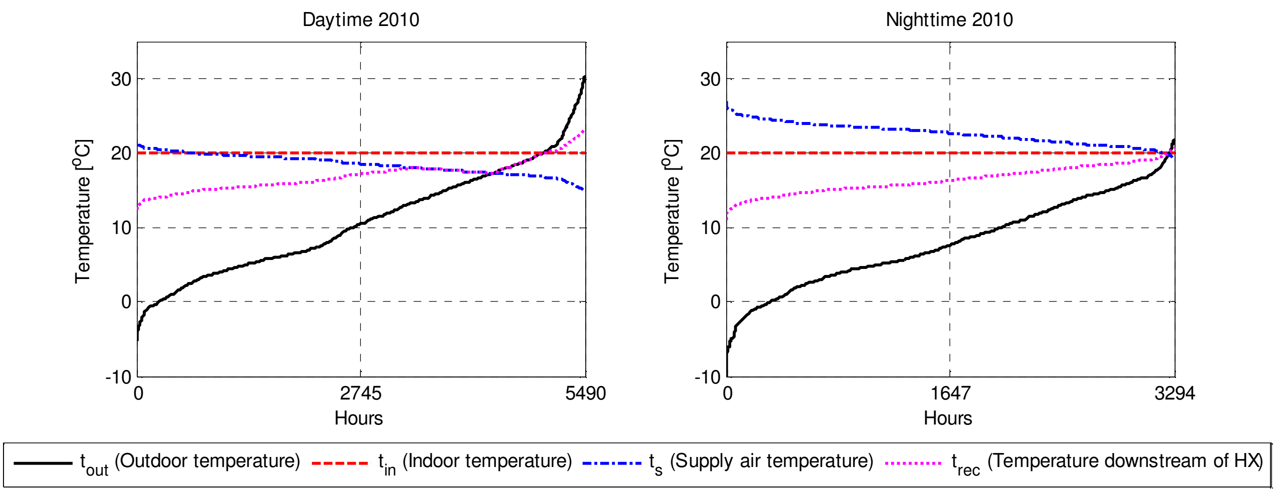


Figure 11 Duration curves of the heat balance for the supermarket of 2010

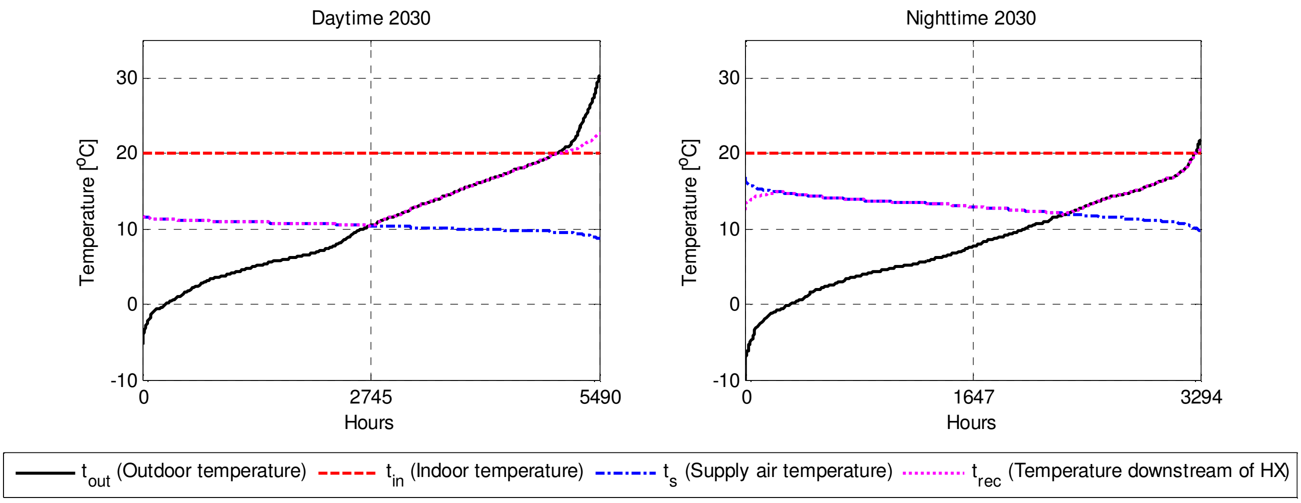


Figure 12 Duration curves of the heat balance for the supermarket of 2030

Cooling for the refrigeration of food

The cooling (extraction of heat) used in the vertical cabinets and horizontal gondolas is expressed as

$$\dot{Q} = L \cdot \dot{q} \cdot \gamma \qquad \text{(eq. 3.8)}$$

L is the length of the cabinet/gondola, $\dot{q}$ is the specific average heat extraction rate and γ is the cooling load correction factor due to indoor humidity (see Figure 3).

Electricity

The demand for electricity equals

$$\dot{W}_{tot} = \dot{W}_{lights/equipment} + \dot{W}_{cabinets} + \dot{V}_{vent} \cdot SFP \qquad \text{(eq.3.9)}$$

$\dot{W}_{lights/equipment}$ is the electricity used by lighting, ovens, plug-in cabinets and electronic equipment. $\dot{W}_{cabinets}$ is electricity used by fans, lighting, anti-sweat heaters and defrost equipment in the conventional cabinets and gondolas. Figure 13 presents the energy needed in the reference supermarket of 2010 and 2030 respectively. Note that the cooling and heating may be produced by electricity, but with this approach the demand is kept technology neutral.

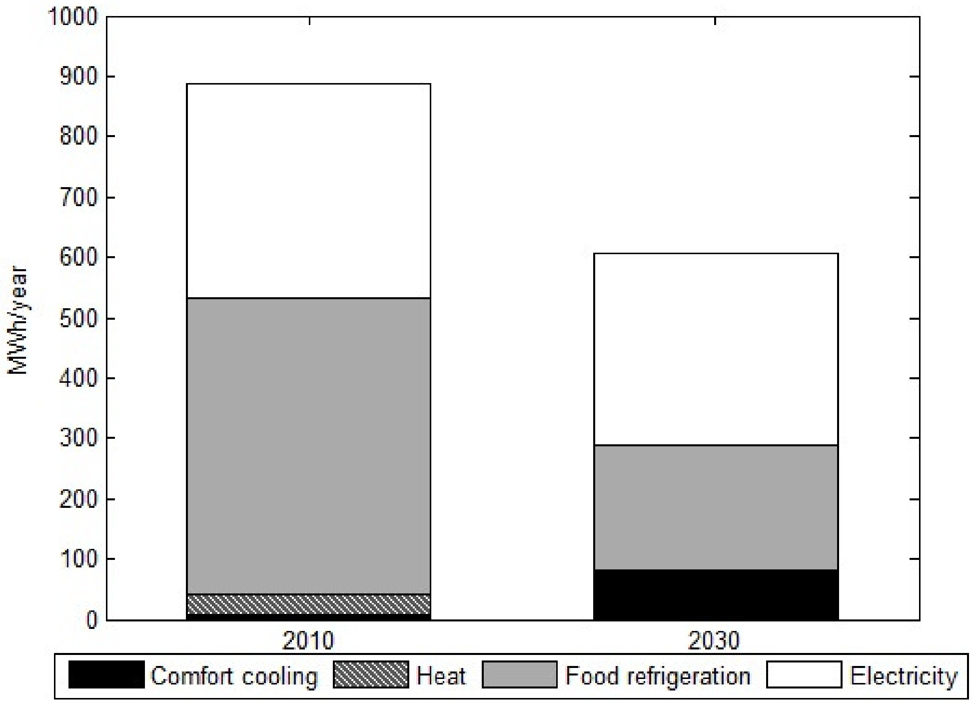


Figure 13 Energy demand in the reference supermarkets

4 Alternative System Designs

In this section, four different ways to meet the energy demands defined in the previous section are presented.

4.1 Selection of Components

Regarding required components, only components that differ substantially between the alternatives are included in the comparison. E.g. costs for cooling coils and heating coils controlling the temperature of the supply air are assumed to be equal for all alternatives. To achieve a fair comparison, it is important to be consistent when selecting components. The choice of manufacturers of components is done with respect to availability of information about performance and prices. Since there are different demands for energy in 2010 compared to 2030, different sizes of components are needed. This makes it necessary to define two sets of components for each alternative.

<u>Compressor</u>

Only semi-hermetic reciprocating compressors are dealt with in this study, since these are very common in supermarket refrigeration applications. The selection of compressors is based on calculations with subcooling of 5 K, superheating of 10 K and R404A as the refrigerant. 70 % of the refrigerants used in Swedish supermarkets in 2003 were R404A (Arias et al. 2004, p. 17[3]). Due to safety of operation, two chillers are used on the chilled- and frozen food side respectively. The selection is done with the calculation software Bitzer Software (Bitzer 2010)[8]. The dimensioning power, presented in Table 3-Table 6, equals the cooling load.

<u>Dry cooler</u>

By using the calculation software AIACalc (Asarums industri AB, 2010)[1], the sufficient dry cooler with lowest cost is selected. The condenser coolant is assumed to be a mixture of water and 30 % ethylene glycol. It is assumed that the coolant inlet and outlet temperatures are 8 and 2 K above the outdoor temperature (38 °C /32 °C in the dimensioning case). Separate dry coolers for comfort cooling and food refrigeration are used. This is common since the comfort cooling is usually operated by the property owner while the food refrigeration is operated by the retailer (Johnsson[27]).

<u>District cooling substation</u>

District cooling substations are selected from the company SWEP International AB. Substations are available in customised sizes. However, quotations of prices were received only for a dimensioning power of 50 kW, 100 kW and 150 kW respectively. Consequently, interpolation and extrapolation are used to estimate the costs of the required substations. The substations include all necessary controllers and electrical components.

<u>Condenser and evaporator</u>

Plate heat exchangers are used as condensers and evaporators. In completely indirect refrigeration systems this is always the case (Arias et al. 2004, p. 31[3]). The calculation software SSP G7 (SWEP International AB, 2010)[21] is used and the sufficient heat exchangers with lowest costs are selected. To calculate the temperature of the refrigerant at the inlet of the condenser, which is required by the calculation software, the isentropic efficiency of the compressor is assumed to be 70 %.

4.2 Alternative 1

In this alternative, district cooling is used both for comfort cooling and condenser cooling. Neither condenser heat recovery nor floating condensing is possible. The air cooler is connected in series with the condensers and the design makes it possible to by-pass the secondary coolant when there is no need for comfort cooling. The condensation temperature is fixed at 20 °C. Figure 14 shows a schematic of the system.

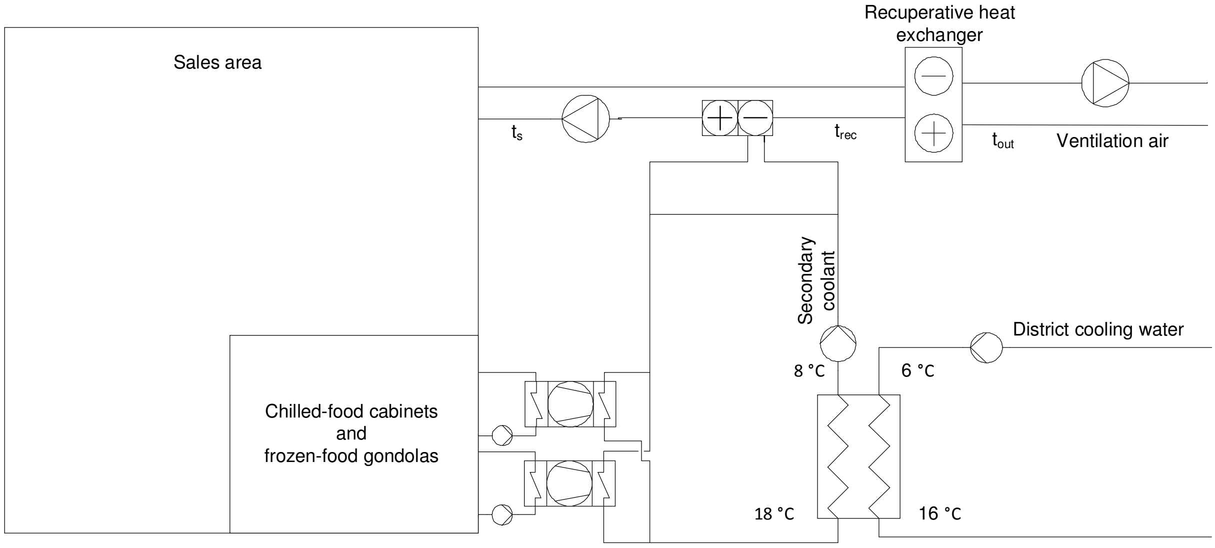


Figure 14 Schematic of Alternative 1

When studying systems involving heat exchangers, it is convenient to illustrate the operation in a T-Q diagram. The purpose of this is to enhance understanding and make sure that no heat is transferred from a lower to a higher temperature fluid. The x-axis represents the amount of heat transferred and the y-axis represents the temperature levels. The two operation modes of alternative 1 are illustrated in a T-Q diagram in Figure 15.

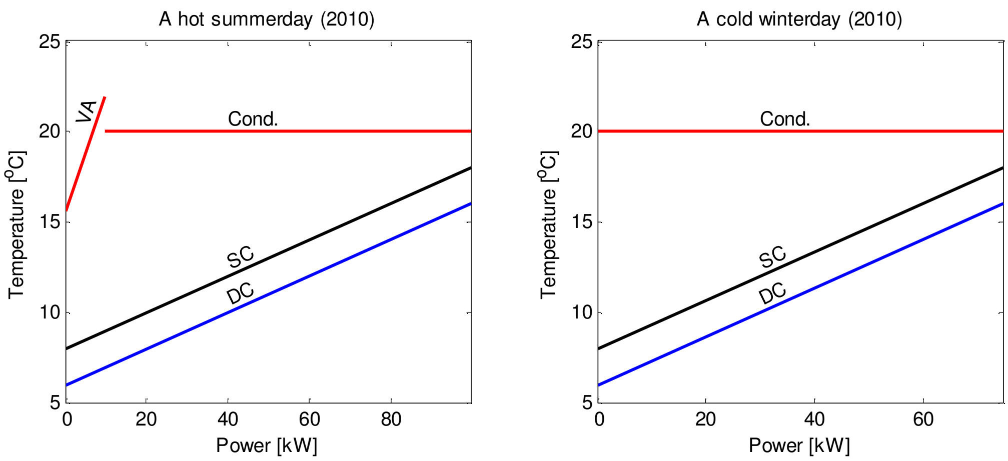


Figure 15 T-Q-diagrams for operation with and without comfort cooling (abbreviations are explained in the designations section)

Required components

The components required specifically in alternative 1 are listed in Table 3. When dimensioning the condensers it is important to take into account that the water inlet temperature is above 8 °C when there is a comfort cooling demand.

Table 3 Required components in alternative 1

Component	Year	Dimensioning power [kW]	Selected model	Cost [kSEK]	Comment
District cooling substation	2010	144	SWEP Substation, 144 kW	94.4	
	2030	126	SWEP Substation, 126 kW	93.8	
Compressor (chilled food)	2010	91	2 x 4J-13.2Y	161.6	
	2030	32	2 x 2CC-4.2Y	51.7	
Compressor (frozen food)	2010	10	2 x 4EC-4.2Y	55.2	
	2030	7	2 x 2CC-3.2Y	49.6	
Condenser (chilled food)	2010	112	2 x B25Tx90	27.8	$t_{SC,in} = 8.5$ °C
	2030	37	2 x B25Tx80	24.5	$t_{SC,in} = 13.8$ °C
Condenser (frozen food)	2010	14	2 x BX8Tx58	13.0	$t_{SC,in} = 8.5$ °C
	2030	10	2 x B15x50	13.1	$t_{SC,in} = 13.8$ °C
			Total: 2010	352.0	
			2030	232.7	

The cost of connecting the supermarket to the district heating system is not included. In 2030, this is justified by the fact that the heat demand is too small to invest in a heat system other than an electric heater. In 2010, however, it is assumed that the supermarket is already in the proximity of the district heating network. A supermarket far away from the district heating network is very unlikely to connect to the district cooling network, and thereby not interesting in this study. Any connection costs are assumed to equal the costs of the condenser heat recovery system in the other alternatives and are therefore not included.

4.3 Alternative 2

In this system, district cooling is not utilized at all. The comfort cooling demand is supplied by an ordinary chiller. When there is a heat demand, condenser heat is recovered by rejecting it in the supply air stream. When there is no heat demand, all condenser heat is rejected in roof-mounted dry coolers.

The ambient outdoor air temperature determines the condensation temperature of the chillers. However, it is not desirable to have the lowest possible condensation temperature since this would cause unacceptably high power consumption in the dry cooler fans. There is an optimization issue to minimize the combined power consumption of dry cooler fans and chiller compressor. According to Johnsson[27], a good compromise is to operate at a condensation temperature of 10 °C above the outdoor temperature. It is important to use speed-controlled fan motors in order to adjust the dry cooler fans to the actual demand.

The power consumption of the dry cooler fans is proportional to the cube of the cooling load in the dry cooler. Equation 4.1 is used to calculate the fan power.

$$\dot{W}_{fan} = \dot{W}_{fan,max} \cdot \left(\frac{\dot{Q}}{\dot{Q}_{max}}\right)^3 \qquad \text{(eq. 4.1)}$$

$\dot{Q}$ is the cooling load, $\dot{Q}_{max}$ is the maximum cooling load and $\dot{W}_{fan,max}$ is the fan power needed at the maximum cooling load, which is obtained from the manufacturers calculation software.

All chillers are operated at a condensation temperature of 10 °C above the outdoor temperature when there is no heat demand. When there is a heat demand, the condensation temperature of the chillers used for condenser heat recovery is 10 °C above the indoor supply air temperature. It is not necessary to have the same condensation temperature in the chilled-food chillers as in the frozen-food chillers. However, the two chilled-food chillers and the two frozen-food chillers respectively are operated at the same condensation temperature. According to Figure 16 it is obvious to choose the frozen-food chillers to recover heat from. Figure 16 shows the heat demand of the supermarket, the available heat in the condensers and the limit under which an electric heater would be more energy efficient (see eq. 2.8). Recovering heat from the chilled-food chillers is approximately as bad as using an electric heater. Recovering heat from the frozen-food chiller is better, but sometimes the heat demand exceeds the amount of available heat. The heat shortages are assumed to be solved with an electric heater. This concerns the supermarket of 2010, in 2030 the heat demand is too small to motivate the investment of a heat recovering system.

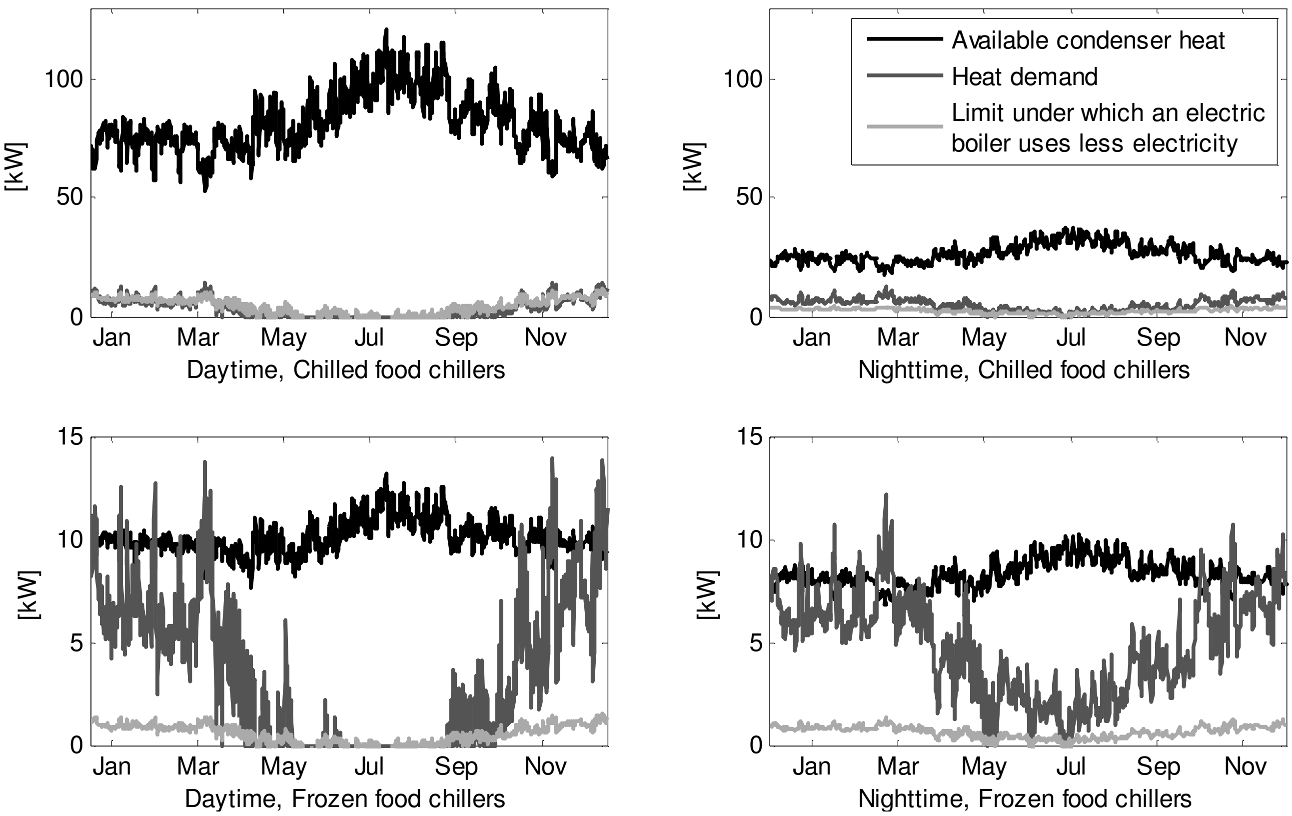


Figure 16 Heat demand, condenser heat and the smallest heat demand needed to have lower electricity consumption in a condenser heat recovery system compared to an electric heater (2010). Note the different scales.

The minimum condensation temperature is set to 5 °C to avoid too low pressure difference in the chillers. Figure 17 shows a schematic of the system and Figure 18 illustrates the operation in a T-Q diagram.

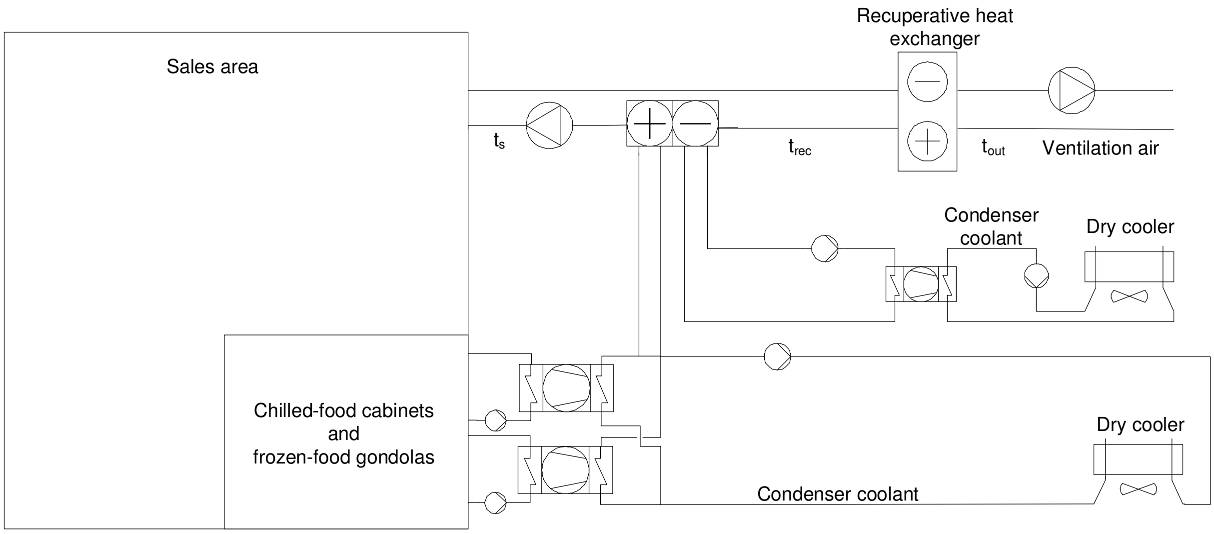


Figure 17 Schematic of alternative 2

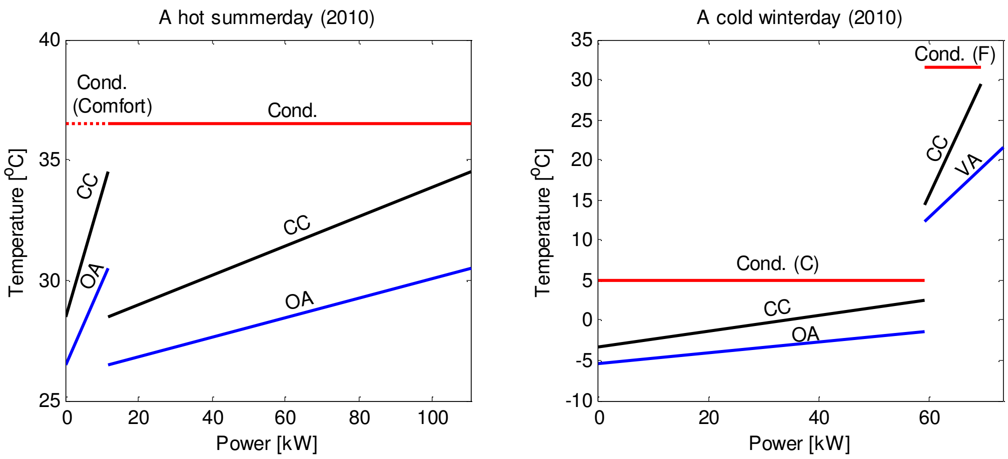


Figure 18 T-Q-diagrams for summer- and wintertime operation respectively (abbreviations are explained in the designations section)

Required components

The components required specifically in alternative 2 are listed in Table 4.

Table 4 Required components in alternative 2

Component	Year	Dimensioning power [kW]	Selected model	Cost [kSEK]	Comment
Compressor (comfort cooling)	2010	22	4EC-6.2Y	31.5	T_2= 6 °C
	2030	80	4G-30.2Y	91.5	
Evaporator (comfort cooling)	2010	22	B25Tx50	8.6	
	2030	80	B50Hx100	37.0	
Condenser (comfort cooling)	2010	28	B25Tx80	12.3	
	2030	99	B120Tx180	49.0	

Compressor (chilled food)	2010	91	2 x 4G-20.2Y	175.9	
	2030	32	2 x 4EC-6.2Y	63.1	
Compressor (frozen food)	2010	10	2 x 4TCS-8.2Y	97.6	
	2030	7	2 x 4DC-5.2Y	69.0	
Condenser (chilled food)	2010	121	2 x B120Tx110	66.9	
	2030	40	2 x B25Tx60	19.6	
Condenser (frozen food)	2010	15	2 x B25Tx20	10.0	
	2030	11	2 x BX8Tx58	13.0	
Dry cooler (refrigeration)	2010	136	XPM-91X-23-4B4-V-900-41	176.8	$W_{fan,max}$=15.0 kW
	2030	51	XPM-91X-2-5B1-V-900-20	73.8	$W_{fan,max}$=5.0 kW
Dry cooler (comfort cooling)	2010	28	X2-D-80Q1A-1-3D-6B4-V-900-12-boostfin	49.5	$W_{fan,max}$=1.8 kW
	2030	99	XPM-91X-4-5B4-V-900-36	133.7	$W_{fan,max}$=10.0 kW
			Total: 2010	629.1	
			2030	549.7	

4.4 Alternative 3

This design is a mix of alternative 1 and 2. The condenser heat is rejected in roof-mounted dry coolers but the comfort cooling demand is achieved by district cooling. Heat is recovered from the frozen-food chiller condenser when there is a heat demand in the supermarket. The condensation temperatures are determined as in alternative 2. Figure 19 shows a schematic of the system and Figure 20 illustrates the operation in a T-Q diagram.

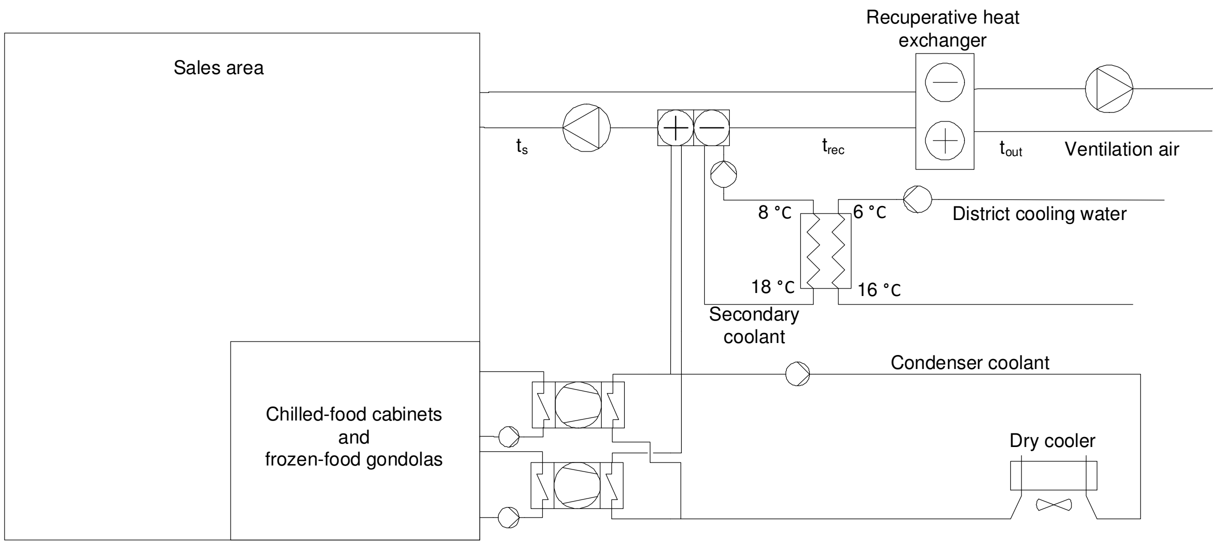


Figure 19 Schematic of alternative 3

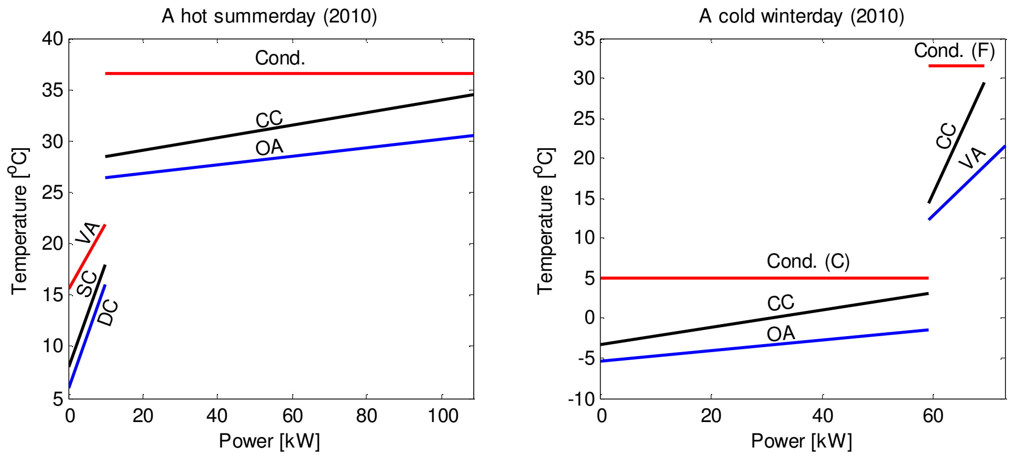


Figure 20 T-Q-diagrams for summer- and wintertime operation respectively (abbreviations are explained in the designations section)

Required components

The components required specifically in alternative 3 are listed in Table 5.

Table 5 Required components in alternative 3

Component	Year	Dimensioning power [kW]	Selected model	Cost [kSEK]	Comment
District cooling substation	2010	22	SWEP Substation, 22 kW	30.9	
	2030	80	SWEP Substation, 80 kW	81.2	
Compressor (chilled food)	2010	91	2 x 4G-20.2Y	175.9	
	2030	32	2 x 4EC-6.2Y	63.1	
Compressor (frozen food)	2010	10	2 x 4TCS-8.2Y	97.6	
	2030	7	2 x 4DC-5.2Y	69.0	
Condenser (chilled food)	2010	121	2 x B120Tx110	66.9	
	2030	40	2 x B25Tx60	19.6	
Condenser (frozen food)	2010	15	2 x B25Tx20	10.0	
	2030	11	2 x BX8Tx58	13.0	
Dry cooler	2010	136	XPM-91X-23-4B4-V-900-41	176.8	$W_{fan,max}$=15.0 kW
	2030	51	XPM-91X-2-5B1-V-900-20	73.8	$W_{fan,max}$=5.0 kW
			Total: 2010	558.1	
			2030	319.7	

4.5 Alternative 4

This system gives priority to low condensation temperatures. Condenser heat recovery is not utilized. As long as the outdoor air temperature is below 10 °C, the condenser heat is rejected in dry coolers allowing a condensation temperature of 10 °C above the outdoor air temperature. When the outdoor air temperature exceeds 10 °C, district cooling is used and the condensation temperature is 20 °C. District cooling is also used to cover the comfort cooling demand. A schematic of the system is presented in Figure 21 and the operation is illustrated in a T-Q diagram in Figure 22.

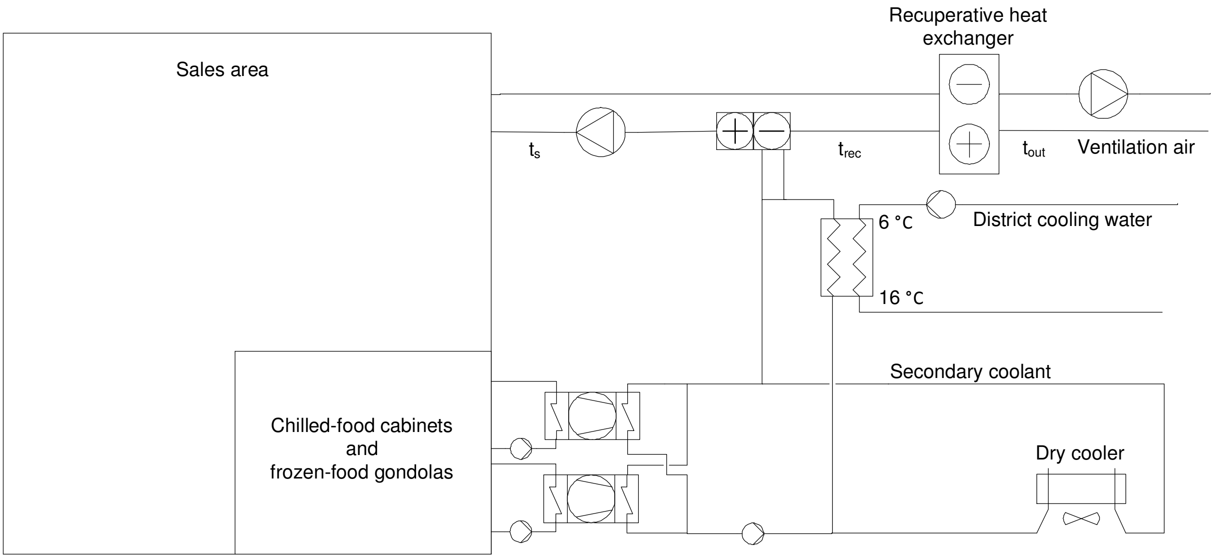


Figure 21 Schematic of alternative 4

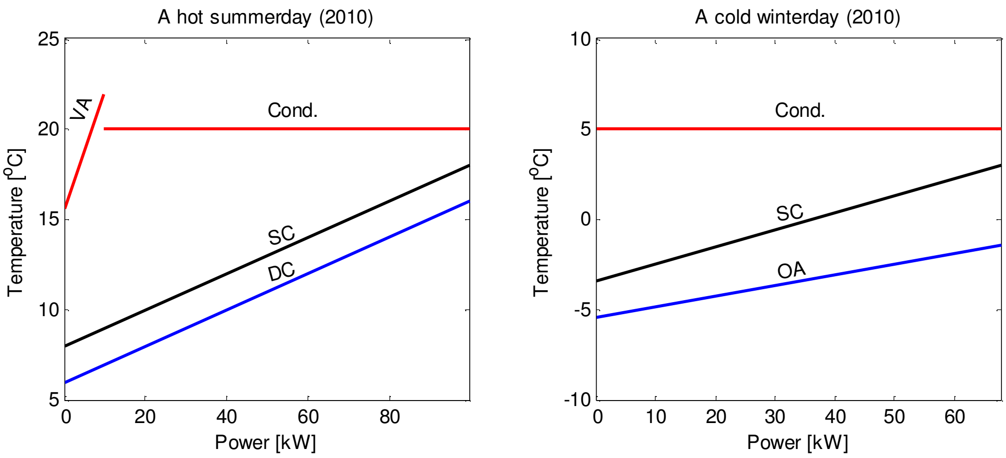


Figure 22 T-Q-diagrams for summer- and wintertime operation respectively (abbreviations are explained in the designations section)

Required components

The components required specifically in alternative 4 are listed in Table 6.

Table 6 Required components in alternative 4

Component	Year	Dimensioning power [kW]	Selected model	Cost [kSEK]	Comment
District cooling substation	2010	144	SWEP Substation, 144 kW	94.4	
	2030	126	SWEP Substation, 126 kW	93.8	
Compressor (chilled food)	2010	91	2 x 4J-13.2Y	161.6	
	2030	32	2 x 2CC-4.2Y	51.7	
Compressor (frozen food)	2010	10	2 x 4EC-4.2Y	55.2	
	2030	7	2x 2CC-3.2Y	49.6	
Condenser (chilled food)	2010	112	2 x B25Tx90	26.9	$t_{cc,in} = 8.5$ °C
	2030	37	2 x B25Tx80	24.5	$t_{cc,in} = 13.8$ °C
Condenser (frozen food)	2010	14	2 x BX8Tx58	13.0	$t_{cc,in} = 8.5$ °C
	2030	10	2 x B15x50	13.1	$t_{cc,in} = 13.8$ °C
Dry cooler	2010	102	XPM-91X-5-4B4-V-900-36	145.1	$W_{fan,max}$=12.5 kW $t_{out} = 10.0$ °C
	2030	39	X2-D91Q2A-1-3E-6B4-V-900-12-boostfin	60.3	$W_{fan,max}$=3.6 kW $t_{out} = 10.0$ °C
			Total: 2010	496.2	
			2030	293.0	

5 Results

5.1 Energy Use

The electricity demand presented in Figure 13 is independent of alternative. It equals 352.9 MWh/year in 2010 and 317.7 MWh/year in 2030 and <u>is excluded in this section from now on.</u> Figure 23-Figure 26 present the remaining demand for energy in the four alternatives respectively.

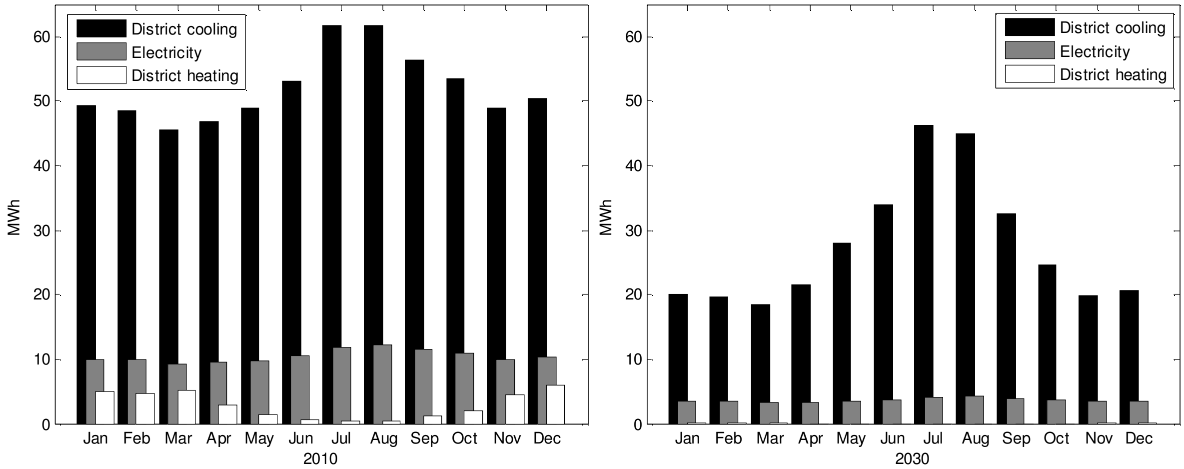


Figure 23 Monthly use of energy for alternative 1 (District cooling used for comfort cooling as well as condenser cooling all year)

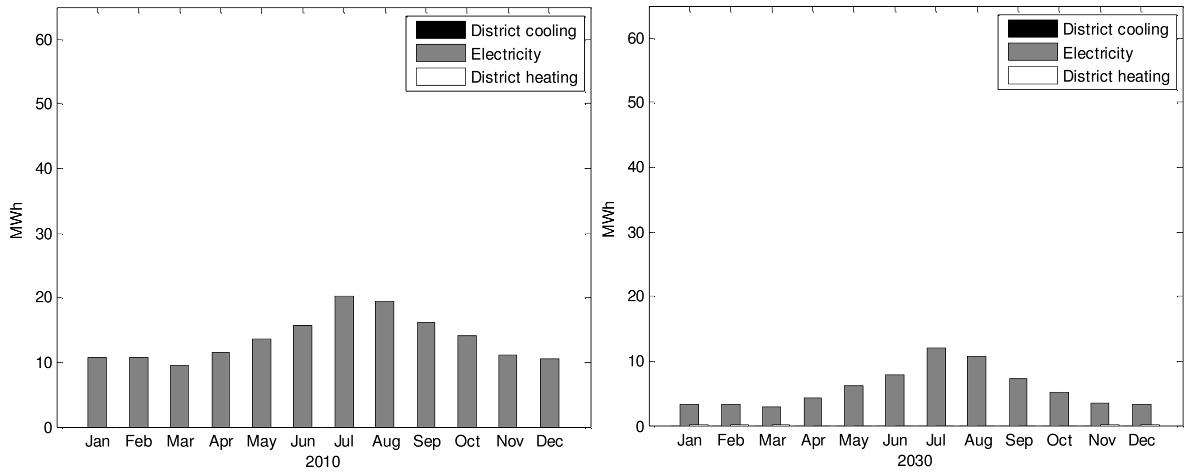


Figure 24 Monthly use of energy for alternative 2 (No use of district cooling)

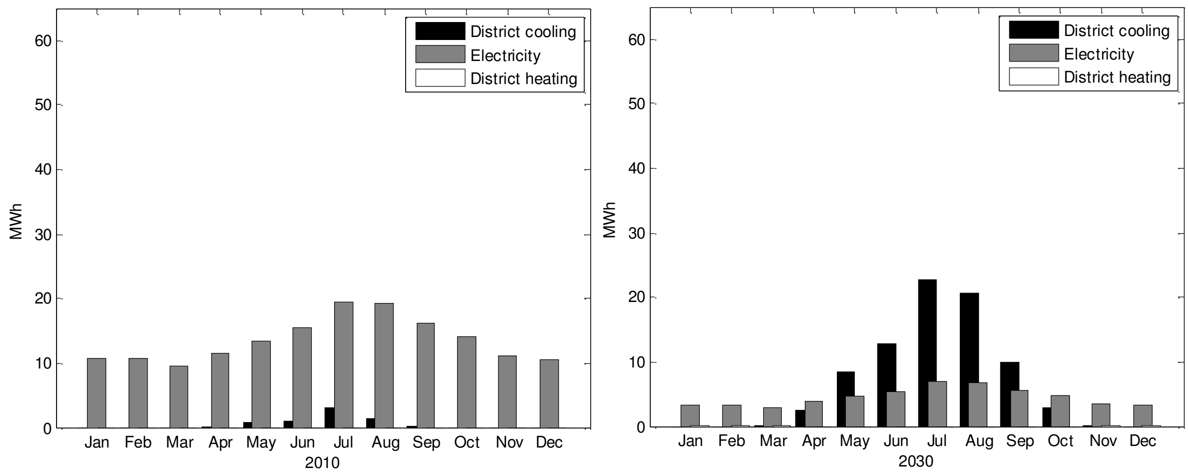


Figure 25 Monthly use of energy for alternative 3 (District cooling used for comfort cooling)

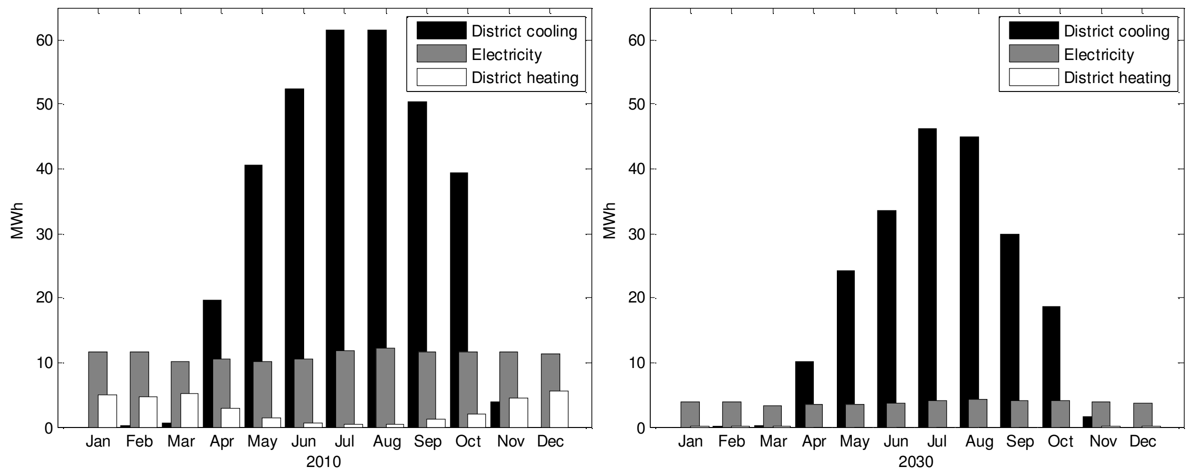


Figure 26 Monthly use of energy for alternative 4 (Using district cooling for comfort cooling as well as condenser cooling when the outdoor air temperature exceeds 10 °C)

The annual use of energy is presented in Table 7 and Table 8 respectively.

Table 7 Annual use of energy 2010 [MWh/year]

	Alternative 1	Alternative 2	Alternative 3	Alternative 4
District heating	34.3	0	0	34.3
District cooling	625.0	0	7.0	330.6
Electricity	126.0	164.7	163.2	135.5

Table 8 Annual use of energy 2030 [MWh/year]

	Alternative 1	Alternative 2	Alternative 3	Alternative 4
Heating	0.2	0.2	0.2	0.2
District cooling	331.0	0	80.1	209.6
Electricity	43.8	70.0	54.8	45.7

The heat demand in 2030 is way too small to invest in a heating system other than an electric heater. However this is equal in all alternatives and is therefore not bothering the comparison.

The difference in average COP of the refrigeration systems depends on the condensation temperatures only. Since the cooling load varies due to changes in weather conditions it is more appropriate to compare the seasonal performance factor, SPF, defined as the total cooling generated during one year divided by the total compressor work during one year. By also including parasitic losses a fair comparative figure is obtained. Since the pump power is assumed to be equal in all alternatives, the parasitic losses in this comparison only include the fan power of the dry coolers. The average COP, the SPF and the SPF including dry cooler fan power are presented in Table 9.

Table 9 Comparison of energy efficiency figures (food cooling, 2030)

	Alternative 1	Alternative 2	Alternative 3	Alternative 4
Average COP	4.73	5.44	5.44	5.67
SPF	4.73	4.64	4.64	5.22
SPF incl. dry cooler fan power	4.73	3.78	3.78	4.54

5.2 Economic Comparison

To be able to compare investment costs with future demands for energy it is necessary to use an economic model. In this study, the net present value method is used. By calculating the net present value (cost) of future demands for energy, these can be added to the investment and enable comparison of the four different alternatives. By setting the net present value of an alternative including district cooling equal to the net present value of the alternative without district cooling a competitive price of district cooling is obtained. The price of district cooling that equalizes the net present value of the two alternatives is calculated according to equation 5.1.

$$C_{DC} = \frac{A_1 - A_2 + I \cdot \left(C_{el} \cdot \left(Q_{el,1} - Q_{el,2}\right) + C_{DH} \cdot \left(Q_{DH,1} - Q_{DH,2}\right)\right)}{Q_{DC,2} \cdot I} \qquad \text{(eq. 5.1)}$$

A = Investment cost
Q_{el} =Annual demand for electricity
Q_{DC} = Annual demand for district cooling
Q_{DH} =Annual demand for district heating
C_{DC}= Competitive price of district cooling
C_{el} = Price of electricity
C_{DH} =Price of district heating
I= Net present value factor

Subscripts 1 and 2 represent an alternative without and with district cooling respectively.

The net present value factor is calculated from

$$I = \frac{1 - (1 + i)^{-n}}{i} \qquad \text{(eq. 5.2)}$$

i is the discount rate and n is the economic life of the investment. The discount rate is set to 7 %, the economic life to 10 years, the price of electricity to 1000 SEK/MWh and the price of district heating to 700 SEK/MWh. The annual energy demands were presented in Table 7 and Table 8 and the investment costs of the four alternatives are summarized in Table 10.

Table 10 Investment costs [kSEK]

	2010	2030
Alternative 1	352.0	232.7
Alternative 2	629.1	549.7
Alternative 3	558.1	319.7
Alternative 4	496.2	293.0

Table 11 presents the calculated price, quantity and load factor of district cooling. The load factor is defined as ratio of the average demand to the peak demand.

Table 11 Price, quantity and load factor of district cooling

	Alternative 1			Alternative 3			Alternative 4		
	Price [SEK/MWh]	Quantity [MWh]	Load factor [%]	Price [SEK/MWh]	Quantity [MWh]	Load factor [%]	Price [SEK/MWh]	Quantity [MWh]	Load factor [%]
2010	87	625.0	49	1658	7.0	4	73	330.6	26
2030	216	331.0	30	599	80.1	11	290	209.6	19

The price presented in Table 11 includes all fees the supermarket would pay for the district cooling, this is often referred to as the specific price of district cooling. In reality, this price is often divided into a price of energy [SEK/MWh], a price of power [SEK/kW·year] and in some cases also an access fee [SEK].

5.3 Sensitivity Analysis

Since the results are valid only under the conditions used in this study, they cannot be regarded as universal and applicable to all supermarkets. By carrying out a sensitivity analysis, the relation between the input parameters and the results is investigated, and more general conclusions can thereby be drawn. The outputs of interest in this analysis are the price and the amount of district cooling. The analyzed input parameters are divided in two categories.

- Parameters that influence the results directly:
  - Discount rate
  - Economic life
  - Price of energy
  - Price of district cooling substation
- Parameters that influence the results indirectly through the demand for energy and components:
  - Internal generation of heat
  - Outdoor temperature
  - Amount of chilled-food cabinets and frozen-food gondolas

The first category does influence only the price of district cooling, not the amount. To avoid an overwhelming amount of information, only the results of the case of 2030 are analyzed. There is one important simplification used in this sensitivity analysis; the price of each component is assumed to be proportional to its dimensioning power. This implies that if the dimensioning power of a condenser is increased by 15 %, the price of that condenser is increased by 15 %. The same approach is used to calculate the maximum fan power in the dry coolers.

Discount rate

The influence of changes in discount rate is presented in Figure 27. Note that the y-axes are shifted unequally. However, the scales are equal in order to make it easy to see how the alternatives differ in sensitivity.

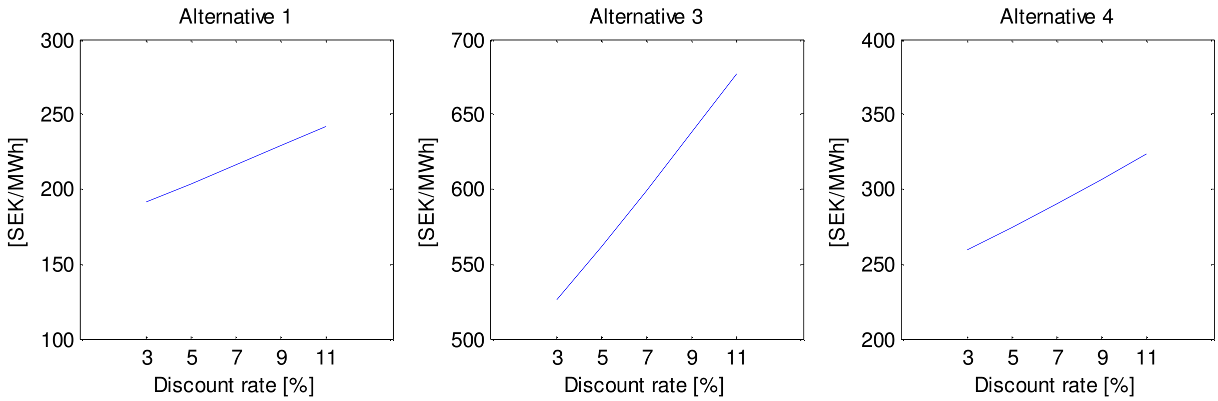


Figure 27 The price of district cooling as a function of the discount rate.

Economic life

The influence of changes in economic life is presented in Figure 28.

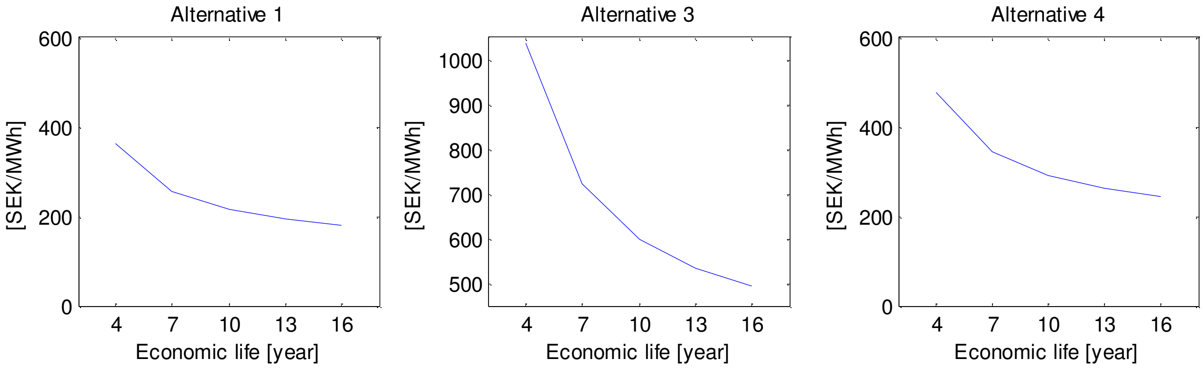


Figure 28 The price of district cooling as a function of the economic life.

Price of energy

Since the supermarket of 2030 is assumed to not use district heating, only the price of electricity is investigated. Two approaches are used to analyze the influence of the electricity price. First, it is assumed that the price of electricity is constant during the lifetime of the investments. Then, it is assumed that the price of energy is increased during the lifetime of the investments by a certain percentage annually. The result of the first approach is presented in Figure 29.

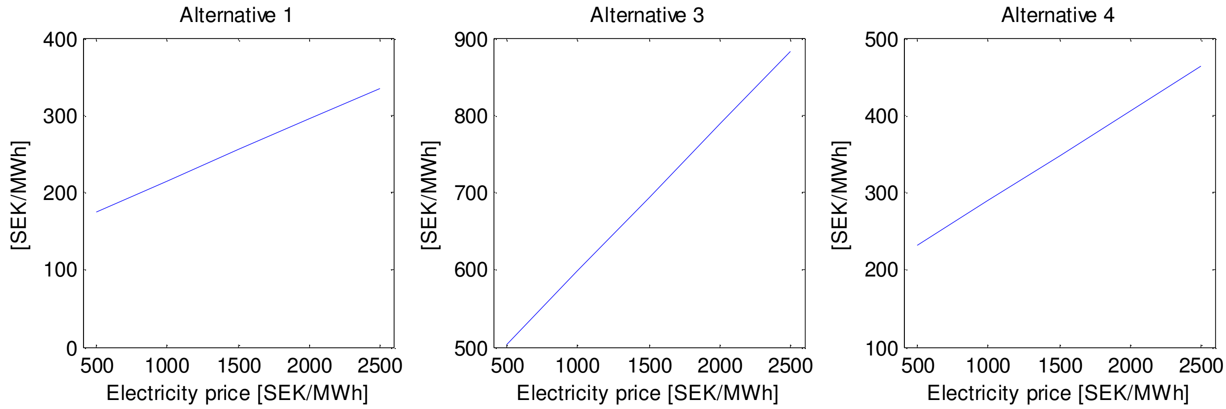


Figure 29 The price of district cooling as a function of the electricity price.

In the approach with increasing prices of energy, the initial price of electricity is kept at 1000 SEK/MWh. It is assumed that the percental increase in the price of electricity equals the percental increase in the price of district cooling. Mathematically, this is obtained by substituting the net present value factor calculated in eq. 5.2 into the one calculated in eq. 5.3 (Nilsson 2003)[20].

$$I_q = \frac{1 - \left(\frac{1+q}{1+i}\right)^n}{\frac{1+i}{1+q} - 1} \qquad \text{(eq. 5.3)}$$

q equals the annual increase in energy prices. The result is the initial price of district cooling (y-axis), while the annual increase is presented on the x-axis.

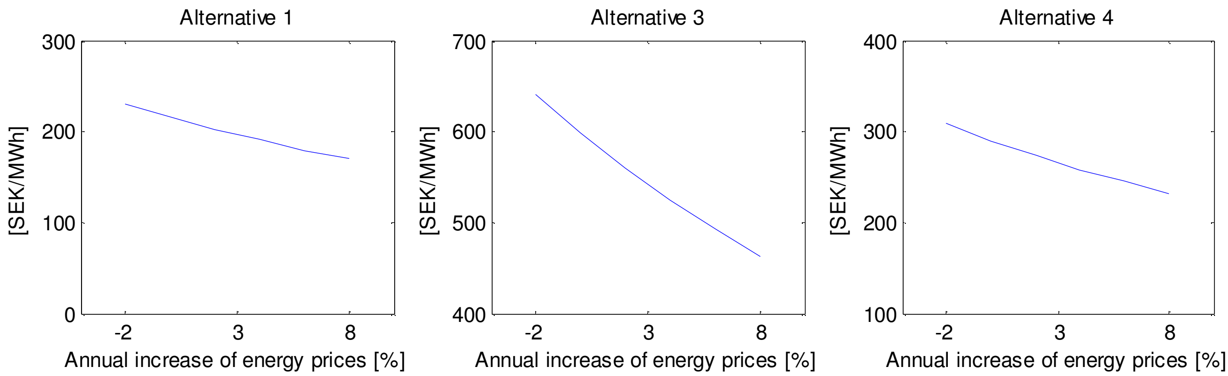


Figure 30 The initial price of district cooling as a function of annual increase of energy prices

Price of district cooling substation

Investigating the price of district cooling substation includes any additional costs that might come around when buying a substation and connecting it to the district cooling network.

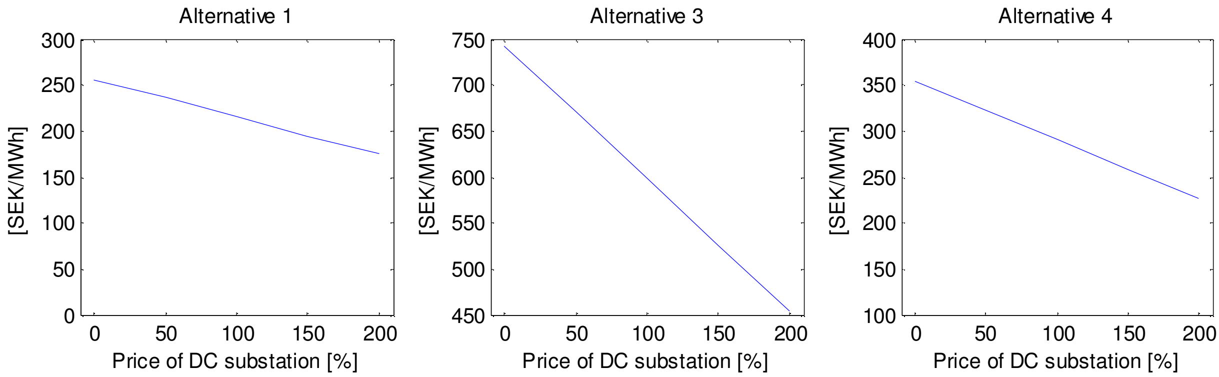


Figure 31 The price of district cooling as a function of the cost of the district cooling substation (compared to the original cost)

Internal generation of heat

The net amount of internally generated heat in the supermarket can differ a lot from one supermarket to another. This parameter covers assumptions made about lighting, customer flow, plug-in cabinets, ovens etc. Figure 32 presents how sensitive the amount and price of district cooling are to changes in the amount of internally generated heat. Practical problems, such as too low supply air temperature when having a vast amount of internally generated heat, are not taken into account.

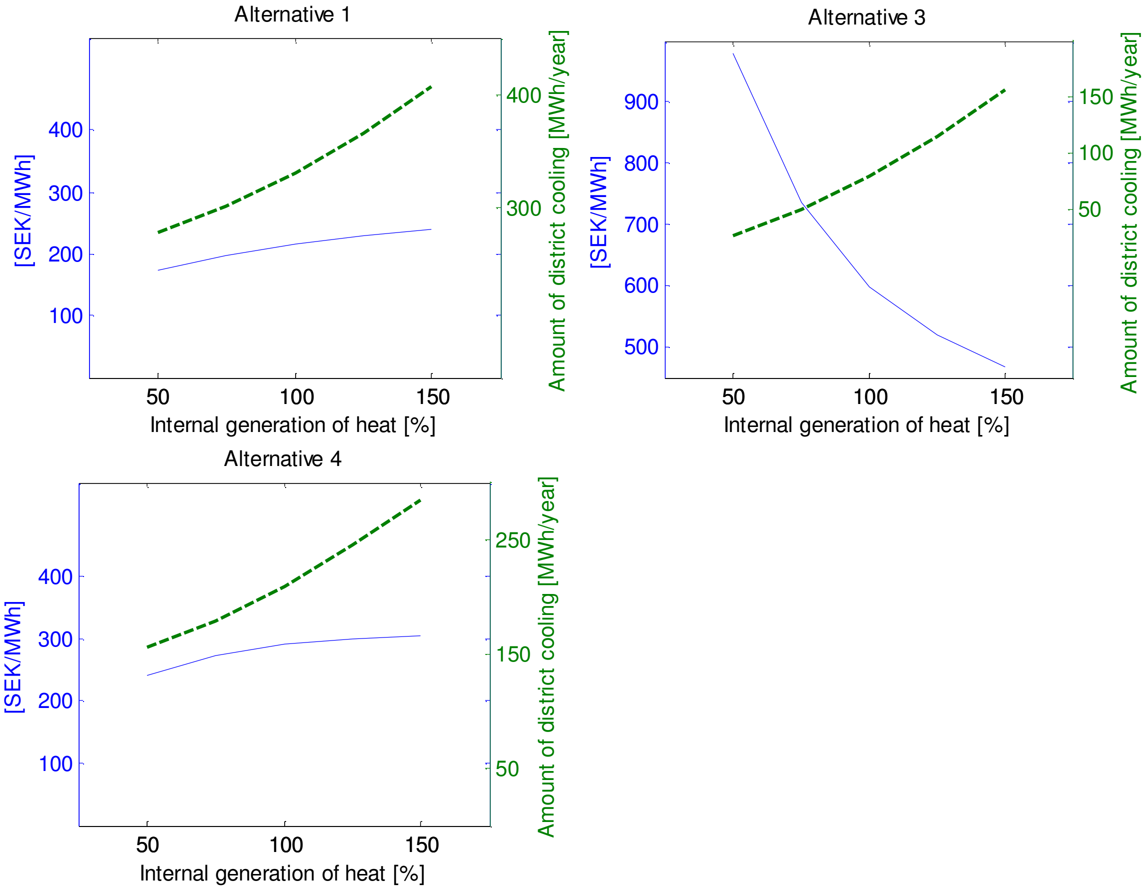


Figure 32 The price and amount of district cooling as a function of the net amount of internally generated heat in the supermarket (compared to the original case)

Outdoor temperature

Analyzing the influence of the outdoor temperature gives an indication of how the situation differs in other geographical locations. The outdoor relative humidity is kept unchanged. A higher outdoor temperature increases the cooling load in the display cabinets since it causes a higher indoor relative humidity. The influence of the outdoor temperature is presented in Figure 33.

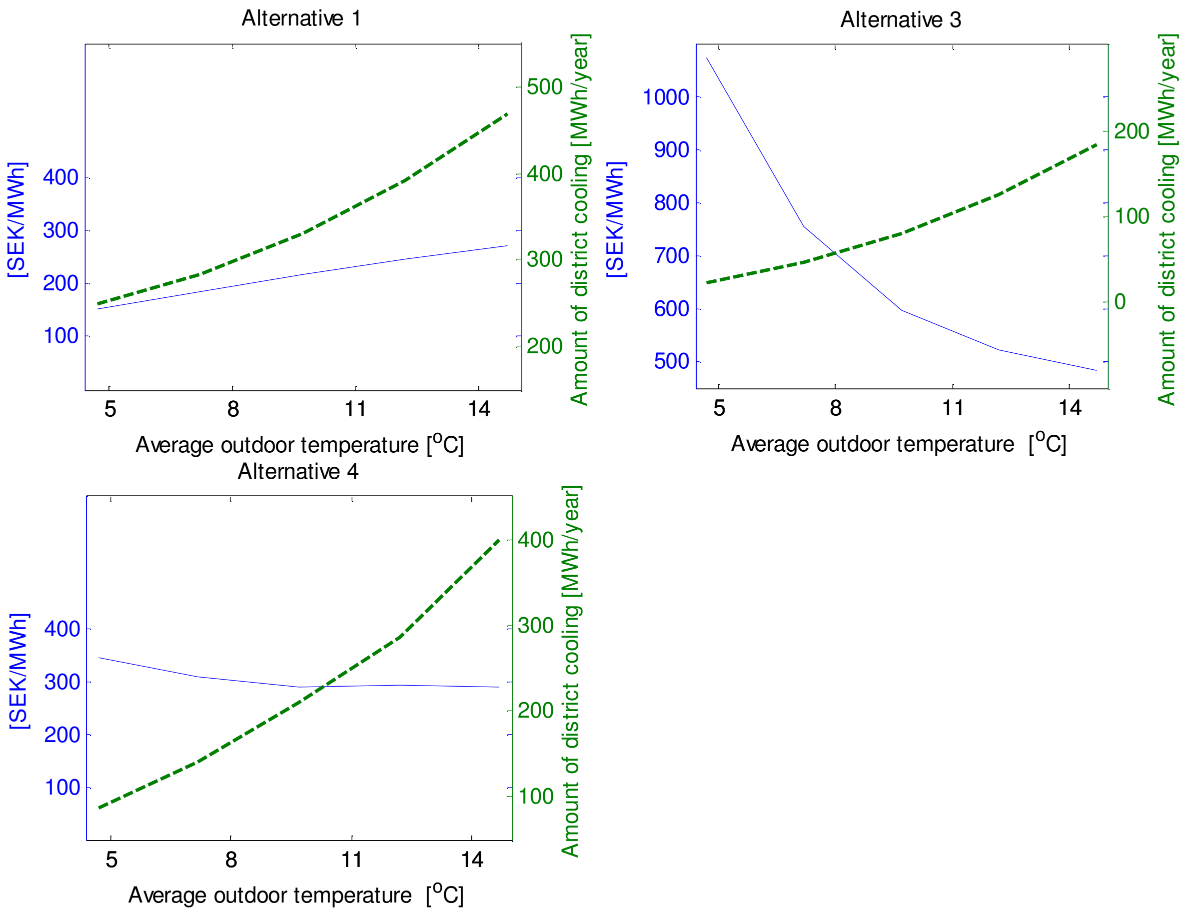


Figure 33 The price and amount of district cooling as a function of the avarage outdoor air temperature

Amount of food cooling

An increased amount of food cooling decreases the demand for comfort cooling and increases the demand for condenser cooling. This is presented in Figure 34.

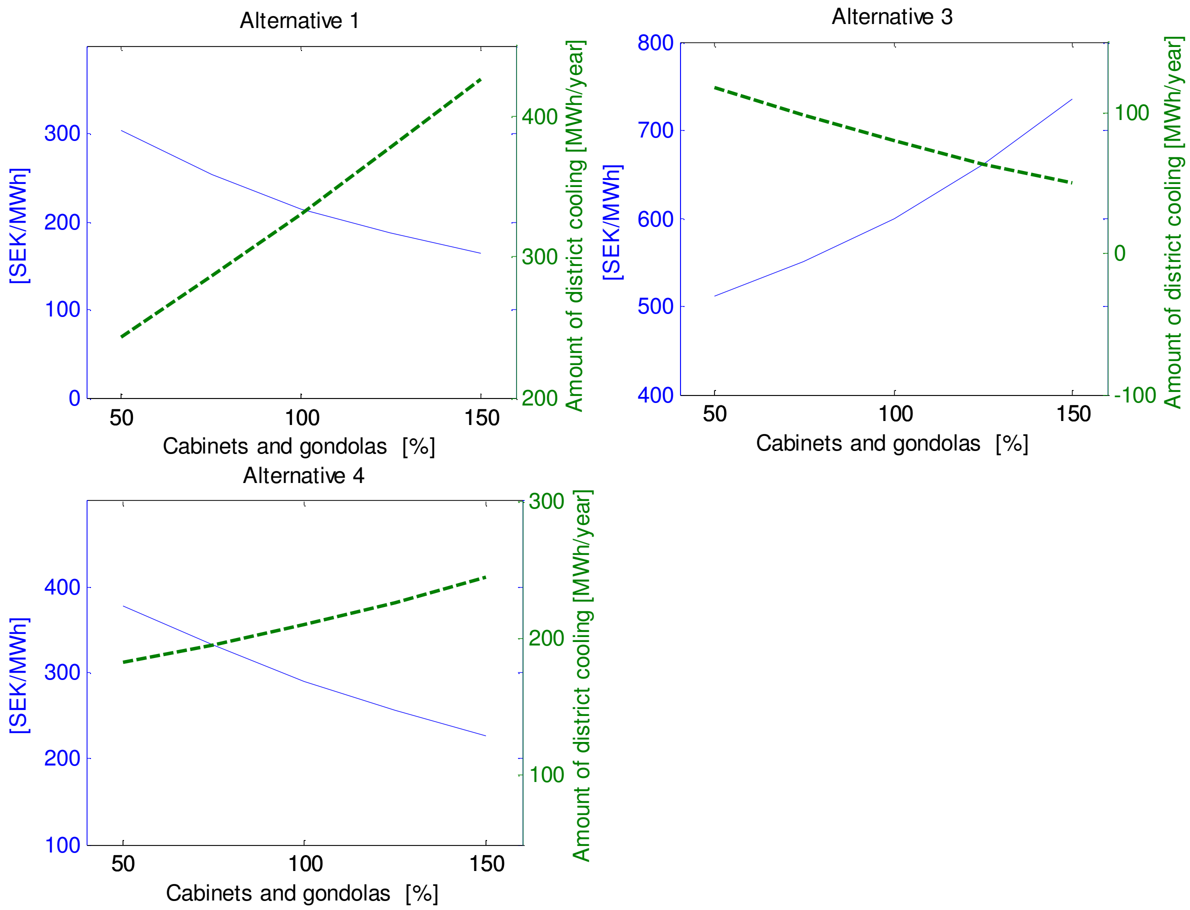


Figure 34 The price and amount of district cooling as a function of the amount of chilled-food cabinets and frozen-food gondolas (Compared to the original case)

6 Discussion

It is difficult to define "a typical supermarket" since there are many parameters involved, of which many can vary in a very wide range. The term "the supermarket of 2030" used in this study may be misleading. It is fully possible to design a supermarket like that today and consequently decrease the energy demand and increase the competitiveness of district cooling.

It is important to keep in mind that the results are valid only under the conditions used in this study. One important simplification made in this study is the constant indoor temperature. In reality, most HVAC systems allow a higher indoor temperature during summer and vice versa, which decreases the demand for comfort cooling and heating. A hint of the error from this simplification is obtained by adjusting the line of indoor temperature in Figure 11 and Figure 12. Another simplification is the disability to vary the ventilation flow rate according to the demand of heating and cooling. If there is a comfort cooling demand when it is cold outside, this could in reality be solved with an increased ventilation flow rate. However, studies have shown that the SFP must be very low in order to make this approach beneficial (Fahlén et al.[12]).

It is reasonable to assume that display cabinets equipped with doors are not as sensitive to changes in indoor humidity as cabinets without doors. In reality, the annual load curve would be flatter with cabinets equipped with doors, which would make the use of dry coolers slightly more beneficial. However, doors are not the only measure to make energy efficient display cabinets and it is hard to predict how sensitive other solutions are to fluctuations of indoor humidity.

It is assumed that the comfort cooling is achieved by cooling the ventilation air before it is supplied into the supermarket. In a system where the indoor air is cooled by chilled beams in the supermarket the results may be different.

The specific electricity demands [kWh_{el}/m^2] of the reference supermarkets are significantly surpassing the average supermarket studied in by The Swedish Energy Agency 2010a[24]. The explanation is that the reference supermarkets are much smaller than the average supermarket, and small supermarkets have higher specific electricity demands than big supermarkets (The Swedish Energy Agency 2010a[24])

The data of outdoor temperature implies an average temperature of 9.7 °C. This is 1.9 °C above the normal average temperature in Göteborg (defined as the average during 1990 - 1961). But only one of the last twenty years had an average temperature below the normal, and the annual average temperature in Göteborg is predicted to rise further according to The Swedish Meteorological and Hydrological Institute.

Costs and electricity consumptions of all pumps in the refrigeration systems are assumed to be equal in the four alternatives and therefore not included in the comparison. A big temperature change in a heat exchanger requires less flow rate (at constant power), and the required pump work is proportional to the cube of the flow rate. The temperature rise of the condenser coolant in the condenser is 6.0 °C without district cooling and 4.2 °C - 10 °C with district cooling (the average is 8.6 °C). This is valid for the supermarket of 2030 and the variation depends on the comfort cooling demand. Consequently it is reasonable to assume that the alternatives with district cooling cooled condensers require larger pumps, but the annual electricity demand of those pumps is less.

Intuitively, alternative 4 seems very expensive since investments are made in both district cooling and dry coolers. The results from this study show quite the opposite. Alternative 4 has a lower investment cost than both alternative 2 and alternative 3. A big part of the explanation is the fact that the chillers do not need to be dimensioned for a condensation temperature of 40 °C.

This study does not include the possibility to sell recovered heat to nearby buildings. This possibility is studied by Tabrizi (2009)[23], and would be of disadvantage to the competitiveness of district cooling.

Regarding the sensitivity analysis, alternative 3 is more sensitive than the other alternatives (in absolute terms). This is true for all parameters. The price of district cooling increases exponentially in alternative 3 when decreasing the amount of internally generated heat or the average outdoor temperature. The reason for this is that the amount of district cooling decreases, and since alternative 3 is a cheaper investment than alternative 2, the savings in investments are allocated on a decreased amount of energy.

District cooling is not competitive in the supermarket of 2010. The results indicate that it is possible to charge a high price for the district cooling if it is used for comfort cooling, but the demand is only 7 MWh/year. In reality, this demand is probably too small to be met at all. In other words, it is not fair to compare the district cooling alternative with a scenario where other comfort cooling equipment is invested in.

The results of the supermarket of 2030 are more interesting. Using district cooling for both comfort cooling and condenser cooling implies a demand of 331 MWh annually and a competitive price of 216 SEK/MWh. Using district cooling only when the outdoor temperature exceeds 10 °C implies a demand of 210 MWh/year and a competitive price of 290 SEK/MWh and using district cooling only for comfort cooling implies a demand of 80 MWh/year and a competitive price of 599 SEK/MWh. In Figure 35, the price of district cooling is plotted against the number of full-load hours during one year (full-load hours are defined as the load factor times the number of hours).

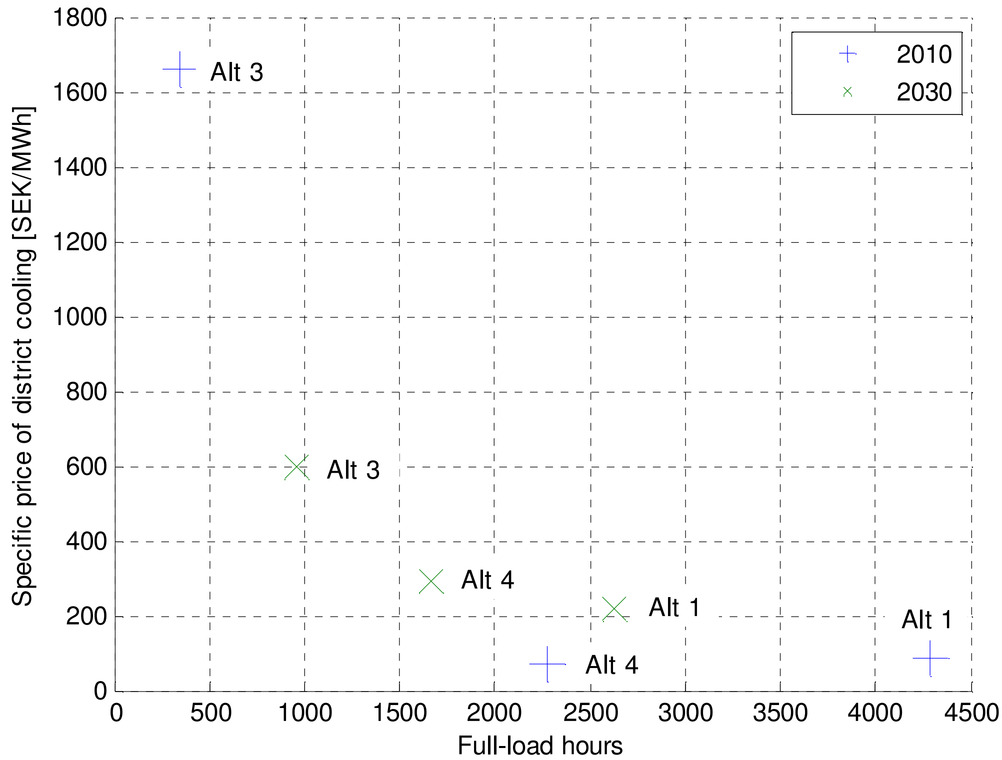


Figure 35 Specific price of district cooling plotted against number of full-load hours per year

The relation between price and number of full-load hours obtained in this study corresponds very well to the relation based on concluded district cooling agreements presented in Folkesson (2009)[13]. The prices of district cooling in the supermarket of 2030 are among the lower prices presented in Folkesson (2009)[13], and it shall be noted that the prices presented in that study does not include any access fees.

Costs of operation and maintenance are not included in this study. It is common to add 2-3 % to the investment cost of a chiller to take this into account, which would be of advantage to the competitiveness of district cooling. In addition to the prices calculated in this study, it is common to value other advantages of district cooling. This includes reliability, simplicity and reduction of noise from dry coolers which also take up a great deal of space. Estimating the value of these factors is beyond the scope of this study.

7 Conclusions

All conclusions drawn in this section are valid only under the conditions used in this study.

Cooling the condensers of a refrigeration system in a supermarket with district cooling does not lead to a lower average condensation temperature compared to a conventional system operating with a floating condensation temperature. District cooling cooled condensers also make it impossible to recover condenser heat. On the other hand, there are three major advantages associated with utilization of district cooling in supermarkets:

- Avoid usage of electricity consuming dry coolers.
- Not need to dimension the refrigeration system according to the maximum outdoor temperature.
- Meet the demand of comfort cooling in an energy efficient way.

The main difference between today's reference supermarket and that of the future is the net amount of internally generated heat. This big increase, which is mainly due to more energy efficient display cabinets, is shown to wipe out the demand for heating and to increase the annual demand for comfort cooling from 10 kWh/m^2 to 120 kWh/m^2 in the reference supermarket. In the meantime, the demand for food cooling decreases from 742 kWh/m^2 to 309 kWh/m^2.

In today's reference supermarket, the district cooling must be very cheap in order to compete with conventional system designs (87 SEK/MWh).

The competitiveness of district cooling is much better in the future reference supermarket. The main reason is the increased comfort cooling demand and the fact that a condenser heat recovery system is useless. Using district cooling in the future reference supermarket for both comfort cooling and condenser cooling implies an annual demand of 331 MWh, a competitive price of 216 SEK/MWh and 2628 full-load hours. Using district cooling only when the outdoor temperature exceeds 10 °C implies a demand of 210 MWh/year, a competitive price of 290 SEK/MWh and 1664 full-load hours. Using it only for comfort cooling implies a demand of 80 MWh/year, a competitive price of 599 SEK/MWh and 964 full-load hours. The competitive price is defined as the price that equalizes the net present value of a system including district cooling with the net present value of a conventional system without district cooling. Neither simplicity, reliability, noise from bulky dry coolers nor costs for operation and maintenance are taken into account.

The future reference supermarket is fully possible to design today and thereby increasing both the energy efficiency of the supermarket and the competitiveness of district cooling. Which alternative to prefer from a district cooling company point of view, is very much a matter of the benefit from having a large amount of full-load hours.

8 References

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Appendix A – Humid Air

Data of outdoor temperature and relative humidity are used as input in this study. Equation A1 and A2 are used to calculate the saturation pressure of water at a certain outdoor temperature, t, when the temperature is above and below 0 °C respectively. Equations A1-A4 are from Claesson, J. (2006, p. 75)[10].

$$p''_{H2O} = e^{12{,}03-\frac{4025}{t+235}} \qquad \text{(eq. A1)}$$

$$p''_{H2O} = e^{17{,}391-\frac{6142{,}83}{t+273{,}15}} \qquad \text{(eq. A2)}$$

The outdoor absolute humidity is then calculated as

$$x_{out} = 0{,}622 \cdot \frac{\varphi \cdot p''_{H2O}}{p - \varphi \cdot p''_{H2O}} \qquad \text{(eq. A3)}$$

p is the total pressure expressed in bar and φ is the outdoor relative humidity. The indoor absolute humidity is assumed to equal the outdoor absolute humidity in the winter and to be 2 $g_{water}/kg_{dry\ air}$ less humid than the outdoor air in the summer. The difference is assumed to increase linearly between summer and winter. This relation is from figure 7.2 in Axell et al. (2004, p. 50)[6]. By substituting the outdoor temperature to the indoor temperature in eq. A1, the saturation pressure of water at the indoor temperature is obtained. Finally, by using eq. A4, the indoor relative humidity is obtained.

$$\varphi_{in} = \frac{x_{in} \cdot p}{p''_{H2O} \cdot (x_{in} + 0{,}622)} \qquad \text{(eq. A4)}$$

Appendix B – Latent Cooling Load

When cooling air, condensation will occur if the temperature of the cooling coil is lower than the dewpoint of the air (Nilsson 2003)[20]. The condensation causes a latent load, which doesn't change the temperature of the air. The total cooling load consists of a sensible part and a latent part according to equation B.1.

$$\dot{Q} = \dot{Q}_s + \dot{Q}_L \qquad \text{(eq. B.1)}$$

$$\dot{Q}_s = \rho \cdot \dot{V}_{vent} \cdot c_{p,a} \cdot (t_1 - t_2) \qquad \text{(eq. B.2)}$$

$$\dot{Q}_L = \rho \cdot \dot{V}_{vent} \cdot (r_a \cdot \Delta x_a + c_{p,v} \cdot (t_1 - t_2) \cdot \Delta x_a) \qquad \text{(eq. B.3)}$$

$\dot{Q}$ = Total cooling load
$\dot{Q}_s$ = Sensible cooling load
$\dot{Q}_L$ = Latent cooling load
ρ = Air density
$\dot{V}_{vent}$ =Air volume flow
$c_{p,a}$ = Specific heat capacity of air
t_1 = Inlet air temperature (t_{rec})
t_2 = Outlet air temperature (t_s)
r_a = vaporization heat of air
Δx_a = change of airflow humidity ratio ($\mathrm{x}_1 - \mathrm{x}_2$)
$c_{p,v}$ = specific heat capacity of water vapour

The change of airflow humidity ratio, Δx_a, is calculated according to equation B.4.

$$\Delta \mathrm{x_a} = \frac{\mathrm{x_1} - \mathrm{x_{sat}}}{\mathrm{t_1} - \mathrm{t_{coil}}} \cdot (\mathrm{t_1} - \mathrm{t_2}) \qquad \text{(eq. B.4)}$$

$\mathrm{x_1}$ = Inlet humidity ratio
$\mathrm{x_{sat}}$ = Saturated humidity ratio at coil temperature
$\mathrm{t_{coil}}$ = Coil temperature

This is efficiently explained in a Mollier-diagram. In figure B1, the state of the air in Göteborg in 2008 is plotted (the blue circles). The red dot represents the cooling coil. The black arrow represents a cooling process from state 1 to state 2. The air is cooled from (x_1,t_1) to (x_2,t_2) in the direction towards (x_{sat},t_{coil}).

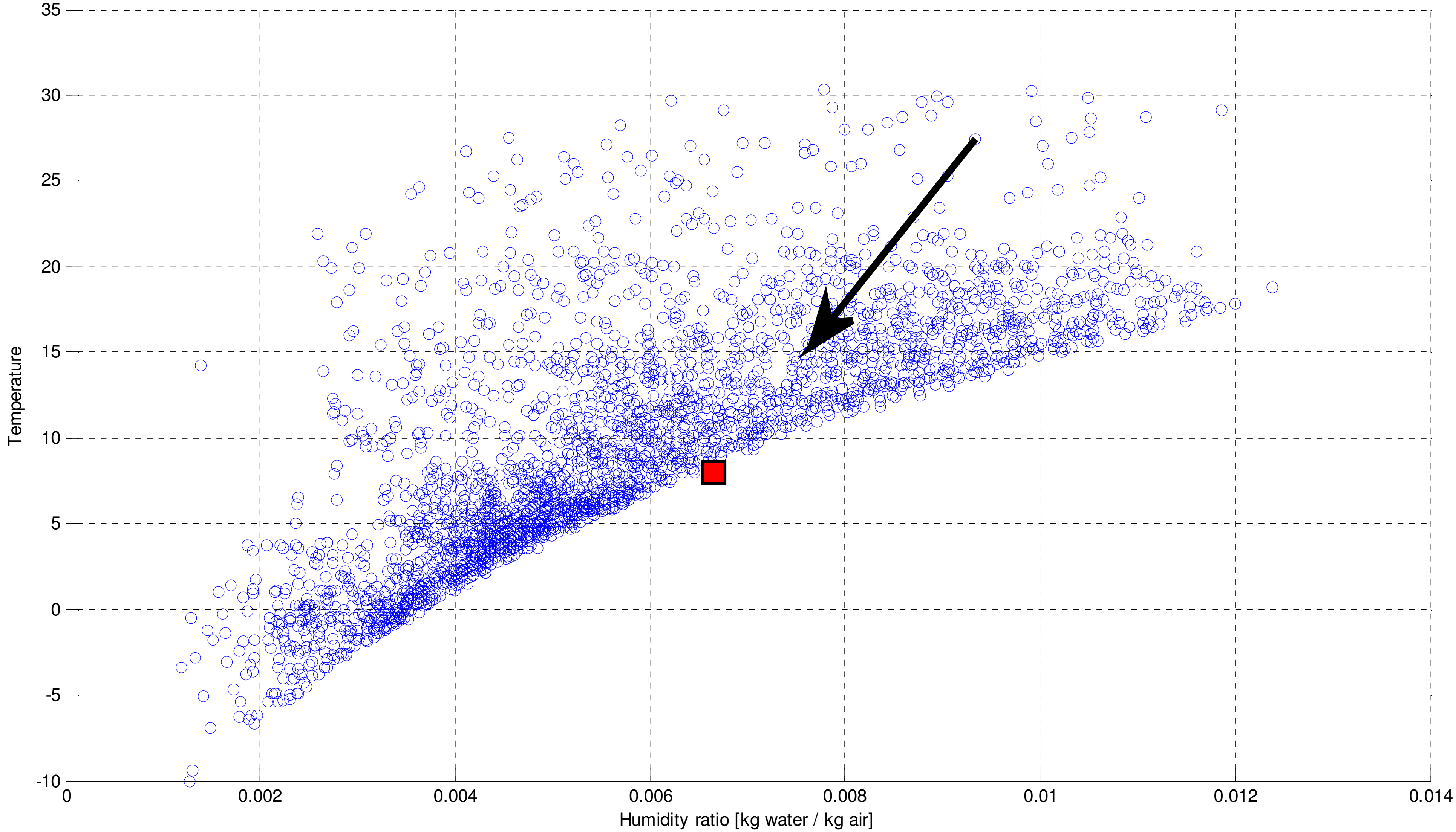

Figure B1 Temperature and humidity in Göteborg during 2008