Energy Usage in Supermarkets

- Modelling and Field Measurements

Doctoral Thesis

by

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Division of Applied Thermodynamics and Refrigeration

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Royal Institute of Technology

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ABSTRACT

This thesis investigates a special type of energy system, namely energy use in supermarkets through modelling, simulations and field studies. A user-friendly computer program, CyberMart, which calculates the total energy performance of a supermarket, is presented. The modelling method described in this thesis has four phases: the first phase is the development of a conceptual model that includes its objectives, the environment and the components of the system, and their interconnections. The second phase is a quantitative model in which the ideas from the conceptual model are transformed into mathematical and physical relationships. The third phase is an evaluation of the model with a sensitivity analysis of its predictions and comparisons between the computer model and results from field measurements. The fourth phase is the model application in which the computer model answers questions identified in the beginning of the modelling process as well as other questions arising throughout the work.

Field measurements in seven different supermarkets in Sweden were carried out to: (i) investigate the most important parameters that influence energy performance in supermarkets, (ii) analyse the operation of new system designs with indirect system implementation in Sweden during recent years, and (iii) validate the computer model.

A thorough sensitivity analysis shows a total sensitivity of 5.6 %, which is a satisfactory result given a 10% change in the majority of input parameters and assumptions, with the exception of outdoor temperatures and solar radiation that were calculated as extreme values in METEONORM. Comparisons between measurements and simulations in five supermarkets also show a good agreement. Measurements and simulation results for a whole year were not possible due to lack of data.

CyberMart opens up perspectives for designers and engineers in the field by providing innovative opportunities for assessment and testing of new energy efficient measures but also for evaluation of different already-installed system designs and components. The implementation of new energy-saving technologies in supermarkets requires an extensive integrated analysis of the energy performances of the refrigeration system, HVAC system, lighting, equipment, and the total energy usage. This analysis should be done over a long period, to evaluate and compare the real energy performance with the theoretical values calculated by CyberMart.

Keywords: Supermarket, Energy Performance, System Analysis, Modelling, Simulation, Field Measurements, Refrigeration Systems, Indirect Systems, Heat Recovery, Floating Condensing, TEWI, LCC.
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1 Introduction

Information and communication technology as well as sustainability concerns are some of the driving forces of development in the world today. In a new era of rapid change, information and communication technology has influenced different sectors of our society. Personal computers, software, computer networks, internet and mobile phones are basic tools in both business and private life. Internet and mobile phones have transformed communication and changed many aspects of our lives. In an increasingly interconnected world, the expansion and development of many industries depends on the efficiency of computers to access data and make decisions. The energy sector has also been affected by information and communication technology. Generation, distribution and utilization of energy have been influenced by personal computers, computer monitoring, control devices, communication equipment, internet, energy models, intelligent buildings, etc. These tools have become important instruments to assist researchers, designers, engineers, technicians and consumers in decision-making, maintenance and validation of new and old systems.

Sustainable development has also influenced the different sectors of society. Sustainable development is a concept where environmental management, social equity and economic growth interact both for the benefit of present and future generations. Energy influences these three dimensions of sustainable development (International Energy Agency 2001). Energy is fundamental to increase productivity and industrial activities, to improve the standard of living of people in the world, especially in developing countries, and to reduce the emission of greenhouse gases. Nevertheless, energy is associated in many aspects with unsustainable development. Global warming and ozone layer depletion are two issues linked with energy production and energy utilization. Johansson and Goldemerg (Johansson 2002) point out the challenge of developing energy-related policies for sustainable development:
It is imperative to find ways to greatly expand energy services, especially to the two billion people who currently rely on traditional forms of energy, as well as for generations to come; this expansion must be achieved in ways that are environmentally sound, as well as safe, affordable, convenient, reliable and equitable.

Stricter legislation to reduce global warming and ozone depletion as well as growing environmental awareness have brought about a revolution in energy generation, distribution and utilization. New more efficient systems and components have been developed to reduce energy consumption and emission of pollutants and greenhouse gases to the environment.

In Sweden, several policies and measures have been implemented to reduce emissions of greenhouse gases from the energy sector, improve efficiency in energy use, promote sustainable development, stimulate the use of renewable energy and increase international cooperation. These measures include carbon dioxide emission and energy taxation, new programmes to improve the efficiency of energy use, encourage the use of renewable forms of energy, sustainable development and reduction in the amount of heating provided by electricity (Swedish Energy Agency 2002).

Stricter environmental legislation to phase out CFC and HCFC refrigerants has also been implemented in Sweden. The Swedish parliament banned new installation with CFC refrigerants as of January 1995 and prohibited the use of these refrigerants from January 2000. For HCFC refrigerants, new installation was banned from 1998. The phase out of CFC and HCFC refrigerants has led to new refrigeration system designs with lower refrigerant charge and the introduction of new refrigerants.

The supermarket sector has been affected by the replacement of CFC and HCFC refrigerants and by a major reassessment of the use of energy. New system solutions with indirect systems have been introduced in supermarkets in Sweden to minimize the charge for refrigerant and refrigerant leakage. Supermarkets in Sweden use approximately 3% of the total electricity consumed in the country, which is equivalent to 1.8 TWh/year. The potential for increasing energy efficiency in stores is large.
Since the energy systems of a supermarket are relatively complex and new ideas and concepts have been introduced in supermarkets to decrease energy usage and minimize refrigerant charge, a computer model may assist designers and engineers with decisions regarding and validation of energy measures that influence energy use in supermarkets. The main objective of the computer model is to evaluate energy efficient measures in supermarkets with a focus on energy usage, environmental impact (TEWI) and economy (Life Cycle Cost) of refrigeration systems.

1.1 Background

In 1998 the Department of Energy Technology of the Royal Institute of Technology of Sweden started a project in co-operation with the two major supermarket chains in Sweden, COOP Sweden and ICA AB - different companies engaged in refrigeration technology and the Swedish Energy Agency. Modelling and field measurements of supermarket energy systems have been undertaken in the research project known as “The Energy Efficient Supermarket” for more than six years. The results have been part of the Swedish contribution to the IEA Annex 26 (Advanced Supermarket Refrigeration/Heat Recovery Systems) under the IEA Implementing Agreement on Heat Pumping Technologies (Baxter 2003).

The overall aim of the project “The Energy Efficient Supermarket” was to develop a simulation model where indoor climate, HVAC system, display cases, cooling and freezing rooms and the refrigeration system of a supermarket can be simulated for one year.

1.2 Purpose

The purpose of this thesis is to investigate and model the energy performance in supermarkets. A user-friendly computer program that can calculate the total energy consumption of a supermarket with reasonable accuracy has been developed. The model describes the properties of the different components in the system when different energy measures are compared. The investments and operational costs are implemented through a Life Cycle Costing with focus on the refrigeration system. The environmental impact from the refrigeration system is characterized through the Total Equiva-
lent Warming Impact measure. Designers and engineers could use the model to make decisions regarding energy improvement.

Field measurements in seven different supermarkets in Sweden were carried out to investigate the most important parameters that influence energy performance in supermarkets, to analyse the operation of new system designs with indirect systems implemented in Sweden during recent years, and to validate the computer model developed.

1.3 Method

The main objective of this thesis is to investigate and model energy performance in supermarkets.

The modelling method applied has four different phases. The first phase is the development of a conceptual model that includes its objectives, the environment and components of the system and their interconnections. The second phase is a quantitative model in which the ideas from the conceptual model are transformed into mathematical and physical relationships. The third phase is an evaluation of the model with a sensitivity analysis of its predictions and with a comparison between the computer model and results from field measurements. The fourth phase is the model application in which the computer model answers the questions identified in the beginning of modelling process.
1.4 Publications

1.4.1 Papers to Conferences and Journals


1.4.2 Report

1.4.3 Other Publications


1.5 Disposition of the Thesis

This thesis has been divided into nine chapters:

- Chapter 1 describes the background, purpose and methods of the thesis, as well as papers and reports presented in conferences, journals, and seminars;

- Chapter 2 gives an introduction to energy usage and environmental impact in supermarkets. An historical summary about supermarkets is presented in this chapter;

- Chapter 3 presents an overview of different refrigeration system designs used in supermarkets, such as the direct system, completely indirect system and partially indirect system;

- Chapter 4 describes results from field measurements in seven supermarkets in Sweden;
• Chapter 5 presents the different phases of the modelling process with a detailed explanation of the conceptual and quantitative models;

• Chapter 6 describes the different windows, input data, interface with users, calculation proceedings, and results from the model;

• Chapter 7 evaluates the model through sensitivity analyses and comparisons between the model and field measurements;

• Chapter 8 describes two applications of the model, which compare heat recovery versus floating condensing and direct versus indirect refrigeration systems;

• Chapter 9 presents conclusions, discussion and future studies.
2 Energy Usage and Environmental Impact in Supermarkets

2.1 Introduction

The history of supermarkets began in the middle of the 18th century, when small food stores opened in different countries. A single person, normally the owner, who oversaw the purchasing and selling of products, ran the stores. At the end of the 18th century, some companies started chains of grocery stores to sell firstly their products and later other kinds of everyday commodities. These stores offered a limited selection of dry goods. At the beginning of the last century, grocery stores carried a range of grocery items, such as dry goods, meat and dairy products. Customers made purchases every day in small quantities at the neighbourhood grocery store, principally milk and bread. Some grocery stores offered credit and home delivery of everyday commodities to their customers (Mitchman 1995).

Around 1930, supermarkets emerged as a new concept of food selling. Their main characteristics were low prices, self-service, impulse buying, big sale areas, high volumes, cash sale and free parking. The emergence of supermarkets was also due to recently developed technologies such as mass communication, refrigeration systems, cash registers and the automobile (Mitchman 1995). Their appearance also transformed the wholesale sector. Wholesalers started groups of independent retailers to develop their own resale products, and to take advantage of large-scale purchasing in order to compete with national chains. Wholesalers were also influenced by the development of road infrastructure and availability of lorries that reduced the cost of transportation and moved wholesale purchasing to industrial areas with easy access to the transport system.
In Sweden, the first cooperative company that purchased and distributed goods commenced around 1850 in Örsundsbro. The first self-service store was introduced in Sweden in 1940. After the Second World War, structural changes of society influenced by development in the United States affected the food distribution system. There were approximately 30,000 grocery stores in Sweden at the beginning of the 1950s. The structural reorganisation of the Swedish food industry reduced the number of stores to 12,000 in 1970, 9,000 in 1980, 8,300 in 1990, and approximately 6,600 stores in 1996. Structural changes in the Swedish food industry also affected wholesaling of merchandise. At the beginning of the 1950s, the three dominant wholesalers in Sweden had 231 distribution stores and an average turnover of five million Swedish crowns per store. In 1986, the same wholesalers had 48 distribution stores and an average turnover of more than 120 million Swedish crowns per store (Svensson 1998).

Development of the food industry has been influenced by some factors that can be divided into three categories: competition, consumer preferences and technology (Mowery 1999). Competition in the food industry has prompted creation of new strategies for survival in an over-saturated market. Some supermarket chains have focused on low prices, offering more store brands and less customer services. Other supermarket chains have extended the sale area with non-food merchandise lines and service departments such as banking, nurseries, florists and pharmacies (Mitchman 1998). Another driving force of supermarket development has been consumer preferences that have influenced marketing strategies and food organization. Some aspects such as ethnic food, more men buying in supermarkets, shorter time allotted for shopping and healthy eating trends have affected promotional strategies and the form of supermarkets. The third influential factor is technology with the incorporation of scanning registers, distribution of information, product flow and information recorded about customers.

The interior design of supermarkets has also changed during recent years. The layout of supermarkets has been designed to make shopping easier but also to increase “impulse shopping”. Shorter shopping time and a growing demand for ready-to-eat food have affected the interior design and circulation pattern of customers in supermarkets. The traditional design induced customers to buy ad-
ditional merchandise through long distances between the entrance and products and between products and check-out (see Figure 2-1). Yet this layout can actually reduce sales since customers may become discontent and decide to shop at convenience stores. Different user-friendlier interior designs have been implemented where products that customers purchase frequently are grouped together in one part of the sale area and frozen products are located close to check-out to avoid defrosting (Mowery 1999). The growing demand for easy-to-prepare foods has increased the sale of frozen food. In Sweden, the purchase of frozen food has increased from about 172,927 metric tons in 1975 to 438,114 metric tons in 2002.

Figure 2-1: Layout of a Traditional Supermarket

The introduction of online shopping in the late 1990s has also affected the supermarket sector. Customers’ growing interest in internet may increase the potential of the electronic supermarket industry. Online retail is also attractive from an economic point of view because e-trade does not require paying for checkout clerks, display cases or parking lots (Johnson 2000). In the U.S., online retail is growing at 35% a year in comparison with about 5% in tradi-
tional retail (Jones 2004). In Sweden, only a few percent of the total customers ordered goods through the internet in 2001. However, a survey by the company SIFO shows that about 20% of customers are interested in online shopping. Three of the four companies that offered online shopping have discontinued this service. The reasons are poor sales and extra costs for order administration, selection of products and transport (Supermarket 2002).

The appearance of superstores has also influenced the supermarket industry. Hypermarkets have incorporated low price products, wider range of food and non-food items and additional services such as banking, nurseries and restaurants. Superstores increase productivity and profitability of the selling space (Mitchman 1995). Hypermarkets have come to dominate the market during the last ten years.

In 1995, the 700 largest superstores in Sweden with a turnover higher than 50 million Swedish crowns per year (1 SEK is equivalent to 0.07 Euro) had about 46% of the total turnover from stores. In 2001, the number of superstores with a turnover higher than 50 million SEK per year increased to 830 stores with about 61% of the total turnover. On the other hand, the approximately 3500 stores with a turnover less than 10 million SEK per year had about 12% of the total turnover in 2001 (see Table 2-1).

<table>
<thead>
<tr>
<th>Year</th>
<th>Turnover: more than 50m SEK</th>
<th>Turnover: 20-50m SEK</th>
<th>Turnover: 10-20m SEK</th>
<th>Turnover: less than 10m SEK</th>
</tr>
</thead>
<tbody>
<tr>
<td>1995</td>
<td>46%</td>
<td>27%</td>
<td>11%</td>
<td>16%</td>
</tr>
<tr>
<td>1996</td>
<td>47%</td>
<td>27%</td>
<td>11%</td>
<td>15%</td>
</tr>
<tr>
<td>1997</td>
<td>49%</td>
<td>26%</td>
<td>10%</td>
<td>15%</td>
</tr>
<tr>
<td>1998</td>
<td>53%</td>
<td>24%</td>
<td>9%</td>
<td>14%</td>
</tr>
<tr>
<td>1999</td>
<td>56%</td>
<td>23%</td>
<td>8%</td>
<td>13%</td>
</tr>
<tr>
<td>2000</td>
<td>57%</td>
<td>23%</td>
<td>8%</td>
<td>12%</td>
</tr>
<tr>
<td>2001</td>
<td>61%</td>
<td>20%</td>
<td>7%</td>
<td>12%</td>
</tr>
<tr>
<td>2002</td>
<td>63%</td>
<td>19%</td>
<td>6%</td>
<td>12%</td>
</tr>
</tbody>
</table>
The number of supermarkets with a turnover less than 10 million SEK per year decreased by about 440 stores during the period 1997–2002 (Supermarket 2000; Supermarket 2002; Supermarket 2003).

Supermarket chains cover the market with different store formats. Superstores, supermarkets, low price stores, conveniences stores and grocery markets compete to attract customers from different local areas. Store formats and average surface of stores are significantly different even in neighbouring countries.

The definition of a superstore is a self-service store with a wider range of food and non-food items. In Sweden, the sale area of a superstore is greater than 2500 m² (Supermarket 2002), in Germany, the area of a superstore is greater than 4000 m² (Harnisch 2003) and in the U.S., the average surface area of hypermarkets is 11500 m² (UNEP, 2002). Typical surface areas of supermarkets and hypermarkets in different countries are presented in Table 2-2 (UNEP, 2002).

<table>
<thead>
<tr>
<th></th>
<th>Brazil</th>
<th>China</th>
<th>France</th>
<th>Japan</th>
<th>USA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Surface Area of Supermarkets (m²)</td>
<td>680</td>
<td>510</td>
<td>1500</td>
<td>1120</td>
<td>4000</td>
</tr>
<tr>
<td>Average Surface Area of Hypermarkets (m²)</td>
<td>3500</td>
<td>6800</td>
<td>6000</td>
<td>8250</td>
<td>11500</td>
</tr>
</tbody>
</table>

The number of supermarkets and hypermarkets are also different depending on the country (see Table 2-3). The economic growth of China in recent years has also influenced the supermarket sector. As an example, the number of small stores, with an average surface area of about 380 m², has increased six times during the last four years (UNEP, 2002).
In Sweden, the number of stores in 2003 was approximately 6100. Four supermarket chains dominated the market until 2003 when Lidl and Netto opened new stores in Sweden (Supermarket 2004). The Swedish grocery market in 2003 is presented in Table 2-4.

### Table 2-3: Number of Supermarkets and Hypermarkets (Source: UNEP, 2002)

<table>
<thead>
<tr>
<th>Region</th>
<th>Number of Supermarkets</th>
<th>Number of Hypermarkets</th>
</tr>
</thead>
<tbody>
<tr>
<td>EU</td>
<td>58134</td>
<td>5410</td>
</tr>
<tr>
<td>Other Europe</td>
<td>8954</td>
<td>492</td>
</tr>
<tr>
<td>USA</td>
<td>40203</td>
<td>2470</td>
</tr>
<tr>
<td>Other America</td>
<td>75441</td>
<td>7287</td>
</tr>
<tr>
<td>China</td>
<td>101200</td>
<td>100</td>
</tr>
<tr>
<td>Japan</td>
<td>14663</td>
<td>1603</td>
</tr>
<tr>
<td>Other Asia</td>
<td>18826</td>
<td>620</td>
</tr>
<tr>
<td>Africa, Oceania</td>
<td>4538</td>
<td>39</td>
</tr>
<tr>
<td>Total</td>
<td>321959</td>
<td>18021</td>
</tr>
</tbody>
</table>

### Table 2-4: Swedish Grocery Market 2003 (Source: Supermarket, 2004)

<table>
<thead>
<tr>
<th>Swedish Grocery Market 2003</th>
<th>Stores</th>
<th>Turnover (mil. SEK)</th>
<th>Percent Turnover</th>
</tr>
</thead>
<tbody>
<tr>
<td>ICA</td>
<td>1791</td>
<td>74913</td>
<td>45.0%</td>
</tr>
<tr>
<td>Kooperationen</td>
<td>879</td>
<td>36709</td>
<td>22.0%</td>
</tr>
<tr>
<td>Axfood</td>
<td>890</td>
<td>36556</td>
<td>21.9%</td>
</tr>
<tr>
<td>Bergendahl Group</td>
<td>129</td>
<td>5303</td>
<td>3.2%</td>
</tr>
<tr>
<td>Netto</td>
<td>28</td>
<td>700</td>
<td>0.4%</td>
</tr>
<tr>
<td>Lidl</td>
<td>28</td>
<td>265</td>
<td>0.2%</td>
</tr>
<tr>
<td>Other Grocery Stores</td>
<td>2310</td>
<td>12187</td>
<td>7.3%</td>
</tr>
<tr>
<td>Total</td>
<td>6060</td>
<td>166633</td>
<td>100%</td>
</tr>
</tbody>
</table>
The supermarket chain ICA has 45% of the total turnover in the Swedish grocery market. Kooperationen and Axfood both have about 22% of the turnover. The other grocery stores, i.e. traffic and services stores, have 7.3% of the turnover.

The majority of the new supermarkets constructed in Sweden up until 2002 were superstores with a sale area greater than 2000 m². This tendency changed in 2003 when Lidl and Netto appeared on the Swedish market (Supermarket 2004). During 2002, the number of new supermarkets constructed was 20 with an average sales area of 2400 m². In 2003, the number of new supermarkets constructed was 59 with an average sales area of 1600 m² (Lidl and Netto constructed 38 of those 59 new supermarkets). Lidl and Netto are low price stores with a limited sale area (the average sales area of Lidl stores is 1600 m² and 700 m² for Netto stores) (Supermarket 2004). Figure 2-2 illustrates the number of new supermarkets in Sweden and their sales areas.

![New Supermarkets in Sweden](image)

Figure 2-2: New Supermarkets in Sweden

An analysis of the overall cost structure of a typical supermarket in Sweden including the product costs and the profit (Lundqvist 2000) is shown in Figure 2-3. Product costs are 76%, 11% are salary costs, 3% are rent costs, 2% are marketing costs, 4% are other costs, 1% is energy cost and 3% is profit.
Figure 2-3: Cost Structure and Profit of a Typical Supermarket

Figure 2-4 presents the overall cost structure of a typical supermarket in Sweden excluding the product costs and the profit. Salary costs are 52%, 11% are rent costs, 10% are marketing costs, 19% are other costs, and 5% is energy cost.

Figure 2-4: Cost Structure of a Typical Supermarket excluding Product Costs
The cost of energy for this supermarket is only 1% of the total turnover. Since the profit is 3% of the turnover, a 50% reduction of energy consumption gives a 15% increase in profit.

2.2 Energy in Sweden

National and EU decisions determine, among other things, the conditions of the energy market in Sweden. The objective of the current energy policy in Sweden is to ensure a reliable supply of electricity and other forms of energy on a competitive market and to create the right conditions for efficient use and cost efficient supply of energy with minimal influence on climate, health and the environment. The Swedish energy balance, presented in Figure 2-5, shows that the total energy supplied in 2002 was about 616 TWh (including 5 TWh electricity import). The energy supplied from heat pumps to the energy system is the output heat. The total final energy use in 2002 was about 401 TWh spread over three user sectors: Industry, Transport, Residential and Services (see Figure 2-6). The total losses were about 215 TWh. The losses in nuclear power are 132 TWh. Electricity and bio-fuels are the most important energy carriers for the industrial sector, oil products are the most important for transport, and district heating are the most important for the residential and services sector (Swedish Energy Agency 2003).

![Figure 2-5: Total Energy Supply in Sweden 2002 (Source: Swedish Energy Agency, 2003)](image-url)
The electricity markets in the Nordic countries and the EU have been transformed from national monopolistic companies into international markets during the last few years.

![Total Energy Use 2002](image)

Sweden was the second Nordic country after Norway to set up a competitive electricity market in 1996. Today, all the Nordic countries except Iceland are integrated in the Nordic electricity market, known as Nord Pool. Germany, Poland and some Baltic states are also active on the Nordic electricity market. The price of electricity in the Nordic countries is determined by the availability of hydropower in Norway and Sweden, nuclear power in Finland and Sweden, and fuel prices.

Electricity and district heating are the most important energy carriers for the residential and services sector. Electricity production in Sweden in 2002 was about 143 TWh. Hydropower and nuclear power produce most of the electricity in Sweden. 46% of the total electricity production, which is about 66 TWh, was supplied by hydropower in 2002. Nuclear power production amounted to 65.6 TWh, which is also 46% of the total electricity production. In addition to nuclear and hydropower, Sweden also operates wind and combustion-based power production. Wind power production has expanded during the last ten years and at the end of 2002, electricity production from wind power was 0.56 TWh, less than 0.4% of
the total electricity production for that year. Combustion-based power production supplied in 2002 was 11 TWh, which is about 8% of the total electricity production (Swedish Energy Agency. 2003).

District heating is centralised production and distribution of hot water to multiple buildings. The heat can be supplied in different ways from boilers, heat pumps, waste heat from industry, geothermal or co-generation plants. In Sweden, district heating provided over 40% of the heating requirements for residential and commercial buildings. The total energy provided to the district heating sector was 55 TWh in 2002. Heat production from biofuels was 35 TWh, which is more than 60% of the total energy provided. Sweden has about 13000 km of distribution mains that supplied 49 TWh of heating in 2002. District cooling is a similar energy system to district heating with a centralised production and distribution of cold water. District cooling is produced in chillers, heat pumps, heat-driven absorption chillers and free cooling from the bottom of the sea or lakes. In Sweden, the length of distribution mains amounts to over 220 km, which supplied about 600 GWh of district cooling in 2002 (Swedish Energy Agency. 2003).

2.3 Energy Usage in Supermarkets

Supermarkets are intensive users of energy in all countries. Electricity consumption in large supermarkets in the US and France is estimated to be 4% of the national electricity use (Orphelin 1997). In the US, typical supermarkets with approximately 3700-5600 m² of sales area consume about 2-3 million kWh annually for total store energy use (Baxter 2003). The national average electricity intensity (the annual electricity use divided by the size of the facility) of a grocery store in the US is about 565 kWh/m² per year (Energy Star 2003). In Sweden, approximately 3% of the electricity consumed is used in supermarkets (1.8 TWh/year) (Sjöberg 1997). A survey made by one of the Swedish supermarket chains shows that the average energy consumption in 256 supermarkets is about 421 kWh/m² per year. The total energy consumption in a hypermarket (about 7000 m²) is about 326 kWh/m² per year while the total energy consumption in small neighbourhood shops (about 600 m²) is about 471 kWh/m² per year (Olsson 1998).
A typical supermarket in Sweden uses between 35-50% of its total electricity consumption for refrigeration equipment (Lundqvist 2000). Typical electricity usage of a grocery store in the US, presented in Figure 2-7, shows that 39% is used for refrigeration, 23% for lighting, 11% for cooling, 4% for ventilation, 13% for heating, 3% for miscellaneous, 2% for water heating and 5% for cooking (Energy Star 2003).
A breakdown of the energy usage from a medium-sized supermarket in Sweden (see Figure 2-8) shows that 47% of the energy is used for medium and low temperature refrigeration, 27% for illumination, 13% for fans and climate control, 3% for the kitchen, 5% for outdoor usage and 5% for other uses (Furberg 2000).

There is a great potential for improvement of energy systems in supermarkets. Typical efficiency improvements may involve refrigeration systems, illumination and HVAC system. Energy-saving technologies such as heat recovery, floating head condensing pressure, defrost control, energy efficient lighting, high efficiency motors, efficient control and energy efficient display cases have been implemented in several supermarkets to reduce energy consumption.

Utilization of heat rejected from condensers to heat service water or the premises in cold climates is a good measure to improve energy usage in supermarkets. Heat recovery leads to a reduction in costs and in the usage of fossil fuels for heating. A drawback with heat reclaiming is the higher condensing temperature that increases energy consumption from refrigeration systems.

The floating condensing system has been implemented in some supermarkets to reduce energy usage by refrigeration systems. The introduction of electronic expansion valves operating over a wide range of pressure drops allows for a low condensing temperature at low ambient temperatures. A reduction of condensing temperatures increases the coefficient of performance of refrigeration systems.

Illumination accounts for about 25% of total electricity used in supermarkets. Cost savings between 25-35% of the electricity consumed for lighting are possible by using the most energy efficient lamp and control system, maximising the use of daylight, painting the surfaces of the room with matt colours of high reflectance (Irish Energy Centre, 1995).

Display cases commonly carry large refrigeration loads, especially vertical open display cabinets. The reason is that this kind of cabinet displays a large amount of food on a small surface in the store with a large open front area (see Figure 2-9). The heat and moisture exchanged between the products in the cabinet and the store environment affect the refrigeration load, defrost and
environment affect the refrigeration load, defrost and condensation on walls and products. Infiltration causes about 60-70% of the cooling load for a typical open vertical display cabinet (Axell, 2002). Installing glass doors in display cases reduces the infiltration and energy consumption of cabinets. The reason for the absence of glass doors in display cases is to avoid placing an obstacle between the customer and the product, which may hinder the customer’s impulse to purchase a new product. Results from a laboratory test that evaluated glass doors on an open five-deck display case show a reduction of the total cooling load of the case by 68% (Faramarzi, 2002).

![Open Vertical Display Cabinet]

High relative humidity increases the frost accumulation in cabinets, which deteriorates the coil performance and reduces the refrigeration capacity of the coil (Datta 1998). To remove this frost, frequent defrost cycles are necessary, which increase energy use. High relative humidity produces more condensation on products and cabinet walls, which increase energy consumption by anti-sweat heaters. A low relative humidity in the supermarket decreases the operating costs of display cases but increases the costs of the air conditioning system (Howell 1993). Dehumidification technologies applied in supermarkets are desiccant cooling, heat pipe-assisted air conditioning systems and dual path systems (Henderson 1999).
Different actors influence the implementation of new energy-saving technologies (Lundqvist 2000). The question is how to address these issues and furthermore, how to identify the key players in the decision process. It is important to understand that the overall objectives of these actors differ considerably. In Figure 2-10, different actors that influence decisions concerning energy-efficient supermarkets are presented.

![Figure 2-10: Different Actors that Influence Decisions concerning Energy-Efficient Supermarkets](image)

The profile and structure of a supermarket chain influence decisions about investment and operation of new equipment in supermarkets. A green profile facilitates the introduction of energy efficiency technologies. A centrally-oriented structure, such as the supermarket chain COOP in Sweden (member of Euro Coop), may make decisions with rationales that are different from a more decentralized organization such as ICA Group.

The shop owner is responsible for the operation of and investment in the supermarket. The owner in supermarket chains with centrally-oriented structures is the owner of the supermarket company while in supermarkets with decentralized organization, the owner is a private company or person. The owner usually makes the principal decisions, choosing between many different potential investments. Since the cost of energy is a low percentage of the overall turnover (see Figure 2-3), a more attractive interior decoration of the shop or better marketing may promise shorter payback figures. Yet energy efficiency is a driving force for a green profile that may attract customers. Relatively short paybacks on simple energy effi-
ciency measures such as night lids/curtains, timers, cleaning of fan coils, etc. are attractive today.

Energy consultants designing supermarkets are making selections among various overall solutions and concepts for the supermarket owner. The consultants are relatively knowledgeable in various system solutions and techniques and the way these affect the energy efficiency of a supermarket, but they cannot handle the complexity on HVAC and refrigeration systems in a changing climate. The influence on the owner is thus considerable. Low life cycle cost is a weak argument in these discussions. Low first cost is often given high priority. Manufacturers of components or sub-systems and service companies are also important players. These companies develop new technologies. Energy-efficient display cases, low-charge central chillers, energy-efficient illumination systems, etc. are typical examples.

Suppliers of foodstuffs such as beverages, milk products, etc. are offering concepts such as the “milk market” - a special low-temperature zone within a supermarket. Customers’ awareness of energy efficiency issues and long-term climate problems are growing and may lead to competitive advantages for those supermarkets that manage to adopt, and present, an energy-efficient profile for the future.

Other key players are energy-labelling institutions such as Environmental Protection Agencies, governmental research councils, refrigerant manufacturers, etc. Education of employers and research support for new energy-efficient technologies are important actions that need to be undertaken.

It is clear that several important actors need to co-operate in order to attain more energy-efficient supermarkets in the future. Since the energy systems of a supermarket are relatively complex, improvements in one subsystem affect other systems. An example is the heat recovery system that influences the design and operation of refrigeration and HVAC systems. Here the cooperation between different actors or divisions in supermarket chains is important to implement and operate energy-saving technologies. Unfortunately, the overall objectives of these divisions or actors differ considerably. A systems approach is necessary where the whole supermarket is considered. The organization of supermarket chains
must take into consideration the supermarket as a system when different energy measures are discussed. In the case of heat recovery, the system approach leads to an increased energy usage from the refrigeration system and decreased energy consumption for the heating system. One sub-system becomes more efficient whereas another becomes less efficient. This measure also influences investment cost, installation costs, maintenance cost and environmental cost of both sub-systems. These costs probably influence the decision of the divisions affected by the implementation of heat recovery in the structure of the supermarket chain in different ways.

2.4 Refrigerant Emissions

The impact of refrigerant leakage on the environment has affected the design of refrigeration systems in supermarkets. Commercial refrigeration is the sector with the largest refrigerant emissions totalling about 185,000 metric tonnes in 2002, which is equivalent to 37% of the worldwide refrigerant emissions (Palandre 2004). Some reasons for the large emissions in supermarkets are the thousands of fittings in the refrigeration system, the large area where the refrigerant is moved and the high refrigerant charges (Gage 1998). The centralised system with a direct system had annual refrigerant emissions in the range of 20-35% during the 1980s (Bivens 2004). Emissions have decreased in the last 20 years due to international and national regulations on refrigerant usage, efficient refrigeration systems and personnel training.

Environmental legislation has, from an historic perspective, never been eased. In Europe, the emission rates have been influenced by the European regulation 2037 that banned both the use of CFC refrigerant and the manufacture of new equipment with HCFC refrigerant from 1 January 2001.

In the Bivens and Gage report, leakage rates for some countries in Europe and the US are presented (Bivens 2004). In the Netherlands, annual emissions from commercial refrigeration decreased to 3.2% of refrigerant charge in 1999. The reduction is due to national mandatory regulations for CFCs, HCFCs, and HFCs, assisted by a certification model known as STEK. The STEK organization cooperates with the government and trade and industry groups in the refrigeration and air conditioning sector. STEK is re-
sponsible for the certification of companies and qualification of personnel who work with refrigeration systems. In Germany, two articles report emission rates in the range of 5-10% from centralised systems with direct systems in supermarkets. In Denmark, strong initiatives to phase out HFC refrigerants completely have been undertaken by the Danish Environmental Protection Agency. The annual emissions from new commercial refrigeration systems have decreased from 20-25% of refrigerant charge to about 10% of charge. In Norway, supermarket chains show annual emissions of about 14% of the refrigerant charge from 220 supermarkets.

Emission rates from supermarkets in the US were not measured early on, but rates between 30% to 50% of the total refrigerant charge were values used by industry (Gage 1998). 349 stores reported annual emission rates between 8-15% in the time period 1993-1994. Two supermarket chains located in the western and eastern parts of the US reported annual refrigerant loss rates in the range of 18-22% of refrigerant charge for 2001 and 2002 (Bivens 2004).

In Sweden, regulation pertaining to CFC, HCFC and HFC refrigerants (presented in Table 2-5) banned new installations with CFC refrigerants from January 1995 and stopped the use of these refrigerants from January 2000. For HCFC refrigerants, new installations were banned from 1998.

Table 2-5: Regulation of CFC, HCFC and HFC Refrigerants in Sweden

<table>
<thead>
<tr>
<th>ASHRAE Number</th>
<th>Type of Refrigerant</th>
<th>Import or New Installation Banned</th>
<th>Refill Banned</th>
<th>Use Banned</th>
</tr>
</thead>
<tbody>
<tr>
<td>R12, R502</td>
<td>CFC</td>
<td>1-Jan-1995</td>
<td>1-Jan-1998</td>
<td>1-Jan-2000</td>
</tr>
<tr>
<td>R22</td>
<td>HCFC</td>
<td>1-Jan-1998</td>
<td>1-Jan-2002</td>
<td></td>
</tr>
<tr>
<td>R134a, R404A</td>
<td>HFC</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Refrigerant distribution in supermarkets in Sweden has varied considerably because of the phase-out of CFC and HCFC refrigerants.
This has been verified in a Master’s Thesis carried out by Engsten and Lindh, who studied refrigerant distribution and emission rates from a supermarket chain in Sweden (Engsten 2002).

In 1996 the dominant refrigerant in supermarkets was R22, making up 40% of refrigerants used in 488 stores (see Figure 2-11). CFC and HCFC refrigerants were 65% of total refrigerant used.

In 2003, refrigerant distribution was completely different. The dominant refrigerant in supermarkets is now R404A, making up 70% of refrigerants used in 371 stores (see Figure 2-12). CFC refrigerants have disappeared from the supermarkets in the study and R22 was only 4% of total refrigerant used. The other refrigerants used in the 371 stores were R407C, R417A and R507.

Emission rates from about 450 stores (average of 508 stores in 1996 and 408 in 2003) belonging to two supermarket chains in Sweden, COOP and ICA, are presented in Table 2-6. The amount of COOP supermarkets included in the report decreased from 488 in 1996 to 371 in 2003. The ICA supermarkets are from the greater Stockholm area. The amount of ICA stores in the study was 20 in 1996 and 32 in 2003.
The total refrigerant charge from 508 supermarkets in 1996 was about 65 metric tonnes. The total refrigerant charge from 403 stores in 2003 decreased to about 47 metric tonnes. The refrigerant charge per store decreased from about 128 kg in 1996 to 117 kg in 2003.

Table 2-6: Emission Rates from about 450 Stores from 1996-2003 in Sweden (Engsten 2002)

<table>
<thead>
<tr>
<th>Year</th>
<th>Stores COOP</th>
<th>Stores ICA</th>
<th>Total Stores</th>
<th>Total Charge [Kg]</th>
<th>Leakage [Kg]</th>
<th>Leakage [%]</th>
<th>Charge /Store [kg]</th>
<th>Leakage /Store [kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1996</td>
<td>488</td>
<td>20</td>
<td>508</td>
<td>65181</td>
<td>7934</td>
<td>12.2</td>
<td>128.3</td>
<td>15.6</td>
</tr>
<tr>
<td>1997</td>
<td>496</td>
<td>28</td>
<td>524</td>
<td>65589</td>
<td>9278</td>
<td>14.1</td>
<td>125.2</td>
<td>17.7</td>
</tr>
<tr>
<td>1998</td>
<td>465</td>
<td>31</td>
<td>496</td>
<td>60556</td>
<td>7986</td>
<td>13.2</td>
<td>122.1</td>
<td>16.1</td>
</tr>
<tr>
<td>1999</td>
<td>452</td>
<td>36</td>
<td>488</td>
<td>57477</td>
<td>7215</td>
<td>12.6</td>
<td>117.8</td>
<td>14.8</td>
</tr>
<tr>
<td>2000</td>
<td>451</td>
<td>37</td>
<td>488</td>
<td>61479</td>
<td>7674</td>
<td>12.5</td>
<td>126.0</td>
<td>15.7</td>
</tr>
<tr>
<td>2001</td>
<td>417</td>
<td>39</td>
<td>456</td>
<td>55545</td>
<td>4784</td>
<td>8.6</td>
<td>121.8</td>
<td>10.5</td>
</tr>
<tr>
<td>2002</td>
<td>389</td>
<td>38</td>
<td>427</td>
<td>50404</td>
<td>4325</td>
<td>8.6</td>
<td>118.0</td>
<td>10.1</td>
</tr>
<tr>
<td>2003</td>
<td>371</td>
<td>32</td>
<td>403</td>
<td>47210</td>
<td>5288</td>
<td>11.2</td>
<td>117.2</td>
<td>13.1</td>
</tr>
</tbody>
</table>
The emission rates, from the stores in the study, decreased from 14.1% of the total refrigerant charge in 1997 to 8.6% of total charge in 2001 (see Figure 2-13 and Table 2-6).

![Total Leakage (~ 450 Stores) 1996-2003](image)

Figure 2-13: Emission Rates from about 450 Stores from 1996-2003 (Engsten 2002)

The reason for the large decrease between 2000 and 2001 might be a new contract between COOP and the service companies. In the contract, the service companies are responsible for the refilled refrigerant and prevention of leakages. In 2003, emissions increased to 11.2% of the total charge. The reason for the increase is unclear (Engsten 2002).

The study also shows that a few stores have large leakage problems. About 50% of all COOP stores in the study have a leakage of 4% or less and 67% of COOP stores have a leakage of less than 10%. On the other hand, the five percent of the stores with the highest leakage have a leakage over 40% and 70-80% of the annual leakage from 1999-2003 was caused by 25% of the stores. Five selected stores with the highest leakage have, or have had during the major part of the studied period, a centralized direct system. Four of the five stores have one refrigeration system for the cabinets and storage in the medium and low temperature levels. When there is a leak, it is not unusual that the total charge of the system leaks out (Engsten 2002).
2.5 TEWI and LCCP

Global environmental impacts such as ozone layer depletion and global warming due to greenhouse gas emissions to the atmosphere have been associated with refrigeration systems during recent years. The influence on the ozone layer from CFC and HCFC refrigerants is well known and stronger legislation has been implemented to decrease the use of these refrigerants. Refrigerants CFC and HCFC are also recognised as greenhouse gases since these primarily substitute the refrigerant HFC. The concept of TEWI is a useful tool when studying the influence of a refrigeration system on global warming. The TEWI combines the direct emissions of CO$_2$ due to refrigerant leakage and refrigerant losses at the end of the system’s life and the indirect emissions of CO$_2$ associated with energy consumption and generation.

The TEWI calculation of a refrigeration system is based on the following relation:

$$\text{TEWI} = \left( M_{\text{losses}} \cdot N + M_{\text{ref}} \cdot (1 - \kappa) \right) \cdot \text{GWP}_{\text{ref}} + \text{RC} \cdot E \cdot N$$

(2.1)

Where $M_{\text{losses}}$ is the refrigerant leakage, $N$ is the lifetime of the refrigeration system, $M_{\text{ref}}$ is the refrigerant charge, $\kappa$ is the recycling factor, GWP$_{\text{ref}}$ is the Global Warming Potential of Refrigerant, RC is the Regional Conversion Factor, which is the emission of CO$_2$ per unit of energy delivered, and E is the annual energy consumption of the equipment.

The two first parts of equation (2.1) are the direct impact, which take into consideration the refrigerant leakage during the lifetime of the system and the refrigerant losses at the end of the system’s life. The second part of the equation is the indirect impact that takes into account the energy used during the lifetime of the refrigeration system and the CO$_2$ emissions from the production of electricity. The CO$_2$ emissions from electricity generation is calculated with a Regional Conversion Factor RC, which is the emission of CO$_2$ per unit of energy delivered in kg CO$_2$/kWh (Sand 1997).

The regional conversion factor varies from country to country due to the efficiency of power plants and regional fuel mix. The aver-
age of CO₂ emissions from a carbon power plant is about 1.11 [kg CO₂/kWh], from an oil power plant is about 0.77 [kg CO₂/kWh], from a gas power plant is about 0.55 [kg CO₂/kWh] and from a nuclear and hydroelectric power plant is 0.00 [kg CO₂/kWh]. The best value of the regional conversion factor for Sweden is about 0.04[kg CO₂/kWh], for Denmark it is about 0.84[kg CO₂/kWh], for Norway about 0.00[kg CO₂/kWh] and for Finland about 0.24[kg CO₂/kWh] (Sand 1997).

Life Cycle Climate Performance is a concept that takes into consideration the effect of production of the refrigerant in the system in addition to the direct impact of leakage and indirect impact of energy used.

Values of Global Warming Potential (GWP) (which is the contribution to global warming of the refrigerant as compared to an equivalent amount of CO₂) and values of emissions and energy used in the production of refrigerant are presented in Table 2-7 for different refrigerants (Spatz 2003).

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R-22</th>
<th>R-410A</th>
<th>R-290</th>
<th>R-404A</th>
</tr>
</thead>
<tbody>
<tr>
<td>GWP</td>
<td>1700</td>
<td>1975</td>
<td>3</td>
<td>3784</td>
</tr>
<tr>
<td>Manufact</td>
<td>390</td>
<td>14</td>
<td>2</td>
<td>18</td>
</tr>
<tr>
<td>Total</td>
<td>2090</td>
<td>1989</td>
<td>5</td>
<td>3802</td>
</tr>
</tbody>
</table>

2.6 Conclusions

Supermarkets are intensive users of energy in several countries. Electricity consumption in large supermarkets in the US and in France is estimated to be 4% of national electricity use. In Sweden, approximately 3% of total electricity consumption is used in supermarkets.

There is great potential for improvement of energy systems in supermarkets. Typical efficiency improvements may involve better refrigeration systems, illumination and HVAC system. Energy-saving technologies such as heat recovery, floating head condensing pressure, defrost control, energy-efficient lighting, high efficiency motors, efficient control and energy-efficient display cases
have been implemented in several supermarkets to reduce energy consumption.

Different, sometimes conflicting, interests influence the implementation of new energy-saving technologies. The cooperation between different actors or divisions in supermarket chains is of great importance in implementing and operating energy-saving technologies. Unfortunately, the overall objectives of these divisions or actors differ considerably. A system approach is necessary where the whole supermarket is considered. Supermarket chains must take into consideration the supermarket as a system when different energy measures are discussed.

Stricter environmental legislation to phase-out CFC and HCFC refrigerants and the impact of refrigerant leakage on the environment have affected the refrigeration system in supermarkets. In Sweden, refrigerant distribution in supermarkets has varied considerably following the phase-out of CFC and HCFC refrigerants. In a study carried out by Engsten and Lindh at the Department of Energy Technology at KTH, the CFC and HCFC refrigerants used in 488 stores in 1996 were about 65% of the total refrigerant installed. In 2003, CFC refrigerants disappeared and HCFC refrigerants were only 4% of the refrigerant used in 371 stores. On the other hand, the emission rates from 524 supermarkets in 1997 was about 14.1% of the total refrigerant charge, while in 2003 the emission rate in 403 stores was about 11.2% of the total refrigerant charge. The study also shows that a few stores have large leakage problems. About 50% of all COOP stores in the study have a leakage of 4% or less and 67% of COOP stores have a leakage of less than 10%. The five percent of stores with the highest leakage, however, have a leakage over 40% and 70-80% of the annual leakage from 1999-2003 was caused by 25% of the stores.
3 Refrigeration Systems in Supermarkets

The purpose of refrigeration systems in supermarkets is to provide storage of and display perishable food prior to sale. Food is stored in walk-in storages before the transfer to display cases in the sale area. There are two principal temperature levels in supermarkets: medium temperature for preservation of chilled food and low temperature for frozen products. Chilled food is maintained between 1°C and 14°C, while frozen food is kept at -12°C to -18°C, depending on the country. The evaporation temperature, for a medium temperature system, varies between -15°C and 5°C, and for a low temperature system, the evaporation temperatures are in the range of -30°C to -40°C. Variations in temperature are dependent upon products, display cases and the chosen refrigeration system (UNEP, 2002).

Three main types of refrigeration systems are used in stores: stand-alone equipment, condensing units and centralised systems (UNEP, 2002). Stand-alone or plug-in equipment is often a display case where the refrigeration system is integrated into the cabinet and the condenser heat is rejected to the sales area of the supermarket. The purpose of plug-in equipment is to display ice cream or cold beverages such as beer or soft drinks.

Condensing units are small-size refrigeration equipment with one or two compressors and a condenser installed on the roof or in a small machine room. Condensing units provide refrigeration to a small group of cabinets installed in convenience stores and small supermarkets.

Centralised systems consist of a central refrigeration unit located in a machine room. There are two types of centralised system: direct and indirect system. In a direct system (DX), racks of compressors in the machine room are connected to the evaporators in the dis-
play cases and to the condensers on the roof by long pipes with refrigerator. In an indirect system, the central refrigeration unit cools a fluid that circulates between the evaporator in the machine room and the display cases in the sales area.

The quest for increased energy efficiency and the phase-out of ozone depleting substances have affected refrigeration system design for supermarkets considerably. The traditional CFC and HCFC refrigerants are replaced today with R404A, R134a, etc. A renewed interest in natural refrigerants such as ammonia, propane and CO$_2$ has resulted in charge minimisation and relatively leak proof systems being placed in machine rooms. Supermarkets appear in all kinds of sizes from small neighbourhood shops to hypermarkets and the choice of overall system solutions vary considerably.

New system solutions with completely or partially indirect systems have been developed and introduced in recent years in supermarkets. This has been done in order to lower the refrigerant charge and, at the same time, minimize potential refrigerant leakage. For example, in a supermarket in the city of Lund in the south of Sweden, a refrigeration system with 500 kg CFC refrigerant was replaced with a new one that uses 36 kg ammonia as refrigerant and CO$_2$ as a secondary refrigerant (Arias 1999).

Water solutions of glycols, alcohols and chlorides have long been used as secondary refrigerants. The increasing interest in indirect systems has led to the development of some new secondary refrigerants based on potassium formate and potassium acetate alone or mixed, which have good heat transfer and pressure drop characteristics. Thermo-physical properties, material compatibility, environmental pollution and toxicity, flammability and handling security and finally cost are the aspects to take into consideration when determining which secondary refrigerant is to be used in a particular application such as in supermarkets (Melinder 2003).

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1 This fluid is known by different names, such as secondary refrigerant, secondary fluid, secondary coolant, heat transfer fluid, or brine. In this thesis, the fluid is referred to either as secondary refrigerant or as brine.
The temperature profile and most important components of the secondary refrigerant circuit are presented in Figure 3-1.

Other very promising developments are phase-changing secondary refrigerants such as CO$_2$ and ice slurries.

Technology with CO$_2$ as secondary refrigerant for low temperature has been implemented since 1995 in the Nordic countries. In Sweden, by the year 2000, 40 supermarkets were using CO$_2$ systems with capacities ranging from 10 to 280 kW (Sawalha 2003). CO$_2$ systems require much lower tube diameters and the pressure drop is negligible when compared to conventional systems. CO$_2$ is a very interesting heat transfer fluid for low temperature systems because of its transport properties and low viscosity at temperatures below $-20^\circ$C. The problem with CO$_2$ is the high pressure required (19.7 bar at $-20^\circ$C).

Ice slurry is another interesting secondary refrigerant. Ice slurry is a mixture of fluid and ice particles that is produced when a water so-
olution of an antifreeze substance, e.g. propylene glycol or ethanol, is cooled below its depressed freezing point (Brandon 2003). This system offers additional advantages with enhanced thermal capacity and thermal storage in the system. Ice slurry has been tested in display cases with promising results (Tassou 2001); (Field 2003). Tests have also been carried out in supermarkets (Ben Lakhdar 2001); (End 2001). Results from a supermarket with ice slurry in France show higher working temperature (–5°C to -4°C instead of -10°C to -8°C for the same display cabinet), better defrost and good distribution of ice slurry. The drawback of ice slurry systems is the high cost of ice generation and accumulation, for which the savings in distribution and accumulation do not compensate (Rivet 2001).

Secondary systems and the minimisation of refrigerant charge may lead to an unwanted increase in the overall energy used. Comparison between the energy consumption of a primary and secondary refrigeration system have been carried out in different studies. Theoretical calculations and practical experiences confirm the increase in energy consumption due to the extra temperature differences and pumps introduced into the system (Clodic 1998; Yunting 2001). Recent theoretical and experimental studies, however, indicate that indirect systems are more energy efficient than direct systems, if properly designed (Lindborg 2000); (Horton 2002); (You 2001); (Faramarzi 2004). More efficient defrosting systems, better part load characteristics and more reliable systems are believed to contribute to energy savings. An evaluation using a concept like Total Equivalent Warming Impact (Sand 1997) may be used to estimate the overall environmental impact of a system. Calculations made for the two extremes: Sweden (nuclear and hydro power) and Denmark (coal and gas) show that the impact from large refrigerant leakages always dominates over CO₂ releases related to the production of electricity (Arias 1999).
3.1 Direct System

The most traditional refrigeration system design in supermarkets is the direct system (Figure 3-2). In direct systems, the refrigerant circulates from the machine room, where the compressor is found, to the display cases in the sales area where it evaporates and absorbs heat. The system requires long pipes to connect the compressors to display cases and to the condensers on the roof. This implies very large refrigerant charges.

The most common direct system in supermarkets is the multiplex refrigeration system, which consists of a rack of compressors operating at the same saturated suction temperature with common suction and discharge refrigeration lines (Baxter, 2003). The amount of refrigerant in a centralised direct system is typically 4-5 kg/kW of refrigeration capacity (Baxter, 2003). Another direct refrigeration system used in supermarkets is the single-compressor condensing system, which provides refrigeration to a small set of display cases (Horton, 2002). The benefits and drawbacks of the direct system are summarized in a strengths, weaknesses, opportunities and threats (SWOT) analysis (see Table 3-1).
### Table 3-1: SWOT of Direct Systems

<table>
<thead>
<tr>
<th><strong>Strengths</strong></th>
<th><strong>Weaknesses</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>- Good efficiency</td>
<td>- Large refrigerant charges</td>
</tr>
<tr>
<td>- Less components than indirect systems</td>
<td>- Built on site: Leakage</td>
</tr>
<tr>
<td>- Lower investment cost than indirect system</td>
<td>- No possibility to use ammonia or HC refrigerants</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Opportunities</strong></th>
<th><strong>Threats</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>- New techniques to make tight systems</td>
<td>- Stiffer legislation for HFC refrigerants leading to charge limits and environmental taxes for leakage</td>
</tr>
<tr>
<td>- Mechanical subcooling&lt;sup&gt;1&lt;/sup&gt;</td>
<td></td>
</tr>
</tbody>
</table>

Another variation of the direct system is the distributed system with a separate rooftop condenser (see Figure 3-3).

**Distributed System DX**

![Distributed System DX Diagram](image)

**Figure 3-3: Distributed System DX**

<sup>1</sup> Mechanical subcooling means that a vapour compression cycle is used for the purpose of providing subcooling to the main refrigeration system (Thornton, 1994).
The distributed system consists of several small compressor racks located in boxes near the display cases. The refrigerant circuit lengths in a distributed system are less and the total refrigerant charge will be about 75% of multiplex systems (Bivens, 2004).

3.2 Indirect System

3.2.1 Completely Indirect System
Refrigeration with indirect systems has been introduced in supermarkets to decrease the refrigerant charge and to minimize potential refrigerant leakage. Indirect systems have many forms; one of them is the completely indirect system. A design with a completely indirect system is presented in Figure 3-4. In this system design, there are two refrigeration systems (chillers) with different brines and levels of temperature.

![Completely Indirect System Diagram](image)

Figure 3-4: System Design 2: Completely Indirect System

The secondary refrigerant in the medium temperature level often has an approach temperature around -8°C and a return temperature around -4°C. A typical value of secondary refrigerant temperature going to deep-freeze display cases is about -32°C and the re-
turn temperature is about -29°C. Secondary refrigerants based on potassium formate, potassium acetate, glycols, alcohols and chlorides are used as secondary refrigerants. CO₂ vapour-liquid might be used as a secondary refrigerant in the low temperature system.

One or two other secondary loops (coolant fluids or dry cooler fluids) are used in the system to transport the heat rejected from the condensers, in the machine room, to two different dry coolers located on the roof of the supermarket. Typical out-going temperature of the coolant fluid is about 32°C and the return temperature is about 36°C.

The waste heat from the condenser can be recovered during the winter with substantial energy savings in cold climates. The benefits and drawbacks of completely indirect systems are summarized in a SWOT analysis in Table 3-2.

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Lower refrigerant charges</td>
<td>- Risk for low energy efficiency</td>
</tr>
<tr>
<td>- Simple and cheaper service</td>
<td>- Pump work</td>
</tr>
<tr>
<td>- Use of natural refrigerants possible</td>
<td>- Risk for corrosion</td>
</tr>
<tr>
<td></td>
<td>- Pipes need to be insulated</td>
</tr>
<tr>
<td>Opportunities</td>
<td>Threats</td>
</tr>
<tr>
<td>- Phase-changing brines: ice slurries and CO₂</td>
<td>- More efficient direct systems</td>
</tr>
</tbody>
</table>

A variant of the design of the completely indirect system that improves its performance is presented in Figure 3-5. In this case, the refrigerant in the low temperature unit is sub-cooled with the brine of the medium temperature unit (Thornton 1994). The temperature of the refrigerant after sub-cooling can be about +5°C.
The effect of sub-cooling on the low temperature system is presented in the P-h diagram for R404A in Figure 3-6.
The influence of sub-cooling on the refrigerating effect depends, among other things, on the kind of refrigerant. For R404A, the increase in refrigeration capacity for the low temperature system can be approximately 30%, as shown in Figure 3-6. Heat from the sub-cooling process is rejected to the medium temperature system, which has a higher coefficient of performance COP \_2 than does the low temperature system. This implies lower energy consumption than in a completely indirect system.

Another variation of the completely indirect system is presented in Figure 3-7. The system design uses district cooling to cool the heat rejected from condensers. In this installation, the display cases contain compressors, condensers and evaporators.

The district cooling return pipe cools the condenser from the display cases through a secondary fluid. The coolant fluid temperatures fluctuate between 12°C and 16°C, which are lower than normal. The condensing temperature of the refrigeration system is about 20°C, which implies lower compressor power in comparison with the other system designs. One disadvantage with this solution
is the price of district cooling that might increase the total energy cost.

3.2.2 Partially Indirect System
The most common partially indirect system in supermarkets in Sweden is shown in Figure 3-8. The heat from the condensers is rejected by a dry cooler on the roof of the supermarket to the environment. The low temperature system has a direct system between the compressors and the deep-freeze display cases, and the medium temperature system has an indirect system between the cabinets and the chiller.

![Diagram of Partially Indirect System](image)

Figure 3-8: System Design 3: Partially Indirect System

The benefits and drawbacks of the partially indirect system are summarized in a SWOT analysis in Table 3-3.
Table 3-3: SWOT of Partially Indirect System

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Good efficiency</td>
<td>- Refrigeration charge larger than completely indirect system</td>
</tr>
<tr>
<td>- Lower charge, less leakage</td>
<td>- Pump work</td>
</tr>
<tr>
<td>- Higher reliability</td>
<td>- Risk of corrosion</td>
</tr>
<tr>
<td></td>
<td>- Pipes need to be insulated</td>
</tr>
<tr>
<td>Opportunities</td>
<td>Threats</td>
</tr>
<tr>
<td>- Mechanical sub-cooling of low</td>
<td>- Stiffer legislation for HFC refrigerants leading to charge limits</td>
</tr>
<tr>
<td>temperature system</td>
<td></td>
</tr>
</tbody>
</table>

Another variation of the partially indirect is the distributed system, which has a secondary loop for heat rejection (Baxter, 2003). The distributed system consists of several small condenser units located in boxes near the display cases. The heat from the condenser is rejected by a coolant fluid loop, which circulates between the condensers and a dry cooler on the roof (see Figure 3-9).

Figure 3-9: Distributed System Indirect
3.2.3 Indirect Cascade System

The cascade system, shown in Figure 3-10 is a favourable solution that avoids the large pressure ratio in the low temperature system obtained in the completely indirect system. The installation operates with two different temperature levels and secondary loops. The temperature of the secondary refrigerant in the medium temperature unit has, as in completely indirect systems, an out-going temperature of about -8°C and a return temperature of about -4°C. The out-going temperature of the secondary refrigerant in the low temperature system is about -32°C and the return temperature is about -28°C.

The condenser heat from the low temperature system is rejected to the secondary refrigerant with the medium temperature. The condensing temperature of the low temperature system is about 0°C, which increases the coefficient of performance of the refrigeration cycle and decreases the energy consumption of the low temperature system. The drawback with this system is the increase of refrigeration capacity and compressor power of the medium temperature system due to the condenser heat from the low temperature system.

![Cascade System A Diagram]

Figure 3-10: System Design 4: Cascade System A
The heat from the other condenser is rejected to the outside through a secondary loop that connects the condenser to a dry cooler located on the roof of the supermarket. The waste heat from the condenser can also be recovered during the winter.

The benefits and drawbacks of cascade system A are summarized in a SWOT analysis in Table 3-4.

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Lower refrigerant charges, less leakage</td>
<td>- Both medium and low temperature interact</td>
</tr>
<tr>
<td>- Simple and cheaper service</td>
<td>- Pump work</td>
</tr>
<tr>
<td>- Natural refrigerants possible</td>
<td>- Risk of corrosion</td>
</tr>
<tr>
<td></td>
<td>- Pipes need to be insulated</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Threats</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Phase-change brines: ice slurries and CO₂</td>
<td>- More efficient direct system</td>
</tr>
</tbody>
</table>

Another design utilizing a cascade system is introduced in Figure 3-11. The secondary refrigerant in the medium temperature unit cools the display cases and the condensers of the low temperature unit as in the previous design. The difference is in the refrigeration system of the low temperature unit contained in the deep-freeze display cases. The purpose of the system is to decrease the pump power of the low temperature system because of the high viscosity of the secondary refrigerant at low temperature that affects the pressure drop of the fluid between the chiller and deep-freeze display cases. The refrigeration systems in deep freeze cabinets are compact boxes with connection to the brine 1 in the medium temperature unit and to the brine 2, which circulates in the deep-freeze cabinet. The system is easy to repair because the service technician can change the box and turn on the machine in few minutes.
The benefits and drawbacks of cascade system B are summarized in a SWOT analysis in Table 3-5.

Table 3-5: SWOT of Cascade System B

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Lower refrigerant charges, less leakage</td>
<td>- Both medium and low temperature interact</td>
</tr>
<tr>
<td>- Simple and cheaper service</td>
<td>- More compressor power required</td>
</tr>
<tr>
<td>- Natural refrigerants possible</td>
<td>- Risk of noise</td>
</tr>
<tr>
<td></td>
<td>- Pipes need to be insulated</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Threats</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Phase-change brines: ice slurries and CO₂</td>
<td>- More efficient direct system</td>
</tr>
</tbody>
</table>
3.3 Carbon Dioxide

Other interesting refrigerant systems using carbon dioxide as refrigerant have been implemented in supermarkets to reduce energy consumption and environmental impact.

3.3.1 Cascade System with CO₂

In order to decrease the pumping power in a low temperature system with CO₂ as secondary refrigerant, a system using CO₂ as refrigerant has been developed. A cascade refrigeration system with CO₂ in the low temperature stage and ammonia, propane or R404A in the medium temperature unit is an interesting solution that has been tested in several supermarkets with promising results (see Figure 3-12).

![Cascade System with CO₂](image)

Figure 3-12: Cascade System with CO₂ in the Low Temperature Stage.

In Denmark, a cascade system operating with propane and CO₂ in the low temperature unit was implemented in two supermarkets. The propane is condensed directly in air-cooled condensers on the roof. The propane refrigeration system has two evaporators; one where the propane exchanges heat with an indirect system with glycol that covers the refrigeration requirement from the medium
temperature unit, and another cascade heat exchanger where the propane evaporates and the CO₂ is condensed. Results from the system show that the energy consumption decreased by about 5% compared to a conventional supermarket while the investment was 20% higher (Christensen 1999).

The benefits and drawbacks of a cascade system with CO₂ are summarized in a SWOT analysis in Table 3-6.

### Table 3-6: SWOT of Cascade System with CO₂

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Natural refrigerant</td>
<td>- Risk accompanying high pressure</td>
</tr>
<tr>
<td>- Lower CO₂ emissions</td>
<td>- Pipes need to be insulated</td>
</tr>
<tr>
<td>- Natural refrigerants possible</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Threats</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Ice slurries as brine in medium temperature unit</td>
<td>- More efficient direct system</td>
</tr>
</tbody>
</table>

#### 3.3.2 CO₂ as the Only Refrigerant

CO₂ as the only refrigerant in the refrigeration system is an important alternative to HFC refrigerants in supermarkets. The CO₂ cycle might be trans-critical or sub-critical depending on ambient temperatures. The trans-critical temperature of CO₂ is 31°C. At higher ambient temperatures, the refrigeration system with CO₂ will operate at temperatures over the critical point. At low ambient temperature, as in cold climates, the operation of the refrigeration system will be in the sub-critical region.

The advantage with CO₂ as the only refrigerant in the refrigeration system in comparison with the cascade system is the absence of the heat exchanger between the low and medium temperature levels. The disadvantage is the high operating pressure of the high stage of the cycle.

A multistage CO₂ system is presented in Figure 3-13. The system has a two-stage compressor on the pressure side, an internal exchanger and a vessel with CO₂ in liquid and vapour state. A pump transports liquid CO₂ to cabinets and cold rooms in the medium
temperature unit. Liquid CO₂ from the vessel goes to the direct expansion evaporator in cabinets and cold rooms in the low temperature level. After the evaporators, the CO₂ is compressed in the low pressure compressor and discharged as gas in the vessel (Schiesaro 2002).

**Multistage System 1 with CO₂**

Another refrigeration system using CO₂ as the only refrigerant (see Figure 3-14), presented by Girotto, Minetto and Nekså, utilizes a two-stage compression in the low temperature unit to reduce the discharge temperature, to reduce the pressure ratio, to improve the isentropic and volumetric efficiencies and to allow the use of a suction - liquid heat exchanger, which increases efficiency (Girotto 2003). The two-stage compression in the low temperature unit has direct rejection of heat to ambient air, which decreases energy consumption and the investment cost in comparison with the cascade concept (Ibid.).
The benefits and drawbacks of a system with CO$_2$ as the only refrigerant are summarized in a SWOT analysis in Table 3-7.

Table 3-7: SWOT of System with CO$_2$ as the Only Refrigerant

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Natural refrigerant</td>
<td>- Trans-critical system</td>
</tr>
<tr>
<td>- Lower CO$_2$ emissions</td>
<td>- Risk accompanying high pressure</td>
</tr>
<tr>
<td></td>
<td>- Pipes need to be insulated</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Threats</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Sub-critical at low ambient temperatures</td>
<td>- More efficient direct system</td>
</tr>
</tbody>
</table>
3.4 Heat Recovery and Floating Condensing in Indirect Systems

Many supermarkets in Sweden utilize heat recovery (or heat reclaim) from condensers as one way to increase the overall energy efficiency of the system. Obviously, this option is only interesting in relatively cold climates such as in northern Europe, Canada, etc. One drawback of heat reclaim is that the condensation temperature must be kept at a level where heat can be transferred to the heating system of the supermarket. The typical required temperature level for the condenser coolant is 38°C after the condenser. This increases energy consumption from compressors but at the same time this leads to a reduction of energy consumption for the heating system (district heating or electric or oil boiler). An alternative to heat recovery is floating condensing temperature that improves the coefficient of performance and decreases the energy consumption of the compressors at lower outdoor temperature. Another option is to utilize both heat recovery and floating condensing pressure depending on the heat requirements of the premises.

One interesting question for heat reclaim is how much energy can be utilized from the condensers in a real system. There are many viewpoints on this subject and practical experiences indicate that 40 – 70% of the necessary heat can be recovered. One reason for this is that the refrigeration system does not operate continuously. Another issue is that the design and operation of the refrigeration system and the HVAC systems not are done by the same people/organization and the communication between them is not always the best.

3.4.1 Heat Recovery Systems

The refrigeration system in supermarkets always rejects large amounts of heat from the condenser to the environment and a new efficient refrigeration system design takes advantage of the rejected heat by using it for air heating. There are three different heat recovery system designs that are the most representative for supermarkets in Sweden.

The first heat recovery system design, presented in Figure 3-15, has an approach temperature around 36°C to the dry cooler and a...
return temperature around 32°C. The approach temperature to the heat exchanger in the air system is around 32°C and the return temperature is around 28°C.

**Heat Recovery System 1**

![Diagram](image)

The circuit between the condenser and the dry cooler is connected with the air system by an extra heat exchanger. An auxiliary heater (electric or oil boiler or district heating) has been connected between the normal heat exchanger and the finned heat exchanger in the air system to ensure that the desired hot water temperature. The reason for the extra heat exchanger is to guarantee the operation of the refrigeration system in case of a ventilation system failure and to separate the responsibilities between contractors when incorrect operations of ventilation or refrigeration systems occur.

The system design has two major disadvantages: the first is that the heat exchanger reduces the approach temperature to the coil in the air system and the second is that when the air supply temperature is reached, the control system by-passes the heat exchanger in the air system and, as is often the case, the heat from the auxiliary heating is rejected to ambient air temperature in the dry cooler.
This system requires a sophisticated control unit to avoid heat losses from the auxiliary heating to the environment through the dry cooler.

The second heat recovery system design is presented in Figure 3-16. The circuit between the condenser and the dry cooler is connected with the air system by an extra heat exchanger as in the first system design. The auxiliary heating is connected to the air system after the heat exchanger that recovers the heat from the condensers. The approach temperature to the dry cooler is around 36°C and the return temperature is around 32°C. The approach temperature to the heat exchanger in the air system is around 32°C and the return temperature is around 28°C. The disadvantage of this system is the reduction of the approach temperature to the coil in the air system as in system 1.

The third heat recovery system design is shown in Figure 3-17. The heat from the condensers is rejected directly to the air system via a heat exchanger. The approach temperature to the heat exchanger is around 36°C, and the return temperature is around
32°C. The auxiliary heating is connected to the air system after the heat exchanger that recovers the heat from the refrigeration system.

3.4.2 Floating Condensing System
A drawback with the heat recovery system is the high condensing temperature that increases the energy consumption of the refrigeration system. In floating condensing systems (see Figure 3-18), the condensing temperature changes with the ambient temperature. The system is possible to implement with electronic expansion valves that are designed to operate over a wider range of pressure drops. At lower outdoor temperatures, the condensing temperature can float down properly. This increases the coefficient of performance, COP₂, and decreases the energy consumption of compressors. To cover the heat requirements of the premises, it is necessary to use an oil boiler or district heating.

Figure 3-17: Heat Recovery System Design 3
3.4.3 Heat Recovery and Floating Condensing System

A novel system with both heat recovery and floating condensing has been designed in Sweden for increased energy efficiency in supermarkets. The main idea is to cover the cooling requirements for cabinets and storages and the heating requirements for the premises. When the heating requirements decrease, one or more chillers should run at lower condensing temperatures. This would reduce the energy consumption of the refrigeration system. Figure 3-199 shows a system with two chillers running at different condensing temperatures. One chiller covers the heating requirements and the other chiller works with the floating condensing temperature.
3.5 Conclusions

The purpose of refrigeration systems in supermarkets is to provide storage and display of perishable foods prior to sale. Food is stored in walk-in storages before transfer to display cases in the sales area. There are two principal temperature levels in supermarkets: medium temperature for preservation of chilled food and low temperature for frozen products.

The quest for increased energy efficiency and the phase-out of ozone depleting substances have affected the refrigeration system design for supermarkets considerably. New system solutions with completely or partially indirect systems have been developed and introduced in supermarkets to lower the refrigerant charge and minimize potential refrigerant leakage.

The increasing interest in indirect systems has led to the development of some new secondary refrigerants based on potassium
formate and potassium acetate alone or mixed since these fluids have good heat transfer and pressure drop characteristics. Other very promising developments are phase-changing secondary refrigerants such as CO₂ and ice slurries. Ice slurry is an interesting secondary refrigerant that adds thermal storage to the system. The technology with CO₂ as secondary refrigerant for the low temperature system has been implemented since 1995 in the Nordic countries. In order to avoid the pumping power in a low temperature system with CO₂ as secondary refrigerant, a system using CO₂ as primary refrigerant has been developed. A cascade refrigeration system with CO₂ in the low temperature stage and ammonia, propane or R404A in the medium temperature unit is an interesting solution with promising results. CO₂ as the only refrigerant in the refrigeration system is an important alternative to HFC refrigerants in supermarkets. The CO₂ cycle may be trans-critical or sub-critical depending on ambient temperatures.

Many supermarkets in Sweden utilize heat recovery to increase the overall energy efficiency of the system. One disadvantage of heat recovery is the high condensation temperature necessary to transfer the heat from the condenser to the heating system of the supermarket. This increases the energy consumption for the refrigeration system but at the same time leads to a reduction of energy consumption for the heating system. One interesting question is how much energy can be utilized from the condensers in a real system. Practical experiences indicate that 40 – 70% of the necessary heat can be recovered. One reason for this is that the refrigeration system does not operate continuously. Another issue is that the design and operation of the refrigeration system and the HVAC systems are not done by the same people/organization and the communication between them is not always satisfactory.

An alternative to heat recovery is floating condensing temperature that improves the coefficient of performance and decreases energy consumption of the compressors at lower outdoor temperature. Another option is to utilize both heat recovery and floating condensing temperature depending on the heating requirements of the premises.
4 Field Measurements

4.1 Overview

New refrigeration system designs with the indirect system have been implemented in supermarkets to reduce the refrigerant charge and to minimize the leakage. In Sweden, the evaluation of these systems has been carried out during short periods and with limited resources. In some cases the evaluation of the refrigeration system took into consideration only air temperatures in cabinets. A more exhaustive examination of systems and components is necessary to evaluate energy efficient technologies and energy consumption in supermarkets during at least one year. Several parameters such as indoor and outdoor temperatures, moisture, air temperature in cabinets, energy usage in lighting, heating, cooling and refrigeration systems should be logged at pre-set time intervals, one hour for example, to see the interaction of different subsystems with the effects of different energy measures.

Field measurements in seven supermarkets in Sweden have been carried out to evaluate new refrigeration system designs with the indirect system, to estimate the influence of different parameters such as outdoor temperature and moisture and to validate the computer model CyberMart. The measurements have been divided into three different periods to reduce the amount of measurements and to cover the most important parameters in supermarkets (Arias 2000).

In Period 1, which covered one year, the following parameters were measured: outdoor temperature, indoor temperature, relative indoor air humidity, secondary refrigerant temperature before and after the chiller, and the compressor power of the medium temperature unit.

In Period 2, which covered one week, the following parameters were measured: the air temperature in the inlet, middle and outlet
of a display case and the compressor power of the low temperature unit.

In Period 3, which covered one hour, the following parameters were measured: the air temperature in the inlet, middle and outlet of a display case.

4.1.1 Field Test 1
The first supermarket tested is in the city of Sala. The total area of the supermarket is 2700 m². The refrigeration system in the supermarket in Sala has a cascade system design (see Figure 4-1 and chapter 3.2.3). The secondary refrigerant in the medium temperature system cools the display cases and the condenser for the chiller of the low temperature system.

The refrigeration system has a refrigeration capacity of about 35 kW in the low temperature unit and about 130 kW in the medium temperature unit. The supermarket has 10 display cases, 3 deep-freeze cabinets and 7 cold rooms. The refrigerant in both temperature units is R404A. The secondary refrigerant in the medium temperature unit is Pekasol 50 with 60% concentration. The secondary refrigerant in the low temperature unit is Pekasol 50 with
90% concentration. The supermarket has a heat recovery system and an air conditioning system. The auxiliary heating is district heating.

4.1.2 Field Test 2
The second supermarket tested is in the city of Hjo. The refrigeration system in the supermarket in Hjo is also a cascade system (see Figure 4-2). In this case, there is some refrigeration by an indirect system in each deep-freeze cabinet. Scroll compressors have been installed in the cabinets to avoid noise. The condensers for these machines and the display cases in the medium temperature system are cooled by a chiller situated in the machine room. There is also a heat recovery system in this supermarket. The total area of the supermarket is about 700 m², the refrigeration system has a refrigeration capacity of about 60 kW in the medium temperature unit and about 13 kW in the low temperature unit. The refrigerant in both temperature units is R404A.

![Figure 4-2: Refrigeration System in Hjo](image-url)
4.1.3 Field Test 3

The third supermarket tested is in Farsta Centrum in the city of Stockholm. At the beginning, the refrigeration system in Farsta Centrum was designed to use district cooling to cool the condenser of the refrigeration equipment in each deep-freeze cabinet, display case and cool storage via a brine loop. The high prices of district cooling in Farsta Centrum forced development of a new solution with a chiller installed on the roof instead of using the district cooling system (see Figure 4-3). The supermarket has been divided, with air curtains, into two different zones: a cold zone for products requiring refrigeration (where all the display cases are located) and a warm zone for non-refrigerated products. The refrigeration system has 46 compressors distributed in the cold zone. The refrigeration capacity of the chiller on the roof is about 200 kW. The total area of the supermarket is 1800 m². The refrigerant used is R404A. The supermarket has no heat recovery system because heating and cooling are included in the rent of the premises.

![Farsta Diagram](image)

Figure 4-3: Refrigeration System Design in Farsta Centrum
4.1.4 Field Test 4
The fourth supermarket tested is in the city of Hedemora. The supermarket in Hedemora has refrigeration with a parallel system design and sub-cooling (see Figure 4-4). The secondary refrigerant in the medium temperature system is propylene glycol and CO2 in the low temperature unit. The refrigeration capacity in the medium temperature unit is 75 kW and 29 kW in the low temperature unit. The total area of the supermarket is about 2000 m². The refrigerant for both temperature units is R404A. The supermarket has a heat recovery system.

![Figure 4-4: Refrigeration System Design in Hedemora](image)

4.1.5 Field Test 5
The fifth supermarket tested is in the city of Västerås. Refrigeration in the supermarket in Västerås consists of a completely indirect system in the medium temperature unit and a partially indirect system in the low temperature unit (see Figure 4-5). The total area of the supermarket is about 2200 m². The system has a refrigeration capacity of about 130 kW in the medium temperature unit and about 32 kW in the low temperature unit. The refrigerant in both temperature units is R404A. The supermarket has a heat recovery system.
system. Two students from Mälardalen University, in cooperation with the supermarket chain ICA and the Department of Energy Technology at KTH, carried out the measurements in Västerås. The focus of these measurements was the performance of the heat recovery system and the consumption of electricity (Skärbo 2002).

4.1.6 Field Test 6

The sixth supermarket tested is located in Täby Centrum in Stockholm. Refrigeration in the supermarket in Täby Centrum is provided by a cascade system. Each deep-freeze cabinet has a compact refrigeration system. The condensers for some of these cabinets are cooled with the brine from the medium temperature system while other cabinets directly reject heat from the condenser to the sales area in order to increase the air temperature around the cabinets (see Figure 4-6). The display cases in the medium temperature system are cooled with a chiller situated in the machine room. The total area of the supermarket is about 2700 m². The system has a refrigeration capacity of about 190 kW in the medium temperature unit and about 32 kW in the low temperature unit. The refrigerant in both temperature units is R404A. The supermarket has a heat recovery system.
The seventh supermarket tested is located in Kista Centrum in Stockholm. Refrigeration in the supermarket in Kista Centrum is also provided by a cascade system. The condensers for some of the cabinets are cooled with the brine from the medium temperature unit, while other cabinets reject heat from the condenser directly into the sales area, as in the supermarket in Täby Centrum (see Figure 4-7). The display cases in the medium temperature unit are cooled with a chiller situated in the machine room. The total area of the supermarket is about 2700 m². The system has a refrigeration capacity of about 140 kW in the medium temperature unit and about 21 kW in the low temperature unit. The refrigerant in both temperature units is R404A. The supermarket has no heat recovery system because heating and cooling are included in the rent of the premises.
4.1.8 Test Instruments

The temperature and relative humidity for air was measured with Tinytag-loggers from Intab. The Tinytag-loggers have a resolution of 12 bits for temperature logging and 8 bits for relative humidity logging. The accuracy of Tinytags is better than +/-0.4°C at 20°C for temperature and better than +/-3% at 25°C for moisture (Intab 2004). The compressor power was measured with Energy-logger ELite 4 from Pacific Science & Technology. The Energy-logger ELite 4 has a resolution of 12 bits and an accuracy better than 1% of reading (Pacific Science & Technology 1999). The temperature and relative humidity have been measured and stored every hour. The compressor power is also measured and stored every hour, but in this case the value represents the compressor power average during the last hour.
4.2 Results

Results from the measurements confirm the importance and the influence of the outdoor temperature, indoor temperature and relative air humidity on compressor power. Figure 4-8 shows the average values during one day for compressor power, indoor temperature and outdoor temperature over the course of one year in the supermarket in Hjo. The variations in compressor power follow the variations in outdoor temperature throughout the whole year, including during the heating season that starts in the middle of October and ends in the middle of March. In Figure 4-8, it is also possible to see the influence of the indoor temperature on compressor power. From 15 December to 4 January, a failure in a fan in the HVAC system resulted in low air temperatures in the supermarket, which in turn strongly influenced the compressor power.

Figure 4-8: Indoor Temperature, Outdoor Temperature and Compressor Power in Hjo

Figure 4-9 presents measurements of indoor temperature in the cold and warm zones, outdoor temperature and compressor power for the chiller in Farsta Centrum. Measurements from Farsta Centrum also show the influence of those temperatures on compressor power.
Results from the measurement of indoor and outdoor temperatures, relative humidity and compressor power in Sala and Hedemora are presented in Figure 4-10 and Figure 4-11 respectively. The diagrams show the influence of moisture on compressor power and the relation between relative humidity and outdoor temperature.
The influence of relative humidity on compressor power is confirmed in Figure 4-12, which presents the outdoor temperature, indoor temperature, relative humidity, in-coming and out-going brine temperatures to and from the chiller and the compressor power over the course of two days.
The 14th of July was a warm and rainy in Sala that affected the moisture in the supermarket and caused the highest compressor power usage during the summer of 1999.

The variations in temperature, moisture and compressor power in Sala during week 32 in August 1999 are shown in Figure 4-13. In both Figure 4-12 and Figure 4-13, it is also possible to see the influence of night covering of display cases and defrosting on compressor power. The cabinets are defrosted with electrical heaters.

According to Figure 4-12 and Figure 4-13, compressor power is reduced 10 to 20% by the night covering of the cabinets. Covering occurs automatically when the supermarket closes at 21.00 and ends at 8.00 in the morning.

In Figure 4-14, the inlet air temperature, average temperature and return air temperature of a display case are presented. The heat extraction rate, according to the manufacturer's technical data for climate class 25°C/60%RH, is 9.0 kW, the storage temperature is +1°C, the inlet air temperature is –2°C and the return air temperature is +5°C. During the period when the supermarket is open, the variation in temperatures corresponds to the cabinet’s technical data. When the cabinet is covered, the inlet air temperature and the return air temperature converge toward the storage temperature.
The measurements in a cabinet during Period 3 (one hour) taken on the 17th of May 2000 are presented in Figure 4-15 and corroborate the technical data of the display case. The temperatures have been measured momentarily and stored every 15 seconds. In the diagram, it is also possible to see the influence of customers and employees on the air temperatures when they approach the cabinet. The influence on inlet air temperature is insignificant.
High indoor relative humidity increases the refrigeration load and compressor power. In Figure 4-16, air temperatures from a cabinet, moisture levels and compressor power over the course of one day in June in Sala are presented. The compressor operates at maximum capacity and cannot maintain the required inlet air temperature in the display cases (below –2°C). The temperature in the cabinet consequently increases to around 5°C.

The influence of high indoor temperatures on cabinets has been studied in Hedemora. The high investment cost for an air-conditioning system and a short period of higher outdoor temperatures during the summer have affected the decision to install AC in many supermarkets in Sweden. Figure 4-17 presents relative humidity, indoor temperature, outdoor temperature and return air temperatures in four different sections of a deep-freeze cabinet during three days in August 2001. The air temperatures in the sections are below –20°C and indoor temperature is around 20°C, as intended.
Figure 4-17: Indoor and Outdoor Temperatures, Indoor Relative Humidity and Air Temperatures in a Deep-freeze Cabinet during Three Days in August 2001 in Hedemora.

Figure 4-18 presents relative humidity, indoor temperature, outdoor temperature, and return air temperatures in the same four sections of cabinet in Hedemora during four days in July 2001.

Figure 4-18: Indoor and Outdoor Temperatures, Indoor Relative Humidity and Air Temperatures in a Deep-freeze Cabinet during Four Days in July 2001 in Hedemora.
The outdoor temperatures were above 30°C during a long part of the day, which affected the indoor temperatures. Indoor temperatures during the day were above 25°C, which is the dimensioning temperature of the display cases. The maximum indoor temperature was 28°C, which occurred on the 5th of July at around 17:00 o’clock. The return air temperatures in three sections were higher than –20°C and one of them had return air temperatures higher than -15°C during the day.

If the temperature of the products is assumed to be 1°C higher than the return air temperature, which is a reasonable assumption, then the temperature of the product should be about –14°C at a return air temperature of -15°C. According to the National Food Administration, which regulates and supervises the food sector in Sweden, the maximum temperature of frozen products should be –18°C. The date for the minimum durability, or use-by date, of frozen food is calculated given a temperature of –18°C. Higher product temperature is accompanied by the risk of the apparition of bacteria in the frozen food, which might cause illness to consumers. The results from Figure 4-17 and Figure 4-18 confirm the necessity of air conditioning during the warmer days in summer.

Customers and employees experience lower indoor temperatures in supermarkets during the winter. For this reason, measurement of indoor temperature and moisture were carried out in the supermarkets in Täby Centrum and Kista Centrum to study the variation in temperature and relative humidity in different zones of the stores over the course of one year.

In Täby Centrum, temperature and relative humidity were measured in two different zones: a warm zone for non-cooled products and a cold zone where all the cabinets and cooled products are located. In Kista Centrum, temperature and relative humidity were also measured in a warm and cold zone but in this case at three different points in the supermarket.

In Täby Centrum Tinytag-loggers were installed at about 1.8 meters height from the floor and about 7 meters from the nearest cabinets in the warm zone and at about 0.5 meter from the floor and 2 meters from the nearest cabinets in the cold zone. Figure 4-19 presents measurements of indoor temperatures in the cold and warm zones, ambient temperature and compressor power for
the chillers of the medium temperature system in Täby Centrum. Figure 4-20 presents measurements of outdoor relative humidity and in the cold and warm zones also in Täby Centrum.

In Figure 4-19, the difference between indoor temperature in the warm and cold zones is about 8°C during the winter and about
3°C during the summer. In Figure 4-20, the relative humidity in the warm zone is about 10% lower than in the cold zone during the winter, while the difference between the relative humidities during summer is insignificant.

In Kista Centrum, Tinytag-loggers were installed at three different points in the supermarket, one in the warm zone and two in the cold zone. The measuring devices in the warm zone were installed at about 1.8 meters height from the floor and about 8 meters from the nearest cabinets. Tinytag-loggers in the cold zone were installed at about 0.4 meter from the floor and 2 meters from the nearest cabinets at one point (the so-called “cold zone low”) and the others at about 1.8 meters height from the floor and 2 meters from the nearest cabinets (the so-called “cold zone high”).

Figure 4-21 presents measurements of indoor temperature in the cold and warm zones, ambient temperature and compressor power for the chillers of the medium temperature unit in Kista Centrum during the course of approximately one year. Figure 4-22 presents measurements of moisture in the cold and warm zones as well as outdoor temperature and relative humidity in Kista Centrum over the course of approximately one year.

![Figure 4-21: Temperatures and Compressor Power in Kista Centrum](image)
In Figure 4-21, the difference between indoor temperature in the warm and cold zones is also about 8°C during the winter and about 3°C during the summer in the supermarket in Kista.

In Figure 4-22, the difference between the relative humidity in the warm zone and in the cold zone low during the winter is also about 10%, while in the summer the relative humidity in the warm zone is about 10% higher than in the cold zone low. The values of relative humidity in the cold zone high during the winter are between the warm zone and cold zone low values. During the summer, the values of relative humidity in the cold zone high and cold zone low are about the same.

Measurements from Täby Centrum and Kista Centrum also corroborate the influence of indoor temperature and moisture on compressor power. The indoor relative humidity in the cold and warm zones in both supermarkets follows the variation in outdoor temperature during the winter period of study. During the summer, the indoor relative humidity varies as a function of both outdoor temperature and relative humidity.
Another important parameter that influences indoor relative humidity is the humidity ratio. The variation in the humidity ratio in the supermarket in Hjo between July 1999 and July 2000 is shown in Figure 4-23.

Values lower than 2 \( \text{gr.H}_2\text{O/kg.dry air} \) were measured at lower outdoor temperature. The humidity ratio affects the relative humidity, which influences energy consumption. At the same time, a low humidity ratio can produce a reduction in weight of fruit, vegetables and other products.

The humidity ratios in cool and warm zones as well as outdoor in Täby Centrum and in Kista Centrum are shown in Figure 4-24 and Figure 4-25 respectively. The average of the humidity ratio during the winter in both supermarkets is about 3[gram H\textsubscript{2}O/kg.dry air]. During the summer, there are some days with humidity ratio values over 9[gram H\textsubscript{2}O/kg.dry air].
The outdoor humidity ratio and the humidity ratios in the cold and warm zones are about the same during the winter period in both supermarkets. During the summer period (end of April to end of September), the variation in humidity ratios between the different zones and outdoor humidity is significant in both supermarkets.
The indoor humidity ratios are lower than the outdoor values because the air conditioning system and cabinets dry the indoor air.

Outdoor temperatures, humidity ratio differences and enthalpy differences between the indoor and outdoor climate in Farsta Centrum are shown in Figure 4-26.

The air enthalpy difference between indoor and outdoor climate is almost a function of the air temperature difference. The difference in humidity ratios is quite small during summer time. One reason for this is because the air conditioning system of the shopping centre, where the supermarket is located, is not sufficient to cover the cooling demand of the building. During warmer days, the employees of the shopping centre open the doors to circulate the indoor air.

Outdoor temperatures, humidity ratio differences and enthalpy differences between the indoor and outdoor climate in Kista Centrum are shown in Figure 4-27.
The air enthalpy difference between indoor and outdoor climate is almost a function of the air temperature difference during the winter period. During the summer period, the difference in air enthalpy is a function of air temperature and humidity ratio differences.

The results from measurements presented up to now have clearly shown the impact of indoor temperature and relative humidity on the refrigeration loads of display cases and cold rooms in supermarkets. Measurements of indoor and outdoor temperatures and indoor relative humidity carried out in the supermarkets in the cities of Sala and Hedemora also point out a relation between the outdoor temperature, the humidity ratio, and the indoor relative humidity (see Figure 4-28, Figure 4-29 and Figure 4-30).

Figure 4-28 presents measurements of indoor and outdoor temperatures and indoor relative humidity from Hedemora and Sala. Figure 4-29 shows the measurements of outdoor and indoor humidity ratio in both cities. Figure 4-30 gives the measurements of outdoor temperature and outdoor humidity ratio in the two cities. The distance between the cities is about 60 km. In Figure 4-28, it is possible to see that the indoor relative humidity in both supermarkets follows the variation in outdoor temperature during the pe-
period of study. At constant indoor temperature, the indoor relative humidity is only a function of the indoor humidity ratio. Figure 4-29 illustrates that the outdoor and indoor humidity ratios in Sala and Hedemora are about the same during the winter period. At the same time, Figure 4-30 shows that the outdoor humidity ratio follows the variation of the outdoor temperature during the winter time.

![Figure 4-28: Indoor and Outdoor Temperatures and Indoor Relative Humidity in Hedemora and Sala](image)

![Figure 4-29: Humidity Ratio in Hedemora and Sala](image)
Lower outdoor temperatures decrease indoor relative humidity, which reduces the refrigeration load of cabinets and cold rooms. Reduction in the refrigeration load of the system reduces the amount of available heat from the condenser and heat recovery. At lower outdoor temperatures, when the heating load is largest, the refrigeration load can decrease by as much as 50-70% in comparison to the nominal refrigeration load with indoor conditions of 22°C and 65% RH.

Field measurements in the supermarket of Sala have been carried out during the month of March 2001 in order to investigate the variation in compressor power and the influence of this on condenser coolant temperatures. The temperatures and relative humidity have been measured momentarily and stored every five minutes. The compressor power is also stored every five minutes, but in this case the value represents the mean power during the last five minutes.

Figure 4-31 and Figure 4-32 show results from the measurement of outdoor temperature, indoor temperature, relative humidity, compressor power, and dry cooler fluid approach and return temperatures on the 3rd and the 11th of March 2001.
On March 3rd, 2001 (Figure 4-31), the average compressor power was 36.4 kW, and on March 11th, it was 42.6 kW. Those values are equivalent to 64% and 77% of the nominal compressor power. The over dimension of the refrigeration system during cold days makes the running time of the compressor very short, which influences the condenser coolant approach temperature. Figure 4-31 and Figure 4-32 show the influence of outdoor temperature and moisture on compressor running time: the low average outdoor temperature (about –10°C) and relative humidity (about 18%) on March 3rd caused a lower compressor running time and a larger variation in condenser coolant approach temperature than on the 11th of March, when the average outdoor temperature was about 5°C and the relative humidity was about 33%.

The percentage of time when the coolant fluid approach temperature was higher than 33°C has been calculated as a measure of the time in which heat from the condenser can be recovered. The results show that on March 11th the percentage of time was 74% while on March 3rd the percentage of time was 48%.
Skärbo and Zanghnaeh carried out measurements in the supermarket in the city of Västerås with a focus on the heat recovery system and electricity consumption (Skärbo 2002). Results from these measurements have been used to validate the model and to study in detail the heat recovery system sub-model.

The heat recovery system in the supermarket in Västerås is of system design 2 shown in Figure 3-16. The refrigeration system consists of two chillers with a common secondary refrigerant, condenser coolant fluid circuits in the medium temperature system and three racks of compressors with a common condenser in the low temperature system.

During the summer, when both chillers are operating, the approach temperature of the coolant fluid from the condenser is around 38°C, which is the dimensioning approach temperature to the extra heat exchanger before the heat exchanger in the ventilation system. Results from measurements carried out on July 11th, 2002 in Västerås are presented in Figure 4-33. The parameters in the figure are the approach coolant fluid temperature from both condensers, the common approach temperature to the extra heat exchanger and the condensing pressure of both compressors. Figure 4-33 shows that the approach coolant fluid temperatures are about 38°C and that the pressures are quite uniform during the
period of measurement. The condensing pressures are about 19 bar, equivalent to a condensing temperature of about 41°C.

Figure 4-33: Approach Coolant Fluid Temperatures and Pressures from Condensers on July 11th in Västerås

Figure 4-34 presents results from measurements carried out on February 21st, 2002 in the same supermarket. The parameters in Figure 4-34 are the approach coolant fluid temperature from both condensers, the common approach temperature to the extra heat exchanger and the condensing pressure of both compressors.

Figure 4-34: Approach Coolant Fluid Temperatures and Pressures from Condensers on February 21st in Västerås
Figure 4-34 shows that when both chillers are operating, the common approach temperature to the extra heat exchanger is about 35°C. When both chillers are stopped, the common approach temperature is about 29°C (the pumps are always running). When one of the chillers is stopped, the common approach temperature is about 32°C and not 35°C as it should be.

The reason for this is given in Figure 4-35. Suppose that chiller 1 is operating and chiller 2 is stopped. The approach coolant fluid temperature from condenser 2 is equal to the return coolant fluid temperature from the extra heat exchanger, which is 29°C. The approach coolant fluid temperature from condenser 1 is about 35°C. The approach coolant fluids from both condensers are mixed at the mixing point shown in Figure 4-35. The approach temperature to the heat exchanger after the mixing point is the average combined temperature of the fluids, that is, 32°C. The same problem with the mix of fluids at different temperatures occurs in the secondary refrigerant circuit.
When one of the compressors in Figure 4-35 is off, the coolant fluid from the heat recovery system circulates through both condensers. The temperature of the coolant fluid after circulation through one of the condensers increases by about 4°C, while the temperature of the fluid after circulation through the other condenser remains changed. The fluids from both condensers will be mixed at the mix point giving a decrease in the total coolant fluid approach temperature.

The supermarket in Västerås had high electricity consumption because of unsatisfactory design and control of the heat recovery system and a low coefficient of performance in the refrigeration system (Skärbo 2002). Figure 4-36 show measurements of evaporating and condensing temperatures in both chillers in the medium temperature system. Both chillers have evaporating temperatures lower than -18°C, which are too low in comparison with the dimensioning value.

The majority of the supermarkets studied have had problems with high energy consumption, high condensing temperatures or low evaporating temperatures (in comparison to set point values), inadequate heat reclaim systems, and unsatisfactory design and control of different components or systems, etc. in the stores.
The refrigeration system in the supermarket in Kista Centrum works at a higher condensing temperature when conditions allow for the use of floating condensing pressure to improve the energy performance of the refrigeration system, since heating and cooling of the premises are included in the rent. The supermarkets in Hjo and in Hedemora have had problems with the indoor climate as shown in Figure 4-8 and Figure 4-18. Furthermore, in the supermarket in Hedemora, the refrigeration charge in the medium temperature unit was too low, which affects the running time of the two compressors in the system.

Results from measurements in Sala show values higher than 9 kW for compressor power in the low temperature unit, which is the nominal value at 0°C condensing temperature and -36°C evaporating temperature (see Figure 4-37). The reason for this was that at the low condensing temperature, three compressor motors failed in the low temperature system. This forced an increase of the condensing temperature to about 20°C, which in turn increased the compressor power (Bjerkhög 2004).

Figure 4-37: Compressor Power in the Low Temperature Unit, Indoor Temperature and Relative Humidity in Sala
4.3 Conclusions

Results from field measurements show, not surprisingly, that outdoor temperature and indoor relative humidity are two important factors to take into consideration when dimensioning refrigeration system design and heat recovery systems.

Measurements carried out in supermarkets in the cities of Sala and Hedemora emphasize the relationship between outdoor temperature, the humidity ratio, and the indoor relative humidity.

The field measurements also illustrate that night covering of display cases and deep-freeze display cases reduce energy consumption in supermarkets by about 10-20%. Night covering is thus an efficient method of reducing infiltration and radiation loss in cabinets.

The measurements also demonstrate that at indoor temperatures above 25°C, the temperature of products in display cases and deep-freeze cabinets increase to levels that affect the date of minimum durability of the food. An air conditioning system is necessary in cold climates to avoid indoor temperatures higher than 25°C, even if the warm season is short.

Measurements in Västerås show a lower supplied temperature to the heating coil in the HVAC system than the dimensioning value, lower condenser heat than expected, and mixing points in the coolant fluid circuit, all of which illustrate an unsatisfactory design and control of the heat recovery system.

The majority of the supermarkets studied have had problems with high energy consumption, high condensing temperatures or low evaporating temperatures in comparison to set point values, inadequate heat reclaim systems, and generally unsatisfactory design and control of different components or systems in the stores.
5 CyberMart, Systems and Models

5.1 Introduction

Many new ideas and concepts have been introduced in supermarkets during the last few years, with the intention of decreasing energy usage and minimizing refrigerant charge. Some of these new ideas and concepts are beneficial while others are less so. The supermarket sector has more or less used the trial and error approach to implement and evaluate these new ideas and concepts. One example is the introduction of heat recovery systems to reduce energy consumption. Experience shows in many cases that heat recovery does not work as it was expected to.

A supermarket is a complex system where many subsystems interact. A system approach must be taken in evaluating the impact of energy efficient measures in different subsystems in the supermarket. It is necessary to implement a systems model in order to predict and evaluate the introduction of new concepts and ideas in supermarkets.

There are many models available for evaluation of energy efficiency, renewable energy and sustainability of buildings. In the Building Energy Software Tools Directory on the homepage of the U.S. Department of Energy, there is information about 292 building software tools. The directory includes databases, spreadsheets, component and system analyses, and whole-building energy performance simulation programs (U.S. Department of Energy 2005). None of the whole-building energy performance simulation programs included in this directory evaluates energy efficiency in supermarkets. Building energy simulation tools are able to model building and HVAC performance but not refrigeration system performance or the interaction between cabinets and the surrounding environment.
Nevertheless, companies and researchers around the world have developed computer programs for calculation of total energy performance in supermarkets such as the Supermarket Simulation Tool (SST) created by EPRI in the United States, Clim Top created by A.M.E.C. in France, a special version of DOE-2 program created by Hirsh & Associates in the U.S., and a model using TRNSYS created by Brunel University in the U.K. Other computer programs focus on the energy performance of the refrigeration system in supermarkets. Two of them are the Supermarket Excel Spreadsheets created by ORNL in the U.S. and Econu Koeling II created by TNO in the Netherlands.

The program SST created by EPRI is an hourly building simulation program with a detailed model of a supermarket refrigeration system. The program uses hourly weather data to predict building loads and HVAC energy use. Detailed display case and cold room models consider the impact of indoor humidity levels on refrigeration loads, defrost requirements and anti-condensate heater operation. At the same time, the cooling and dehumidification provided by the cabinets is taken into consideration in the heat and moisture balance calculation. SST considers the use of heat recovery from the refrigeration system for space heating. The interface of the software includes several features that allow the user to quickly assemble the components necessary to define the refrigeration, HVAC and building envelope systems. Components of a system such as compressors or cabinets are represented as “icons” and can be defined or modified by clicking on the item. The performance of each component is modelled using algorithms consistent with available data from manufacturers (Khattar 2000).

The program Clim Top created by A.M.E.C. (a French association promoting energy management in supermarkets) is an hourly building simulation program with focus on HVAC energy consumption in large supermarkets. The tool has a user-friendly interface, especially designed for users working with energy saving in supermarkets. Excel was chosen as the interface. The inputs are organised in four categories: Location, Building, Occupation and Equipment. The user can select from two levels of detail according to his or her level of knowledge. The simplified level proposes typical default values as a help resource. Special attention has been paid to display case descriptions because of their influence on the global energy balance. Outputs are also presented in Excel. Vari-
ous tables and charts allow for comparison of simulation results with real consumption and also to compare the simulations with one another (Orphelin 1997).

The special version of the DOE-2 program created by Hirsh & Associates is an adaptation for analysis of annual energy performance in supermarkets. DOE-2 is a building simulation program that predicts the hourly energy use and cost of a building given hourly weather information and a description of the building and its HVAC equipment and utility rate structure. A user can provide a simple or detailed description of building design or alternative design options and obtain the proposed building’s energy consumption, interior environmental conditions and energy operation cost (Lawrence Berkeley National Laboratory 2005). This special version of DOE-2 is used to model the implementation of energy efficient measures for display cases on the whole store level. The program takes into consideration the energy saved by improved display cases and the corresponding change in space loads experienced by the building’s HVAC system (Walker 2003).

The program created by Brunel University in the U.K. is based on a large number of component models, which have been linked together within a TRNSYS environment. Major component models include the compressor, air-cooled condenser, thermostatic expansion device, display cabinet and control (Ge 2000). TRNSYS is a transient systems simulation program with a modular structure. It recognizes a system description language in which the user specifies the components that constitute the system and the manner in which they are connected. The TRNSYS library includes many of the components commonly found in thermal and electrical energy systems, as well as component routines to handle input of weather data or other time-dependent forcing functions and output of simulation results (S.E.I. 2005).

The program Econu Koeling II created by TNO in the Netherlands is an instrument for comparing offers for refrigeration installation in supermarkets from different suppliers. Econu Koeling II calculates the energy consumption during a year based on details given in the offer. The program transforms an offer received by a supplier into an energy consumption figure. Econu Koeling II is most suited to DX systems but it can also accommodate indirect systems. The program allows no choice of ambient conditions.
The outdoor temperature distribution used is a typical long-term average for the Netherlands. The ambient temperature period is divided into intervals with occurrence distribution from so-called binned analysis. Econu Koeling II comes with a database containing all cabinets from the Eurovent directory of certified products. The program provides reporting on energy consumption per rack and the energy usage of compressors, condenser fans, direct cabinet consumption and others, such as plug-in cabinets (Van der Sluis 2004).

The ORNL Supermarket Excel Spreadsheet program has also been developed to provide the information needed to make decisions affecting energy and refrigerant requirements for multiplexed direct expansion, secondary loop and distributed refrigeration systems. The program uses outdoor temperature bins to compute energy consumption by compressor and condenser fan power, display cases, cold rooms and average power for secondary loop pumps. The user can select the outdoor conditions from among 237 U.S. cities. The ORNL Supermarket Excel Spreadsheets program has 5 predefined low temperature and 8 predefined medium temperature loads. The program has a module for detailed calculations of pressure drop for direct and indirect systems. All calculations are performed using Microsoft Excel spreadsheets and a data interface helps users to enter data (Fischer 2000).

Table 5-1 shows an overview of calculation possibilities from the different computer programs for calculation of total energy performance in supermarkets presented above. The items specified are calculation of indoor conditions, HVAC system, direct refrigeration system, indirect refrigeration system, cascade indirect system, TEWI and LCC.

None of the described computer programs compares different refrigeration system designs with indirect systems such as completely, partially or cascade indirect system, or calculates either TEWI or LCC. Thus a new model is needed to evaluate energy usage, economy and environmental impact in supermarkets. The CyberMart model (Cyber in Table 5-1) calculates the energy performance of the different components in the system, such as the HVAC system, direct refrigeration system, indirect refrigeration system, cabinets, lighting, equipment, etc.
Table 5-1: Overview of Calculation Possibilities in the Different Programs that Calculate Total Energy Performance in Supermarkets

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<th>SST</th>
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<th>Brunel</th>
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<td>No</td>
<td>Yes</td>
</tr>
</tbody>
</table>

5.2 The Modelling Process

The modelling process is the creation of an abstraction of reality. What constitutes the reality of interest or “the system of study” can, of course, be argued and various definitions are given in available literature. The more general description from thermodynamics does not include the specific purpose of a system; one good example is the system description by (Kotas 1995):

A system is an identifiable collection of matter whose behaviour is the subject of study. For identification, the system is enclosed by a system boundary, which may be purely imaginary or may coincide with a real boundary.
In other literature, the idea of the system may also include system objectives and management perspectives. One such formulation of what constitutes a system is given by (Churchman 1968):

A system is a set of parts coordinated to accomplish a set of goals.

The latter definition takes into consideration the purpose of the system. What is especially interesting with these two references is the way the two authors handle the system’s surroundings. Kotas simply states that:

Everything outside the boundary of the system is called the surrounding.

Churchman, on the other hand, realizes the difficulty of defining the system boundary. He suggests two questions to identify a suitable system boundary for a study:

Does it matter relative to my objectives?

If the answer to this question is “yes”, it is worth considering the second question:

Can I do something about it?

If the answer to the second question is “no”, then it is in the environment.

The modelling process consists of different phases, from deciding on the reality of interest for the study according to the objectives of the study itself to completion of the definitive model that answers the questions of the overall objectives identified in the beginning of modelling process. Four phases in the modelling process have been identified. The first phase is the development of a conceptual model of the reality of interest based on the objective of the model. The conceptual model includes the objectives of the model, the environment of the system and the components of the system and their interconnections.
The second phase in the modelling process is the development of a quantitative model based on the conceptual model. In the quantitative model, the different conceptions and ideas from the conceptual model are transformed into physical and mathematical relationships that can be implemented in a computer (Grant 1997). The computer model solves the different equations from the mathematical models during an appointed time period.

The third phase in the modelling process is the evaluation of the computer model developed in the second phase of the modelling process (see Figure 5-3).
The model evaluation has two different parts. The first part is a comparison between the computer model and results from measurements carried out in a part of the system of study. The second part is a sensitivity analysis of model predictions.
The fourth phase of the modelling process is the “model application.” At this stage of the process, the computer model attempts to answer the questions identified in the beginning of modelling process (see Figure 5-4).

The accuracy of the answers from the model depends on the complexity of the model. The original objectives of the model and the end-user’s competence put constraints on the model and limit the detail of input, for example. A program that is too complex for the intended user has requirements for input that do not improve the quality of the answers. On the other hand, a program that is too simple limits the quality of the answers. A balance must be struck between the required level of modelling detail and the appropriate level of answer accuracy that is required (Weinstein 1985).

A relationship between the level of detail and the size of model is presented in Figure 5-5 (Lundqvist 2005).

Very detailed models have been developed for specific studies of components in new technologies or research. In contrast, rules of thumb are used for conceptual studies and rough estimations of new systems. Today’s modelling abilities allow development of de-
tailed but simplified models once the concept has been decided upon.

5.3 Conceptual Model
The conceptual model includes a formulation of (i) the objectives of the model, (ii) the definition of the environment of the system, (iii) an identification of components of the system and (iv) their interconnections.

The main objective of this particular systems model is a user-friendly computer program that can calculate the total energy consumption of a supermarket with reasonable accuracy. The model should describe the properties of the different components in the system when different energy measures are compared. The investments and operational costs should be calculated through Life Cycle Cost with focus on the refrigeration system. The environmental impact from the refrigeration system should be characterized through the Total Equivalent Warming Impact measure. Another objective of the model selected is to have as little input data as possible and yet produce reliable results since the users of the program are designers and engineers from different companies involved in the implementation of new systems and energy efficiency measures in supermarkets. They are looking for rapid answers to their questions.

The various boundaries of the system “supermarket” are presented in Figure 5-6. The figure shows different subsystems in supermarket such as the HVAC system, refrigeration system, cabinet system and heating sources. Estimation of the energy requirement in a supermarket is based on the interrelatedness between the different subsystems and their energy demand.
Figure 5-7 shows a conceptual schema of the different subsystems in a supermarket and their interconnections.
Outdoor climate is an important factor for energy usage in supermarkets. It affects the indoor climate and the performance of the refrigeration system in several ways. The outdoor temperature, relative humidity, solar irradiation and wind speed all influence the indoor climate through the building envelope, the ventilation system and infiltration.

The heat or cooling gains through the building envelope, i.e. the walls, floor, roof and windows, depend on the thermal properties and the structure of the components. Specific heat capacity, density and thermal conductivity of the walls, floor, roof and windows affect the heat transfer and storage of energy in the building structure.

The ventilation system supplies air from the outside to the inside of the supermarket in order to provide comfort and acceptable indoor air quality. The air supplied to the supermarket is heated or cooled in the HVAC system according to the desired indoor conditions.

Infiltration is the air leakage from the outside to the inside of supermarket through exterior doors and windows. Infiltration is caused by the temperature difference between indoor and outdoor air and wind velocities.

In addition to heat or cooling gains, ventilation and infiltration through the building envelope, the indoor conditions in a supermarket are affected by lighting, equipment, occupants, cabinets and the control system. A conceptual model of the indoor climate is presented in Figure 5-8.
Lighting and equipment account for an important part of the total energy performance in supermarkets. Moreover, lighting and equipment emit heat that affects the heating or cooling loads.

People in supermarkets are customers and employees who give off heat, moisture and carbon dioxide. The heat and moisture emitted by people affect the heating and cooling requirements. The carbon dioxide influences the volume of air from the outside needed to cover the ventilation requirements. The number of occupants varies according to the profile of the supermarket with a daily and weekly pattern.

The cabinets exchange heat and moisture with the indoor air of the store. Cabinets require proper indoor conditions in order to reduce refrigeration loads and to avoid frost on products and coils. The cold air from the cabinets escaping to the surroundings affects heating and cooling requirements in the supermarket.

The control system of the supermarket manages the operating scheme of lighting, equipment, the HVAC system and set point temperatures when the supermarket is either open or closed and when the season is summer or winter.

The main objective of the HVAC system in supermarkets is to maintain the desired comfort level and indoor air quality conditions. The traditional HVAC system is a central plant located in the machine room consisting of heating and cooling systems, an air...
The typical heating system consists of an oil boiler, electric boiler or district heating system connected to the HVAC system by a heat exchanger. The cooling system is a chiller or district cooling connected to the HVAC system by a heat exchanger.

The air from the inside of the store is sometimes re-circulated to improve the energy performance. The quantity of re-circulated air is related to ventilation requirements for fresh air, which in turn is a function of the number of occupants in the supermarket.

A rotary heat exchanger is another type of heat recovery system. The heat contained in the air extracted from the store is reutilised to heat or cool the fresh air supplied to the indoor area.

The ventilation system usually has two fans: one for air supplied and another for air extraction. The energy demand of the fans is due to volume flow of the air and pressure drops in the distribution system.

The main goal of the refrigeration system in supermarkets is to provide storage for and display perishable food prior to sale. The refrigeration system has normally two temperature levels: medium temperature for preservation of fresh food and low temperature for frozen products. The refrigeration system consists basically of
four components: a condenser, an evaporator, a compressor and an expansion device. The performance of refrigeration machinery depends on the condensing and evaporating temperatures. The condensing temperature is affected by the outdoor conditions, while cooling loads from cabinets and cold rooms influence the evaporating temperature. High indoor temperature and relative humidity increase the cooling load in cabinets and storages. High indoor relative humidity enhances the defrost energy.

Heat recovery from condensers for heating the premises is one way to increase overall energy efficiency in supermarkets. An alternative to heat recovery is floating condensing pressure that improves the coefficient of performance of the refrigeration system during periods of low outdoor temperature.

5.4 Quantitative Model

The second phase of system analysis is the translation of the different components and subsystems defined in the conceptual model into a series of mathematical equations that are solved in a computer program. The mathematical equations and the necessary information for the different subsystems and components used in the integrated system model are explained in the following sub-chapters. The calculation procedures, communication with the user and the structure of simulations are presented in chapter 6.

5.4.1 Building Model

There are many principles for calculation of energy requirements in buildings. The complexity of the model varies according to the application and the objectives of the model. There are three fundamentally different types of building energy modelling techniques: steady state, quasi-steady state and dynamic. Steady state models assume that there is no energy storage during the time period or temperature condition under consideration. These models are based on time averaged temperature differences between the indoor and outdoor conditions, and all properties and variables are assumed constant for each calculation condition. The quasi-steady state model takes into consideration transient effects from weather, equipment use, occupancy profile and the storage and release of energy. The calculation period for quasi-steady state models can be any time interval, but one hour or a typical day from
each month are often used. Dynamic models are based on time intervals that are less than one hour in order to represent the continuous time variation of the properties of interest in the building (Hunn 1996).

ASHRAE suggests two methods for calculation of cooling and heating loads. The first method is the Heat Balance Method that can be viewed as four different processes. The first process is an outside face heat balance involving solar radiation, convective flux with outside air and conductive flux. The second process is a wall conduction process where two methods, a finite difference procedure and conduction transfer function, have been used widely to model the wall conduction. The third process is the inside heat balance concerning the radiation between different surfaces, the convective flux to zone air and the conductive flux. The fourth process is an air heat balance involving the convective part of surfaces, internal loads and air from infiltration and the HVAC system. The second method to calculate cooling and heating loads suggested is the Radiant Time Series Method (RTS), which is a simplification of the heat balance method. These methods are based on the assumption of steady periodic conditions, i.e. that weather, occupancy and internal loads are identical to those of preceding days. The RTS method is suitable for peak design load calculation, but it should not be used for annual energy calculations (ASHRAE 2001).

The model used to calculate the heating and cooling loads in CyberMart is an adaptation of a model suggested by Tor Helge Dokka (Dokka 2001). The model is based on the heat balances of room air, room surfaces and building structure. The model assumes that the variation in room temperature is negligible, that the temperatures of surfaces are the same and that thermal loads such as outdoor temperature, solar radiation and internal gains are constant during each time interval (1 hour in this case). The walls, floor and roof are modelled as equivalent RC-circuit elements where the construction’s internal heat capacity is concentrated in the middle of an accumulating layer in the internal part of the construction in contact with the room air. The sales and office areas in the supermarket were assumed to be one zone for the calculation of heating and cooling loads.
The room air heat balance is the heat gains to the room air minus the heat losses from the room air:

\[
\dot{Q}_{\text{con}} - H_{\text{ven}} \cdot (T_{\text{sup}} - T_{\text{room}}) - H_{\text{win}} \cdot (T_{\text{room}} - T_{\text{out}}) - H_{\text{con}} \cdot (T_{\text{room}} - T_{\text{surf}}) - \dot{Q}_{\text{gr}} = 0
\]  

(5.1)

\(\dot{Q}_{\text{con}}\) is defined as the convective part of the heat gains from people, equipment, lighting, cabinets and solar radiation through windows in the supermarket:

\[
\dot{Q}_{\text{con}} = \dot{Q}_{\text{solecon}} + \dot{Q}_{\text{percon}} + \dot{Q}_{\text{eqcon}} + \dot{Q}_{\text{lightcon}} - \dot{Q}_{\text{cabconv}}
\]  

(5.2)

\(H_{\text{ven}}\) is the specific loss for ventilation air flow, which is defined as the product of mass flow in the ventilation system and the specific heat capacity of air:

\[
H_{\text{ven}} = \rho_{\text{air}} \cdot c_{\text{p}_{\text{air}}} \cdot \dot{V}_{\text{ven}}
\]  

(5.3)

\(H_{\text{win}}\) is the specific losses for windows and infiltrations:

\[
H_{\text{win}} = H_{\text{win}} + H_{\text{inf}}
\]  

(5.4)

Where \(H_{\text{win}}\) is the product of the heat transfer coefficient and the area of the windows:

\[
H_{\text{win}} = \sum (U_{\text{win}} \cdot A_{\text{win}})
\]  

(5.5)

And \(H_{\text{inf}}\) is the product of the mass flow from infiltration and the specific heat capacity of air:

\[
H_{\text{inf}} = \rho_{\text{air}} \cdot c_{\text{p}_{\text{air}}} \cdot \dot{V}_{\text{inf}}
\]  

(5.6)

The specific loss of convection between the air and the surfaces, \(H_{\text{con}}\), is defined as the product of the heat transfer coefficient of convection and the area of the surfaces:

\[
H_{\text{con}} = \sum (h_{\text{con}} \cdot A)
\]  

(5.7)
The heat losses to the ground, \( Q_{gr} \), have been calculated according to International Standard ISO 13370 (ISO 13370 1998).

The room surfaces heat balance is defined as the heat gains minus the heat losses from the room surfaces:

\[
\dot{Q}_{\text{rad}} - H_a \cdot (T_{\text{surf}} - T_{\text{acc}}) + H_{\text{con}} \cdot (T_{\text{room}} - T_{\text{surf}}) = 0 \quad (5.8)
\]

\( \dot{Q}_{\text{rad}} \) is defined as the radiative part of the heat gains from people, equipment, lighting, cabinets and solar radiation through windows in the supermarket:

\[
\dot{Q}_{\text{rad}} = \dot{Q}_{\text{solrad}} + \dot{Q}_{\text{perrad}} + \dot{Q}_{\text{eqlrad}} + \dot{Q}_{\text{lighrad}} - \dot{Q}_{\text{cabrad}} \quad (5.9)
\]

\( H_a \) is the specific structure loss, which is defined as the heat transfer coefficient of the accumulating layer and the surface area:

\[
H_a = \sum (U_a \cdot A) \quad (5.10)
\]

The heat balance of the building structure is defined as the heat gains to the structure minus the accumulated heat in the structure minus the heat losses from the structure:

\[
H_a \cdot (T_{\text{surf}} - T_{\text{acc}}) - H_{\text{out}} \cdot (T_{\text{acc}} - T_{\text{out}}) = C_s \cdot \frac{dT_{\text{acc}}}{dt} \quad (5.11)
\]

\( H_{\text{out}} \) is the specific loss from the accumulating layer to the outside. It is defined as the sum of the products of the heat transfer coefficient from the accumulating layer to the outside, \( U_{\text{out}} \), and the surface area:

\[
H_{\text{out}} = \sum (U_{\text{out}} \cdot A) \quad (5.12)
\]
\( C_a \) is the total heat capacity of the building that it is defined as the sum of the products of the specific heat capacity, restricted to the accumulating layer of the construction, and the area of the walls, roof and floor:

\[
C_a = \sum (C'_a \cdot A)
\]  

(5.13)

\( C'_a \) is the product of the specific heat, density and thickness of the accumulating layer, \( dac \):

\[
C'_a = c_p \cdot \rho \cdot dac
\]  

(5.14)

The thickness of the accumulating layer is shown in Figure 5-11. Generally, the accumulating layer can be estimated with the following formula (Dokka 2001):

\[
dac = 1.95 \cdot \sqrt{\frac{\lambda}{\rho \cdot c_p}}
\]  

(5.15)

Figure 5-11: Thickness of the Accumulating Layer of a Wall
From equation (5.1)

\[ T_{room} = \frac{\dot{Q}_{air} + H_{con} \cdot T_{surf}}{H_{air}} \]  

(5.16)

Where

\[ \dot{Q}_{air} = \dot{Q}_{con} + H_{ven} \cdot T_{sup} + H_{winf} \cdot T_{out} - \dot{Q}_{gr} \]  

(5.17)

\[ H_{air} = H_{winf} + H_{con} + H_{ven} \]  

(5.18)

From equation (5.8)

\[ T_{surf} = \frac{\dot{Q}_{rad} + H_{con} \cdot \frac{\dot{Q}_{air}}{H_{air}} + H_{a} \cdot T_{acc}}{H_{s}} \]  

(5.19)

Where

\[ H_{s} = H_{con} + H_{a} - \frac{H_{con}^{2}}{H_{air}} \]  

(5.20)

Substituting equations (5.16) and (5.19) in equation (5.11)

\[ \frac{dT_{acc}}{dt} = \frac{H_{a}}{H_{s}} \left( \frac{\dot{Q}_{rad}}{H_{air}} + \frac{H_{con}}{H_{air}} \cdot \dot{Q}_{air} \right) + \frac{H_{out} \cdot T_{out}}{C_{a}} - \left( \frac{H_{a} + H_{out} - \frac{H_{con}^{2}}{H_{air}}}{C_{a}} \right) \cdot T_{acc} \]  

(5.21)

From equation (5.21)

\[ \frac{dT_{acc}}{dt} = \frac{T_{sa} - T_{acc}}{\tau} \]  

(5.22)
Where the stationary temperature of the structure $T_\infty$ is defined as:

$$T_\infty = \frac{\dot{Q}_{\text{acc}}}{H_{\text{acc}}}$$  \hspace{1cm} (5.23)

and

$$\dot{Q}_{\text{acc}} = \frac{H_s}{H_s} \left( \dot{Q}_{\text{rad}} + \frac{H_{\text{con}}}{H_{\text{air}}} \dot{Q}_{\text{air}} \right) + H_{\text{out}} \cdot T_{\text{out}}$$  \hspace{1cm} (5.24)

$$H_{\text{acc}} = H_a + H_{\text{out}} - \frac{H_s^2}{H_s}$$  \hspace{1cm} (5.25)

The time constant $\tau$ is defined as

$$\tau = \frac{C_s}{H_{\text{acc}}}$$  \hspace{1cm} (5.26)

The solution of the differential equation in (5.22) is

$$T_{\text{acc}}(t) = T_\infty + \left( T_{\text{acc}}(t) - T_\infty \right) \exp \left( \frac{-t}{\tau} \right)$$  \hspace{1cm} (5.27)

\textit{5.4.2 Outdoor Climate}

The outdoor climate data used in CyberMart has been generated from the software METEONORM (Remund 2005), which supplies climate data at any location in the world hourly.

METEONORM uses the average values from the period 1961-1990 for different weather stations according to the World Meteorological Institute (WMO) climate normal.

The parameters: air temperature, relative humidity, wind speed, beam and diffuse radiation on horizontal surface, height of sun, solar azimuth and cloud cover fraction, have been simulated in METEONORM for each locality in CyberMart.
The effects of convection and solar radiation on the external surfaces have been combined into an equivalent temperature (Hagentoft 2001), defined as:

\[ T_{eq} = T_{out} + \left( \frac{I_{sol} \cdot \alpha_{sol} + \left( T_{sky} - T_{out} \right) \cdot \alpha_{rad}}{\alpha_{e}} \right) \]  \hspace{1cm} (5.28)

Where \( \alpha_{sol} \) is the absorptivity for solar radiation, \( \alpha_{rad} \) is the radiative heat transfer coefficient and \( \alpha_{e} \) is the effective heat transfer coefficient that is the sum of the radiative and convective heat transfer coefficients.

The total solar irradiation, \( I_{sol} \), of a surface is calculated as the sum of the direct irradiation, \( I_{DN} \cos(\theta) \), plus the diffuse irradiation, \( I_{d\theta} \), plus the reflected solar irradiation, \( I_{refl} \) (ASHRAE 2001):

\[ I_{sol} = I_{DN} \cdot \cos(\theta) + I_{d\theta} + I_{refl} \]  \hspace{1cm} (5.29)

The angle of incidence, \( \theta \), between the normal of the different surfaces and the direct solar beam is given for vertical surfaces as:

\[ \cos(\theta) = \cos(\beta) \cdot \cos(\gamma) \]  \hspace{1cm} (5.30)

and for horizontal surfaces as

\[ \cos(\theta) = \sin(\beta) \]  \hspace{1cm} (5.31)

\( \beta \) is the solar altitude and \( \gamma \) is the surface-solar azimuth. The surface-solar azimuth is defined as the difference between the solar azimuth and the surface azimuth. The influence of cloudy days on surfaces has been calculated according to (Sandberg 1973).

The radiative heat transfer coefficient (Petersson 2001) is given by:

\[ \alpha_{r} = 4 \cdot \varepsilon \cdot \sigma \cdot \left( \left( T_{sky} + T_{out} \right) / 2 \right)^{3} \]  \hspace{1cm} (5.32)
Where $\varepsilon$ is the emissivity of the material and $\sigma$ is the Stefan-Boltzman constant. The sky temperature has been calculated according to (Hagentoft 2001) as:

$$T_{\text{sky}} = 1.2 \cdot T_{\text{out}} - 14 \quad \text{For horizontal surface, clear sky} \quad (5.33)$$

$$T_{\text{sky}} = 1.1 \cdot T_{\text{out}} - 5 \quad \text{For vertical surface, clear sky} \quad (5.34)$$

$$T_{\text{sky}} = T_{\text{out}} \quad \text{For cloudy sky} \quad (5.35)$$

The convective heat transfer coefficient (Hagentoft 2001) is given by:

$$\alpha_c = 5 + 4.5 \cdot u - 0.14 \cdot u^2 \quad (5.36)$$

Where $u$ is the wind speed in m/s.

5.4.3 HVAC System

The different components of the HVAC system, simulated in CyberMart, are shown in Figure 5-12. These components are: a rotary heat exchanger, a bypass for re-circulation of return air, a coil for air cooling and two coils for air heating - one for heat recovery from the condenser and the other for auxiliary heating.
The temperature and humidity ratio are calculated for each component and conditions of the air supplied are dependent on the outdoor conditions and indoor requirements. The mass flow of fresh air is dependent on the concentration of carbon dioxide in the supermarket. The supplied and return mass flow rates have been assumed equivalent. The thermo-physical properties of the air have been calculated according to (Ekroth 1994) and (Hansen 1997).

The humidity ratio and the concentration of carbon dioxide in the room have been calculated from a mass balance of air in the room (Dokka 2001). For the humidity ratio, X:

\[
\dot{m}_{\text{sup}} \cdot (X_{\text{sup}} - X_{\text{room}}) - \dot{m}_{\text{inf}} \cdot (X_{\text{room}} - X_{\text{out}}) + g_w = V \cdot \frac{dX_{\text{room}}}{dt}
\]

Where \( g_w \) is the generation of water vapor from people, cabinets and equipment and \( \frac{dX_{\text{room}}}{dt} \) is the accumulation of humidity in the supermarket.

And for the concentration of carbon dioxide, \( CC \),

\[
\dot{m}_{\text{sup}} \cdot (CC_{\text{sup}} - CC_{\text{room}}) - \dot{m}_{\text{inf}} \cdot (CC_{\text{room}} - CC_{\text{out}}) + g_{\text{CO2}} = V \cdot \frac{dCC_{\text{room}}}{dt}
\]

Where \( g_{\text{CO2}} \) is the generation of carbon dioxide from people and \( \frac{dCC_{\text{room}}}{dt} \) is the accumulation of carbon dioxide in the supermarket.

The conditions of the air after the rotary heat exchanger are dependent on the temperature efficiency of the heat exchanger. The temperature and humidity ratio of the air after the rotary heat exchanger are calculated as:

\[
T_2 = T_i + \eta_{\text{hex}} \cdot (T_i - T_r)
\]

\[
X_2 = X_i + \eta_{\text{hex}} \cdot (X_i - X_r)
\]

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Where $\eta_{\text{RHEX}}$ is the temperature efficiency of the rotary heat exchanger and $T_1$, $T_2$, $T_7$, $X_1$, $X_2$, and $X_7$ are temperatures and humidity ratios before and after the rotating heat exchanger according to Figure 5-12.

The mass flows of the re-circulation and fresh air are determined from the mass concentration of carbon dioxide in the supermarket. The chosen limit for CO$_2$ concentration in the supermarket is 800 ppm. The conditions of the air, after the re-circulation point, are calculated from a heat balance at the mix point:

\[
\dot{m}_{\text{sup}} \cdot h_3 = \dot{m}_{\text{fresh}} \cdot h_2 + \dot{m}_{\text{recir}} \cdot h_7
\]

\[
\dot{m}_{\text{sup}} \cdot X_3 = \dot{m}_{\text{fresh}} \cdot X_2 + \dot{m}_{\text{recir}} \cdot X_7
\]

Where $\dot{m}_{\text{sup}}$, $\dot{m}_{\text{fresh}}$, $\dot{m}_{\text{recir}}$ are the mass flow of supplied, return and fresh air and $h_2$, $h_3$, $h_7$, $X_2$, $X_3$, and $X_7$ are temperatures and humidity ratios before and after the mixing point according to Figure 5-12.

Influence of the fan on air temperature, after the mixing point, has been calculated from:

\[
\dot{Q}_{\text{fan}} = \dot{m}_{\text{sup}} \cdot c_{p_{\text{air}}} \cdot \Delta T = \dot{m}_{\text{sup}} \cdot c_{p_{\text{air}}} \cdot (T_4 - T_3)
\]

and $X_4 = X_3$

Where the increase in temperature over the fan, $\Delta T$, has been assumed equal to 1°C, and $T_3$, $T_4$, $X_3$ and $X_4$ are temperatures and humidity ratios before and after the fan.

The temperature and humidity ratio of the air after the cooling coil, $T_5$ and $X_5$, are calculated according to the cooling requirement of the supermarket, the temperature of the coil and the maximum cooling coil capacity given by the user in the program.

The cooling load of the air conditioning system consists of two parts: a sensible and a latent part.
When the air dew point temperature before the cooling coil is lower than the coil temperature, the cooling load of the air conditioning system is defined as:

\[ Q_{cl} = \dot{m}_{sup} \cdot (c_{p,air} \cdot (T_5 - T_4) + r_0 \cdot (X_5 - X_4)) \]  \hspace{1cm} (5.44)

Where \( r_0 \) is the heat of evaporation for water at 0°C, \( \dot{m}_{sup} \cdot c_{p,air} \cdot (T_5 - T_4) \) is the sensible part and \( \dot{m}_{sup} \cdot r_0 \cdot (X_5 - X_4) \) is the latent part of the cooling load.

When the air dew point temperature before the cooling coil is equal to the coil temperature, then the cooling load is dependent only on the sensible part:

\[ Q_{cl} = \dot{m}_{sup} \cdot c_{p,air} \cdot (T_5 - T_4) \]  \hspace{1cm} (5.45)

and \( X_5 = X_4 \)

The temperature of the air, \( T_6 \), after the heating coils is calculated according to the heating requirement of the supermarket. The heating load of the air conditioning system is calculated as:

\[ Q_{heatingload} = \dot{m}_{sup} \cdot c_{p,air} \cdot (T_6 - T_5) \]  \hspace{1cm} (5.46)

and \( X_6 = X_5 \)

5.4.4 Refrigeration System

5.4.4.1 Indirect System Modelling

The components of the indirect system that are simulated in CyberMart are presented in Figure 5-13.

The most important device in the model is the chiller that is represented by four components: condenser, expansion valve, evaporator and compressor. The model of the chiller is based on (Bourdouxhe 1994). The components that directly affect the operating conditions of the condenser and the evaporator are cabinets and dry coolers.
5.4.4.1 Cabinets and Deep-Freeze Cabinets

The performance data of the cabinets and deep-freeze cabinets in CyberMart have been taken from manufacturer data sheets. The average air temperature, inlet and return air temperatures, evaporating temperature, electrical data for fans, heating wires, defrost heaters and light, coil volume, diameter of tubes and refrigeration loads at 22°C – 65% RH and at 25°C – 60% RH for each cabinet have been put into a database. The refrigeration loads in display cases are dependent on indoor conditions in the supermarket; a higher indoor temperature and relative humidity increase the cooling demand and the energy requirement. An energy balance of an open vertical cabinet is shown in Figure 5-14 where heat losses from infiltration, radiation, conduction, lighting, the fan, heating wires and defrost are presented. Figure 5-14 also shows the interaction between the indoor conditions in the supermarket and the interior conditions in vertical cabinet. The heat losses dependent on the ambient conditions in supermarkets are from infiltration, radiation, conduction and defrost. The losses from infiltration are about 66% of the total refrigeration load at 25°C indoor temperature.
The effect of indoor temperature on the refrigeration load in vertical display cases is presented in Figure 5-15. The values in the diagram have been calculated according to the heat balance in Figure 5-14 at different indoor temperatures (Fahlén 1999).
The diagram shows that the refrigeration load will be halved at 14°C. At the same temperature in the cabinet and in the surroundings, the refrigeration load is equivalent to heat from the illumination, heating wires and fan, which is about 18% of the total refrigeration load at 25°C.

The effect of the indoor temperature on the refrigeration load in a horizontal deep-freeze cabinet has been calculated according to the same reasoning as for vertical display cases.

An energy balance of a horizontal cabinet is shown in Figure 5-16 where heat losses from infiltration, radiation, conduction, lighting, the fan, heating wires and defrost are presented. The heat losses dependent on the ambient conditions in supermarkets are from infiltration, radiation, conduction and defrost. The losses from radiation are about 46% of the total refrigeration load at 25°C indoor temperature.

At an indoor temperature of 25°C, the refrigeration load is equivalent to 100%. At the same temperature in the cabinet and the surroundings, the refrigeration load is equivalent to the heat from heating wires, defrost and the fan, which is about 20% of the total refrigeration load at 25°C.

Figure 5-16: Energy Balance of a Horizontal Display Case
The influence of indoor relative humidity on the refrigeration load of display cases has been calculated with a correction factor TP according to (Howell 1993). The correction factor TP is a relation between the refrigeration load at a selected store relative humidity (RH) and the refrigeration load at 55% store relative humidity. The factor TP has been defined, when the display case temperature TC is known, as:

$$TP = 1 - (55 - RH) \cdot [D + E \cdot TC + F \cdot TC^2 + G \cdot TC^3] \quad (5.47)$$

The coefficients D, E, F and G for vertical display cases have been defined by (Howell 1993) as:

D = 7.3867 E-3

E = 6.510 E-5

F = -4.6164 E-7

G = 7.2405 E-8

And the coefficients D, E, F and G for horizontal display cases are

D = 6.577 E-3
E = 4.8668 E-5
F = 5.3562 E-7
G = 3.7393 E-9

5.4.4.1.2 Cold Storage

The capacity demand for cold storage is due to four factors: heat transmission, exchange of air, cooling or freezing of products and internal heat generation (Granryd 2003).

Heat transmission through walls, floor and ceiling is dependent on the overall heat transfer coefficient and the temperature difference between the room and the surroundings. The heat transmission has been defined as:

\[
\dot{Q}_{tr} = \sum \left( U_{crn} \cdot A_{crn} \cdot (t_{sur} - t_{cr}) \right)
\]  

(5.48)

The exchange of air in cold rooms depends on the frequency of door openings and the size of the room. The exchange of air increases the refrigeration load of the room. The influence of incoming air in the room can be calculated as:

\[
\dot{Q}_{airex} = \dot{V}_{airex} \cdot \rho \cdot \left( h_{sur} - h_{room} \right)
\]  

(5.49)

\( \dot{V}_{airex} \) is an average volume flow of incoming air that is defined as (Granryd 2003):

\[
\dot{V}_{airex} = V_{room} \cdot \frac{nd}{24 \cdot 3600}
\]  

(5.50)

nd is the number of air exchanges in the room per 24 hours.

Temperatures and the frequency of door openings influence the number of air exchanges. Results from experiments are presented in Table 5-2 (Granryd 2003).
The refrigerant load required to cool or freeze products is dependent on the mass flow rate and the enthalpy difference of the products:

\[ \dot{Q}_{\text{pr}} = \dot{m}_{\text{pr}} \cdot \left( h_{\text{prin}} - h_{\text{prcr}} \right) \]  

The enthalpy difference for freezer rooms has been assumed to be 45 [kJ/kg], which is the average between the enthalpies of different products at temperatures -15°C and -18°C. In the same manner, the enthalpy difference for the cold room has been assumed to be 55 [kJ/kg], which is the average between the enthalpies of different products at temperatures 17°C and 1°C. The mass flow has been assumed to be 20 kg/m³ per 24 hours (Bäckström 1970) for cold rooms and 15 kg/m³ per 24 hours for freezer rooms.

Internal heat generation from lighting and people also affects the refrigeration load of the cold room. The heat generated by lighting has been assumed to be 15 W/m² and the heat from people to be 200 W.

### 5.4.4.1.3 Dry Cooler and Other Heat Exchangers

The dry cooler and other heat exchangers have been modelled as counter flow heat exchangers. The effectiveness of the counter flow heat exchanger is defined in (Incropera 1996) as:

\[ \varepsilon = \frac{1 - e^{-\text{NTU}(1-C_r)}}{1 - C_r \cdot e^{-\text{NTU}(1-C_r)}} \]  

NTU is the number of transfer units defined for the counter flow heat exchanger as:

\[ \text{NTU}_{\text{HEX}} = \frac{U A_{\text{HEX}}}{C_{\text{min}}} \]
$C_{\text{min}}$ is defined as the minimum of the hot and cold fluid heat capacity rates. The cold heat capacity rate is the product of the mass flow and the heat capacity of the cold fluid:

$$C_c = \dot{m}_c \cdot c_p \tag{5.54}$$

The hot heat capacity rate is the product of the mass flow and the heat capacity of the hot fluid:

$$C_h = \dot{m}_h \cdot c_p \tag{5.55}$$

The heat capacity ratio, $C_r$, in equation (5.52) is defined as the ratio between the minimum and maximum fluid heat capacity rates between the hot and cold fluid heat capacity rates:

$$C_r = \frac{C_{\text{min}}}{C_{\text{max}}} \tag{5.56}$$

### 5.4.4.1.4 Pressure Drop and Pumps

The pressure drops in the secondary refrigerant loop and in the dry cooler fluid loop for both the medium and low temperature units are dependent on the type of fluid used and on the thermophysical properties of the fluid. The Reynolds number, $Re$, is the decisive parameter to decide the type of flow.

The Reynolds number is defined as

$$Re = \frac{\omega \cdot di}{\nu} \tag{5.57}$$

Values of the Reynolds number above 2000 - 2300 generally mean turbulent flow. $Re$ lower than this value mean laminar flow. The pressure drop can be calculated with the following equation (Melinder 1997):

$$\Delta pf = f_i \cdot \rho \cdot w^2 \cdot L \frac{\omega}{di} \tag{5.58}$$
The friction factor, $f_t$, for turbulent fluid is calculated according to (Gnielinski 1976):

$$f_t = \frac{0.5}{\left(0.79 \cdot \ln(Re) - 1.64\right)^{0.5}}$$

(5.59)

And for laminar flow the friction factor is defined as

$$f_l = \frac{32}{Re}$$

(5.60)

The condensers and evaporators in indirect systems are often plate heat exchangers. The pressure drops in the brine side of the evaporator and the dry cooler fluid side of the condenser have been estimated according to the following equation (Hewitt 1998):

$$\Delta P_{\text{phex}} = \frac{2 \cdot f_{\text{phex}} \cdot N_p \cdot \dot{G}^2 \cdot L}{\rho \cdot d e}$$

(5.61)

The friction factor, $f_{\text{phex}}$, is calculated from performance data.

The pump power has been calculated by

$$p_{\text{pump}} = \frac{\Delta P_{\text{tot}} \cdot \dot{V}}{\eta_{\text{pump}}}$$

(5.62)

$$\Delta P_{\text{tot}} = \Delta P_f + \Delta P_{\text{phex}} + \Delta P_{\text{cabinet}}$$

(5.63)

$\Delta P_{\text{cabinet}}$ is the pressure drop in the cabinet that has been calculated using equation (5.58) and data from manufacturers.

The pressure drop of carbon dioxide used in the low temperature unit as secondary refrigerant has been calculated according to equation (5.58). The friction factor was calculated according to equation (5.59). For two-phase flow, the homogenous model, which assumes that vapor and liquid velocities in two-phase flow

---

1 In the original paper by Gnielinski, the equation is quoted as G.K. Filonenko.
are equal, was used to calculate the pressure drop (Spatz 2003). In the homogenous model, the pressure drop is calculated as in single-phase flow but density and viscosity are obtained by:

\[
\rho_{\text{hom}} = \frac{1}{\frac{x_g}{\rho_g} + \frac{x_l}{\rho_l}}
\]

(5.64)

\[
\mu = \frac{1}{\frac{x_g}{\mu_g} + \frac{x_l}{\mu_l}}
\]

(5.65)

### 5.4.4.1.5 Condenser and Evaporator Model

The condenser and evaporator used in CyberMart for indirect systems are plate heat exchangers. The refrigeration side of the evaporator and condenser has been assumed to have a constant temperature. The properties of the secondary refrigerant and the coolant fluid have been taken from (Melinder 1997) and data from manufacturers. The effectiveness of the condenser and evaporator is defined as (Incropera 1996):

\[
\varepsilon = 1 - e^{(-\text{NTU})}
\]

(5.66)

\text{NTU} is the number of transfer units defined for the evaporator as:

\[
\text{NTU}_{\text{evap}} = \frac{UA_{\text{evap}}}{\dot{m}_{br} \cdot cP_{br}}
\]

(5.67)

And for the condenser as

\[
\text{NTU}_{\text{cond}} = \frac{UA_{\text{cond}}}{\dot{m}_{coolfl} \cdot cP_{coolfl}}
\]

(5.68)

The values of \(UA_{\text{evap}}\) and \(UA_{\text{cond}}\) have also been taken from manufacturer performance data and assumed to be constant in the model during the simulation. The values of \(UA_{\text{evap}}\) and \(UA_{\text{cond}}\) in the database correspond to the refrigerant R404A and Propylene Glycol at 35% concentration as brine or coolant fluid in the
evaporator or the condenser respectively. In order to generalize the behavior of the heat exchanger, the following procedure has been implemented to calculate a new UA value for secondary refrigerants or dry cooler fluids other than Propylene Glycol:

\[
\frac{1}{U_{\text{evap}}} = \frac{1}{\alpha_{\text{ref}}} + \frac{d}{\lambda_{\text{met}}} + \frac{1}{\alpha_{\text{br}}} \quad (5.69)
\]

The heat transfer coefficient of the refrigerant \( \alpha_{\text{ref}} \) has been assumed constant for the calculation of the new UA value. The heat transfer coefficient \( \alpha_{\text{br}} \) of the new brine has been calculated with the following relation defined in (Hewitt 1998) as:

\[
\alpha_{\text{br}} = \frac{N_{u} \cdot \lambda_{\text{br}}}{d \cdot e} \quad (5.70)
\]

Nu is the Nusselt number defined for the plate heat exchanger and turbulent flow as (Hewitt 1998):

\[
N_{u} = 0.2 \cdot Re^{n_{h}} \cdot Pr^{-0.4} \cdot \left( \frac{\mu_{\text{br}}}{\mu_{wa}} \right)^{0.17} \quad (5.71)
\]

The exponent \( n_{h} \) was calculated from performance data.

And for laminar flow as

\[
N_{u} = 0.29 \cdot (Re \cdot Pr)^{0.4} \cdot \left( \frac{\mu_{\text{br}}}{\mu_{wa}} \right)^{0.1} \quad (5.72)
\]

The Prandtl number, Pr, is defined as

\[
Pr = \frac{\mu_{\text{br}} \cdot c_{p_{\text{br}}}}{\lambda_{\text{br}}} \quad (5.73)
\]

Values of \( UA_{\text{cond}} \) and \( UA_{\text{evap}} \) for two fluids other than propylene glycol have been taken from manufacturer performance data to validate the calculation method of new UA values. The values
The UA values of two evaporators with capacities of 50kW and 95kW and two condensers with capacities of 67kW and 134kW were calculated. For the evaporators, the differences between the manufacturers’ and calculated values, using pekasol, are 3.3% and 3.1%, and using ethanol, are 1.4% and 2.2%. For the condensers, the differences between the manufacturers’ and calculated values, using pekasol, are 6.5% and 5.3%, and using ethanol, are 0.5% and 0.6%.

In general the differences between the manufacturers’ and calculated values are lower than 6.5%, which is a satisfactory result for the calculation method of new UA values.

The influence of UA values on the refrigeration system and on the total energy performance in the supermarket is estimated in Chapter 7.

<table>
<thead>
<tr>
<th>Table 5-3: Comparison of Calculated UA Values with Those of Manufacturers</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>Evap 50kW</td>
</tr>
<tr>
<td>A= 8.7m²</td>
</tr>
<tr>
<td>U</td>
</tr>
<tr>
<td>UA</td>
</tr>
<tr>
<td>Error</td>
</tr>
<tr>
<td>Evap 95kW</td>
</tr>
<tr>
<td>A= 13.2m²</td>
</tr>
<tr>
<td>U</td>
</tr>
<tr>
<td>UA</td>
</tr>
<tr>
<td>Error</td>
</tr>
<tr>
<td>Condenser</td>
</tr>
<tr>
<td>---------------</td>
</tr>
<tr>
<td>67kW</td>
</tr>
<tr>
<td>A= 6.5m²</td>
</tr>
<tr>
<td>U</td>
</tr>
<tr>
<td>UA</td>
</tr>
<tr>
<td>Error</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Condenser</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>134kW</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A= 15.5m²</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>U</td>
<td>1202</td>
<td>1337</td>
<td>1266</td>
<td>1183</td>
</tr>
<tr>
<td>UA</td>
<td>18583</td>
<td>20670</td>
<td>19572</td>
<td>18289</td>
</tr>
<tr>
<td>Error</td>
<td>5.3%</td>
<td>0.6%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### 5.4.4.1.6 Compressor Model

The compressor model has been developed from performance data. Compressor manufacturers have developed different software where compressor power, refrigeration capacity and mass flow are given for different evaporating and condensing temperatures. The compressor power and refrigeration capacity data has been introduced into a database as two matrices of three rows and three columns equivalent to condensing temperatures of 30°C, 40°C and 50°C and evaporating temperatures of -15°C, -10°C and -5°C. The matrix of compressor 2N-5.2Y is shown in Figure 5-18.

The compressor power is dependent on the variation of the indoor and outdoor climate, which directly or indirectly will influence the refrigeration load. The capacity and COP of the refrigeration system is also dependent on the boundary conditions of the condenser. The simulation of the compressor power is done by interpolation and/or extrapolation between the condensing and evaporating temperatures.
5.4.4.1.7 Defrost

The conditions of the air in a cold storage or in a display cabinet affect the refrigeration capacity of the coils. At surface temperatures lower than the dew point temperature of the air, the water vapor in the air will condense on surfaces, and at surface temperature lower than 0°C, there will be frost deposited on the surface. The frost can have different frost depth, which depends on temperature and humidity in the air.

The rate of frost deposit on the surface, $m_{\text{frost}}$, is defined as (Granryd 2003):

$$ m_{\text{frost}} = \frac{\dot{Q}_d}{r + S} \quad (5.74) $$

Where $S$ is the latent heat of fusion of water from liquid to solid ice, $r$ is the latent heat of vaporization and $\dot{Q}_d$ is the latent heat contribution of the total refrigeration capacity.

$$ \dot{Q}_{\text{tot}} = \dot{Q}_d + \dot{Q}_c \quad (5.75) $$

The latent heat is the heat due to mass transfer between the surrounding air and the surface and is defined as:

$$ \dot{Q}_d = \alpha_d \cdot A \cdot \Delta T \quad (5.76) $$
Where $\alpha_d$ is the diffusion heat transfer coefficient due to the mass transfer, which is defined for surface temperature higher than 0°C as (Granryd 2003):

$$\alpha_d \equiv 1520 \cdot \frac{\Delta p_{\text{H}_2\text{O}}}{\Delta T} \cdot \alpha_c$$

(5.77)

And for surface temperature lower than 0°C as

$$\alpha_d \equiv 1740 \cdot \frac{\Delta p_{\text{H}_2\text{O}}}{\Delta T} \cdot \alpha_c$$

(5.78)

$\Delta p_{\text{H}_2\text{O}}$ is the difference in partial pressure between the surrounding air and the surface.

The total refrigeration capacity can also be defined as:

$$\dot{Q}_{\text{tot}} = (\alpha_d + \alpha_c) \cdot \Lambda \cdot \Delta T$$

(5.79)

From equation (5.79) and (5.76), it is possible to define the latent heat as a function of the total refrigeration capacity:

$$\dot{Q}_d = \dot{Q}_{\text{tot}} \frac{(\alpha_d / \alpha_c)}{(1 + \alpha_d / \alpha_c)}$$

(5.80)

From equation (5.80) and (5.74), it is possible to calculate the rate of frost deposit on the surface as a function of the total refrigeration capacity:

$$\dot{m}_{\text{frost}} = \dot{Q}_{\text{tot}} \frac{(\alpha_d / \alpha_c)}{(r + S)(1 + \alpha_d / \alpha_c)}$$

(5.81)

The influence of the indoor relative humidity on defrost in display cases has been calculated with a correction factor DP according to (Howell 1993). The correction factor DP is a relation between the moisture exchange across the air curtain of the cabinet at a selected store relative humidity (RH) and moisture exchange across the air curtain of the cabinet at 55% store relative humidity.
The factor $DP$ has been defined, when the display case temperature $TC$ is known, as:

$$DP = 1 - (55 - RH) \cdot [D + E \cdot TC + F \cdot TC^2 + G \cdot TC^3]$$  \hspace{1cm} (5.82)

The coefficients $D$, $E$, $F$ and $G$ for vertical display cases have been defined as (Howell 1993):

- $D = 1.9030 \times 10^{-2}$
- $E = 5.4239 \times 10^{-5}$
- $F = -3.4944 \times 10^{-7}$
- $G = 1.2786 \times 10^{-7}$

And the coefficients $D$, $E$, $F$ and $G$ for horizontal display cases are

- $D = 1.9032 \times 10^{-2}$
- $E = 1.8607 \times 10^{-5}$
- $F = 1.4947 \times 10^{-6}$
- $G = 7.5506 \times 10^{-8}$

5.4.4.2 Direct System Modelling

The traditional refrigeration system design in supermarkets is the direct system. The characteristics of the system are long lines of refrigerant between the compressor, evaporator and condenser, which affect the total refrigerant charge in the system and the potential for refrigerant leakage.

Figure 5-19 presents the components of the direct system that are simulated in CyberMart.
5.4.4.2.1 Condenser and Evaporator Model

The condensers are of the air-cooled condenser type and the evaporators are the fincoils in cabinets and in the air coolers. The refrigeration side of the evaporator and condenser has been assumed to have constant temperature.

The effectiveness of the condenser and evaporator has already been defined in equation (5.66).

The number of transfer units, NTU, is defined for the evaporator as (Incropera 1996):

\[ NTU_{\text{evap}} = \frac{UA_{\text{evap}}}{m_{\text{air}} \cdot c_{p_{\text{air}}}} \]  
(5.83)

And for the condenser as

\[ NTU_{\text{cond}} = \frac{UA_{\text{cond}}}{m_{\text{air}} \cdot c_{p_{\text{air}}}} \]  
(5.84)
The values of $U A_{\text{cond}}$ and $U A_{\text{evap}}$ have been taken from manufacturer performance data.

### 5.4.4.2.2 Pressure Drop in Gas and Liquid Lines

The pressure drops in vapor and liquid lines in the refrigeration system can be calculated from equation (5.58).

The friction factor can be found as a function of the Reynolds number in equation (5.59) and (5.60). It is often convenient to express the pressure drop as an equivalent temperature drop for the refrigerant in order to see the effect of the pressure drop on the evaporating temperature.

Granryd has used the Clapeyron equation to convert the pressure drop in Pa to an equivalent change in saturation temperature (Granryd 2003):

$$\Delta t'' = T_{\text{abs}} \cdot \frac{(v'' - v')}{r} \cdot \Delta p \quad (5.85)$$

### 5.4.5 Life Cycle Cost (LCC)

The Life Cycle Cost is a method that evaluates present and future costs from the investments in and operation and maintenance of a project over its entire life cycle. To calculate the LCC, it is necessary to compute the present value of all costs occurring during the period of study (usually related to the lifespan of the project).

The costs during the life cycle are of two types: single costs and annually recurring costs. The single costs occur one or more times during the study period, for example, investment cost, repair cost, etc. The annually recurring costs occur regularly every year, for example, energy cost, annual maintenance cost, etc.

The present value of single costs can be calculated using the Single Present Value formula:

$$SCP_n = \frac{SC_n}{\left[1+(i-pc)\right]^n} \quad (5.86)$$
Where \( SCP_n \) is the present value of a single cost, \( SC_n \), after \( n \) years. \( SC_n \) is the single cost after \( n \) years and \( pe \) is the inflation or the price escalation.

The present value of annually recurrent costs can be calculated using Uniform Present Value formula:

\[
RCP_n = RC_n \cdot \frac{1 - \left[1 + (i - pe)^{-n}\right]}{(i - pe)} \tag{5.87}
\]

Where \( RCP_n \) is the present value of an annually recurrent cost, \( RC_n \), and \( RC_n \) is the annually recurrent cost.

The LCC formula (Nordell 2001) used in CyberMart is

\[
LCC_{TOTAL} = Inv + LCC_{Energy} + LCC_{OM&R} + LCC_{Environment} + LCC_{others} \tag{5.88}
\]

Where \( Inv \) is the present value of investment costs that include, for instance, costs of products, installation, administration, etc. \( LCC_{Energy} \) is the present value of annual energy cost. \( LCC_{OM&R} \) is the present value of non-fuel operating, maintenance and repair cost. \( LCC_{Environment} \) is the present value of environmental costs. \( LCC_{Others} \) is the present value of other costs calculated.

5.4.6 Total Equivalent Warming Impact (TEWI)

The TEWI calculation of a refrigeration system used in CyberMart is based on the following equation:

\[
TEWI = M_{losses} \cdot N \cdot GWP_{ref} + RC \cdot E \cdot N \tag{5.89}
\]

Where \( M_{losses} \) is the refrigerant leakage, \( N \) is the period of study in years, \( GWP_{ref} \) is the global warming potential of the refrigerant, \( RC \) is the regional conversion factor, and \( E \) is the annual energy consumption of the equipment.
5.5 Conclusions

The supermarket sector has more or less used the trial and error approach to implement and evaluate new ideas and concepts such as indirect systems, heat recovery, floating condensing, etc. A computer model is needed to estimate the impact of energy efficient measures in different subsystems in the supermarket.

An overview of six different models that calculate energy performance in supermarkets was presented in the first part of this chapter. None of the computer programs described can compare different refrigeration system designs with indirect systems such as the completely, partially or cascade indirect system, nor can they calculate TEWI or LCC.

For these reasons a systems model, CyberMart, is needed in order to predict and evaluate the introduction of new concepts and ideas in supermarkets, to compare different refrigeration system designs with indirect systems and to calculate TEWI and LCC.

Four phases have been identified in the modelling process: a conceptual model, a quantitative model, an evaluation of the model and an application of the model. Two phases, the conceptual model and the quantitative model, have been presented in this chapter.

Based on the objectives of the conceptual model, the different subsystems in a supermarket such as outdoor climate, indoor climate, infiltration, HVAC system, cabinets, storages, refrigeration system, lighting, equipment, occupants, heat recovery system, defrost, control system and their interconnections are presented.

In the quantitative model, the different components and subsystems defined in the conceptual model are transformed into a series of mathematical equations that are solved by CyberMart, which is presented in the next chapter.
The calculation process of the different modules in CyberMart, presented in Figure 6-1, is explained in this chapter. CyberMart has been built in Delphi, which is an application development product for writing Windows applications. Delphi is an object-oriented programming language with a database function.
others are kept in a database that is used by different modules in the program in order to simulate the variations in the refrigeration system under different conditions. The supermarket in Sala is selected as an example to show the different windows, input data, interface with users, calculation proceedings and results from the model. A list with the input data required in the program is presented in Appendix A.

6.1 Main Window

The main window has a menu with three different pull down menus: File, Window and Help, a toolbar with four different buttons: Open, Save, Save as and Help, and a box with the project name, the reference and OK button (see Figure 6-2).

![Figure 6-2: Main Window](image)

The File menu has four different options: Open, Save, Save as and Exit. These options work as in other standard Windows applications.

The Window menu has nine different options: System, Input, Building, Pressure Drop Med., Pressure Drop Low, Energy, Refrigeration System,
These options give the user the possibility to move quickly between the different opened windows in the program.

In the main window there are eight buttons. The first one opens the window for Building. The other seven buttons are equivalent to the seven different refrigeration system designs, which are possible to simulate and compare in CyberMart:

- Direct System
- Completely Indirect System
- Partially Indirect System
- Cascade system A
- Cascade system B
- Parallel System with mechanical sub-cooling
- District cooling

6.2 Building

The input data required in the Building window (see Figure 6-3) is the location, building and HVAC conditions.

![Figure 6-3: Building Window](image)
In the Building window there are eight buttons: Main Program, Climatic Conditions, Building Envelope, Ventilation, Heat Sources, Opening hours, Heating Air Conditioning and Energy Calculation.

The window for climatic conditions has two diagrams, a scrollbar list with cities and a button Calculate. The user selects the locality of the supermarket in the scrollbar. The button Calculate starts the calculation of the climate data according to the flow diagram in Figure 6-4.

![Flow Diagram for Calculation of Climatic Conditions](Image)

With the locality as input, the program first retrieves a text file with the hourly air temperature, relative humidity, wind speed, beam and diffuse radiation on a horizontal surface, height of sun, solar
azimuth and cloud cover fraction calculated in METEONORM. With the values from the text file, the program calculates hourly values of humidity ratio, solar radiation, sky temperature and equivalent temperature. The program also calculates daily average values of outdoor temperature, relative humidity and humidity ratio. The values of climatic conditions are saved in different vectors. When the calculation has finished, a dialog window with four buttons: Year, 15 Mars, 15 Jul. and 15 Dec. will appear (see Figure 6-5). In the diagram to the left, measurements of air temperature, relative humidity and humidity ratios from the locality selected are presented. The diagram to the right shows results of the solar radiation during one year and on 15 Mars, 15 Jul. and 15 Dec.

The Building Envelope window is an input data window for the dimensions of the supermarket, walls, roof, floor and windows. The required input data is length, width, height and sales area of the supermarket; the area, direction (S, SE, E, NE, N, NW, W, SW), side and type of construction for the walls, roof and floor; and area, type and shield of the windows.

The types of construction modelled in the program are heavy, medium and light. The heavy wall has, from the inside: 120 mm concrete, 150 mm insulation and 80 mm concrete. The medium wall...
has, from the inside: 10 mm stucco, 200 mm light concrete, 150 mm insulation and 10 mm stucco. The light wall has: 25 mm gypsum, 200 mm cross bars, 150 mm insulation and 120 mm brick. The radio button *edge insulation* opens a new window where it is possible to select between horizontal or vertical edge insulation of the ground.

The types and shield of windows in the program have been taken from (Dokka 1995). Figure 6-6 shows the input values from the Building Envelope window. The input values for the supermarket in Sala have been obtained from (Bjerkhög 2004).

![Building Envelope Window](image)

**Figure 6-6: Building Envelope Window**

The Ventilation window is an input data window where the user will provide information on air volume flow when the supermarket is open or closed, during winter and summer and the pressure drop when the supermarket is open or closed. A daily profile of the air volume flow gives the user the possibility to change the air volume flow that is supplied to the supermarket during the day.

The user must also provide information on the infiltration in the supermarket in air changes per hour. The default value is 0.3 changes per hour when the supermarket is open and 0.1 when it is closed. These values are probably conservative in new supermarkets.
There are three different ventilation systems in CyberMart. The user can select from among rotary heat exchanger, re-circulated return air or both rotary heat exchanger and re-circulated return air together. For the ventilation systems with re-circulated return air and both rotary heat exchanger and re-circulated return air together, the flow of fresh air is dependent on the carbon dioxide in the supermarket. When the concentration of CO₂ is higher than 800 ppm, the flow of fresh air increases and the flow of re-circulated air decreases. If the radio button “Outdoor Air” is clicked, then the supplied airflow when the supermarket is closed is only re-circulated air. The input values in the ventilation window for the supermarket in Sala are presented in Figure 6-7 (Bjerkhög 2004).

![Ventilation Window](image)

Figure 6-7: Ventilation Window

The Heat Sources window is another input data window. The input data required in this interface is: lighting and equipment when the supermarket is open or closed, the vapor production from equipment, the litres per day of services water heating, compressor power and heat dissipated from plug-in cabinets, and the weekly and daily profiles of occupants in the supermarket. The heat sources input data used in Sala is shown in Figure 6-8.
The Opening Hours window requires input of opening hours Monday to Friday, Saturday and Sunday. The opening hours in the supermarket in Sala are presented in Figure 6-9.
The Heating and Air Conditioning window is an input data window for the heating and cooling system. The input data required in this window are: the set point indoor temperature when the supermarket is open or closed, the control temperature when the winter season ends (end winter) and the summer season starts (start summer), the cost for electricity and the type of heating system (district heating or oil).

The radio button Heating opens two other radio buttons where the user can select Heat Recovery and/or Floating Condensing. The radio button Air Conditioning opens a dialog window where the user can select between a chiller or district cooling. The chiller opens a new window where the user will give the temperature of the water (brine).

The input values in the Heating and Air Conditioning window for the supermarket in Sala are presented in Figure 6-10.

![Figure 6-10: Heating and Air Conditioning Window](image-url)
6.3 Refrigeration System Designs

The next step in CyberMart is the selection of the refrigeration system design. Seven refrigeration system designs are possible to simulate and compare in CyberMart: direct system, completely indirect system, partially indirect system, cascade system A, cascade system B, parallel system with mechanical sub-cooling, and the system where the condensers are cooled with the return pipe of district cooling.

For Direct Systems, the first input data is the number of racks of compressors and the kind of compressor in the medium and low temperature systems.

For Indirect Systems, the input data is the kind of compressor, the coolant fluid and the secondary refrigerant for the medium and low temperature systems.

The refrigeration system design in the supermarket in Sala is a cascade system A. In the Cascade System A window, there is a picture where the refrigeration system design is explained. The input data for the refrigeration system in Sala is presented in Figure 6-11.

![Figure 6-11: Cascade System A (Sala Example)]
The compressors are reciprocating in the medium temperature unit and scroll in the low temperature unit. The coolant fluid and the secondary refrigerant for the medium temperature unit are pekasol 50 (60%); the secondary refrigerant for the low temperature unit is pekasol 50 (90%).

The Back button closes the cascade system A window and opens the main window. The Next button opens the Input window for selection of cabinets and cold rooms in the supermarket.

6.3.1 Cabinets and Cold Rooms
In the Input window, the user will select between different models of display cases and deep-freeze display cases and give the dimension of cold rooms in the supermarket. The Input window varies according to the system design. Direct systems and Partially indirect systems have their own input window and completely indirect systems, cascade system A, cascade system B, parallel system with sub-cooling and district cooling or other have another. The reason is that direct system designs have different racks of compressors with diverse cabinets or cold rooms.

The Input window for direct and partially indirect systems has two dialogue windows for specification of the number of cabinets, deep-freeze cabinets and cold storages for each rack of compressors on the medium and low temperature units (see Figure 6-12).

In a new dialogue window, the user must select the model of the cabinets and the dimension of the cold rooms. CyberMart has today a database with about 180 cabinets from Carrier and Wica with data on length, refrigeration capacity, power of fans, light, heating wire, defrost heater, inlet air temperature, outlet air temperature, temperature in cabinet, evaporation temperature, supplied brine temperature, return brine temperature and pressure drop for propylene glycol at 35% concentration. Additional cabinets can be specified as a text file and added by the user. Storages are dimensioned by area and storage temperature.
The input window for completely indirect systems, cascade system A, cascade system B, parallel system with mechanical sub-cooling, and district cooling has a dialogue window with four buttons: *Cabinets*, *DF Cabinets* (deep-freeze cabinets), *Storage* and *DF Storage* (deep-freeze storage), and four boxes for the specification of the number of cabinets, deep-freeze cabinets and storages (see Figure 6-13).

Each button opens a new dialogue window where the user can select the model of the cabinets and also give the dimensions of the cold rooms. Figure 6-14 shows the cabinet window with the display cases in Sala (Electrolux cabinets are Carrier today).
Figure 6-13: Completely Indirect System, Cascade System A, Cascade System B, Parallel System with Sub-cooling and District Cooling or Other

Figure 6-14: Cabinet Window
When the cabinets and storages have been selected, the user clicks the button *Calculate* and the program will start the calculation of refrigeration capacity according to the flow diagram in Figure 6-15.

![Flow Diagram of Input Window](image)

Results of the total dimensioning refrigeration capacity for medium and low temperature systems and refrigeration capacity for
each cabinet and storage are presented in the program. The Next button opens the window that determines the pressure drop of the medium temperature unit.

6.3.2 Pressure Drop of the Medium Temperature System

The Pressure Drop window of the medium temperature unit differs according to the system design. Direct systems have one pressure drop window and completely and partially indirect systems, cascade system A, cascade system B, parallel systems with mechanical sub-cooling and district cooling have another.

![Pressure Drop Window](Figure 6-16: Window of Pressure Drop for Direct Systems)

The Pressure Drop window for direct systems has a scrollbar list with the dimensions of the pipes in the database and a matrix with seven columns for input data of suction, discharge and liquid lines for each rack of compressors (see Figure 6-16). The first column in the matrix shows the suction, discharge and liquid lines for each compressor. The input data in the second column is the value of refrigeration capacity for each rack. The input in the other columns is length of lines, dimension of pipes, elbows, other equivalent lengths and price per meter of each line.
When the dimensions of the different pipes have been entered, the user will click the button *Calculate* and the program will start the calculation of pressure drop according to the flow chart in Figure 6-17. The results are presented in a matrix where the values of refrigeration capacity, velocity, pressure drop and its equivalent tem-
perature drop are shown for the suction, discharge and liquid lines of each rack (see Figure 6-16).

The Pressure Drop window for the other system designs has four dialogue windows for information on brine, dry cooler fluid, main pipe and branch pipes. In the first window, the input data is the concentration of the brine in the medium temperature unit and pipe type of the tube between the evaporator and the display cases. In the second window, the input data is the length, dimension, type and price of the pipe between the condenser and dry cooler and the concentration of the coolant fluid. If the system design is district cooling, the input data is the concentration of the brine in the medium temperature system and the pipe type.

The number of parts in the main pipe is the input data in the third window. The button *Main Pipe* opens a new window with a scroll-bar list with pipes in the database and a matrix with five columns for dimensioning of the main pipe. The first column of the matrix shows the number of parts in the main pipe. Input data in the second column is the refrigeration capacity of each part in the main pipe.
Pipe. Diameter, length, length equivalent and price of the pipe are the input data for the other columns in the matrix.

START

Input data:
Dimension of pipes,
Refrigeration capacity

Read cabinets data from input window

Brine = Propylene Glycol 35% Conc.

Yes

No

Assuming diameter of tube in the coil of cabinet

Calculate Reynolds number and length of tube in coil for propylene glycol 35% conc

Calculate Reynolds number and pressure drop in cabinet for new fluid

1
Calculate Heat transfer coefficient for new fluid in evaporator and condenser

Brine = Propylene Glycol 35% Conc.

Yes

Calculate Heat transfer coefficient for propylene glycol 35% conc. in evaporator and condensor

No

Calculate Heat transfer coefficient for new fluid in evaporator and condensor

Calculate pressure drop and volume flow in different part of main pipe with brine

Calculate Total pressure drop from main pipe with brine, evaporator and cabinet

Read input data for dry cooler in database: Heat transfer coefficients, area, volume, pressure drop, energy cost, air flow

1

2
Figure 6-19: Flow Diagram of Pressure Drop Window for Indirect Systems

When the dimensioning of the main pipe is finished the Calculate button will start the calculation of the pressure drop in the medium temperature unit according to the flow diagram in Figure 6-19.
The results are presented in a matrix where the values of refrigeration capacity, diameter, volume flow and pressure drop of the brine main pipe and the dry cooler fluid pipe are included. The input values in the Pressure Drop window for the supermarket in Sala and the results are presented in Figure 6-18. The Next button opens the window that determines the pressure drop of the medium temperature system.

6.3.3 Pressure Drop of the Low Temperature System

The Pressure Drop window of the low temperature system varies according to the system design. Direct systems and partially indirect systems have one pressure drop window while completely indirect systems, cascade system A, cascade system B, parallel systems with mechanical sub-cooling and district cooling or other have another.

![Pressure Drop Window of the Low Temperature System](image)

The pressure drop window of the low temperature level for direct systems and partially indirect systems has a scrollbar list with pipes in the database and a matrix for input data on suction, discharge and liquid lines for each rack of compressors, as in the medium
temperature unit (see Figure 6-20). The input data required in the matrix is the refrigeration capacity for each rack, the length of lines, dimension of pipes, elbows, other equivalent lengths and price per meter of each line. The calculation procedure of the pressure drop in low temperature unit for direct systems is according to the flow diagram in Figure 6-17.

The pressure drop window of the low temperature system for the other system designs has four dialogue windows for information on brine, dry cooler fluid, main pipe and branch pipe. In the first window, the input data is the concentration of the brine in the low temperature unit and pipe type of the tube between the evaporator and the display cases. In the second window, the input data is the length, dimension, type and price of the pipe between the condenser and dry cooler and the concentration of the coolant fluid (note: this window does not appear for cascade systems). The number of parts in the main pipe is the input data in the third window. The button Main Pipe opens a new window with a scrollbar list with pipes in the database and a matrix for dimensioning of the main pipe as in the medium temperature unit. The input data in
the matrix is the refrigeration capacity of each part in the main pipe, the diameter, length, length equivalent and price of the pipe.

The calculation procedures of pressure drop in the low temperature unit for indirect systems are according to the flow diagram for the medium temperature unit presented in Figure 6-19. The secondary refrigerant used as reference for calculation of the heat exchangers in the low temperature unit is pekasol 50 (90%). The results are presented in a matrix where the values of refrigeration capacity, diameter, volume flow and pressure drop of the brine main pipe and the coolant fluid pipe are included.

An indirect system with carbon dioxide as secondary refrigerant in the low temperature unit is a special issue considered in Cyber-Mart. The pressure drop window for CO₂ is shown in Figure 6-22. The input data needed is capacity, diameter, length, length equivalent and price of liquid and return lines of CO₂.

<table>
<thead>
<tr>
<th>CO₂ lines</th>
<th>Capacity</th>
<th>Diameter</th>
<th>Length</th>
<th>Other Eq.</th>
<th>Price</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid Line</td>
<td>35060</td>
<td>11/8</td>
<td>40</td>
<td>20</td>
<td>400</td>
</tr>
<tr>
<td>Return Line</td>
<td>35060</td>
<td>11/8</td>
<td>40</td>
<td>20</td>
<td>400</td>
</tr>
</tbody>
</table>

Copper pipe
2/8

The calculation procedure of pressure drop in CO₂ systems is done according to the flow diagram presented in Figure 6-24.
The results of pressure drop for CO$_2$ are presented in a matrix where the values of refrigeration capacity, diameter, pressure drop
and temperature drops in the liquid and return lines are included. The Next button opens a new window for calculation of system’s energy consumption.

### 6.4 Energy Calculation

The energy performance calculation of the supermarket is started from this window. The input values required for calculation of total energy usage in the store are the values given and calculated in the different windows presented above.

In the Energy Calculation window, there are six different buttons: **Pressure Drop, New Refrigeration System Design, Building, Refrigeration System, Save** and **Calculate** (see Figure 6-24).

![Figure 6-24: Energy Calculation Window](image)

The button **Calculate** starts the calculation of total energy performance of the supermarket, which is the sum of the energy performance of the refrigeration system, HVAC system, lighting, equipment and plug-in cabinets. The calculation procedure is presented in Figure 6-25.
Calculate values of ventilation flow, occupants, infiltration, lighting, equipment.

Calculate specific loss for windows, internal convection, accumulating layer, structure, and heat capacity of building.

For $i = 1$ To 8760

Read from vectors values of: $T_{out}$, $RH_{out}$, Wind, Beam, Solar radiation, humidity ratio

Compute supermarket open or close, and season (heating, cooling or in between) and setpoint temperatures

Calculate values of ventilation flow, occupants, infiltration, lighting, equipment.

Calculate specific loss for ventilation, infiltrations, air, indoor surfaces, time constant.

Calculate generation of CO2 and generation of vapour

Calculate heat from occupants, and heat to the ground

1
1. Read from vectors values of:
   - heat from cabinets and
   - condensers and frost from cabinets at time i-1

Calculate equivalent temperature for walls and roof

Calculate convective and radiative part of heat gains

Start HVAC system with outdoor temperature and humidity ratio

Assuming air supplied temperature

New Tair supplied

Rotary heat exchanger

Yes

Calculate air temperature and humidity ratio after rotary heat exchanger, heat recovered in rotary heat exchanger

No

Air recirculation

Yes

Calculate volume flow of fresh air and recirculated air

No

If CO2

Calculate air enthalpy, temperature and humidity ratio after recirculation.
Calculate air temperature and humidity ratio after heat recovery coil, and heat recovered from condensers.

Compute heat from condenser at time i - 1 and supplied temperature from coolant fluid.

Calculate air temperature and humidity ratio after heat recovery coil, and heat recovered from condensers.

Tair after heating coil > Tair supplied

Tair after heating coil = Tair supplied

Heat Recovery from condensers

No

Yes

Compute max heat capacity from heating system.

Calculate air temperature and humidity ratio after heating system coil, and heat consumed from heating system.

Tair after HR coil < Tair supplied

Yes

No
Compute max cooling capacity from cooling system

Calculate temperature and dew point temperature of cooling coil

Calculate air temperature and humidity ratio after cooling system coil

Calculate energy consumed in cooling system

Tair after cooling coil < Tair supplied

Tair after cooling coil = Tair supplied

Calculate heat gain to air and structure

Calculate structure and surface temperatures

Calculate room temperature
New Tair supplied

Tair supplied = Tair supplied + 0.2

Tair supplied = Tair supplied - 0.2

Troom = Tsetpoint

Calculates carbon dioxide in supermarket

Tair fresh = Tair supplied * 1.2

If CO2 > 800 ppm

Start Refrigeration system
Compute indoor temperature and relative humidity

Compute heat demand of building

Read cabinets data from input window

Direct system

No

Read input data for air cooled condenser in database: Heat transfer coefficients, area, volume, air volume flow

Calculate evaporator heat transfer coefficient and air mass flow from cabinets

Indirect system

No

Read heat exchangers data from medium and low pressure drops windows

Compute refrigeration capacity for medium and low temperature levels

Yes

5

6
Compute type of compressor from system design window

Read input data for compressors in database: Refrigeration Capacity and compressor power at different evaporating and condensing temperatures

Save values of compressors in matrixes

Compute secondary refrigerants and coolant fluids

Indirect system

Compute air mass flow from dry cooler and a constant outdoor temperature of 27°C

Compute air mass flow from dry cooler and the outdoor temperature

Calculate refrigeration load and frost from cabinets and storages according with indoor temperature and relative humidity

Ind. Syst.

Yes

No

Indirect system

Float. Cond.

Floating

Condensing

Float. Cond.

Yes

No
Calculate mass flow of secondary refrigerant and coolant fluid

Assuming brine temperature after evaporator (TbroA) and coolant fluid temperature after condenser (TcloA)

Calculate coolant fluid temperature after dry cooler and brine temperature after cabinets and storages

Calculate effectiveness of evaporator and condenser

Assuming nominal refrigeration capacity as refrigeration capacity (Q2A)

Calculate condensing and evaporating temperatures with mass flows, brine and coolant fluid temperatures and effectiveness of heat exchangers

Compute refrigeration capacity (Q2B) and compressor power from compressor matrix and evaporating and condensing temperatures

Q2A = Q2B

Calculate run time (relation between refrigeration load from cabinets and storage (Q2load) and refrigeration capacity from compressor (Q2B))

Calculate compressor power from run time and condenser heat
Compute air mass flow from condenser and constant outdoor temperature of 27°C

Calculate refrigeration load and frost from cabinets and storages according with indoor temperature and relative humidity

Calculate brine temperature after evaporator (TbroB) and coolant fluid temperature after condenser (TcloB)

If Ind. Syst.

Direct system

Save comp. power and cond. heat at constant cond. temp

Heat Rec. and Float. Cond.

TbroA = TbroB
TcloA = TcloB

Yes

No

If Dir. Syst.

Tbro = Tbro
Tclo = Tclo

Yes

No

Calculate refrigeration load and frost from cabinets and storages according with indoor temperature and relative humidity

Compute air mass flow from condenser and the outdoor temperature

Floating Condensing

Yes

No

Compute air mass flow from condenser and constant outdoor temperature of 27°C
Compute air mass flow from condenser and constant outdoor temperature of 27°C

Compute refrigeration capacity (Q2comp) and compressor power from compressor matrix, temperature drop in suction line and evaporating and condensing temperatures

Assuming nominal refrigeration capacity as refrigeration capacity (Q2A)

Calculate condensing and evaporating temperatures with air mass flows, air temperatures and effectiveness of heat exchangers

Compute air mass flow from condenser and constant outdoor temperature of 27°C

If

Q2A = Q2B

Yes

Calculate run time (relation between refrigeration load from cabinets and storage (Q2load) and refrigeration capacity from compressor (Q2B))

Calculate compressor power from run time, condenser heat and air temperature after evaporator and condenser

If

Dir. Syst.

No

Q2A = Q2B

10
Save values of refrigeration load, compressor power, condenser heat, condensing and evaporating temperatures in vectors

Heat Rec. and Float. Cond.

Yes

Assuming two chillers in medium and two in low temperature levels

Calculate compressor power and condenser heat according with heat demand from building

Save values of compressor power and condenser heat, in vectors

End Refrigeration system

New day

Yes

Calculate hourly average of indoor temp., relative humidity and humidity ratio, heating and cooling demand, air supplied temperature,

Calculate hourly average of refrigeration load, compressor power, condenser heat, condensing and evaporating temperatures

Save the calculated daily values in vectors
Calculate yearly energy consumption from pumps, fans, lighting, equipments, plug-in cabinets and compressors

Calculate yearly energy consumption from refrigeration system and heating and cooling systems

Compute cost of electricity, district heating, district cooling and oil

Calculate yearly energy cost from fans, lighting, equipments, plug-in cabinets, refrigeration system and heating and cooling systems

Save calculated yearly values of energy consumption and energy cost

Show values of energy consumption and energy cost from fans, lighting, equipments, plug-in cabinets, refrigeration system and heating and cooling systems

Show diagrams with hourly and daily values of indoor and outdoor condition and heating and cooling loads from building

Figure 6-25: Flow Diagram of Energy Calculation
The process for the calculation of energy performance in the supermarket presented in Figure 6-25 shows the different steps from input values and assumptions used in the program, to calculation of indoor conditions, heating and cooling loads from the building, refrigeration loads from cabinets and storages, compressor power and condenser heat from the refrigeration system, energy performance of lighting, equipment, plug-in cabinets, fans, pumps, the refrigeration system and heating and cooling systems and yearly cost. The flow diagram does not present the interconnection between the different modules in the program (it is therefore quite simplified).

The *Pressure Drop* button closes the Energy Calculation window and opens the Pressure Drop window of the low temperature unit.

The *New Refrigeration System Design* button closes the Energy Calculation window and opens the main window where it is possible to select a new system design for comparison.

The *Building* button closes the Energy Calculation window and opens the Building window where it is possible to select air conditioning, heat recovery, floating condensing, etc., in order to study the influence of various building parameters.

The *Refrigeration System* button closes the Energy Calculation window and opens the Refrigeration System window. This window show results from the first and last refrigeration system designs simulated.

The *Save* button opens a window where you can save the results from the first and last system designs simulated. The parameters that are saved are hour, outdoor temperature, indoor temperature, indoor relative humidity, total compressor power, compressor power of the medium and low temperature system, refrigeration capacity and condenser power of the first and last system designs simulated.

When the calculation has finished, a dialogue window with five buttons and results from energy performance equivalent to the button *Energy Year* will appear (see Figure 6-26). The results presented are the energy consumption from heating and cooling sys-
tems, lighting, rotary heat exchanger, fans, service water heating and the refrigeration system of the first and last systems simulated.

The button *Temp Year* shows a diagram with results of indoor and outdoor climate over the course of one year (Figure 6-27). The parameters in the diagram are outdoor temperature, outdoor relative humidity, outdoor humidity ratio, indoor temperatures, indoor relative humidity, indoor humidity ratio and air supplied temperature.

The button *Load Year*, shows a diagram with results of the heating and cooling loads of the supermarket during the course of one year.

The button *Cost Year* opens results from the cost of the different components such as heating, air conditioning, lighting, rotary heat exchanger, fans, service water heating and refrigeration system of the first and last system designs.
The button *Temp. and Load Day* opens two diagrams with results of indoor climate and loads over the course of one day. The user can select the day of interest in the scrollbar to the right of the button.

![Supermarket Energy Calculation](image)

**Figure 6-27: Energy Calculation**

### 6.4.1 Result Refrigeration System

In the Refrigeration System window, there is a dialogue window with a diagram and ten buttons: *Power, Medium, Low, Energy, LCC, TEWI, Report, Close, Energy Calculation Back and New system* (see Figure 6-28). The diagram shows results of the compressor power from the medium and low temperature systems during the course of one year, which is equivalent to clicking the button *Power*.

The button *Medium* opens a diagram with results of the refrigeration capacity, condenser power, condensing and evaporating temperatures of the medium temperature system, outdoor temperature, indoor temperature and relative air humidity in the supermarket over the course of one year.

The button *Low* opens a diagram with results of the refrigeration capacity, condenser power, condensing and evaporating temperatures of the low temperature system, outdoor temperature, indoor
temperature and relative air humidity in the supermarket during the course of one year.

The button *Energy* opens a table with results of the energy consumption of the refrigeration system over the course of one year for the first and last systems simulated. The table shows results from the energy consumption of compressors, pumps, fans and others (lighting and defrost from manufacturer performance data) for the medium and low temperature systems and the total energy consumption of the refrigeration system.

![Refrigeration System Window](image)

Figure 6-28: Refrigeration System Window

The button *LCC* opens a window for calculation of the Life Cycle Cost of the first and last refrigeration systems simulated.

The button *TEWI* opens a window for calculation of Total Equivalent Warming Impact of the first and last refrigeration systems simulated.

The button *Report* opens a window with a report of the results of total energy consumption, total cost, LCC and TEWI from the first and last refrigeration systems simulated.

The *Close* button closes the CyberMart program.
6.4.2 LCC

In the LCC window, there is a matrix and six buttons: Back, Inv.Cost of the first refrigeration system simulated, Inv.Cost of last refrigeration system simulated, LCCothers, LCCtot and Calculate.

If the investment costs of the system’s design are unknown, Inv.Cost will help the user with this. The button Inv.Cost opens a window with information on the different components in the refrigeration system to help calculate the investment cost of the refrigeration system (see Figure 6-29).

![Figure 6-29: LCC Investment Window](image)

The button LCCothers opens a window for calculation of the LCC of operation, maintenance, repair and others. The input data required for this calculation is the period of service (which is the same as the study period), interest rate, expected inflation, annually recurring cost and single cost of operation, maintenance, repair and others, and the number of years after the investment that this
cost occurs. The button *Calculate* starts the calculation of the LCCothers according to equations (5.86) and (5.87).

Figure 6-30: LCCothers Window

The button *LCCtot* opens a window for calculation of the total life cycle cost. The input data necessary in this window is the period of study, interest rate, annual price increase of electricity and electricity price. In the other columns, the results will appear from the calculation of investment cost, energy application and LCCothers of both system designs. The user can write the value of investment costs and LCCothers directly in the window. The button *Calculate* starts the calculation of the LCCtot according to equations (5.86), (5.87) and (5.88). When the calculation is finished, a new window with results of energy cost, discount factor based on the interest rate, LCCenergy and LCCtot of both refrigeration systems is opened (see Figure 6-31). The district cooling system design has two other conditions: the price of the district cooling and the annual price increase of district cooling. The results of the consumption and cost from the district cooling are presented beside the energy cost.

The button *Back* closes the LCC window and opens the Refrigeration System window.
6.4.3 TEWI

The TEWI button opens a window for calculation of the total estimated equivalent warming impact, TEWI. The input data required is the working life of the refrigeration machine, the loss rate of refrigerant during one year in percentage and the regional conversion factor that is a measure of CO₂ produced in electricity generation.

The button Calculate starts the calculation of the TEWI according to equation (5.89). When the calculation is finished, a window with different results and the buttons TEWI and Diagram is opened (see Figure 6-32).
Figure 6-32: TEWI Window

The results presented are the refrigerant charge in the system, the GWP (global warming potential) of the refrigerant, the direct emission of CO₂ from the refrigerant loss during the working life of the refrigeration machine, the indirect emission of CO₂ from the energy usage by the refrigeration machine and the total emission of CO₂ that is the sum of the direct and indirect emissions of CO₂ of each refrigeration system design compared.

The button Diagram opens a diagram with results of direct, indirect and total emissions of CO₂ of both refrigeration systems. The Back button closes the TEWI window and opens the Refrigeration System window.

6.4.4 Report

The button Report opens a window with a report of results from total energy performance, total costs, LCC, TEWI and energy consumption of the refrigeration system designs simulated in Cyber-Mart.

In the report, there are ten buttons to zoom, move, print, save, open and close the report.
6.5 Conclusions

A computer model known as CyberMart, which has the ability to simulate building heating and cooling loads, the HVAC system and seven different refrigeration systems, has been presented in this chapter. The properties of the different components in the model have been taken from performance data of different manufacturers. The intended users of the program are designers and engineers from different companies involved in the implementation of new systems and energy efficient measures in supermarkets.

The supermarket in Sala was selected as an example to show the different windows, input data, interface with users, calculation proceedings and results from the model.
The different windows in CyberMart can be divided into three groups: input data, calculation and results. Some windows belong to two different groups. The input data windows are: the Building window, the Refrigeration System window, the Cabinets and Storages window and the Pressure Drop windows. The calculation windows are: the Cabinets and Storages window, the Pressure Drop windows, the Energy Calculation window, the TEWI window and the LCC window. The results windows are the Energy Calculation window, the Refrigeration System window, the TEWI window, the LCC window and the report.

One objective of the model was to have as little input data as possible and yet produce reliable results. A list with the input data required in the program presented in the Appendix A illustrates that the amount of input data is small in proportion to the level of calculation and results presented in the program. Some components such as condensers, evaporators, compressors and dry coolers are selected directly in the database by the program to reduce the amount of input data. Other parameters such as heat from people, efficiency of pumps and fans, etc., have been assumed as constants in the program to reduce the input data. The sales and office area in the supermarket were assumed to be one zone for the calculation of heating and cooling loads, also for the sake of reducing the input data.

It is now time to see if the third and fourth phases of the modelling process can be undertaken.
7 Evaluation of CyberMart

The third phase in the modelling process is the evaluation of the model developed in relation with the overall objectives identified in the beginning of the modelling process. The evaluation of CyberMart has two different parts. The first part is a sensitivity analysis of model predictions. The second part is a comparison between results from the computer model and results from measurements carried out in five supermarkets in Sweden.

7.1 Sensitivity Analysis

Sensitivity analysis is an important technique for evaluating the variation or uncertainties that the model may have with regard to its predictions (Macdonald 2001). The differential sensitivity analysis (DSA) is used to consider the sensitivity of the program CyberMart. The DSA method, which is explained by (Lomas 1992), enables the sensitivity of the results to input parameter changes. A base case is simulated with the best approximation for the parameters in consideration. Then the simulation is performed when one input parameter IP is changed to IP + ΔIP and the other input parameters remain the same as in the base case. The change in the parameter predicted, ΔPP, corresponds to a direct measure of the effect of the change in the input parameter IP + ΔIP. The ΔPP is the sensitivity of PP to an uncertainty in input IP.

\[ ΔPP = PP_{IP+ΔIP} - PP_{BaseCase} \]  

(6.1)

Where \( PP_{IP+ΔIP} \) is the result of the simulation from the modified parameter IP, and \( PP_{BaseCase} \) is the result of the simulation from the base case.
The total sensitivity is dependent on the uncertainties in all input parameters and it is estimated from the quadrature sum of the changes in the parameters predicted:

\[
\Delta P_{TOT} = \sqrt{\sum_{i=1}^{1} \Delta P_i^2}
\]

(6.2)

The supermarket simulated as a base case is the supermarket in Sala. The refrigeration system used in the simulations is partially indirect in the low temperature unit and completely indirect in the medium temperature unit. The supermarket simulated in the base case has no heat recovery or floating condensing system to fully capture the effect of the change of input parameters on the heating system.

The input parameters and assumptions studied in the sensitivity analysis are: outdoor temperature, solar radiation, relative humidity, heat transfer coefficient of walls, air volume flow, infiltrations, lighting, capacity of equipment, vapor water production from equipment, efficiency of rotary heat exchanger, occupants, pressure drop in the ventilation system, supermarket opening hours (one hour more every day), set point temperature during heating season, set point temperature during cooling season, heat transfer coefficient of windows, windows shield, efficiency of fans in the ventilation system, minimum flow of fresh air, the indoor heat transfer coefficient of convection, heat and metabolism from occupants, properties of secondary refrigerant and coolant fluids, refrigeration load from cabinets in the medium and low temperature units, values of refrigeration capacity and compressor power from matrices, UA values of evaporators and condensers, pump power of secondary refrigerant and coolant fluid, fans in refrigeration systems (fans in cabinet, cold rooms, and dry cooler), defrost and compressor efficiency in medium and low temperature systems.

The change in the input parameters and assumptions was 10%. The change in outdoor temperature and solar radiation was calculated in the software METEONORM (Remund 2005) as extreme values.

The individual and total sensitivities of the different input parameters and assumptions are presented in Table 7-1.
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Basic Case</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1215</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Outdoor Temp. Max.</td>
<td>112</td>
<td>4</td>
<td>542</td>
<td>525</td>
<td>1183</td>
<td>32.2</td>
<td>2.7%</td>
</tr>
<tr>
<td>Solar Rad. Max.</td>
<td>147</td>
<td>2</td>
<td>541</td>
<td>525</td>
<td>1215</td>
<td>0.0</td>
<td>0.0%</td>
</tr>
<tr>
<td>Outdoor RH +10%</td>
<td>151</td>
<td>2</td>
<td>544</td>
<td>525</td>
<td>1222</td>
<td>-7.0</td>
<td>-0.6%</td>
</tr>
<tr>
<td>Wall (U and C) +10%</td>
<td>149</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1216</td>
<td>-1.0</td>
<td>-0.1%</td>
</tr>
<tr>
<td>Vent. Vol. Flow +10%</td>
<td>144</td>
<td>2</td>
<td>540</td>
<td>537</td>
<td>1223</td>
<td>-8.2</td>
<td>-0.7%</td>
</tr>
<tr>
<td>Infiltrations +10%</td>
<td>157</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1224</td>
<td>-9.0</td>
<td>-0.7%</td>
</tr>
<tr>
<td>Lighting +10%</td>
<td>125</td>
<td>3</td>
<td>541</td>
<td>556</td>
<td>1225</td>
<td>-9.8</td>
<td>-0.8%</td>
</tr>
<tr>
<td>Equipment +10%</td>
<td>144</td>
<td>2</td>
<td>540</td>
<td>531</td>
<td>1217</td>
<td>-2.2</td>
<td>-0.2%</td>
</tr>
<tr>
<td>Equip. H2O Prod. +10%</td>
<td>147</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1214</td>
<td>1.0</td>
<td>0.1%</td>
</tr>
<tr>
<td>Efficiency rhex +10%</td>
<td>154</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1221</td>
<td>-6.0</td>
<td>-0.5%</td>
</tr>
<tr>
<td>Occupants +10%</td>
<td>146</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1213</td>
<td>1.8</td>
<td>0.1%</td>
</tr>
<tr>
<td>Press. Drop Vent +10%</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>537</td>
<td>1227</td>
<td>-12.0</td>
<td>-1.0%</td>
</tr>
<tr>
<td>----------------------------------</td>
<td>--------------</td>
<td>--------------</td>
<td>-------------------</td>
<td>------------</td>
<td>--------------</td>
<td>-------------</td>
<td>-------</td>
</tr>
<tr>
<td>Open 1hr More per Day</td>
<td>133</td>
<td>2</td>
<td>546</td>
<td>556</td>
<td>1237</td>
<td>-22.4</td>
<td>-1.8%</td>
</tr>
<tr>
<td>Set Point T. Heat. -0.5°C</td>
<td>132</td>
<td>2</td>
<td>538</td>
<td>535</td>
<td>1207</td>
<td>8.0</td>
<td>0.7%</td>
</tr>
<tr>
<td>Set Point T. Cool. +0.5°C</td>
<td>144</td>
<td>2</td>
<td>543</td>
<td>535</td>
<td>1224</td>
<td>-8.6</td>
<td>-0.7%</td>
</tr>
<tr>
<td>U Windows +10%</td>
<td>149</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1216</td>
<td>-1.0</td>
<td>-0.1%</td>
</tr>
<tr>
<td>Window Shield +10%</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1215</td>
<td>0.0</td>
<td>0.0%</td>
</tr>
<tr>
<td>Effic. Fan Vent +10%</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>514</td>
<td>1204</td>
<td>11.0</td>
<td>0.9%</td>
</tr>
<tr>
<td>hconv Indoor +10%</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1215</td>
<td>-0.2</td>
<td>0.0%</td>
</tr>
<tr>
<td>Met Person +10%</td>
<td>146</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1213</td>
<td>1.8</td>
<td>0.1%</td>
</tr>
<tr>
<td>Brine Prop. +10%</td>
<td>148</td>
<td>2</td>
<td>528</td>
<td>525</td>
<td>1203</td>
<td>12.0</td>
<td>1.0%</td>
</tr>
<tr>
<td>Cab. Load Med. +10%</td>
<td>164</td>
<td>2</td>
<td>554</td>
<td>525</td>
<td>1245</td>
<td>-29.6</td>
<td>-2.4%</td>
</tr>
<tr>
<td>Cab. Load Low +10%</td>
<td>155</td>
<td>2</td>
<td>550</td>
<td>525</td>
<td>1232</td>
<td>-16.8</td>
<td>-1.4%</td>
</tr>
<tr>
<td>Comp. Med. +10%</td>
<td>148</td>
<td>2</td>
<td>547</td>
<td>525</td>
<td>1222</td>
<td>-7.0</td>
<td>-0.6%</td>
</tr>
<tr>
<td>Comp. Low +10%</td>
<td>148</td>
<td>2</td>
<td>547</td>
<td>525</td>
<td>1222</td>
<td>-7.0</td>
<td>-0.6%</td>
</tr>
<tr>
<td>UA Evap. Med. 10%</td>
<td>148</td>
<td>2</td>
<td>537</td>
<td>525</td>
<td>1212</td>
<td>3.0</td>
<td>0.2%</td>
</tr>
<tr>
<td>------------------</td>
<td>--------------</td>
<td>--------------</td>
<td>----------------</td>
<td>------------</td>
<td>--------------</td>
<td>-------------</td>
<td>-------</td>
</tr>
<tr>
<td>UA Cond. Med. +10%</td>
<td>148</td>
<td>2</td>
<td>537</td>
<td>525</td>
<td>1212</td>
<td>3.0</td>
<td>0.2%</td>
</tr>
<tr>
<td>UA Cond. Low +10%</td>
<td>148</td>
<td>2</td>
<td>537</td>
<td>525</td>
<td>1212</td>
<td>3.0</td>
<td>0.2%</td>
</tr>
<tr>
<td>UA Evap. Low +10%</td>
<td>148</td>
<td>2</td>
<td>539</td>
<td>525</td>
<td>1214</td>
<td>1.0</td>
<td>0.1%</td>
</tr>
<tr>
<td>Pump Brine Med. +10%</td>
<td>148</td>
<td>2</td>
<td>541</td>
<td>525</td>
<td>1216</td>
<td>-1.0</td>
<td>-0.1%</td>
</tr>
<tr>
<td>Pump Cool Med. +10%</td>
<td>148</td>
<td>2</td>
<td>541</td>
<td>525</td>
<td>1216</td>
<td>-1.0</td>
<td>-0.1%</td>
</tr>
<tr>
<td>Pump Cool Low +10%</td>
<td>148</td>
<td>2</td>
<td>540</td>
<td>525</td>
<td>1215</td>
<td>0.0</td>
<td>0.0%</td>
</tr>
<tr>
<td>Fans Ref. Sys. Med. +10%</td>
<td>148</td>
<td>2</td>
<td>544</td>
<td>525</td>
<td>1219</td>
<td>-4.0</td>
<td>-0.3%</td>
</tr>
<tr>
<td>Fan Ref. Sys. Low +10%</td>
<td>148</td>
<td>2</td>
<td>542</td>
<td>525</td>
<td>1217</td>
<td>-2.0</td>
<td>-0.2%</td>
</tr>
<tr>
<td>Defrost +10%</td>
<td>148</td>
<td>2</td>
<td>541</td>
<td>525</td>
<td>1216</td>
<td>-1.0</td>
<td>-0.1%</td>
</tr>
<tr>
<td>Comp. Effic. Med. -10%</td>
<td>148</td>
<td>2</td>
<td>563</td>
<td>525</td>
<td>1238</td>
<td>-23.0</td>
<td>-1.9%</td>
</tr>
<tr>
<td>Comp. Effic. Low -10%</td>
<td>148</td>
<td>2</td>
<td>560</td>
<td>525</td>
<td>1235</td>
<td>-20.0</td>
<td>-1.6%</td>
</tr>
<tr>
<td>Total Sensitivity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>68.3</td>
<td></td>
</tr>
<tr>
<td>Percent</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>5.6%</td>
<td></td>
</tr>
</tbody>
</table>

193
The sensitivity analysis shows that the outdoor temperature, calculation of refrigeration loads from cabinets, compressor efficiency and the open time of the supermarket are the parameters with important influence on the predicted energy performance of the supermarkets. The outdoor temperature has the largest effect on the predicted energy performance.

The total sensitivity of 5.6% is a satisfactory result since the changes in the majority of input parameters and assumptions was 10% with exception of outdoor temperature and solar radiation that were calculated as extreme values in METEONORM.

7.2 Simulation of Five Supermarkets and Comparison with Field Measurements

Five supermarkets - Sala, Hedemora, Västerås, Hjo, and Täby Centrum - have been simulated. Measurements and simulation results for a whole year were not possible for two supermarkets due to lack of data.

The input data used in the simulation is the data presented in Chapter 4 and from personal communication with (Bjerkhög 2004) and (Jansson 2002). The condensing temperature of the refrigeration system in the low temperature system in Sala was increased in the simulations according to the real value in the store.

Figure 7-1 illustrates results for indoor and outdoor temperatures and indoor relative humidity obtained from simulation and field measurements from the supermarket in the city of Sala during the course of one year.
Figure 7-1: Comparison of Indoor and Outdoor Temperatures and Indoor Relative Humidity in Sala during One Year.

Figure 7-2 presents results of outdoor temperature and compressor power of the medium temperature unit from simulation and field measurements from the supermarket in Sala over the course of one year.

Figure 7-2: Comparison of Outdoor Temperature and Compressor Power in Sala over One Year.
Figure 7-3 and Figure 7-4 present results of indoor and outdoor temperatures and compressor power from simulation and field measurements from the supermarket in Sala during two days in July and August respectively.

Figure 7-3 Comparison of Indoor and Outdoor Temperatures and Compressor Power in Sala during Two Days in July

Figure 7-4: Comparison of Indoor and Outdoor Temperatures and Compressor Power in Sala during Two Days in August
The differences in outdoor temperature and indoor relative humidity between simulation and measurements are due to different weather conditions. The weather data in CyberMart focuses on the prediction of energy use. The results of outdoor temperature and indoor relative humidity from measurements are values that fluctuate every year as does the total energy consumption.

Values of outdoor temperature from measurements and values of outdoor relative humidity and solar radiation from Stockholm (SLB analys 2004) were used in CyberMart as climate data to simulate the supermarket in Sala. The purpose was to compare the results of indoor relative humidity from simulation and measurements using about the same outdoor conditions. Those results are presented in Figure 7-5. It can be seen that there is a good correlation between measured and predicted indoor relative humidity.
moisture, compressor power and total energy consumption in the supermarket in Sala.

Table 7-2: Total Energy Performance in Sala from Measurements and CyberMart

<table>
<thead>
<tr>
<th></th>
<th>Supermarket in Sala</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Consumption</td>
<td>MWh / year</td>
</tr>
<tr>
<td>Measurement</td>
<td>CyberMart</td>
</tr>
<tr>
<td>Electricity</td>
<td>1179</td>
</tr>
<tr>
<td>Heating</td>
<td>53</td>
</tr>
</tbody>
</table>

Results from simulations and field measurements from the supermarket in Hjo are presented in Figure 7-6. The figure illustrates results of indoor and outdoor temperatures and indoor relative humidity during the course of one year.

![Comparison Measurement - CyberMart Supermarket in Hjo during One Year](image)

Figure 7-6: Comparison of Indoor and Outdoor Temperatures and Indoor Relative Humidity in Hjo during One Year

Figure 7-7 presents results of outdoor temperature and compressor power of the medium temperature system from simulation and field measurements from the supermarket in Hjo over the course of one year.
Figure 7-7: Comparison of Outdoor Temperature and Compressor Power in Hjo during One Year

Figure 7-8 presents measurements of indoor and outdoor temperatures and compressor power from simulation and field measurements from the supermarket in Hjo during two days in February.

Figure 7-8: Comparison of Outdoor Temperature and Compressor Power in Hjo during Two Days in February
Results of total energy consumption in the supermarket in Hjo are presented in Table 7-3. The measured consumptions of electricity and district heating are compared with results from CyberMart during one year.

<table>
<thead>
<tr>
<th>Energy Consumption</th>
<th>Measurement</th>
<th>CyberMart</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity</td>
<td>610</td>
<td>586</td>
</tr>
<tr>
<td>Heating</td>
<td>80</td>
<td>49</td>
</tr>
</tbody>
</table>

The differences in compressor power, indoor temperature and moisture between simulations and measurements in the supermarket in Hjo are depicted in Figure 7-6, Figure 7-7 and Figure 7-8. The yearly energy balance in Hjo presented in Table 7-3 differs by 8%. It is clear that the general characteristics are relatively well recreated in the program.

Figure 7-9 illustrates results of indoor and outdoor temperatures and indoor relative humidity obtained from simulation and field measurements from the supermarket in the city of Hedemora during the course of one year.

Figure 7-10 presents results of compressor powers in the medium and low temperature units from simulation and field measurements from the supermarket in Hedemora over the course of one year.
Figure 7-9: Comparison of Indoor and Outdoor Temperatures and Indoor Relative Humidity in Hedemora during One Year

Figure 7-10: Comparison of Compressor Power in the Medium and Low Temperature Units in Hedemora during One Year

Figure 7-11 shows the comparison of outdoor temperature and compressor power in the low temperature unit in Hedemora during two days in January.
The measured consumption of electricity and district heating during one year is compared with results from CyberMart in Table 7-4.

Table 7-4: Total Energy Performance in Hedemora from Measurements and CyberMart

<table>
<thead>
<tr>
<th>Supermarket in Hedemora</th>
<th>Energy Consumption</th>
<th>MWh / year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement</td>
<td>Electricity</td>
<td>850</td>
</tr>
<tr>
<td></td>
<td>Heating</td>
<td>180</td>
</tr>
<tr>
<td>CyberMart</td>
<td>Electricity</td>
<td>819</td>
</tr>
<tr>
<td></td>
<td>Heating</td>
<td>160</td>
</tr>
</tbody>
</table>

Table 7-4 and Figure 7-9, Figure 7-10 and Figure 7-11 show that there is good agreement between the measured and simulated values of indoor and outdoor temperatures, compressor power in the low temperature unit and total energy consumption in Hedemora. Nevertheless, the compressor power in the medium temperature system presented in Figure 7-10 shows important differences between the measured and predicted values. This could be due to a too low refrigeration charge in the system that affects the run time of the two compressors in the medium temperature system.
Results from simulations and field measurements from the supermarket in Täby Centrum are presented in Figure 7-12. The figure shows results of indoor and outdoor temperatures and indoor relative humidity over the course of one year.

Figure 7-12: Comparison of Indoor and Outdoor Temperatures and Indoor Relative Humidity in Täby Centrum during One Year

Figure 7-13: Comparison of Outdoor Temperatures and Compressor Power in Täby Centrum during One Year
Figure 7-13 presents results of outdoor temperature and compressor power in the medium temperature unit from simulation and field measurements in Täby Centrum during the course of one year.

The simulated values of indoor relative humidity presented in Figure 7-12 and those of compressor power in the medium temperature system presented in Figure 7-13 show under-predicted values in comparison with the measured values. The difference between the predicted and measured values of compressor power can be due to the difference between the predicted and measured values of relative humidity.

To illustrate the result of the variation between climate data in CyberMart and real data values, outdoor temperature, outdoor relative humidity and solar radiation from Stockholm (SLB analysis 2004) were used in CyberMart as climate data to see their influence on indoor relative humidity and compressor power in Täby Centrum. Results of indoor and outdoor temperatures, indoor relative humidity and compressor power from measurements and CyberMart using the same climate data during one year are presented in Figure 7-14 and Figure 7-15.
Figure 7-14 shows that there is a good correlation between measured and predicted indoor relative humidity.

![Comparison Measurement - CyberMart Täby Centrum one Year with Same Climate Data](image)

On the other hand, Figure 7-15 shows that the model underpredicts the values of compressor power, especially during the summer period. This could be due to higher condensing temperatures during summer than suggested by the outdoor temperature measurements, to unsatisfactory operation of the compressors or to a poor ability of the model to predict the compressor power in Täby Centrum.

![Figure 7-15: Comparison of Outdoor Temperatures and Compressor Power in Täby Centrum during One Year using the Same Climate Data](image)

Table 7-5 Total Energy Performance in Täby Centrum from Measurements and CyberMart

<table>
<thead>
<tr>
<th>Supermarket in Täby Centrum</th>
<th>Energy Consumption</th>
<th>MWh /year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>1390</td>
<td>1300</td>
</tr>
<tr>
<td>Heating</td>
<td>No Information</td>
<td>30</td>
</tr>
</tbody>
</table>

The measured electricity consumption is compared with results from CyberMart over the course of one year in Table 7-5. The
consumption of district heating in Täby Centrum is difficult to estimate because heat recovered from the condenser is distributed to the whole shopping centre and heating and cooling are included in the rent of the premises.

The measurements carried out in Västerås focused on the function of the heat recovery system and total energy consumption. For this reason, the period of measurement was intensive but limited to a few days in summer and winter. Measurement of electricity consumption of the refrigeration system, auxiliary heating (electric boiler), ventilation system, lighting and the total were carried out monthly over the course of two years by (Lundqvist 2003).

Figure 7-16 presents results from simulation of indoor and outdoor temperatures and indoor relative humidity during one year in Västerås.

Figure 7-17 presents results from simulation of compressor power in medium and low temperature units and total compressor power during one year in Västerås.
Table 7-6 shows measured and simulated values of electricity consumption in Västerås.

Table 7-6: Total Energy Performance in Västerås from Measurements and CyberMart

<table>
<thead>
<tr>
<th>Supermarket in Västerås</th>
<th>Energy Consumption in [MWh/year]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Measurement</td>
</tr>
<tr>
<td>Year</td>
<td>2001</td>
</tr>
<tr>
<td>Refriger. System</td>
<td>672</td>
</tr>
<tr>
<td>Aux. Heating</td>
<td>103</td>
</tr>
<tr>
<td>Fan Ventilation</td>
<td>97</td>
</tr>
<tr>
<td>Lighting</td>
<td>235</td>
</tr>
<tr>
<td>Others</td>
<td>380</td>
</tr>
<tr>
<td>Total Energy</td>
<td>1487</td>
</tr>
</tbody>
</table>

The model predicts the electricity consumption of auxiliary heating, ventilation system and lighting with good accuracy in comparison with measured values. The predicted value of electricity con-
consumption from the refrigeration system differs from the measured value probably because of the low evaporating temperature indicated by the measurement in Västerås (see Figure 4-36). The difference between measured and simulated values of total energy consumption is due to the “others” item, which includes equipment, plug-in cabinets, etc. There are no explanations about the high electricity usage of these items.

Table 7-7 presents a summary of the energy usage in the five supermarkets in which measurements were compared with results from simulations.

<table>
<thead>
<tr>
<th></th>
<th>Energy Usage</th>
<th>MWh / year</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Measurement</td>
<td>CyberMart</td>
</tr>
<tr>
<td><strong>Sala</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>1179</td>
<td>1149</td>
</tr>
<tr>
<td>District Heating</td>
<td>53</td>
<td>39</td>
</tr>
<tr>
<td><strong>Hjo</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>610</td>
<td>586</td>
</tr>
<tr>
<td>District Heating</td>
<td>80</td>
<td>49</td>
</tr>
<tr>
<td><strong>Hedemora</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>850</td>
<td>819</td>
</tr>
<tr>
<td>District Heating</td>
<td>180</td>
<td>160</td>
</tr>
<tr>
<td><strong>Täby Centrum</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>1390</td>
<td>1300</td>
</tr>
<tr>
<td>District Heating</td>
<td>No Information</td>
<td>30</td>
</tr>
</tbody>
</table>
### Energy Usage (MWh / year)

<table>
<thead>
<tr>
<th>Measurement</th>
<th>CyberMart</th>
</tr>
</thead>
<tbody>
<tr>
<td>Västerås</td>
<td></td>
</tr>
<tr>
<td>Electricity</td>
<td>1329</td>
</tr>
<tr>
<td>Heating</td>
<td>94</td>
</tr>
</tbody>
</table>

#### 7.3 Conclusions

The computer model CyberMart, which has the ability to simulate building heating and cooling loads, HVAC and seven different refrigeration systems, has been evaluated through a sensitivity analysis and through comparisons between results from the computer model and measurements carried out in five supermarkets in Sweden.

The sensitivity analysis shows that the outdoor temperature, refrigeration loads from cabinets, compressor efficiencies and the opening time of the supermarket are the parameters with the greatest influence on the predicted energy performance of the supermarkets. The total sensitivity of 5.6% is a satisfactory result since the changes in the majority of input parameters and assumptions were 10% with the exception of outdoor temperature and solar radiation that were calculated as extreme values in METEONORM.

Comparisons between measurements and simulations in five supermarkets show reasonably good agreement. Results of outdoor and indoor temperatures, indoor relative humidity, compressor power and total energy consumption in Sala and Hjo illustrate acceptable correlation between simulations and measurements.

Results from measured and simulated values of indoor and outdoor temperatures, compressor power in the low temperature unit and total energy consumption in Hedemora also show good agreement; however, values of compressor power on the medium temperature unit present important differences between the measured and predicted values probably due to a low refrigeration charge in the real system.
In Täby Centrum, the simulated values of indoor relative humidity and of compressor power in the medium temperature system reveal under-predicted values in comparison with the measured values. Simulations using real values of outdoor relative humidity and temperature, however, indicate good agreement between predicted and measured values of indoor relative humidity. The model under-predicts the values of compressor power especially during the summer period. This could be due to higher condensing temperatures during summer than suggested by the outdoor temperature measurement, unsatisfactory operation of compressors or poor ability of the model to predict the compressor power.

In Västerås, the yearly electricity consumption of the refrigeration system, auxiliary heating, ventilation system, lighting and the total were compared with simulated values. The model predicts the electricity consumption of auxiliary heating, the ventilation system and lighting well. The predicted value of electricity consumption of the refrigeration systems differs from the measured value probably due to unnecessarily low evaporating temperatures in the medium temperature system. The difference between the measured and simulated values of total energy usage is due to the high electricity consumption from equipment, plug-in cabinets and others.

The sensitivity analysis and the comparison between simulations and measurements in five supermarkets also showed that outdoor relative humidity and temperature are the parameters with the greatest influence on the predicted energy performance of the supermarkets. Prediction of climate data is a complex issue due to the yearly variation in climatic conditions. The climate data predicted in METEONORM and applied in CyberMart uses the average values from the period 1961-1990 for the different weather stations. The average values of climate data under-predict extreme values of outdoor temperatures and relative humidity as simulations and measurements in Sala and Täby Centrum illustrated. Simulations using real values of outdoor relative humidity and temperature, however, indicate good agreement between predicted and measured values of indoor relative humidity.
8 Applications of CyberMart

8.1 Introduction

The fourth phase of the modelling process is the model application. At this stage, the computer model answers the questions identified in the beginning of the modelling process, such as questions about energy performance of different system designs, performance of heat recovery or floating condensing systems, performance of air conditioning systems and their influence on the indoor relative humidity, performance of secondary refrigerants and coolant fluids, performance of energy effective cabinets, etc. Two subjects have been selected as examples to be addressed by the model in this chapter: heat recovery vs. floating condensing systems and direct vs. indirect systems.

8.2 Heat Recovery vs. Floating Condensing

One idea behind the development of the CyberMart software was to facilitate comparison of different solutions for the same supermarket, such as a heat recovery system or a floating condensing system.

Heat recovery, floating condensing and both heat recovery and floating condensing systems have been simulated in CyberMart for the specific supermarket. The supermarket studied in the program is the supermarket in Sala. The different components of the HVAC system in Sala are: a rotary heat exchanger, a bypass for recirculation of the return air and three coils for cooling, heat recovery and auxiliary heating. Results from simulation of the available heat from the condensers and heat requirement of the supermarket in Sala are presented in Figure 8-1.
It is clear from Figure 8-1 that the available condenser heat is more than sufficient to cover the heating requirements of the building. Practical experience indicates that the energy consumption of the auxiliary heating is considerable, which points to an unsatisfactory recovery of the available heat from condensers. A reason for this could be poor design of the heat recovery systems and on-off operation of the refrigeration systems.

Figure 8-2 presents results from simulation of the total compressor power for the three different cases of interest here: heat recovery, floating condensing and simultaneous heat recovery and floating condensing. The compressor power of the refrigeration machine working with floating condensing is reduced to around 50%, at lower outdoor temperatures, in comparison with the compressor power of the refrigeration machine working with heat recovery.
The energy consumption of the three systems, presented in Table 8-1, shows that the system with both heat recovery and floating condensing is the most energy effective system for the supermarket in Sala.

Table 8-1: Comparison between Heat Recovery, Floating Condensing and both Heat Recovery and Floating Condensing

<table>
<thead>
<tr>
<th></th>
<th>Heat Rec.</th>
<th>Float. Cond.</th>
<th>HR and FC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Rotary Heat Exchanger [MWh]</td>
<td>60</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Heat Recovery [MWh]</td>
<td>149</td>
<td>0</td>
<td>149</td>
</tr>
<tr>
<td>Heating System [MWh]</td>
<td>0</td>
<td>149</td>
<td>0</td>
</tr>
<tr>
<td>Heating Requirements [MWh]</td>
<td>209</td>
<td>209</td>
<td>209</td>
</tr>
<tr>
<td>Service Water Heating [MWh]</td>
<td>39</td>
<td>39</td>
<td>39</td>
</tr>
<tr>
<td>Total Heat [MWh]</td>
<td>39</td>
<td>188</td>
<td>39</td>
</tr>
<tr>
<td>Cost District Heating [EUR/kWh]</td>
<td>0.055</td>
<td>0.055</td>
<td>0.055</td>
</tr>
<tr>
<td>Total Heating Cost [EUR/year]</td>
<td>2100</td>
<td>10300</td>
<td>2100</td>
</tr>
<tr>
<td>Energy Compressors [MWh]</td>
<td>512</td>
<td>374</td>
<td>429</td>
</tr>
<tr>
<td>Total Refrigeration system [MWh]</td>
<td>659</td>
<td>521</td>
<td>577</td>
</tr>
<tr>
<td>Total Electricity [MWh]</td>
<td>1149</td>
<td>1011</td>
<td>1067</td>
</tr>
<tr>
<td>Cost Electricity [EUR/kWh]</td>
<td>0.055</td>
<td>0.055</td>
<td>0.055</td>
</tr>
<tr>
<td>Total Electricity Cost [EUR/year]</td>
<td>63100</td>
<td>55600</td>
<td>58600</td>
</tr>
<tr>
<td>Total Cost [EUR/year]</td>
<td>65200</td>
<td>65900</td>
<td>60700</td>
</tr>
</tbody>
</table>
Simulations of different supermarkets with different refrigeration capacities and different outdoor climates in the cities of Luleå, Stockholm and Malmö were carried out in CyberMart to compare heat recovery, floating condensing and simultaneous heat recovery and floating condensing systems at different conditions. The auxiliary heating system was assumed to be an electric boiler.

Figure 8-3 illustrates results from simulation of a supermarket that has the same area and refrigeration capacities as the supermarket in Sala but in three different climates. Figure 8-4 presents results from simulation of a supermarket with a sale area of 500m² and refrigeration capacities of 35kW and 8kW in the medium and low temperature systems respectively.

![Figure 8-3: Simulation of a Supermarket in Three Different Cities with a Sales Area of 2700m² and Different Systems](image)

The first three columns in Figure 8-3 and Figure 8-4 give the total energy costs for the supermarkets working at a constant condensing temperature of 40°C without heat recovery or floating condensing systems (Cond40C) in three different cities in Sweden: Luleå, Stockholm and Malmö. The second three columns give the total energy costs for the supermarkets with heat recovery systems (HeatRec) in the same cities. The third three columns give the total energy costs for the supermarkets with floating condensing systems (FloatCond), also in the same cities. The last three columns give the total energy costs for the supermarkets with combined...
heat recovery and floating condensing systems (HRandFC), again in the same cities.

Both Figure 8-3 and Figure 8-4 show that the system working with both heat recovery and floating condensing is the most energy effective of the four systems presented.

Figure 8-5 presents savings in percent of the systems with heat recovery, floating condensing and both heat recovery and floating condensing in comparison to the system with constant condensing temperature at 40°C without heat recovery or floating condensing.

Figure 8-5 shows more clearly that the system working with both heat recovery and floating condensing is the most energy effective of the four systems presented. Figure 8-5 also shows that in the supermarket with an area of 2700m², the system working with heat recovery is more energy efficient than the system working with floating condensing in Luleå and Stockholm, while floating condensing is more efficient than heat recovery in Malmö. In the supermarket with an area of 500m², the system working with floating condensing is more energy effective than the system working with heat recovery in the three cities.
Figure 8-5: Savings in Comparison to a System with Constant Condensing Temperature at 40°C without Heat Recovery or Floating Condensing

Figure 8-6 also presents savings in percent from the systems with heat recovery, floating condensing and both heat recovery and floating condensing in comparison to the system with constant condensing temperature at 40°C without heat recovery or floating condensing. For this case, the supermarkets with sales areas of 2700m² have been simulated at different refrigeration capacities: 110kW and 45kW, 90kW and 35kW, and 75kW and 25kW in the medium and low temperature systems.

Figure 8-6 also shows that the system working with both heat recovery and floating condensing is the most energy effective of the four systems. Figure 8-6 illustrates that in the supermarket with refrigeration capacities 110kW and 45kW, the system working with heat recovery is more energy efficient than the system working with floating condensing in the three cities.

In the supermarket with refrigeration capacities 75kW and 25kW, the system working with heat recovery is more energy efficient than the system working with floating condensing in Luleå, while floating condensing is more efficient than heat recovery in Stockholm and Malmö.
Savings in Comparison with No Heat Rec. or Float.Cond at Different Refrigeration Capacities

Figure 8-6: Savings in Comparison to a System with Constant Condensing Temperature at 40°C without Heat Recovery or Floating Condensing at Different Refrigeration Capacities

Figure 8-7 illustrates results from simulation of the supermarket with a sales area of 2700m² at different energy prices. The systems simulated were heat recovery, floating condensing, both heat recovery and floating condensing and constant condensing temperature at 40°C without heat recovery or floating condensing.

Area: 2700; Ref.System: Medium 90kW, Low 35kW

Figure 8-7: Simulation of a Supermarket with a Sales Area of 2700m² at Different Energy Prices
8.3 Direct vs. Indirect Systems

Comparison between the energy consumption of direct and indirect refrigeration systems has been carried out in different studies (Clodic 1998; Yunting 2001); (Lindborg 2000); (Horton 2002); (You 2001); and (Faramarzi 2004) with different results. In Cybermart, it is possible to compare different refrigeration system designs in the same supermarket. Two refrigeration systems, one with a direct system and another with a completely indirect system and mechanical sub-cooling, were simulated in CyberMart to compare these systems at different conditions.

The supermarket in Sala was selected as the basis for such a comparison. The period of study was 15 years, and the refrigerant in both system designs was R404A. Propylene glycol, with a concentration of 35%, was used as secondary refrigerant in the medium temperature unit and as coolant fluid. The secondary refrigerant in the low temperature unit was carbon dioxide.

Two comparisons were carried out between direct and indirect refrigeration systems. In the first comparison, both system designs recovered heat from the condenser for heating of the premises. In the second comparison, the direct system had a heat recovery system while the completely indirect system with mechanical sub-cooling had both heat recovery and floating condensing.

Figure 8-8 presents results from the first comparison when both system designs had heat recovery systems. The figure illustrates values of total compressor power and compressor power of the medium and low temperature units in both direct and indirect systems during the course of one year. The compressor power in the low temperature unit is lower in the indirect system because of the sub-cooling from the medium temperature unit. For the same reason, the compressor power in the medium temperature unit is higher in the indirect system. The total compressor power of both system designs is about the same.
Table 8-2 presents results from the simulations of the heating system, cooling system, refrigeration system, others (lighting, equipment and fans from the ventilation system) and total energy consumption for both refrigeration system designs.

Table 8-2: Total Energy Consumption from Simulations of Direct and Indirect Systems

<table>
<thead>
<tr>
<th></th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Heat Recovery</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td>Heating System</td>
<td>[MWh/year] 38</td>
<td>46</td>
</tr>
<tr>
<td>Cooling System</td>
<td>[MWh/year] 3</td>
<td>3</td>
</tr>
<tr>
<td>Refrigeration System</td>
<td>[MWh/year] 486</td>
<td>509</td>
</tr>
<tr>
<td>Others</td>
<td>[MWh/year] 487</td>
<td>487</td>
</tr>
<tr>
<td>Total Energy</td>
<td>[MWh/year] 1 014</td>
<td>1 045</td>
</tr>
</tbody>
</table>
Table 8-3 illustrates results of TEWI from simulations of direct and indirect systems. The leakage was 5% per year, the refrigerant charges were 500 kg for the direct system and 55 kg for the indirect system and the regional conversion factor used was 0.04 [kg CO₂ eq./kWh], which is the best value for Sweden (Sand 1997).

<table>
<thead>
<tr>
<th></th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEWI Refrigeration System</td>
<td>Heat Recovery</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td>Direct Emission [kg CO₂ eq.]</td>
<td>1 222 500</td>
<td>134 500</td>
</tr>
<tr>
<td>Indirect Emission [kg CO₂ eq.]</td>
<td>222 800</td>
<td>242 000</td>
</tr>
<tr>
<td>Total Emission [kg CO₂ eq.]</td>
<td>1 445 300</td>
<td>376 500</td>
</tr>
</tbody>
</table>

The influence of the regional conversion factor in TEWI is presented in Table 8-4. In this case, the regional conversion factor used was 0.51 [kg CO₂ eq./kWh], which is the best value for Europe (Sand 1997).

<table>
<thead>
<tr>
<th></th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEWI Refrigeration System</td>
<td>Heat Recovery</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td>Direct Emission [kg CO₂ eq.]</td>
<td>1 222 500</td>
<td>134 500</td>
</tr>
<tr>
<td>Indirect Emission [kg CO₂ eq.]</td>
<td>2 840 700</td>
<td>3 085 500</td>
</tr>
<tr>
<td>Total Emission [kg CO₂ eq.]</td>
<td>4 063 200</td>
<td>3 220 000</td>
</tr>
</tbody>
</table>
Table 8-5 illustrates results of LCC from simulations of direct and indirect systems. The period of study was 15 years, the interest rate was 4%, the annual price increase of electricity was 1% and the cost of electricity was 0.6 SEK (equivalent to about 0.07 Euro). The operating, maintenance and repair cost during one year for the direct system was 12,000 Euro and 9,000 Euro for the indirect system. The investment cost for the direct system is assumed to be 10% cheaper than for the indirect system according to Bjerkhög, who is responsible for the implementation of a new refrigeration system in the supermarket chain COOP Sweden (part of Koopera- tion) (Bjerkhög 2004).

Table 8-5: LCC from Simulations of Direct and Indirect Systems

<table>
<thead>
<tr>
<th></th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LCC Sub-cooling</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td>Refrigeration System</td>
<td>Heat Recovery</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td>Investment Costs [EUR]</td>
<td>300 000</td>
<td>333 000</td>
</tr>
<tr>
<td>LCC Energy [EUR]</td>
<td>387 000</td>
<td>406 000</td>
</tr>
<tr>
<td>LCC Others [EUR]</td>
<td>146 000</td>
<td>106 000</td>
</tr>
<tr>
<td>LCC Total [EUR]</td>
<td>833 000</td>
<td>845 000</td>
</tr>
</tbody>
</table>

A new case where the same two refrigeration system designs were again simulated in the same supermarket is presented. Here, the indirect system has both heat recovery and floating condensing.

Figure 8-9 presents results from simulations of compressor power in both direct and indirect systems over the course of one year. The figure illustrates values of medium and low temperature units and total compressor power.
Figure 8-9: Compressor Power from Direct and Indirect Systems

Table 8-6 presents results from simulation of the heating system, cooling system, refrigeration system, others (lighting, equipment and fans from the ventilation system) and total energy consumption for both refrigeration system designs.

Table 8-6: Total Energy Consumption from Simulations of Direct and Indirect Systems

<table>
<thead>
<tr>
<th></th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Heat Recovery</td>
<td>Heat Recovery and Floating Condensing</td>
</tr>
<tr>
<td>Heating System [MWh/year]</td>
<td>38</td>
<td>46</td>
</tr>
<tr>
<td>Cooling System [MWh/year]</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Refrigeration System [MWh/year]</td>
<td>486</td>
<td>443</td>
</tr>
<tr>
<td>Others [MWh/year]</td>
<td>487</td>
<td>487</td>
</tr>
<tr>
<td>Total Energy [MWh/year]</td>
<td>1014</td>
<td>979</td>
</tr>
</tbody>
</table>
Table 8-7 illustrates results of TEWI from simulations of direct and indirect systems. The period of study, refrigerant, leakage and conversion factor applied in the simulations were the same as those presented in Table 8-3.

**Table 8-7: TEWI from Simulations of Direct and Indirect Systems at a Regional Conversion Factor of 0.04 kg CO2/kWh for Sweden**

<table>
<thead>
<tr>
<th>TEWI Refrigeration System</th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Sub-cooling</td>
<td></td>
</tr>
<tr>
<td>Direct Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>1 222 500</td>
<td>134 500</td>
</tr>
<tr>
<td>Indirect Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>222 800</td>
<td>202 400</td>
</tr>
<tr>
<td>Total Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>1 445 300</td>
<td>336 900</td>
</tr>
</tbody>
</table>

Table 8-8 presents results of TEWI from simulations of direct and indirect systems using the regional conversion factor 0.51 kg CO2eq/kWh, which is the best value for Europe (Sand 1997).

**Table 8-8: TEWI from Simulations of Direct and Indirect Systems at a Regional Conversion Factor of 0.51 kg CO2/kWh for Europe**

<table>
<thead>
<tr>
<th>TEWI Refrigeration System</th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Sub-cooling</td>
<td></td>
</tr>
<tr>
<td>Direct Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>1 222 500</td>
<td>134 500</td>
</tr>
<tr>
<td>Indirect Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>222 800</td>
<td>202 400</td>
</tr>
<tr>
<td>Total Emission</td>
<td></td>
<td></td>
</tr>
<tr>
<td>[kg CO2 eq.]</td>
<td>1 445 300</td>
<td>336 900</td>
</tr>
</tbody>
</table>
Table 8-9 illustrates results of LCC from simulations of direct and indirect systems. The period of study, interest rate, annual price increase of electricity, cost of electricity and operating, maintenance and repair cost were the same as those applied in Table 8-5. The indirect system simulated was implemented with both heat recovery and floating condensing, which has a higher investment cost than the system with heat recovery only (Bjerkhög 2004).

<table>
<thead>
<tr>
<th>LCC Refrigeration System</th>
<th>Direct System</th>
<th>Indirect System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Sub-cooling</td>
<td>Heat Recovery</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Heat Recovery</td>
</tr>
<tr>
<td></td>
<td></td>
<td>and Floating</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Condensing</td>
</tr>
<tr>
<td>Investment Costs [EUR]</td>
<td>300 000</td>
<td>367 000</td>
</tr>
<tr>
<td>LCC Energy [EUR]</td>
<td>387 000</td>
<td>353 000</td>
</tr>
<tr>
<td>LCC Others [EUR]</td>
<td>146 000</td>
<td>106 000</td>
</tr>
<tr>
<td>LCC Total [EUR]</td>
<td>833 000</td>
<td>826 000</td>
</tr>
</tbody>
</table>

Results of total energy consumption, TEWI and LCC from simulations of direct and indirect systems presented in Table 8-2, Table 8-3, Table 8-5, Table 8-6, Table 8-7 and Table 8-9 show that an indirect system with heat recovery and floating condensing is the most energy efficient, has the lowest environmental impact and the lowest cost during a study period of 15 years. This assertion is valid for the supermarket in Sala for the circumstances simulated. Other conditions such as climate, dimension of building or refrigeration capacities may give different results.
8.4 Conclusions

In the fourth phase of modelling, the computer model answers the questions identified in the beginning of the modelling process. Two questions: comparisons of heat recovery vs. floating condensing systems and direct vs. indirect systems were selected as examples because the answers to these questions require a systems approach where the impacts among different subsystems in the supermarket are evaluated. The supermarket in Sala was selected as the basis for such an evaluation.

For the comparison between heat recovery and floating condensing, four different systems were simulated: one with constant condensing temperature of 40°C without heat recovery or floating condensing, one with heat recovery only, one with floating condensing only and one with both heat recovery and floating condensing. These systems were simulated in two different supermarkets, with different refrigeration capacities and in different climates (Luleå, Stockholm and Malmö). According to the results from calculations using CyberMart, the best energy savings is achieved with systems using both heat recovery and floating condensing.

Simulations of direct and indirect refrigeration systems show that an indirect system with heat recovery and floating condensing is the most energy effective system, has the lowest impact on the environment and the lowest cost during a study period of 15 years. This assertion is valid for the supermarket in Sala for the circumstances simulated.

Both examples presented in this chapter have answered the questions formulated in the beginning of the modelling process. The systems approach and the interaction between the different submodels in the supermarket are well recreated in the program.
9 Conclusions, Discussion, and Future Studies

9.1 Conclusions

9.1.1 Energy Usage and Environmental Impact
Supermarkets are intensive users of energy in most countries. In Sweden, approximately 3% of the electric energy consumed is used in supermarkets (1.8 TWh/year). However, there is a great potential for improvement of the energy systems in supermarkets. Typical efficiency improvements may involve more efficient refrigeration, illumination and HVAC systems. Energy-saving technologies such as heat recovery, floating condensing temperature, defrost control, energy efficient lighting, high efficiency motors, efficient control and energy efficient display cases have been implemented in several supermarkets to reduce energy consumption.

Stricter environmental legislation enforcing a phase-out of CFC and HCFC refrigerants and, more recently, the impact of refrigerant leakage on the environment have affected the refrigeration system in supermarkets. New system solutions with completely and partially indirect systems have been developed and introduced in supermarkets to lower the refrigerant charge and to minimize potential refrigerant leakage.

The increasing interest in indirect systems has led to the development of new secondary refrigerants based on potassium formate and potassium acetate alone or mixed since these fluids have good heat transfer and pressure drop characteristics. Other very promising developments are phase-changing secondary refrigerants such as CO$_2$ and ice slurries. Ice slurry is an interesting secondary refrigerant that adds thermal storage to the system.

CO$_2$ as the only refrigerant in the refrigeration system is an important alternative to HFC refrigerants in supermarkets. The CO$_2$ cy-
Many supermarkets in Sweden utilize heat recovery to increase the overall energy efficiency of the system. One disadvantage of heat recovery is the high condensation temperature necessary to transfer the heat from the condenser to the heating system of the supermarket. This increases the energy consumption for the refrigeration system but, at the same time, leads to a reduction of the energy consumption for the heating system. An alternative to heat recovery is floating condensing temperature that improves the coefficient of performance and decreases the energy consumption of the compressors at lower outdoor temperatures. Another option is to utilize both heat recovery and floating condensing temperature depending on the heating requirements of the premises.

9.1.2 Field Measurements

Results from field measurements in seven supermarkets show that outdoor temperature and indoor relative humidity are two important factors to take into consideration when dimensioning refrigeration and heat recovery systems in supermarkets. Lower outdoor temperature and relative humidity lower the compressor power considerably.

Measurements carried out in supermarkets in the cities of Sala and Hedemora emphasize the relationships between the outdoor temperature, humidity ratio and indoor relative humidity.

Field measurements also illustrate that night covering of display cases and deep-freeze display cases reduces the energy consumption in supermarkets by about 10-20%. Night covering is thus an efficient method of reducing infiltration and radiation loss in cabinets.

Indoor temperatures above 25°C increase the temperature of products in display cases and deep-freeze cabinets to levels that affect the date for minimum durability of food. An air conditioning system is necessary to avoid indoor temperatures higher than 25°C, even if the warm season is short.
9.1.3 Modelling and Simulations

A computer program known as CyberMart that has the ability to simulate building heating and cooling loads, the HVAC system and seven different refrigeration systems in supermarkets has been created. The intended users of the program are designers and engineers from different companies involved in the implementation of new systems and energy efficient measures in supermarkets.

A thorough sensitivity analysis shows that the outdoor temperature, refrigeration loads from cabinets, compressor efficiencies and opening hours of the supermarket are parameters that strongly influence the predicted energy performance of the supermarkets. The total sensitivity of 5.6% is a satisfactory result since the changes in the majority of input parameters and assumptions was 10% with the exception of outdoor temperatures and solar radiation, which were calculated as extreme values in METEONORM.

Comparisons between measurements and simulations in five supermarkets show reasonably good agreement. Comparative results from simulations and measurements of outdoor and indoor temperatures, indoor relative humidity, compressor power and total energy consumption in Sala and Hjo illustrate acceptable correlation.

Results from measured and simulated values of indoor and outdoor temperatures, compressor power in the low temperature unit and total energy consumption in Hedemora also show good agreement; however, values of compressor power in the medium temperature unit reveal important differences between the measured and predicted values, probably due to too low refrigeration charge in the system.

In Täby Centrum, the simulated values of indoor relative humidity and compressor power in the medium temperature unit were under-predicted in comparison with the measured values. Simulations using real measured values of outdoor temperature and relative humidity, however, showed good agreement between predicted and measured values of indoor relative humidity. The model under-predicted the values of compressor power especially during the summer period. This could be due to higher condensing temperatures during summer than suggested by the outdoor temperature measurement or to unsatisfactory operation of compressors.
In Västerås, the yearly electricity consumption from the refrigeration system, auxiliary heating, ventilation system, lighting and the total were compared with simulated values. The model predicted the electricity consumption of auxiliary heating, the ventilation system and lighting well. The predicted value of electricity consumption from refrigeration systems differs from the measured value probably because of low evaporating temperatures in the medium temperature unit. The difference between measured and simulated values of total energy consumption is due to the high electricity consumption of equipment, plug-in cabinets and others.

According to the results from CyberMart, the highest potential for energy saving is by using a system with both heat recovery and floating condensing. In theory, the necessary heat can always be supplied from the condensers. Practical experience shows that, in real systems, only a part of the available condenser heat is recovered. There are many reasons for this, but the most important are mixing points in the coolant fluid system, on/off regulation, poorly designed heat recovery systems and separated control system between refrigeration and HVAC systems.

Simulations of direct and indirect refrigeration systems in a supermarket in the city of Sala in Sweden indicate that an indirect system with heat recovery and floating condensing is the most energy efficient system, has the lowest impact on the environment and the lowest cost during a study period of 15 years. This assertion is valid for the supermarket in Sala for the circumstances simulated.

9.2 Discussion

CyberMart has demonstrated its capability of analyzing different energy measures in supermarkets, as shown by the sensitivity analysis and the comparisons between field measurements and simulations.

There are many energy-saving technologies such as heat recovery, floating condensing temperature or energy efficient lighting already implemented in supermarkets to reduce energy consumption. The cooperation between different actors or divisions in supermarket chains and equipment manufacturers is of great importance for implementation and operation of energy-saving technologies. Unfortunately, the overall objective of these divisions or actors differs
considerably. A systems approach is necessary where the whole supermarket is considered. The supermarket chains must take into consideration that a supermarket is a complex system when different energy measures are discussed. The program CyberMart provides new perspectives for supermarkets by giving designers and engineers in the field opportunities for assessment and testing of new energy efficient measures and for evaluation of different already-installed system designs and components. Higher energy consumption from a refrigeration system, HVAC system or lighting should be easy to avoid using the results from CyberMart as references.

Many designers and engineers are looking for rules of thumb by which to give a rapid answer about installation of energy efficient measures. Simulations from CyberMart demonstrate that each supermarket is unique. It has a particular energy efficient solution according to its location, refrigeration system design, amount of cabinets, HVAC system, lighting, equipment, plug-in cabinets, occupants, costs and environmental impact. These rules of thumb are likely to be ignored when both energy efficient measures and economy are evaluated. The relatively high investment cost of intensely energy-saving technologies and hard competition between the different supermarket chains make the introduction of these technologies in supermarkets difficult. However, a particular energy efficient solution for each supermarket is possible to achieve only when there is a balance between LCC, TEWI and performance.

9.3 Future Studies

The model and field measurements presented in this thesis provide a platform for future studies about energy performance in supermarkets. Next steps to improve the model would incorporate multiple zone simulations with at least three different zones for sales areas, offices and receiving areas. The model would also include new natural refrigerants such as ammonia, propane and carbon dioxide. Carbon dioxide as a refrigerant has been installed in some supermarkets in Sweden, and there is great interest from designers and supermarket chains in estimating and comparing refrigeration system designs running with CO₂. There is also a demand to increase the amount of input data in the program to give the user the possibility of selecting different components such as compressors,
heat exchangers, etc. Two or three different levels of detail, according to the level of knowledge of the user, would probably be a satisfactory solution.

The majority of the supermarkets studied have had problems with higher energy consumption, higher condensing temperatures or lower evaporating temperatures than the set point values, inadequate heat reclaim systems and unsatisfactory design and control of different components or systems in the stores. The implementation of new energy-saving technologies in supermarkets requires an extensive analysis of energy performance of refrigeration systems, HVAC systems, lighting, equipment and of total energy consumption. This analysis should be carried out over a long period of time (at least one year) to evaluate and compare the real energy performance with the theoretical values calculated using Cyber-Mart.

The modelling approach used throughout this thesis can also be adapted to other types of systems. Many of the sub-models developed can be used to model energy consumption of ice hockey rinks, restaurants, swimming pool buildings and indoor skiing arenas.
## Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Area</td>
<td>m²</td>
</tr>
<tr>
<td>a</td>
<td>Thermal Diffusivity of Material</td>
<td>m²/s</td>
</tr>
<tr>
<td>C</td>
<td>Concentration of Carbon Dioxide</td>
<td>ppm</td>
</tr>
<tr>
<td>cp</td>
<td>Specific Heat Capacity of Air</td>
<td>J/kg°C</td>
</tr>
<tr>
<td>C</td>
<td>Heat Capacity of Structure</td>
<td>Wh/m²K</td>
</tr>
<tr>
<td>Cr</td>
<td>Heat Capacity Ratio</td>
<td>-</td>
</tr>
<tr>
<td>CC</td>
<td>Concentration of Carbon Dioxide</td>
<td>ppm</td>
</tr>
<tr>
<td>d</td>
<td>Thickness</td>
<td>m</td>
</tr>
<tr>
<td>de</td>
<td>Effective Diameter</td>
<td>m</td>
</tr>
<tr>
<td>di</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>dac</td>
<td>Thickness of the Accumulating Layer</td>
<td>m</td>
</tr>
<tr>
<td>E</td>
<td>Annual Energy Consumption</td>
<td>kWh/year</td>
</tr>
<tr>
<td>f</td>
<td>Friction Factor</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>Generation of Vapor or CO₂</td>
<td>g/h</td>
</tr>
<tr>
<td>G</td>
<td>Mass Velocity</td>
<td>kg/m²/s</td>
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<tr>
<td>GWP</td>
<td>Global Warming Potential</td>
<td>kg CO₂/kg refrigerant</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy</td>
<td>J/kg</td>
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<td>H</td>
<td>Specific Loss</td>
<td>W/K</td>
</tr>
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<td>Unit</td>
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<tr>
<td>i</td>
<td>Discount Rate</td>
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<td>I</td>
<td>Solar Irradiation</td>
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<td>Inv</td>
<td>Present Value of Investment Costs</td>
<td>SEK</td>
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<tr>
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<tr>
<td>L</td>
<td>Length</td>
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<td>LCC</td>
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<td>LCC_{ENERGY}</td>
<td>Present Value of Annual Energy Costs</td>
<td>SEK</td>
</tr>
<tr>
<td>LCC_{OM&amp;R}</td>
<td>Present Value of Non-fuel Operating and Repair Cost</td>
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<tr>
<td>LCC_{Envir}</td>
<td>Present Value of Environmental Costs</td>
<td>SEK</td>
</tr>
<tr>
<td>LCC_{Others}</td>
<td>Present Value of Other Costs</td>
<td>SEK</td>
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<td>M</td>
<td>Refrigerant Leakage during One Year</td>
<td>kg/year</td>
</tr>
<tr>
<td>m</td>
<td>Mass Flow</td>
<td>kg/s</td>
</tr>
<tr>
<td>n</td>
<td>Number of Years</td>
<td>-</td>
</tr>
<tr>
<td>N</td>
<td>Equipment Operation Time</td>
<td>year</td>
</tr>
<tr>
<td>nh</td>
<td>Exponent of Plate Heat Exchanger</td>
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</tr>
<tr>
<td>nd</td>
<td>Number of Air Exchanges per 24 Hours in a Cold Room</td>
<td>h</td>
</tr>
<tr>
<td>Np</td>
<td>Number of Passes in Plate Heat Exchanger</td>
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</tr>
<tr>
<td>NTU</td>
<td>Number of Transfer Units</td>
<td>-</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
<td>Bar</td>
</tr>
<tr>
<td>pe</td>
<td>Inflation or Price Escalation</td>
<td>%</td>
</tr>
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<td>Pp</td>
<td>Pump Power</td>
<td>W</td>
</tr>
<tr>
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<td>Description</td>
<td>Unit</td>
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<td>--------</td>
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<td>PP</td>
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<tr>
<td>$\dot{Q}$</td>
<td>Heat</td>
<td>W</td>
</tr>
<tr>
<td>$r$</td>
<td>Latent Heat of Vaporization</td>
<td>Kg/°K</td>
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<tr>
<td>Re</td>
<td>Reynolds Number</td>
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<td>RC</td>
<td>Annually Recurrent Cost</td>
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<tr>
<td>RCF</td>
<td>Regional Conversion Factor</td>
<td>kgCO₂/kWh</td>
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<tr>
<td>RCP</td>
<td>Present Value of RC</td>
<td>r</td>
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<tr>
<td>S</td>
<td>Latent Heat of Fusion of Water from Liquid to Solid Ice</td>
<td>KJ/kg</td>
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<tr>
<td>SC</td>
<td>Single Cost</td>
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<td>SCP</td>
<td>Present Value of a Single Cost</td>
<td>r</td>
</tr>
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<td>T</td>
<td>Temperature</td>
<td>°C</td>
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<tr>
<td>Tabs</td>
<td>Absolute Temperature</td>
<td>°K</td>
</tr>
<tr>
<td>TO</td>
<td>Oscillation Period</td>
<td>hr</td>
</tr>
<tr>
<td>U</td>
<td>Heat Transfer Coefficient</td>
<td>W/(m²K)</td>
</tr>
<tr>
<td>u</td>
<td>Wind Speed</td>
<td>m/s</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>Volume Flow</td>
<td>m³/s</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
<td>m³</td>
</tr>
<tr>
<td>$v''$</td>
<td>Specific Volume for Saturated Vapor</td>
<td>m³/kg</td>
</tr>
<tr>
<td>$v'$</td>
<td>Specific Volume for Saturated Liquid</td>
<td>m³/kg</td>
</tr>
<tr>
<td>w</td>
<td>Velocity of Fluid</td>
<td>m/s</td>
</tr>
</tbody>
</table>
X  Humidity Ratio  gr/kg dry air

**Greek**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>α</td>
<td>Heat Transfer Coefficient</td>
<td>W/(m² K)</td>
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<tr>
<td>αₘₐₜₜ</td>
<td>Absorptivity for Solar Radiation</td>
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<tr>
<td>Δ</td>
<td>Difference</td>
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<tr>
<td>ρ</td>
<td>Density</td>
<td>kg/m³</td>
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<tr>
<td>η</td>
<td>Efficiency</td>
<td>-</td>
</tr>
<tr>
<td>ε</td>
<td>Effectiveness of HEX</td>
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</tr>
<tr>
<td>λ</td>
<td>Thermal Conductivity</td>
<td>W/(m K)</td>
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<td>ε</td>
<td>Emissivity of Material</td>
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<tr>
<td>σ</td>
<td>Stefan-Boltzman Constant</td>
<td>W/m²K⁴</td>
</tr>
<tr>
<td>ν</td>
<td>Cinematic Viscosity</td>
<td>m²/s</td>
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<tr>
<td>τ</td>
<td>Time Constant</td>
<td>h</td>
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<tr>
<td>θ</td>
<td>Angle of Incidence</td>
<td>rad</td>
</tr>
<tr>
<td>β</td>
<td>Solar Altitude</td>
<td>rad</td>
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<td>γ</td>
<td>Surface-Solar Azimuth</td>
<td>rad</td>
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<tr>
<td>κ</td>
<td>Recycling factor.</td>
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**Subscripts**

<table>
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<th>Description</th>
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<tbody>
<tr>
<td>a</td>
<td>Structure</td>
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<tr>
<td>acc</td>
<td>Accumulating Layer</td>
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<tr>
<td>air</td>
<td>Air</td>
</tr>
<tr>
<td>airex</td>
<td>Air Exchanges in a Cold Room</td>
</tr>
<tr>
<td>br</td>
<td>Secondary Refrigerant</td>
</tr>
<tr>
<td>cabcon</td>
<td>Convective Part of the Heat Gains from Cabinets</td>
</tr>
<tr>
<td>cl</td>
<td>Cooling Load</td>
</tr>
<tr>
<td>coolfl</td>
<td>Cooling Fluid</td>
</tr>
<tr>
<td>cr</td>
<td>Cold Room</td>
</tr>
<tr>
<td>crn</td>
<td>Walls, Floor and Ceiling in a Cold Room</td>
</tr>
<tr>
<td>CO2</td>
<td>Carbon Dioxide</td>
</tr>
<tr>
<td>con</td>
<td>Convective</td>
</tr>
<tr>
<td>cond</td>
<td>Condenser</td>
</tr>
<tr>
<td>d</td>
<td>Latent or Diffusion</td>
</tr>
<tr>
<td>DN</td>
<td>Direct Radiation</td>
</tr>
<tr>
<td>dθ</td>
<td>Diffuse Radiation</td>
</tr>
<tr>
<td>e</td>
<td>Effective</td>
</tr>
<tr>
<td>eq</td>
<td>Equivalent</td>
</tr>
<tr>
<td>eqcon</td>
<td>Convective Part of Heat Gains from Equipment</td>
</tr>
<tr>
<td>evap</td>
<td>Evaporator</td>
</tr>
<tr>
<td>fresh</td>
<td>Fresh Air</td>
</tr>
<tr>
<td>g</td>
<td>Gas</td>
</tr>
</tbody>
</table>
gr   Ground
hl   Heating load
inf  Infiltration
l    Liquid
lightcon Convective Part of the Heat Gains from Lighting
losses Leakage
met  Metal
out  Outside
percon Convective Part of the Heat Gains from People
phex Plate Heat Exchanger
pr   Product
prin Product before Cooling or Freezing Process
prcr Product after Cooling or Freezing Process
rad  Radiative
recir Re-circulation
ref  Refrigerant
refl Reflected
rhex Rotary Heat Exchanger
room Supermarket
sky  Sky
sol  Solar
solcon Convective Part of Solar Gain

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Appendix A: Input Data of CyberMart

The input data required in CyberMart has been divided according to the different windows in the program:

Building

Location:

Dimensions of Building Envelope
Length:
Width:
Height:
Sales Area:

Walls
Area:
Direction:
Inside or Outside:
Kind of Construction (heavy, medium or light):
Dimension of Windows
  Area:
  Kind of Window (two or three glass panes):
  Kind of Window shield (curtain, venetian blind, etc.):

Roof
Area:
Inside or Outside:
Kind of Construction (heavy, medium or light):

Floor
Area:
Inside or Outside:
Kind of Construction (heavy, medium or light):
Opening Hours from Monday to Sunday:

Ventilation

Air Volume Flow in m³/hour

Winter Time and Supermarket is Open:
Winter Time and Supermarket is Closed:
Summer Time and Supermarket is Open:
Summer Time and Supermarket is Closed:

Pressure Drop in Pa

When the Supermarket is Open:
When the Supermarket is Closed:

Ventilation with Rotary Heat Exchanger and/or Re-circulation of Return Air:

Infiltration in Air Changes per Hour

When the Supermarket is Open:
When the Supermarket is Closed:

Heat Sources

Lighting in Watts per Square Meter (W/m²)

When the Supermarket is Open:
When the Supermarket is Closed:

Equipment in Watts (W)

When the Supermarket is Open:
When the Supermarket is Closed:

Vapor Production in Gram per Hour (gr/h)

When the Supermarket is Open:
When the Supermarket is Closed:

Service Water in Liters per Day (lt/day):

Occupants: Max Occupants per Day:

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Heating and Air Conditioning Systems

Heating System:

Heat Recovery from Condenser Heat, Floating Condensing Temperature or Both Heat Recovery and Floating Condensing:

Heating or Auxiliary Heating System: District Heating, Oil Boiler, Electric Boiler and their Cost:

Set-up Temperatures in the Supermarkets
Winter Time and Supermarket is Open:
Winter Time and Supermarket is Closed:
Summer Time and Supermarket is Open:
Summer Time and Supermarket is Closed:

Coolant Temperature (Indirect System) or Condensing Temperature (Direct System) at Heat Recovery:

Air Conditioning System
Water Temperature of the Chiller or District Cooling and its Cost:

Cost of Electricity:

Refrigeration System

The input data required in the Refrigeration System window is dependent on the refrigeration system design.

Compressor Medium Temperature Unit:
Compressor Low Temperature Unit:
Coolant Fluid:
Secondary Refrigerant Medium Temperature System
Secondary Refrigerant Low Temperature System
Night Cover Cabinets:
Night Cover Deep-freeze Cabinets:

Models of Display Cases and Deep-freeze Display Cases
There are 180 cabinets from Carrier (Electrolux) and Wica in the database of CyberMart. Additional cabinets can be specified as a text file and added by the user in the program. The data needed in the text file is:

Refrigeration Capacity at 22°C and 65% RH:
Refrigeration Capacity at 25°C and 60% RH:
Length:
Power of Fans:
Power of Light:
Power of Heating Wires:
Power of Defrost Heater:
Air Temperature at the Inlet of the Cabinet:
Air Temperature at the Outlet of the Cabinet:
Air Temperature in the Cabinet:
Evaporation Temperature:
Supplied Brine Temperature:
Return Brine Temperature:
Pressure Drop for Propylene Glycol:

Pipe Dimensions for Indirect Systems
Pipe Dimensions of the Secondary Refrigerant in the Medium and Low Temperature Units
Kind of Pipe (plastic, copper, steel):
Number of Parts of the Main Pipe:
Each Part of the Main Pipe Required:
Capacity:
Diameter:
Length:
Length Equivalent:
Price of the Pipe (in SEK/m):

Pipe Dimensions of the Coolant Fluid in the Medium and Low Temperature Units
Kind of Pipe (plastic, copper, steel):
Capacity:
Diameter:
Length:
Length Equivalent:
Price of the Pipe (in SEK/m)

Pipe Dimensions for Direct Systems
Each Compressor or Rack of Compressors Required for Suction, Discharge and Liquid Lines
Capacity:
Diameter:
Length:
Length Equivalent:
Price of the Pipe (in SEK/m):

LCC

The input data necessary for the calculation of LCC is:

Investment Costs of the Different Components in the Refrigeration System or the Total Investment Cost:
Cost of Operation, Maintenance, Repair and Others:
Period of Study:
Interest Rate:
Annual Price Increase of Electricity:
Electricity Price:
In the other columns, the results will appear from the calculation of investment cost, energy application and

TEWI

The input data required for the calculation of TEWI is:

Working Life of the Refrigeration Machine:
Loss Rate of Refrigerant during One Year in Percent:
Regional Conversion Factor:
References


Christensen, K. G. (1999). *Use of CO<sub>2</sub> as Primary and Secondary Refrigerant in Supermarket Applications*. 20<sup>th</sup> International Congress of Refrigeration, Sydney, Australia, IIR.


Jones, J. (2004). The Virtual Grocery Store: A proposal to Improve the Quality of Life for Retail Customers through a Virtual Environment. *Architecture*. Virginia, USA, Virginia Polytechnic Institute and State University: 82.


Rivet, P. (2001). *Ice Slurries for Indirect Cooling on Retail Sector: Results on Site Experimentation*. Third Workshop in Ice Slurries, Horw / Lucerne, Switzerland.


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