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Control of Refrigeration Systems for Trade-off between Energy Consumption and Food Quality Loss

Ph.D. Thesis

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Preface

This thesis is submitted in partial fulfilment of the requirements for the Ph.D degree at Automation & Control, Department of Electronic Systems, Aalborg University, Denmark. The work has been carried out in the period from October 2004 to December 2007 under the supervision of Professor Jakob Stoustup and Associate Professor Henrik Rasmussen from Aalborg university, and Associate Professor Arne Jakobsen from Technical University of Denmark.

This project is sponsored by Center for Model Based Control (CMBC) and Danish Ministry of Science and Technology (DMST) under Grant: 2002-603/4001-93. CMBC consists of seven industrial companies, three academic partners and one GTS (authorized technical service institute). Automation & Control, Department of Electronic Systems from Aalborg University, is the host institute for this Ph.D project, and also one of the three academic partners in the center. The refrigeration research activities within CMBC were initiated by Danfoss A/S, which is one of the seven industrial companies, co-sponsoring the center.

Aalborg University, Decemeber 2007
Junping Cai

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I am grateful to the Department of Advanced Engineering headed by Jürgen Süss for hosting me in Danfoss A/S, Nordborg, and all the support from Ph.D Jürgen Süss, Ph.D Claus Thybo, Ph.D Lars F. S. Larsen, Jan Prins, Klaus Lambers, Viggo Maegaard in this department. I like to thank Christian Bendtsen from Danfoss Electronic Control & Sensors, Ph.D Bjarne D. Rasmussen from Grundfos Management A/S, Denmark, Ph.D Morten J. Skovrup from Institute for Product Development, Technical University of Denmark (DTU), Denmark. Ph.D Jørgen Risum from BioCentrum, DTU. Ph.D John B. Jørgensen from Informatics and Mathematical Modeling, DTU. Ph.D Paw Dalgaard from Danish Institute for Fisheries Research, Denmark.

I greatly appreciate all the knowledge sharing from Ph.D Tommy Mølbak, Babak Mataji, Ph.D Dennis Bonne, Ph.D Sten Bay Jørgensen, Martin Riisgaard-Jensen, Ph.D Bodil Redcke, Ph.D Jørgen Knudsen, Ph.D Bao Lin, Ph.D Lars Henrik Hansen within CMBC.

Finally, I owe special gratitude to my family in China, my husband Tom B. Pedersen and my son Xuze for their love and continuous support, for sharing all the difficulties and happiness during this Ph.D study.

Abstract

In supermarkets, control strategies determine both the energy consumption of refrigeration systems and the quality loss of refrigerated foodstuffs. The question is, what can be done to optimize the balance between quality loss and energy consumption? This thesis tries to answer this question by applying two main optimization strategies to traditional refrigeration systems. The first strategy is a new defrost-on-demand scheme, which based on an objective function between quality loss and energy consumption, continuously seeks an optimal time interval for defrosting in dynamic situation. The second strategy is through utilization of the thermal mass of the refrigerated foodstuffs, the day-night temperature variation and the capacity control of the compressor, to realize a trade-off between system energy consumption and food quality loss.

During the past decade, concerns about food quality and food safety have been raised among consumers again and again. However, the food quality has never been included in the control loop of refrigeration systems and used as an active decision factor in order to design new controllers or to optimize the existing ones.

This thesis, for the first time, introduces a food quality model to the control loop of refrigeration systems. Thereby, investigations regarding the quality temperature relation of foodstuffs, and especially the detrimental effect of different defrost schemes on food quality can be carried out. Both food quality loss and system energy consumption are included in the considered objective function for optimization. Based on this objective, several novel optimal approaches are proposed throughout the thesis.

One example is a new defrost-on-demand scheme. It uses a feedback loop consisting of an on-line updating and estimation by an Extended Kalman Filter and a model-based optimization. The scheme automatically adjusts the time interval for the defrost cycles with varying operating conditions, and it continuously seeks the optimal time interval, featuring either an energy optimal time, or a trade-off between the energy consumption and the food quality loss. This adaptive approach is compared with traditional defrost schemes and is found to be able to reduce energy consumption significantly.

The potential benefit of using the refrigerated foodstuffs as thermal energy storage and shifting the cooling load to favorable operating conditions is investigated by solving a dynamic optimization problem. Significant saving on energy by utilizing the day-night temperature variation and foodstuffs' temperature variation tolerance is demonstrated. Furthermore, the Pareto optimality trade-off curve between system energy consumption and food quality loss is obtained.

To minimize the side effects of defrosting on the quality of refrigerated foodstuffs, a new defrost scheme is proposed. By utilizing the thermal mass of the air and the products inside the cabinet, the optimization scheme forces the compressor to work harder and to cool down more prior to the scheduled defrosts. This guarantees the temperature of the

product after the defrost cycles to be within the controlled safety level.

To prevent refrigerated foodstuffs in supermarkets from being discarded on hot days, an optimization strategy by the Model Predictive Control (MPC) is proposed. It deals with the specific condition of extremely high outdoor temperatures that causes compressors to saturate, and to work at maximum capacity. In a traditional control, refrigerated foodstuffs will suffer from a consequential higher temperature storage, which is detrimental to the food quality, and in worst case they have to be discarded according to the regulation from authorities. This will cause the shop owner a big economic loss. By utilizing the thermal mass of foodstuffs and their relative slow temperature change, Model Predictive Control foresees the situation, it will use more compressor power to cool down the foodstuffs in advance and prevent the high temperature storage from happening, thus saving the foodstuffs from being discarded.

Resume

I supermarkeder er reguleringsstrategierne afgørende for både energiforbruget i kølesystemerne og tabet af kvalitet i nedkølede fødevarer. Spørgsmålet er, hvad der kan gøres for at optimere balancen mellem kvalitetstab og energiforbrug? Denne Ph.D.-afhandling forsøger at give svaret på dette spørgsmål ved at implementere to hovedoptimeringsstrategier til det traditionelle kølesystem. Den første strategi er en ny afrimning-efter-behov metode, som baseret på en kostfunktion mellem kvalitetstab og energiforbrug, kontinuerligt søger at optimere tidsintervallet for afrimning under dynamiske betingelser. Den anden strategi er, gennem udnyttelse af fødevarernes termiske masse, variationerne i dag- og nattetemperaturerne, samt kapacitets regulering af kompressoren, at realisere et trade-off mellem systemets energiforbrug og fødevarernes kvalitetstab.

Gennem det seneste årti er bekymringerne om fødevarekvaliteten kommet op blandt forbrugerne igen og igen. Men fødevarekvaliteten er aldrig blevet betragtet som en del af kølesystemernes regulering og brugt som en besluttende faktor ved design af nye regulatorer eller optimering af de eksisterende.

Denne afhandling introducerer, for første gang, en fødevarekvalitets-model til kølesystemers regulering. Hermed kan der udføres undersøgelser af fødevarers kvalitets- og temperaturrelationer og specielt den skadelige effekt af forskellige afrimningsstrategier. Både fødevarernes kvalitetstab og systemets energiforbrug er inkluderet i den målrettede kostfunktion, som er taget i betragtning. Baseret på dette mål er adskillige nyskabende metoder foreslået i forløbet af denne afhandling.

Et eksempel er en ny afrimning-efter-behov metode. Den bruger en tilbagekoblingssløjfe bestående af en on-line opdatering og estimering ved hjælp af et Extended Kalman Filter og en model-baseret optimering. Metoden justerer automatisk tidsintervallet for afrimningscykluserne ved varierende operationelle betingelser og det søger kontinuerligt det optimale tidsinterval, enten som den energi-optimale tid eller som et trade-off mellem energiforbrug og tab af fødevarekvalitet. Denne tilpassende metode er fundet i stand til at reducere energiforbruget betydeligt, sammenlignet med traditionelle afrimningsmetode.

Den potentielle fordel af at bruge kølede fødevarer som termisk energilager og at flytte kølebelastningen til operationelt fordelagtige betingelser er undersøgt ved at løse et dynamisk optimeringsproblem. Der er eftervist betydelige energibesparelser ved udnyttelse af dag-nat temperatur variationerne, samt fødevarernes temperatur-variations tolerance. Yderligere er der opnået en Pareto trade-off kurve mellem systemets energiforbrug og fødevarernes kvalitetstab.

For at minimere afrimningens sideeffekter på kvaliteten af de kølede fødevarer er der foreslået en ny afrimningsstrategi. Ved at udnytte den termiske masse af luft og produkter i kabinettet tvinger afrimningsstrategien kompressoren til at arbejde hårdere og at

nedkøle mere før den planlagte afrimning. Dette garanterer at produkternes temperatur holdes inden for de kontrollerede sikkerhedsniveauer.

For at forebygge kassering af fødevarer i supermarkederne på varme dage foreslås en Model Predictive Control (MPC) optimeringsstrategi. Den opererer med de specifikke betingelser af ekstremt høje udendørstemperaturer, som medfører at kompressorerne udnyttes fuldt ud og arbejder på maksimal kapacitet. Ved en traditionel regulering er konsekvensen at fødevarerne lider under højere temperaturer som er skadelige for kvaliteten og i værste fald er det nødvendigt at kassere dem i henhold til myndighedernes regulativer. Dette påfører butiksejerne store økonomiske tab. Ved at udnytte den termiske masse af fødevarerne og deres relativt langsomme temperaturændringer, forudser Model Predictive Control situationen, og den vil bruge mere kompressorenergi på forhånd til at nedkøle fødevarerne og forebygge høje lagringstemperaturer. Herved spares fødevarerne fra at blive kasseret.

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Chapter 1

Introduction

1.1. Project background

This project is sponsored by the Center for Model Based Control (CMBC) and the Danish Ministry of Science and Technology (DMST) under Grant: 2002-603/4001-93.

CMBC consists of seven industrial companies, three academic partners and one GTS (authorized technical service institute). The overall target of CMBC is to strengthen the competitiveness of Danish industries with the applications of model based monitoring and control. It aims at improving the efficiency of industrial process and integrating model based monitoring and control into products [CMBC, 2004].

Automation and Control, Department of Electronic Systems from Aalborg University, is the host institute for this Ph.D project, and also one of the three academic partners in the center, with expertise in advanced control design.

The refrigeration research activities within CMBC were initiated by Danfoss A/S, which is one of the seven industrial companies, co-sponsoring the center. Danfoss is a world leader in controls, compressors, variable frequency drives, and solutions for refrigeration and air-conditioning, heating, and motion controls. Danfoss meets customers' increasing demands and higher standards through the approach, focused on bringing higher levels of engineering innovation and collaboration to the challenges of energy efficiency and environmental responsibility for its customers in the industries in which they operate [Danfoss, 2006].

1.2. Research object and objective

The research object in this Ph.D project is supermarket refrigeration systems. The layout of a typical supermarket is shown in Figure 1.1, where refrigerated display cases and storage areas are located around the store perimeter. The most commonly used refrigeration system for supermarkets today is the multiplex direct expansion (DX) system [Baxter, 2003], where all display cases and cold store rooms use direct expansion air refrigerant coils that are connected to the system compressors in a remote machine room, located in the back or on the roof of the store.

The objective is to investigate, further develop and specialize the control methods, which can be applied to a refrigeration system, to deal with the problem characterized by non-linear dynamics subjected to constraints and saturation, and seek a systematic way of optimizing the objective function with two criteria: system energy consumption and refrigerated food quality loss. This optimizing strategy should be realized by the

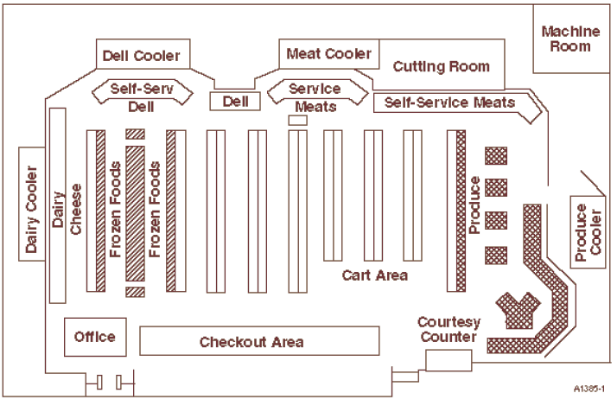


Figure 1.1: Layout of a typical modern supermarket [Baxter, 2003].

application of model based control on the top of the existing traditional control. For this purpose, several models have to be developed, but the purpose of modeling is for control, so the model itself should be relatively simple but still capture the main dynamics from an input/output point of view. The implementation and overall project prospective are illustrated in Figure 1.2, where MPC is the abbreviation for Model Predictive Control.

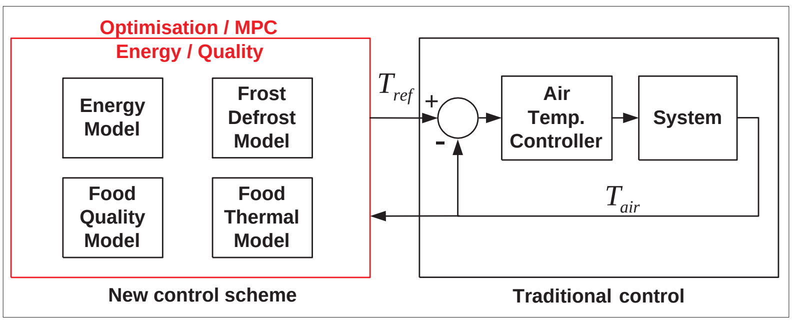


Figure 1.2: Implementation and overall project prospective.

1.3. Motivations

As we mentioned above, in this project, we will focus on an optimal control with respects to system energy consumption and food quality loss. The driving force behind this project will be analyzed on social, legal, economic and technical aspects, as shown in Figure 1.3. It is difficult to distinguish clearly the interests between different parties, and what we elaborate here is only the main concerns for the project.

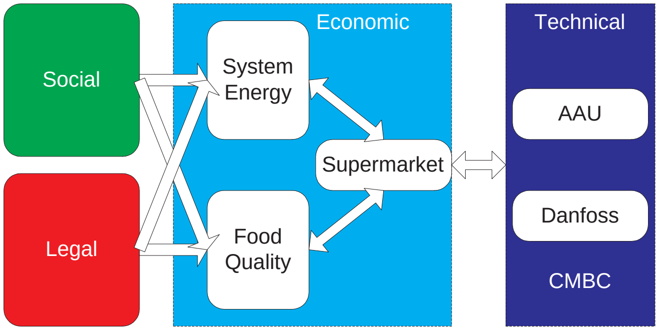


Figure 1.3: Motivations behind the project.

1.3.1. Social concerns on energy

Supermarkets are one of the most energy-intensive types of commercial buildings. Significant energy is used to maintain chilled and frozen food in both product display cases and storage refrigerators.

There is a wide range in size of supermarkets. In Europe stores range in size from about 500 m^2 to 3,000 m^2 or somewhat larger. Stores are typically larger in Canada and the US, ranging from a minimum of about 1,000 m^2 to 10,000 m^2. Plant capacities range from 30∼60 kW for small markets to over 400 kW for the largest stores. Similarly, annual energy use ranges from about 100,000 kWh/y for the smaller stores to 1.5 million kWh/y or more for the largest [Baxter, 2003]. Refrigeration is the largest component of supermarket energy use, accounting for half or more of the store totals, see Figure 1.4, where SCE is the abbreviation for Southern California Edison.

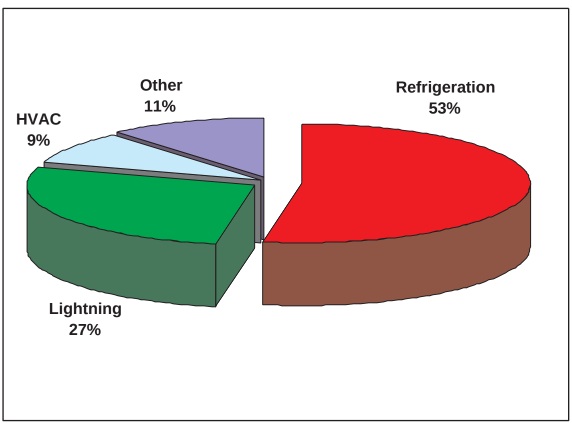


Figure 1.4: Electricity use in supermarket, where refrigeration accounts for more than half of the total energy consumption (source SCE's Field Monitored Data).

1.3.2. Legal issues regarding energy

The US Environmental Protection Agency (EPA) released the first voluntary energy consumption standard for commercial refrigerators and freezers in 2001. Several states, led by California, have since released legislation mandating maximum energy consumption, with effective dates from March 2003 through January 2008. The standards typically start less stringent than Energy Star (A United States government program to promote energy efficient consumer products), but over time achieve the Energy Star recommended limits. The states are also releasing similar legislation to limit the allowable energy consumption of commercial ice cream freezers, vending machines, ice machines, walk-in refrigerators and freezers. In 2005, the Air-Conditioning and Refrigeration Institute (ARI) released a proposal to the federal government to nationalize energy consumption standards for commercial refrigerators and freezers. The effective date in this proposal was recommended to be January 2010, with the final energy consumption standards equal to both the original Energy Star proposals and the final California standards [Lehman, 2006].

According to ARI, the efficiency levels contained in the law will reduce peak power needs by an estimated 8,000 MW by 2020, which is equivalent to the output of 27 new power plants of 300 MW each.

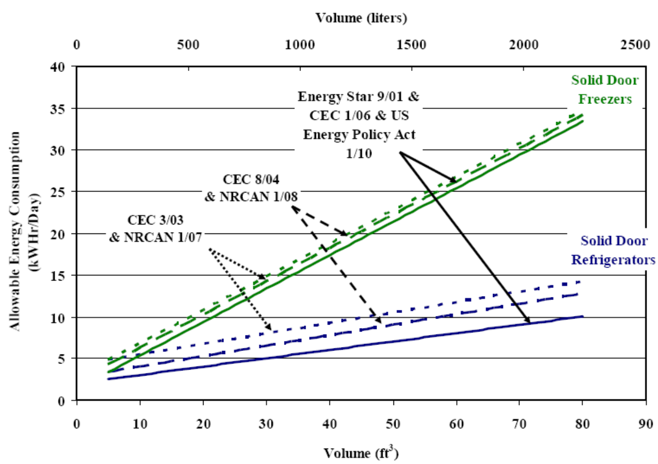


Figure 1.5: Energy Star, CEC, NRCAN, and US Energy Policy Act consumption limits [Lehman, 2006].

Figure 1.5 shows the electricity limit set by Energy Star, California Energy Commission (CEC), Natural Resource Canada (NRCAN) and US Energy Policy Act, and their effective dates. Figure 1.6 shows a sample size of 25 ft^3 (707.9 l) units, the energy reduction over time imposed by CEC.

Manufacturers of all commercial refrigeration equipment must be aware of these standards, and must prepare their products to meet these increasingly more stringent standards.

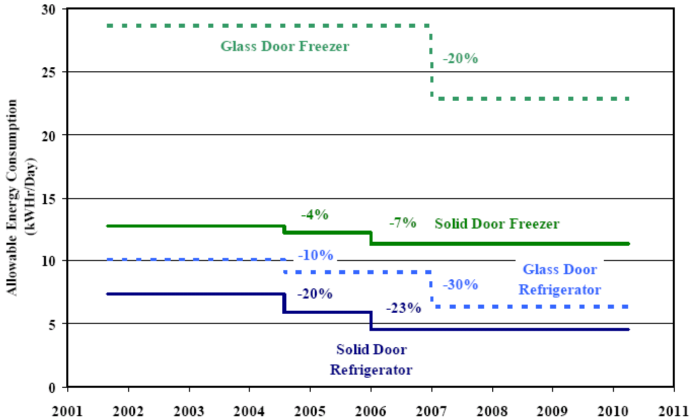


Figure 1.6: CEC Standards on energy consumption for all 25 ft^3 (707.9 l) units [Lehman, 2006].

1.3.3. Social concerns on food quality and safety

During the last decade, concerns about food quality and food safety have been raised among consumers. Several sector-wide crises, like the BSE crisis, dioxin crisis, classical swine fever and foot and mouth disease in Europe have fueled these concerns. Consumers in industrialized countries have become more aware of potential food hazards through greater media coverage [Unnevehr, 2000] [Opara and Mazaud, 2001].

According to the statistics from CDC (Centers for Disease Control), food-borne illness cause:

- 76 million sick per year.
- 325,000 hospitalized per year.
- 5,000 deaths per year.
- Food-borne illness cost U.S an estimated 22 billion dollar per year.

Trienekens et al. [Trienekens et al., 2005] pointed out that one of the major trends related to food quality and safety in the EU is "Consumer awareness regarding food safety and quality". Opportunities for the food industry will be:

- Traceability to gain consumer confidence.
- Knowledge and experience dissemination through the chain.
- Monitoring to ensure safety and quality of food.
- Possibilities for self regulation.

1.3.4. Legal issues regarding food quality and safety

Food quality and safety standard

Three most important generic quality assurance systems in the food sector are Good Agricultural Practices (GAP), Hazard Analysis of Critical Control Points (HACCP) and International Standard Organization (ISO) [Trienekens, 2006].

- GAP systems include a set of guidelines for agricultural practices aiming at assuring minimum standards for production and storage.

- HACCP is a systematic approach to the identification, evaluation and control of those steps in food manufacturing that are critical to product safety. Currently HACCP principles are the basis of most food quality and safety assurance systems (Codex Alimentarius, EU and US food legislation, most private standards). HACCP aims at control of hazards instead of end-of-pipe inspection.

- ISO standards. Recently ISO 22000 has been launched as a new standard covering the whole food supply chain and including attention to HACCP and GAP requirements.

Standard distribution in EU and Denmark

In EU, since 1998, HACCP is obligatory for all companies in the food chain, except for the primary producer. In many countries standards have been developed which even go a step further. For example, in Denmark a HACCP norm is accepted that includes specific attention to provision of management information. This system -DS 3027 [DanishStandards, 2007] is on the forefront of quality system development and is one of the pillars of the new ISO standard 22000 on food quality. Denmark is also in front of developing principles for self-monitoring [Trienekens, 2006].

Self-monitoring is also known as own-control. Its authorization is ensured by trade codes or approval from an authorized laboratory under The Danish Veterinary and Food Administration (DVFA). For instance, DSK (the Danish Grocers Association) in cooperation with Coop Denmark and Danish Supermarket, the leading Danish retail multiples, has developed a trade code for retail business [Esbjerg and Bruun, 2003].

For example, it recommends the following storage temperature for supermarkets:

- Frozen food, the maximum temperature is -18°C.

- Eggs, the maximum temperature is +12°C.

- Store packed minced meat, or minced meat bound with flour and egg, the maximum temperature is +5°C.

- Fresh fish and products, the maximum temperature is +2°C.

- Milk, the maximum temperature is +5°C.

1.3.5. Economic concerns

The economic impact of refrigeration technology throughout the world is already very impressive, and more significant than is generally believed. While the yearly investment in machinery and equipment may approach US$100 billion, the value of products treated by refrigeration is perhaps ten times this amount [Lorentzen, 1987].

United Nations Environment Programme (UNEP) estimates give an indication of the amount of refrigeration in use [UNEP, 2006]. Worldwide, some 760 MT of foodstuffs are refrigerated annually, and 300 MT are in refrigerated storage at any one time. Commercial cold stores have a capacity of 300 million m^3 [MBE, 2007].

Supermarkets have a bottom-line reason for being on the leading edge of energy-related research. They operate on the thinnest of profit margins, in a highly competitive industry. One way to generate more dollars of profit is to increase store sizes, so as to sell more products. The other way to do it is to reduce operating costs [Powell, 2002]. Just as important, operating in such a market demands focus on customer satisfaction and customer loyalty to obtain positive net operating profits. Investigation by Kristensen et al. [Kristensen et al., 2001] showed that the most important drivers of customer satisfaction and customer loyalty is product quality.

Good refrigeration performance and management help to keep food at the correct temperatures and minimize electricity costs for refrigeration. Besides providing the proper training to employees with the best practises, the owners of the supermarket will push the manufacturer of refrigeration systems, to provide energy efficient and food quality reliable refrigeration facility.

Food quality reliable means that the system should not only perform well in normal conditions, but also be able to keep the risk of food spoilage at a minimum level, when it operates in abnormal conditions, such as defrosting, or during extremely hot days, which are known problems for most of refrigeration systems today.

1.3.6. Technical prospective

Danfoss is one of world leading suppliers for commercial refrigeration systems. In order to keep its market share and leading position, Danfoss should always stand in the front line of technology innovation. A good collaboration between Danfoss and Aalborg University, utilizing both academic and technical expertise, will provide a state-of-the-art solution.

Summing up, the analysis from social, legal, economic and technical aspects shows a strong demand on a new optimal control technology for commercial refrigeration systems, which can achieve among others, a weighted objective with system energy efficiency and food quality reliability.

1.4. Simple introduction to commercial refrigeration systems and refrigeration cycle

The refrigeration system used in supermarkets is to remove the heat from a cold reservoir such as display cabinets, to a hot reservoir such as surroundings by a vapor compression cycle, in order to keep the food inside the display cabinet at a desired low temperature, in order to slow down the food decay.

There are various layouts of refrigeration systems in supermarkets, while the most commonly used today is the multiplex direct expansion (DX) system, as shown in Figure 1.7, where the condenser is placed on the roof of the shop building, to exchange the heat with surrounding air.

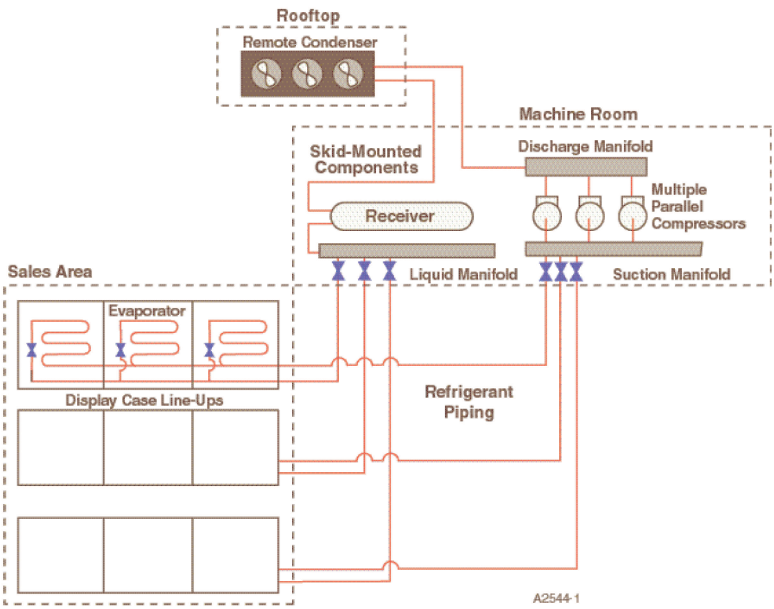


Figure 1.7: Multiplex refrigeration system [Baxter, 2003].

1.4.1. Component and process

A simplified layout of a basic refrigeration system is shown in Figure 1.8. It consists of four main components: compressor, condenser, evaporator and expansion valve. Besides there are evaporator fans and condenser fans, which move the air cross the coil of a heat exchanger to increase heat transfer.

A vapor compression cycle composes of four processes: compression, condensation, expansion and evaporation, shown in Figure 1.9, and the figure also shows the $\log(p) - h$ diagram of the cycle.

Compression 1-2: Where a compressor compresses the entered refrigerant in gas phase from low temperature and pressure into high pressure. Ideally compression is an isentropic (reversible and adiabatic) process (from point 1 to 2s), while the actual work (from point 1 to 2) is always expressed by isentropic efficiency, which is used to express the energy efficiency of a compressor by comparing its energy consumption with the energy consumption necessary for an ideal compression process. Isentropic efficiency can have values between 0 and 1. Typical values are 0.4 to 0.5 for small hermetic compressors, 0.5 to 0.7 for semi-hermetic compressors and 0.5 to 0.8 for large

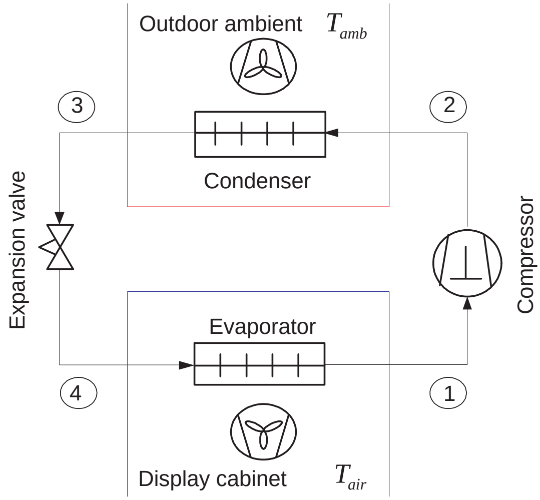


Figure 1.8: A simplified layout of a refrigeration plant (components).

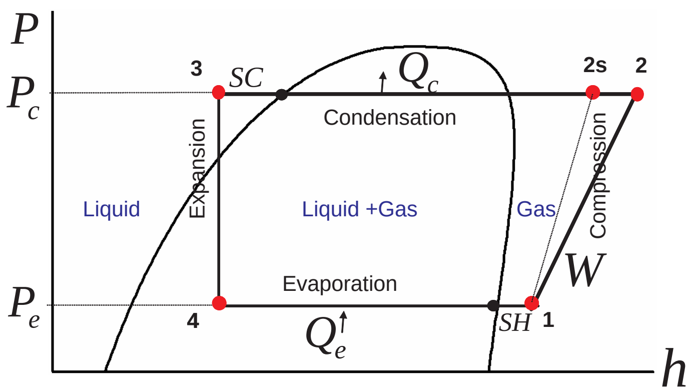


Figure 1.9: Vapor compression cycle in a $\log(p) - h$ diagram.

open compressors. The work of compressor W can be expressed as the following:

$$W = \dot{m}_{ref} \cdot (h_2 - h_1)$$
$$= \dot{m}_{ref} \cdot (h_{2s} - h_1)/\eta_{is}$$

Where $\dot{m}_{ref}$ is the refrigerant mass flow rate, η_{is} is the isentropic efficiency, and h is for enthalpy, subscript 1, 2, and $2s$ are for different state points.

Condensation 2-3: Where a condenser is used to exchange the heat from the hot gas phase refrigerant to a cold surrounding by condensation, and the gaseous refrigerant change to liquid, and finally little sub-cooling (SC), ensure all the refrigerant is in liquid state before it enters the expansion valve. During the condensation process, heat rejected to surroundings Q_c is expressed as the following:

$$Q_c = \dot{m}_{ref} \cdot (h_2 - h_3)$$

Adiabatic expansion 3-4: Where an expansion valve separates high pressure part with low pressure part, with high pressure drop, the liquid refrigerant partially evaporates and becomes a gas liquid mixture with low temperature and pressure before entering evaporator.

Evaporation 4-1: Where an evaporator exchanges the heat with a hot source, with heat absorbed, low temperature mixture of the refrigerant evaporates at a constant pressure, and becomes a gas, at the end of the evaporator, all refrigerant evaporates and temperature increase a little, so called super heat (*SH*), which ensures only the gas refrigerant enters the compressor again. During this process, the cooling capacity Q_e is expressed as the following:

$$Q_e = \dot{m}_{ref} \cdot (h_1 - h_4)$$

1.4.2. System Coefficient Of Performance (COP)

COP is the ratio between the refrigeration capacity of the evaporator and the power consumption of the compressor, as expressed as the following:

$$COP = \frac{Q_e}{W}$$

1.4.3. Normal operation

In a simple control structure, there are mainly the following controls: condenser pressure control, suction pressure control, super-heat control and air temperature control. The condenser pressure is controlled by turning on and off a number of condenser fans or by fans equipped with frequency converters to provide a continuous control. The suction pressure is controlled by turning on and off the compressors. The super-heat is normally controlled by a mechanical thermal expansion valve together with an electronic on/off inlet valve. The display cabinet air temperature is normally controlled by a hysteresis controller that opens and closes the inlet valve to control the flow of refrigerant into evaporator, thus keeping the air temperature within the specific bands.

1.4.4. Frosting and defrosting

Frost formation and defrost methods: Frosting is a well known and undesirable phenomenon on evaporator coils. It happens whenever the surface temperature of the evaporator is below 0°C and humid air passes by. Frost decreases the performance of the heat exchanger. In order to maintain a satisfactory performance, defrosting needs to be done regularly.

The most common defrost methods for medium and low temperature applications are:

- Off-cycle defrost: This is the simplest defrost methods: refrigeration is stopped, evaporator fans continue to move room air over the frosted coil surface, which warms and melts the frost.

- Hot/cool gas defrost: During a hot/cool gas defrost, the normal supply of cold refrigerant is stopped. The former involves the circulation the hot gas from the compressor discharge manifold directly to the display cabinet, and the latter utilizes cooler gas from the liquid receiver. The cool or hot gas condenses in the evaporator, releasing heat which melts the ice from the coil.

- Electric defrost: This approach uses the electric heater embedded on the fin surface to supply heat, warm and melt the frost.

However, not all methods can be used in all circumstances. Normally, for medium temperature applications, the cheapest and least energy consuming means of defrosting is off-cycle. This method of defrosting is widely used in the industry. For low temperature applications, the use of gas defrosting is more energy efficient than electric defrost, but the considerable extra capital costs lead to a long payback period. Consequently, electric defrost is the most commonly used defrost method for low temperature applications.

Defrost schemes: In a simple taxonomy, there are the following two classes of defrost schemes.

- Scheduled defrost: Initiating the defrost cycle by a timer, normally with a fixed number of defrost cycles per day. Defrost is terminated either based on a fixed time or on a stop temperature while with a maximum defrost time as a security.

- Defrost-on-demand: Initiating the defrost cycle only when necessary. This approach normally uses one parameter to initiate and terminate the defrost process, such parameters could be: air pressure difference across the evaporator, temperature differential between the air and the evaporator surface, fan power sensing etc.

1.5. Ideas and review of previous related work

In this project, we will look closely into supermarket refrigeration systems, focusing on the optimal control with a balanced system energy consumption and food quality loss. Even though they are in the same refrigeration process, to ease understanding, we will talk about our ideas and related work separately, and under two situations: normal operation, frosting and defrosting.

1.5.1. Normal operation

Under normal operation, our idea is to investigate and utilize the possibility of using refrigerated foodstuffs as a thermal mass to shift the cooling loads into more favorable operational periods, such as low energy cost period, lower outdoor temperature etc, allowing an opportunity to either reduce system operating cost, without sacrificing food quality; or reduce the food quality loss by utilizing cheaper power; or prevent food from being discarded under extreme situations, where the normal systems fail. Contradictory to the traditional control of refrigeration systems, food quality is used here as an active decision factor.

The idea of exploiting thermal mass storage as a means for saving energy has been suggested in other areas, such as in the HVAC system of commercial buildings, and many years application has proved a big saving potential on energy and electric bills. Braun [Braun, 2003] presented a method by optimizing zone temperature setpoint to minimize the utility cost over a billing period, and demonstrated a significant saving potential by utilizing building thermal mass as one passive thermal storage inventory. His analysis showed that the savings are sensitive to many factors, including utility rates, type of equipment, occupancy schedule, building construction, climate conditions and control strategy, etc.

Ice storage technology as one active thermal storage inventory has also been used in HVAC systems and commercial cooling plants for years, the main idea is to use the ice as a kind of battery, produce ice when it is cheap during the night on a low rate, and use the ice for cooling during the peak period, see examples [Henze et al., 1997] and [Henze and Krarti, 1998].

A combination of both active thermal storage (ice storage) and passive thermal storage (building) was investigated by Zhou et al. [Zhou et al., 2005]. They revealed that by proper control of charge and discharge these two thermal storage batteries, the utility cost saving is significantly greater than either storage, but less than the sum of the individual savings.

This idea has also been applied in industrial refrigeration, when the quantity of food is huge. For example, Altwies and Reindl [Altwies and Reindl, 1999] presented a strategy of using stored frozen vegetables as thermal mass to shift the cooling loads in a refrigerated warehouse with full capacity of 22 million kg of frozen vegetables. Their simulation and experiment showed that full demand shifting (completely shut down of all refrigeration equipment during on peak time period) reduces the operating cost by 53%, but requires the greatest installed refrigeration capacity; load levering (meet the refrigeration load by running the equipment at a constant level through out a 24 h period) saves less energy, but requires least installed equipment capacity; combination of both full demand shifting and load levering leads to a saving of 37% on energy. Contradictory to conventional controls, which try to keep the food temperature at a constant value, a maximum of -18°C, they controlled the surface temperature (worst case) of frozen food to stay below -18°C, but allowed it to vary. But there is no food quality models involved, no direct quantitative monitoring of food quality change during the process.

While in supermarket refrigeration systems, this idea has not been utilized till now. This may be due to the facts that smaller thermal mass of foodstuffs inside the display cabinet compared with buildings or foodstuffs in industrial warehouse; or the difficulties of food temperature measurement, and the complexities of food quality modeling; or the difficulties of time-varying cooling load prediction; or the common sense that fluctuating temperature is detrimental to food quality (even though it is not always true). So the optimization of refrigeration systems is mainly focusing on optimizing the system without considering the thermal storage, without exploiting knowledge of future load situations.

Control and optimization of refrigeration systems

Assume that during the design stage, you have done thorough research and investigations based on cooling demand, legislations from authorities regarding the utilization and the leakage of refrigerant, and life cost analysis, such that you have selected a right system among traditional multiplex direct expansion system, distributed compressor systems, secondary loop systems, or self-contained display cases [Baxter, 2003], [Walker and Baxter, 2003], charged the system with efficient, environmental friendly and cost optimal working fluids [Usta and Ileri, 1999] [Bhattacharyya et al., 2005], and selected the right components [Gordon et al., 1997] [Shiba and Bejan, 2001], then the next thing that you can do and should do to achieve an optimal system performance is to have a better control.

With only one on/off actuator as in a domestic refrigerator, there is little scope for optimization of the performance. While with the recent advance in automation technology, most of industrial refrigeration plants have a number of actuators, such as electric motors, variable speed fans, which has made the refrigeration system more drivable, and also means it is possible to provide the refrigeration power for a desired temperature by various combinations of compressor speed, evaporator fan and condenser fan speed, temperatures setpoints, number of compressors, or number of condensers etc, so called capacity control. Research on capacity control of refrigeration systems and heat pumps was reviewed by Wong and James [Wong and James, 1988]. In paper [Wong and James, 1989], six main types of capacity control systems were identified: on/off control, two speed compressor, variable speed compressor, cylinder unloading, suction throttling and hot gas bypass. Based on the investigation of the influence of those control systems on the performance of refrigeration systems, they concluded that this order represents the most energy efficient control, however with the exception of variable speed the reverse order provides better set point accuracy.

With the help of computer technology, advanced control theory and efficient optimization algorithm, on-line real time optimization is becoming more feasible and applicable, for example, a computer-based energy management system has been installed in the world's largest integrated nylon plant to optimize the refrigeration systems, and proved a substantial energy cost reduction [Cho and Norden, 1982].

Jakobsen et al. [Jakobsen et al., 2001] argued that there are two corrective actions to improve the system energy efficiency, one is by component maintenance and replacement, to keep the individual components at their best available efficiency; the other way is to use system intelligent capacity control, to achieve an optimal power consumption. The sensitivity analysis regarding total power consumption to deviation of different active components operating from their optimal value showed that total power consumption is quite insensitive to relatively small deviations of the capacities of the auxiliary equipment from their optimal values, but the relationship is quite progressive, which means that doubling their capacity leads to a significant increase of the total power consumption. Larsen and Thybo [Larsen and Thybo, 2004] presented an algorithm that by steady state setpoint optimization, gradient based method, to drive a refrigeration system to a power optimal condition when the outdoor temperature changes. In their optimizations, they all restricted the cabinet temperature to a constant value according to the legislation of food storage.

Biotechnology

Modeling: In order to control and monitor food quality change during the process, we need to know its deterioration mechanism. In conventional controls, we only measure and control the cabinet air temperature, while it is the food temperature that determines the food quality (if we assume the food is properly packaged, so the other environmental factor, such as air humidity or air pressure, etc can be neglected). To convert the air temperature to the food temperature, we need to model the heat and mass transfer in refrigerated foodstuffs. A review paper regarding heat and mass transfer in frozen foods was presented by Pham [Pham, 2005]. As freezing and chilling are very important operations in the food industry, there has been a lot of research carried out on modeling of freezing and chilling process, see example [Davey and Pham, 2006], they developed a model for predicting the dynamic product heat load and weight loss during beef chilling by using a finite element analysis.

Regarding the food quality, most of the research is based on experiments and data fittings [Badii and Howell, 2002] [Aubourg et al., 2005]. Few physical models exist, for example, Dalgaard [Dalgaard, 2005] developed a software- Seafood Spoilage and Safety Predictor (SSSP) to predict the shelf-life of seafoods, and the growth of bacteria on seafoods. They are based on Relative Rate of Spoilage (RRS) models and Microbial Spoilage (MS) models, and use direct product temperature profiles.

Recent years, Time Temperature Indictor (TTI), as a simple and effective way of measuring cumulative temperature histories of products along the cold chain (refrigeration from processor to receiver to warehouse to retail to consumer), has become very popular [Toukis et al., 1991], but they are only used as passive recording and monitoring, not for active control, food quality has never been included in the refrigeration control loop, and used as an active decision parameter.

Temperature fluctuation, is that safe for food? The advantages and benefits of load shifting by using refrigerated food as thermal mass are cost savings and reduced on-peak demand for the electric utility. Since it uses the dynamic interrelation of the food and air temperature, this strategy will for sure cause the temperature fluctuation of refrigerated foodstuffs, possible disadvantages include risk to the product quality, shelf life, and nutrient content need to be investigated before such kind of strategy is implemented. Energy and cost savings will be of no use if they lead to degradation in product quality.

Experiments on the influence of fluctuating temperatures were reviewed by Ulrich [Ulrich, 1981] and little evidence of any reduction in keeping quality due to temperature fluctuations was reported at temperatures colder than -18°C.

Gormley et al. [Gormley et al., 2002] tested the samples of frozen raw salmon, smoked mackerel, stewed pork pieces, ice cream, pizza, hollandaise sauce, strawberries, and blanched broccoli by subjecting them to temperature fluctuations below the freezing point, involving three temperature fluctuation cycles of -30°C to -10°C to -30°C on consecutive weeks, followed by storage at a constant -30°C for 8 months. The samples were compared with duplicate sets held at a constant -30°C (control), result showed that the temperature regimes had a minimal effect on texture, color, water-holding capacity and drip loss on thawing for most of the products.

Studies by Ashby et al. [Ashby et al., 1979], Aparicio-Cuesta and Garcia-Moreno [Aparicio-Cuesta and Garcia-Moreno, 1988], Boggs et al. [Boggs et al., 1960], Gortner et al. [Gortner et al., 1948], Hustrulid and Winter [Hustrulid and Winter, 1943] and Woodroof and Shelor [Woodroof and Shelor, 1947] involved a variety of food products (spinach, cauliflower, peas, strawberries, and others). Each study investigated effect of temperature fluctuations over ranges of interest for each particular project. The studies that never allowed frozen products to exceed 0°C consistently, obtained results showing, little to no degradation attributable to fluctuations in the storage temperature. The studies allowing extreme temperature fluctuations with freeze and thaw cycles reported significant quality losses.

All these investigations indicate that for frozen food, constant storage temperature is not critical, if the product is properly selected, prepared and packaged, and if the storage temperature never cause phase change.

1.5.2. Frosting and defrosting

Under frosting and defrosting, one of our ideas is to use food quality as an active decision factor, together with system energy, to determine an optimal cooling time interval between defrost cycles. A new defrost-on-demand scheme. This scheme should automatically adjust the time interval between defrost cycles with varying operating conditions, continuously seeking an optimal time interval, featuring either an energy optimal time, or a trade-off between energy consumption and food quality loss; another idea is to use thermal mass of foodstuffs to cool down more before the scheduled defrost, in order to minimize the quality deterioration of refrigerated foodstuffs as a side effect of defrosting.

Frost build up is a complex process even on a flat plate, and it is affected by many factors, such as air flow velocity, air temperature, air humidity ratio, and plate temperature etc, so most of the research is based on experiments. Sahin [Sahin, 1994] concluded that the thickness of the frost layer is affected primarily by the air humidity ratio, plate temperature, and air temperature, while the effect of Reynolds number is less significant. High humidity ratio, Reynolds number, and temperature difference between the air stream and the plate all yield high mass deposition rates. Frost density depends primarily on frost surface temperature, besides the other parameters. Analytical or numerical modeling exists with different limitations, see example [Yang and Lee, 2005], they proposed a mathematical model to predict the frost properties, heat and mass transfer within the frost layer formed on a cold plate. In their study, laminar flow equations for moist air and empirical correlations for local frost properties are employed to predict the frost layer growth.

Frost growth on real evaporator become more complex, for example, a commonly used fined tube evaporator, material, size, shape and distribution of fins on the tube, layout or structure of the tube plate, all influence the air flow, heat transfer rate, frost growth, and frost distribution or even blockage, compare with modeling of frost formation on flat plates, limited modeling existed, see example [Yang et al., 2006].

While in a refrigeration system, evaporator never works alone, besides coordinating with compressors, condensers and expansion valves, it works closely with fans. Each fan has its own characteristic curve, which related the static pressure and the air flow rate, it must match the system pressure drop under frosting, to determine its actual operation point. This on another side, will influence the fan efficiency and its power consumption. When the air flow rate decreases with time due to frost build up, the overall heat transfer coefficient between the air and the evaporator will decrease, in order to maintain the same cooling demand, the temperature drop of the air across the coils must increase, this in turn will cause the evaporating temperature to decrease. The drop in the evaporating temperature can be appreciable, and cause an increased power consumption to the plant [Stoecker, 1957]. All those operations are interdependent, so predicting the system performance under frosting condition is definitely not trivial, for example see [Chen et al., 2003], in this paper, a validated numerical model for frost growth on heat exchanger fins from Chen et al. [Chen et al., 2000] was modified to simulate a fan-supplied finned heat exchanger under frosting conditions. It was found that frost growth on the heat exchanger causes a dramatic drop in the fin heat rate, airflow rate, and fin efficiency while the pressure drop increases. For another example see [Martinez-Frias and Aceves, 1999], they incorporated the frost formation model into the evaporator subroutine of an existing heat pump model, to calculate performance losses due to frosting as a function of weather conditions, and time of operation since the last evaporator defrost. Performance loss calculation included the effect of air pressure drop through the evaporator, and the reduction in evaporator temperature caused by the growth of the frost layer. Results showed frost formation parameters and heat pump COP as a function of time and ambient conditions. It indicated that there is a range of ambient temperatures and humidities, in which frosting effects are most severe, and this range was explored to calculate heat pump operating conditions.

Frosting affects the performance of the heat exchanger by decreasing the effective air flow area and increasing the thermal resistance between the warm air and the cold refrigerant inside the evaporator. This decrease in free flow areas results in a lower air flow and an increased pressure drop across the coils. This performance degradation will become severe with time if nothing is done, in the worst situation, the system will be iced up and break down. In order to maintain a satisfactory performance, evaporators need to be defrosted regularly. A literature survey on frosting and defrosting of air coils refers [Fahlen, 1996b]. In his Ph.D thesis [Fahlen, 1996a], Fahlen proposed a new approach on how to obtain practical values of frost density and thermal conductivity from experimental investigations of ordinary air coils. The problems of defrost controls were discussed, and a number of alternative methods were compared.

Scheduled defrost is a traditional means of managing defrost cycles in supermarkets. It is simple and uses a low cost controller, so it is the most commonly used defrost scheme in today's supermarkets, but the time schedule is normally determined based on experience and observation, most cases, based on worst case conditions. It is configured during the commission stage and can not automatically adapt to the varying shop conditions under which the system is working, so the time between two defrost cycles can be either too long or too short. An improved version on a time based defrost is called adaptive defrost [Allard and Heinzen, 1988], upon initial installation, the controller starts with a default timed-based defrost program, then, as it learns from previous defrosting history, gradually adjusts the frequency of rest cycles.

Due to the disadvantages related with scheduled defrost, over the years, there have been many attempts to develop defrost-on-demand schemes, that is, initiating the defrost cycle only when necessary, see examples [Llewelyn, 1984] and [Muller, 1975], they proposed to use thermal conductivity of ice and fan power sensing as defrost signal respectively. These approaches typically involved installation of additional sensors to detect frost build up. It normally uses one parameter to initiate and terminate the defrost cycle. The threshold of this detected parameter is determined mainly to ensure a safe operation, or maintain the degradation in the performance of the system within fixed limits over the whole range of operating conditions. For example, a defrost initiation pressure was selected to limit the degradation in performance due to frosting to less than 10% of the steady-state value [Tassou and Marquand, 1987]. Without extra sensors, another defrost-on-demand scheme based on fault detection was proposed by Thybo and Izadi-Zamanabadi [Thybo and Izadi-Zamanabadi, 2004], it is based on the phenomenon of reduced air circulation due to frost build up. In the existing defrost-on-demand schemes, no energy optimality is guaranteed.

None of the existing schemes have used food quality as a decision factor.

When defrosting the system, no matter what kind of defrost methods it is applying, due to the fixed head cost, such as warming fins, warming tubes, heat loss etc, the more frequent defrost, the more energy can be wasted. It is estimated that only 15∼25% of total defrosting energy is actually carried out by the frost condensate leaving the drain pan [Niederer, 1976].

During defrosting, the air temperature inside a display cabinet will normally increase, and so will the food temperature. Depending on the defrost method, the food temperature will usually stay out of the normally controlled range for a period of time, this is harmful to the food quality [Cai et al., 2006c], especially when they suffer a number of defrosting cycles during the storage. While in today's defrosting schemes, no optimization strategy is implemented to minimize this detrimental effect, expect by some complex physical rearrangement of hot air circulation, or using double evaporators, etc.

1.6. Contributions

The main contributions in this project are illustrated in Figure 1.10, numbered from 1 to 5. This project requires a lot of modeling work, such as the heat transfer modeling (convert the measured air temperature T_{air} to the food temperature T_{food}), food quality modeling (modeling of food quality loss Q_{food} during different processes), frosting and defrosting modeling (not shown in the figure), and energy modeling (modeling of system energy consumption E_{sys}), while the main purpose of the modeling is to identify the problem, design a new controller and build a new control architecture, in order to optimize the existing control.

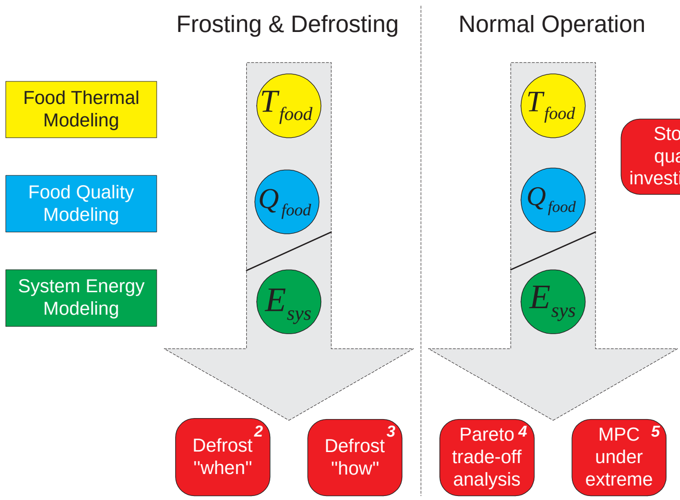


Figure 1.10: Main contributions in this project.

1.6.1. Contribution 1

Investigation of the detrimental effect of different defrost schemes on refrigerated food quality. For this purpose, a dynamic heat transfer model is developed to deal with phase change problems with time varying boundary conditions, in order to convert the traditionally measured air temperature to the product temperature [Cai et al., 2006a]

[Cai et al., 2006b]. Quality model of refrigerated foodstuffs based on the food temperature is developed to evaluate the quality loss during different operations [Cai et al., 2006c] [Cai et al., 2006d].

1.6.2. Contribution 2

Proposal of a new defrost-on-demand scheme, focusing on scheduling defrosting based on an objective function with two parameters: system energy consumption and refrigerated food quality loss. System energy includes two parts: one is the extra energy used to compensate the degraded system performance due to frost build up, another is the defrost direct energy. It uses a feedback loop consisting of on-line model updating and estimation as well as a model based optimization. This scheme automatically adjusts the time interval between defrost cycles with varying operating conditions, continuously seeking an optimal time interval, featuring either an energy optimal time, or a trade-off between energy consumption and food quality loss. This adaptive approach is compared with traditional defrost schemes, found to be able to reduce energy consumption significantly [Cai et al., 2008d] [Cai et al., 2008c].

1.6.3. Contribution 3

Proposal of a new defrost scheme, focusing on how to utilize thermal mass of refrigerated foodstuffs, cool down more before the scheduled defrost, in order to minimize their quality deterioration as a side effect of defrosting [Cai and Stoustrup, 2008].

1.6.4. Contribution 4

Identification of the potential by using refrigerated food as thermal mass to shift refrigeration load, to achieve a reduced system operating cost with guaranteed food safety and quality, and a reduced peak power consumption. A Pareto optimal curve is found by the off-line optimization [Cai et al., 2008a]. The optimality can also be achieved by a predictive control strategy, such as MPC.

1.6.5. Contribution 5

Proposal of an optimization strategy by utilizing the thermal mass of refrigerated foodstuffs and their relatively slow temperature change, as well as the excellent features of MPC, to prevent the refrigerated foodstuffs from being discarded during hot days, when the traditional control fails [Cai et al., 2008b]. It illustrated the potential of linear MPC in dealing with nonlinearities, handling constraints and saturation, and realizing multiple control objectives.

1.7. Outline of thesis (Publications)

The remaining part of the thesis is organized as the following (most of the papers here are in the extended version of the publications):

Summary of Work. This chapter summarizes the work in this Ph.D project. It aims at giving fast facts regarding the problems which need to be solved, how to solve, then, the conclusions, and their application prospective. The application prospective is added in this chapter, with the purpose of giving refrigerating engineers some insights and general application guidelines.

Overall Conclusions and Recommendations. This chapter gives a follow-up of the project objective, and some recommendations.

Dynamic Heat Transfer Model of Refrigerated Foodstuffs. This paper discusses the dynamic heat transfer model of foodstuffs inside a display cabinet, one-dimensional dynamic model is developed, an Explicit Finite Difference Method is applied, to handle the unsteady heat transfer problem with phase change, as well as time varying boundary condition. The influence of different factors such as air velocity, type of food, and food packaging are investigated, the question such as what kind of food is more sensitive to the surrounding temperature change is answered. This model can serve as a prerequisite for modeling of food quality changes, thus enabling the possibility of improving the control of supermarket refrigeration systems [Cai et al., 2006a] [Cai et al., 2006b].

Quality Model of Foodstuffs in a Refrigerated Display Cabinet. This paper discusses quality modeling of foodstuffs, different scenarios of defrost schemes are simulated, questions such as how the defrost temperature and duration influence the food temperature, thus the food quality, as well as what is the optimal defrost scheme from a food quality point of view are answered. This will serve as a prerequisite for designing an optimal control scheme for commercial refrigeration systems, aiming at optimizing a weighed cost function of both food quality and overall energy consumption of systems [Cai et al., 2006c] [Cai et al., 2006d].

An Active Defrost Scheme for Balancing Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems. This paper introduces food quality as a new parameter, together with energy, to determine an optimal cooling time between defrost cycles. A new defrost-on-demand control scheme is proposed. It uses a feedback loop consisting of on-line model updating and estimation as well as dynamic optimization. This scheme automatically adjusts the time interval between defrost cycles with varying operating conditions, continuously seeking an optimal time interval, featuring either an energy optimal time, or a trade off between energy consumption and food quality loss. This adaptive approach is compared with traditional defrost control schemes, found to be able to reduce energy consumption significantly [Cai et al., 2008d] [Cai et al., 2008c].

Minimizing Quality Deteriorations of Refrigerated Foodstuffs as a Side Effect of Defrosting. This paper proposes an optimization scheme for traditional refrigeration systems with hysteresis controllers and scheduled defrosts. It aims at minimizing the side effect of defrost cycles on the storage quality of refrigerated foodstuffs in supermarkets. By utilizing the thermal mass of air and products inside a display cabinet, this optimization scheme forces the compressor to work harder and cool down more prior to the scheduled defrosts, thus guaranteeing the product temperature after defrost cycles still to be within a controlled safe level [Cai and Stoustrup, 2008].

On the Trade-off between Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems. This paper studies the trade-off between energy consumption and food quality loss, at varying ambient conditions, in supermarket refrigeration systems. Compared with the traditional operation with pressure control, a large potential for energy savings without extra loss of food quality is demonstrated. We also show that by utilizing the relatively slow dynamics of the food temperature, compared with the air temperature, we are able to further lower both the energy consumption and the peak value of power requirement. The Pareto optimal curve is found by off-line optimization [Cai et al., 2008a].

Preventing Refrigerated Foodstuffs in Supermarkets from Being Discarded on Hot Days by MPC. This paper presents an optimization strategy for supermarket refrigeration systems. It deals with one special condition when the extremely high outdoor temperature causes the compressor to saturate, and work at its maximum capacity. In a traditional control, refrigerated foodstuffs inside display cabinets will suffer from a consequential higher temperature storage, which is detrimental to the food quality, and in worst cases they have to be discarded according to the regulation from authorities. This will cause a big economic loss to the shop owner. By utilizing the thermal mass of foodstuffs and their relative slow temperature change, Model Predictive Control (MPC), foreseeing this situation, it will use more compressor power to cool down the foodstuffs in advance, preventing the high temperature storage from happening, thus saving them from being discarded [Cai et al., 2008b].

Chapter 2

Summary of work

This chapter summarizes the work in this Ph.D project. It aims at giving fast facts regarding the problems which need to be solved, how to solve, then, the conclusions, and their application prospective. Here the application prospective is added, with the purpose of giving refrigerating engineers some insights and general application guidelines. Detailed problem descriptions and solutions can be found from individual papers.

This chapter is organized as follows: Section 2.1, dynamic heat transfer model of refrigerated foodstuffs. Section 2.2, quality model of foodstuffs in a refrigerated display cabinet. Section 2.3, an active defrost scheme for balancing energy consumption and food quality loss in supermarket refrigeration systems. Section 2.4, minimizing quality deteriorations of refrigerated foodstuffs as a side effect of defrosting. Section 2.5, on the trade-off between energy consumption and food quality loss in supermarket refrigeration systems. Section 2.6, preventing refrigerated foodstuffs in supermarkets from being discarded on hot days by MPC.

2.1. Dynamic Heat Transfer Model of Refrigerated Foodstuffs

2.1.1. Problem

Food quality is mainly influenced by the food temperature. In a traditional refrigeration system, there is no direct food temperature measurement. In order to convert the measured air temperature inside a display cabinet to the food temperature, we developed food thermal models.

In our study, there are two types of models, one type is a finite difference model, which is used to investigate the temperature distribution for different layers inside the food during defrosting; another is a lumped-capacitance model, which is used when the temperature gradients inside the food are so small that they can be ignored and treated as isothermal. Fish products are selected as case studies due to their short shelf life and high temperature sensitivities.

Heat transfer model can be used as a tool to evaluate the effect of different parameters (such as air velocity inside the cabinet, packaging etc.) on the product temperature change. It is a prerequisite for modeling of food quality changes, which will be introduced into the objective function, for optimizing the control of supermarket refrigeration systems.

2.1.2. Solution

Unsteady state conduction refers to a class of problems in which the temperature of the conduction region varies with time, such as the case in which the boundary condition varies with time such that neither a steady state nor a periodic behavior is ultimately attained.

Governing equation

$$\nabla\left(k(T)\cdot\nabla T\right)=\rho\cdot C_p(T)\cdot\frac{\partial T}{\partial t}$$

Where T is temperature, t is time, $k(T)$, ρ, $C_p(T)$ are so called thermophysical properties of foodstuffs: thermal conductivity, density and specific heat capacity.

Convective boundary condition

$$k(T)\cdot\frac{\partial T}{\partial n}=h\cdot(T-T_\infty)$$

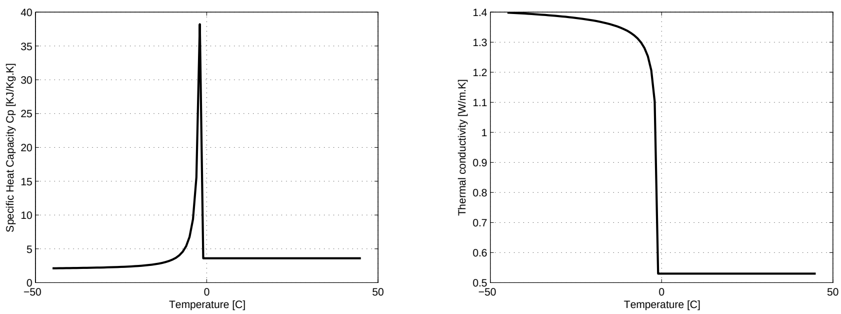
Where n is the outward-facing normal vector on the body surface, h is the convective heat transfer coefficients, and T_∞ is the ambient temperature.

Initial condition

$$T=T_0$$

Where T_0 is a known temperature.

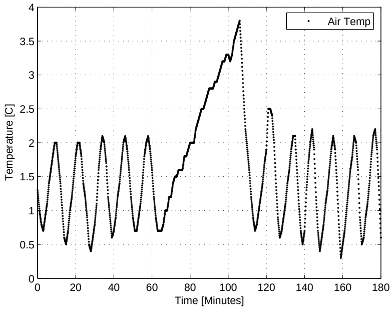
Difficulties in modeling are the temperature-dependent thermophysical properties of foodstuffs, as shown in Figure 2.1, and time varying boundary conditions, as shown in Figure 2.2.



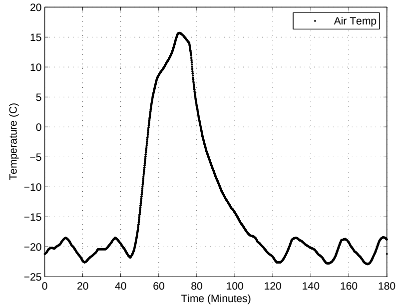
(a) Specific heat capacity C_p (b) Thermal conductivity k

Figure 2.1: Dependence of thermophysical properties with temperature (lean cod products)

To solve this problem, an Explicit Finite Difference Method is applied. Interpolation of air temperature at the boundary is done by a linear interpolation between the sampled data. Temperature-dependent thermophysical properties are evaluated by lagging the evaluation by one time step.

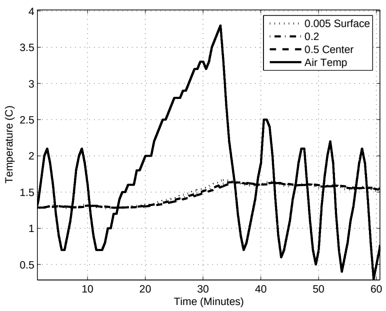


(a) Fresh fish products storage

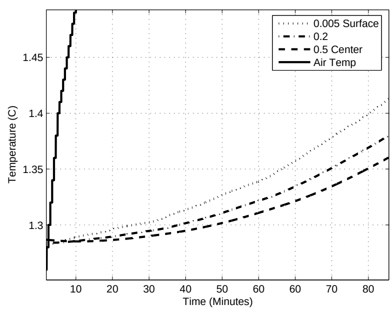


(b) Frozen fish products storage

Figure 2.2: Air temperature profile with one defrost cycle (boundary condition).



(a) Full profile



(b) Partially zoomed

Figure 2.3: Time temperature profile for fresh lean fish products (cod) at different depth.

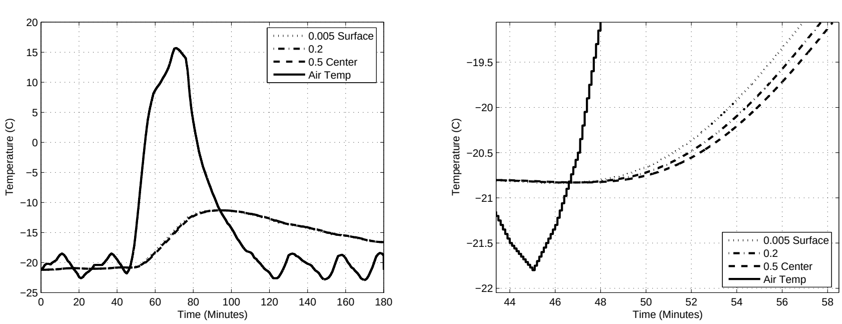
2.1.3. Conclusions

Conclusion 1: For fresh lean fish products such as cod, in the defrost cycle, the surface temperature increases about 0.4°C, and the maximum temperature difference between the surface and center is less than 0.1°C, as shown in Figure 2.3; for the frozen lean fish products, in the defrost cycle, the surface temperature increases 9.9°C, and the maximum temperature difference between the surface and center is around 0.6°C, as shown in Figure 2.4.

Conclusion 2: Air velocity inside the display cabinet and packaging of food influence the heat transfer coefficient h and U respectively, thus influencing the heat transfer from the air to the food. Generally speaking, a proper packaging can to some extend protect the foodstuffs from suffering environmental temperature variations.

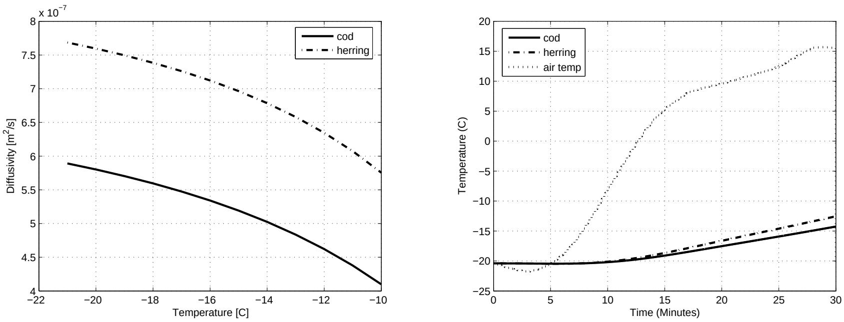
Conclusion 3: Water content of foodstuffs plays a vital role in the freezing process and storage period, as most of the thermophysical properties are calculated based on the water content of the product, and properties of ice and water.

Conclusion 4: Different types of fish products have different sensitivities to the



(a) Full profile (b) Partially zoomed

Figure 2.4: Time temperature profile for frozen fish lean products (cod) at different depth.



(a) Diffusivity (b) Air and product center temperature

Figure 2.5: Time temperature profile for different types of fish products.

temperature changes during defrosting, due to different specific heat capacity C_p, conductivity k and density ρ etc, the overall effect is reflected from the thermal diffusivity α, which is defined as $\alpha = k/(C_p \cdot \rho)$. Diffusivity and the time temperature profile for frozen fatty fish products herring and lean fish products cod are shown in Figure 2.5. We can see that the product with higher diffusivity is more sensitive to the temperature change.

2.1.4. Application perspective

In unsteady-state (or transient) heat transfer calculations, when should we use a finite difference (or finite element) model, and when should we use a lumped-capacitance model?

Estimation of dimensionless Biot number B_i can give us a basic idea. It is defined as follows:

$$B_i = \frac{h \cdot L}{k}$$

Where L is characteristic length, which is commonly defined as the volume of the body divided by the surface area of the body, h and k are the heat transfer coefficient and thermal conductivity of products respectively.

Values of the Biot number smaller than 0.1 imply that the heat conduction inside the body is much faster than the heat conduction away from its surface, and temperature gradients are negligible inside. This can indicate the applicability (or inapplicability) of certain methods of solving transient heat transfer problems. For example, a Biot number less than 0.1 typically indicates less than 5% error will present when assuming a lumped-capacitance model of transient heat transfer.

2.2. Quality Model of Foodstuffs in a Refrigerated Display Cabinet

2.2.1. Problem

To realize an overall optimization objective with a balanced system energy consumption and food quality loss, we need to model the quality loss during different processes. It will be used as a new decision factor when we propose a new defrost-on-demand scheme, and also be used when we optimize the control of the existing refrigeration system. All the product quality modeling is based on product temperature.

2.2.2. Solution

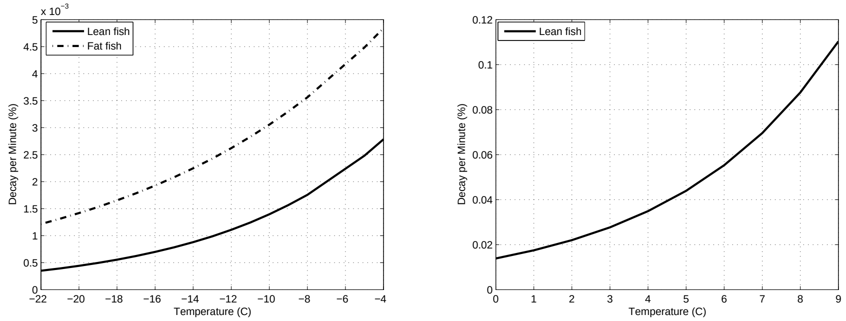
Food quality is modeled based on shelf life calculations. The definition of shelf life depends on the limiting factors for the product. It is often microbial spoilage or decolorization as in chilled storage, or it could be loss of vitamins or color as in frozen storage.

The investigation regarding whether the product has changed with respect to the fresh product is often carried out by triangular sensory tests. For frozen food, product starts with a quality of 100% and end with 0%. Here a product of 0% quality, by this definition, is a very good product. The 0% quality is determined by the fact that a set of assessors looking for difference just find it with significance. The normal consumer would not detect the changes. For chilled food, in contrast to frozen food, the quality from the start is defined as 100% and 0% when the product is unsuitable for sale. In food processing the actual reaction rate is interesting, but only as a means of obtaining the most interesting information: the integral effect, i.e. the accumulated effect after some processing steps or storage periods with varying conditions.

Accumulated quality loss is modeled as follows:

$$Q_{food,loss} = \int \frac{100}{D_{ref}} \cdot \exp\left(\frac{T_{food} - T_{ref}}{z}\right) dt$$

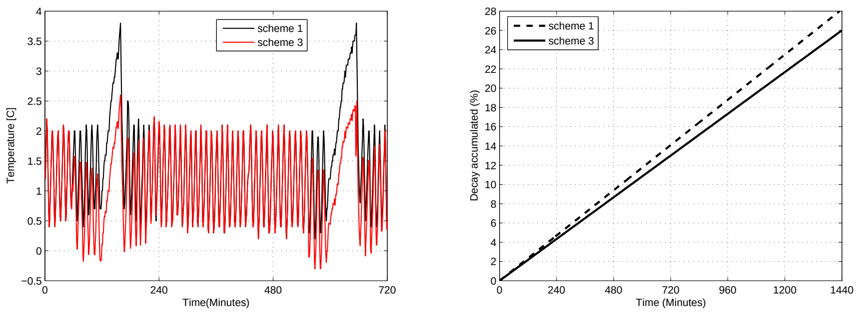
Where z value is the product temperature sensitivity indicator, D_{ref} is the time for loss of quality at the reference temperature T_{ref}, t is the storage time here.



(a) Frozen fat and lean fish products

(b) Fresh lean fish products

Figure 2.6: Minute quality loss in % as a function of temperature.



(a) Air temperature for defrost scheme 1 and 3 (zoomed)

(b) Accumulated daily quality decay for scheme 1 and 3

Figure 2.7: Different defrost schemes and their accumulated daily quality decay.

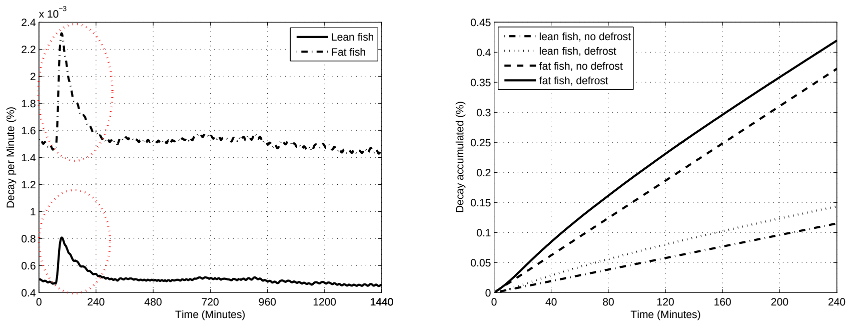
2.2.3. Conclusions

Conclusion 1: The relation between quality decay rate and temperature is non-linear, high temperature has relatively higher deterioration effect, as shown in Figure 2.6. Figure 2.6a shows the quality decay per minute in % for frozen fat and lean fish products, Figure 2.6b shows the quality decay for fresh lean fish products (chilled storage).

Conclusion 2: The frequency of defrosting and time duration of defrost cycles influence the quality decay of foodstuffs inside the display cabinets. Generally speaking, less frequent defrosting with a relatively longer duration gives a better storage quality.

Conclusion 3: Lower the air temperature before the scheduled defrost to cool down the food in advance, thus reducing the peak value of air and food temperature variation, has the potential of minimizing the deterioration effect of defrosting on the quality of refrigerated foodstuffs , as shown in Figure 2.7.

Conclusion 4: Fatty fish products have in general higher decay rates, but it does not mean that they are more sensitive to the temperature change during defrosting, as shown



(a) Fat vs. Lean

(b) Defrost vs. Non-defrost

Figure 2.8: Frozen fish products decay and sensitivities to defrosting.

in Figure 2.8. For fat fish products, if there is no defrosting during this 4 h, decay is 0.37%, with defrost is 0.42%, defrosting increase the decay about 13.5%; for lean fish products, if there is no defrosting during this 4 h, decay is 0.12%, with defrost is 0.14%, defrosting increase the decay about 14.3%.

2.2.4. Application perspective

Food quality is characterized by many factors, such as color, texture, appearance, taste, and smell etc. For chilled and frozen food, changes of these factors are determined by many different reaction mechanisms. Modeling of one single aspect such as the microbial growth is important, but modeling the quality decay in % based on sensory tests as done above usually best approximates the overall quality state of the food.

This model can be applied in different processes, to evaluate the effect of different control strategies on the storage quality of refrigerated foodstuffs. On the one hand, we could use the food quality as a new decision factor, together with system energy, to introduce a new defrost-on-demand control scheme; on the other hand, we could use the small range linearity of the non-linear food quality decay with temperature (temperature variation tolerance), to ensure our optimization on the existing system brings no extra damage to the food, or utilize this characteristic to further decrease the energy consumption.

The non-linear relation between food quality decay and temperature (or temperature variation tolerance) is quite different from one type of products to another, so attentions should be given when dealing with meat, milk, or fish products, etc., and special attentions should be given to the food without packaging, and with high water content.

2.3. An Active Defrost Scheme for Balancing Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems

2.3.1. Problem

In a simple taxonomy, there are two classes of defrost schemes: scheduled defrost, defrost-on-demand. Scheduled defrost is simple and uses a low cost controller, so it is the most commonly used defrost scheme in today's supermarkets, but the time schedule is normally determined based on experience and observation, most cases, based on worst case conditions. It is configured during the commission stage and can not automatically adapt to the varying shop conditions under which the system is working, so the time between two defrost cycles can be either too long or too short. Existing defrost-on-demand schemes typically involve the installation of additional sensors to detect frost build-up, and use one parameter to initiate and terminate the defrost cycle. The threshold of this detected parameter is determined mainly to ensure a safe operation, or maintain the degradation in the performance of the system within fixed limits over the whole range of operating conditions. No energy optimality is guaranteed. None of the existing schemes have used food quality as a decision factor.

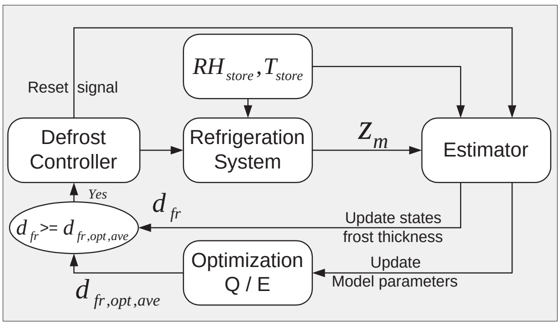


Figure 2.9: On-line new defrost-on-demand control scheme

2.3.2. Solution

To overcome the shortcomings of the current defrost schemes, and realize an objective with a balanced system energy consumption and food quality loss, we propose a new defrost-on-demand control scheme, see Figure 2.9. It uses a feedback loop consisting of an on-line model updating and estimation by an estimator, as well as a model based optimization. z_m is the measured output, the difference between the measurement and prediction is used for on-line updating of model states and parameters. Here we assume that the store temperature T_{store} and relative humidity RH_{store} are measured, such as at every half or at every full hour, depending on the stability of store indoor conditions. $d_{fr,opt,ave}$ is the average optimal frost thickness for defrosting generated by the optimization, but defrost will only be initiated when the estimated frost thickness d_{fr}

from the estimator is equal to or larger than $d_{fr,opt,ave}$. This initiating signal is sent to the normal defrost controller, defrosting starts. Defrosting is terminated in the normal way. After the defrost is complete, a reset signal is sent to the estimator, and the process for the next defrost cycle starts again.

The optimization objective is described as the following:

$$\min_{t_{opt}} E(t) + k \cdot Q_{food,loss}(t)$$

Where $E(t)$ is the system energy consumption, which includes two parts, one is the energy used direct for defrosting, another is the extra energy used for compensating the degraded system efficiency due to frost build-up. $Q_{food,loss}(t)$ is the foodstuff's quality loss, k is a weighing factor based on costs or shop owners' priorities, t_{opt} is the optimal cooling time between defrosting, see Figure 2.10.

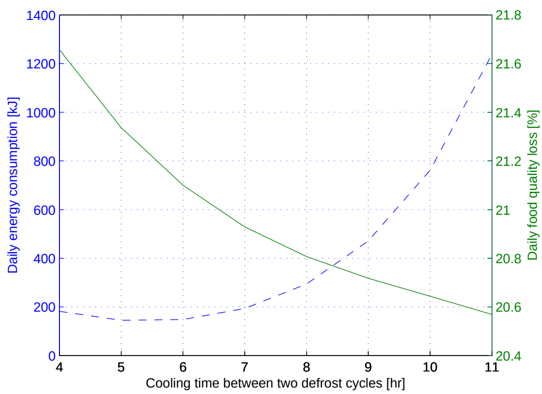


Figure 2.10: Daily energy consumption and food quality loss as a function of cooling time between defrosting.

From the modeling, we know that the optimal frost thickness for defrosting $d_{fr,opt}$ is a function of store temperature T_{store}, relative humidity RH_{store}, and parameters for cabin, fan, evaporator, compressor, as well as t_{opt}, as described as follows:

$$d_{fr,opt} = f(RH_{store}, T_{store}, P_{cab}, P_{fan}, P_{evap}, P_{comp}, t_{opt})$$

To determine an optimal frost thickness threshold under dynamic situations is not easy; a simple method is proposed as follows, and shown in Figure 2.11. It is approximated by the average value of optimal frost thickness under the whole working range. Simulation results under the following conditions: [20°C, 25°C] [50%, 60%] showed that by using the average frost thickness threshold to initiate defrosting, the maximal energy loss compared with using the true optimal value is less than 4%. The alternative using a detailed model for determining the value given by the objective function has from a control point of view a serious lack of robustness, and realistic modeling errors could easily cause larger deviations than 4%. Moreover these 4% in worst-case is insignificant relative to the potential savings demonstrated by Figure 2.12 below.

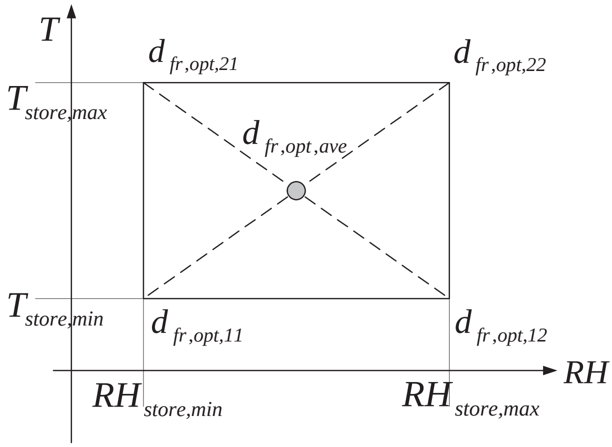


Figure 2.11: The optimal frost thickness threshold for defrosting is approximated by the average frost thickness threshold.

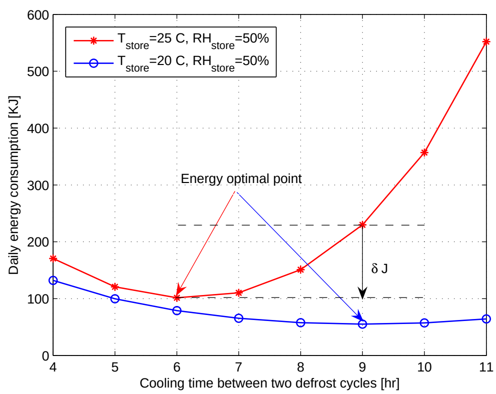


Figure 2.12: Potential gains on energy by the new defrost-on-demand control scheme.

2.3.3. Conclusions

Conclusion 1: Adaptivity: The approach suggested uses real time disturbance measurements (store temperature and humidity) for on-line model updating, estimation and optimization, continually seeking an optimal time for defrosting under dynamic conditions. This adaptive approach is compared with traditional defrost schemes, found to be able to reduce energy consumption significantly, as shown in Figure 2.12.

Conclusion 2: Optimality: The approach suggested is based on an optimality condition which is a weighted function between system energy consumption and food quality loss, so the resulting closed-loop system will always operate on the Pareto-optimal trade-off curve between system energy consumption and food quality loss.

Conclusion 3: Feasibility: The proposed method is a model based control method, and introduction of an estimator can avoid some special sensors, such as fiber optic sensors, differential pressure sensors, or air velocity sensors, etc. Those sensors are normally expensive, and often some reliability and feasibility problems are associated with the complex and unreliable sensing methods. The estimator can infer those values of interest from some available measurements, such as by one or more existing thermal sensors in the system. Thus the new controller can be implemented directly on the top of existing systems, no physical rearrangements or extra components installations are required.

Conclusion 4: To determine an optimal frost thickness threshold under dynamic situations is not easy; a simple method is proposed in the paper. It is approximated by the average value of optimal frost thickness under the whole working range. Simulation results under the following conditions: [20°C, 25°C] [50%, 60%] showed that by using the average frost thickness threshold to initiate defrost, the maximal energy loss compared with using the true optimal value is less than 4%. The alternative using a detailed model for determining the value given by objective function has from a control point of view a serious lack of robustness, and realistic modeling errors could easily cause larger deviations than 4%. Moreover these 4% in worst-case is insignificant relative to the potential savings demonstrated.

Conclusion 5: Modeling of frost formation under dynamic situations is very difficult, while using the energy correlations to estimate the frost formation rate is quite simple, but still captures the main dynamical features seen from an input/output point of view.

Conclusion 6: When the foodstuffs are properly packaged, infiltration is the only source for frost formation. It is heavily influenced by the shop temperature and relative humidity. Generally speaking, high shop temperature and relative humidity give more infiltration load to the system, a higher frost formation rate, and a shorter energy optimal cooling time between the defrost. Integrating the air conditioning system and the refrigeration system into one unit, and investigating the overall energy efficiency in general gives a satisfactory result.

2.3.4. Application perspective

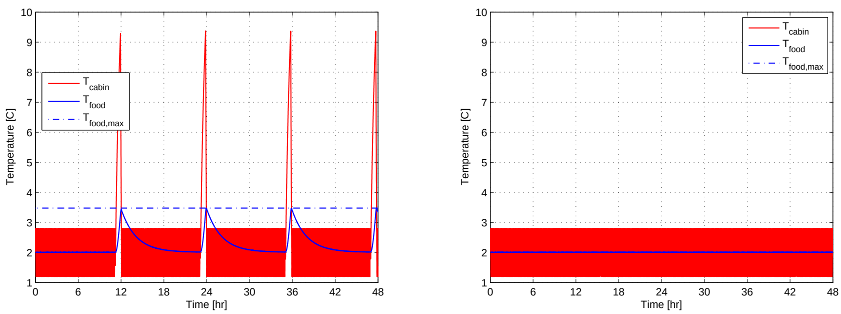
Generally speaking, this new defrost-on-demand scheme is especially suitable for a supermarket, which has dramatic indoor climate change. Implicitly, it is more suitable for the supermarket which has no air conditioning systems.

The energy optimal cooling time between defrost cycles and the potential gains by using this new scheme can be influenced by many different factors, such as the evaporator fan, the geometry of the heat exchanger including the fin spacing and the fin thickness, etc. For the evaporator fan, if it can run with the constant speed (with frequency convertor/constant speed control), the heat transfer performance will not degrade so dramatically as shown with the normal fan, so the energy optimal cooling time will be generally longer. Different fan curves (same type of fans or different types of fans, such as centrifugal or axial fans) will also cause the difference on optimal cooling time and potential gains. For the fin, generally speaking, larger fin spacing and thinner fin could prolong the time for ice blockage, and a longer optimal cooling time, but larger fin spacing will decrease the number of fins per unit of flow area, which decreases the total heat rate, so the fin spacing and fin thickness should be considered together.

2.4. Minimizing Quality Deteriorations of Refrigerated Foodstuffs as a Side Effect of Defrosting

2.4.1. Problem

Commercial refrigeration systems, during normal operations, air temperature inside display cabinets is normally controlled within a specific upper and lower bound by a hysteresis controller. This is sufficient to maintain the product temperature within a recommended level. During defrosting, air temperature inside cabinets will rise, and so will the food temperature. Sometimes the temperature is so high that it even violates the temperature regulation from food authorities. This higher than normal temperature storage will cause an extra quality loss to the food products, see Figure 2.13 and 2.14. Here we use a cabinet for storing and selling fresh lean fish products as an example, the recommended maximum storage temperature is +2°C.



(a) Normal defrost (b) No defrost

Figure 2.13: Air and food temperature profile.

2.4.2. Solution

A model of a refrigeration system with a hysteresis controller and scheduled defrosts is implemented in *gPROMS*®. Optimization is done by dynamic optimization, with the following objective function, where W_C is the power consumption of the compressor.

$$\min_{(T_{bound,lower},T_{bound,upper})} J$$

$$\text{where} \qquad J = \int_{t_0}^{t_f} W_C(t)dt$$

$$\text{subject to} \qquad T_{food} \leq T_{food,max}$$

Here we limit the maximum temperature of the products after defrost cycles to be 2°C, and block the first 6 h after each defrost cycle to be normal setting, and thereafter a new

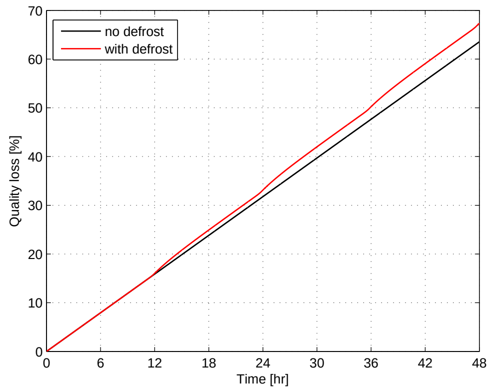


Figure 2.14: Comparison of food quality loss in % during 48 h, with and without defrost.

time invariant input setting for optimization, result is shown in Figure 2.15. A compari-

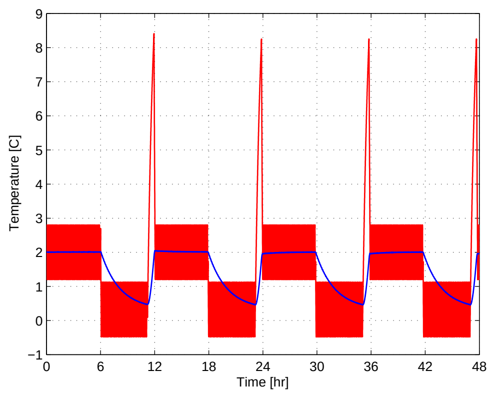


Figure 2.15: Optimization: food temperature is never allowed to exceed its maximum allowable value.

son of food temperature under the normal defrost cycle and the optimization solution is shown in Figure 2.16, and a comparison of quality loss and energy consumption in two days is shown in Figure 2.17.

2.4.3. Conclusions

Conclusion 1: During defrosting, air temperature inside cabinets will rise, and so will the food temperature. Sometimes the temperature is so high that it even violates the regulation from food authorities. This higher than normal temperature storage will cause an extra quality loss to food products. To solve the problem, one way is to modify the system, realizing frost free, thus no defrosting is needed. Another way is to modify the control of the system, minimizing the risk of defrosting to the food quality.

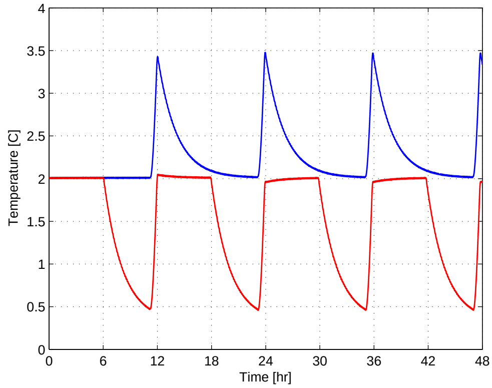
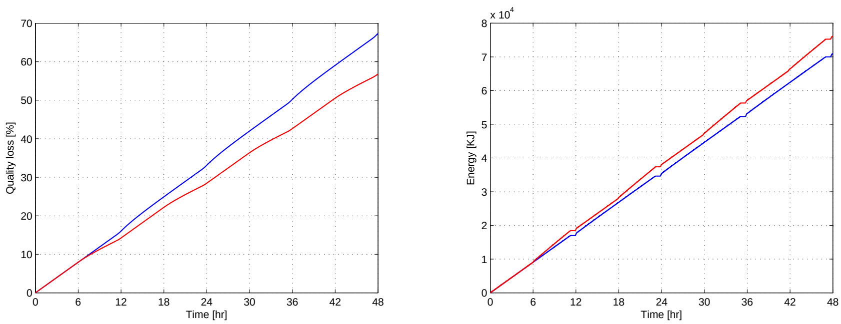


Figure 2.16: Comparison of food temperature under normal defrost and optimization.



(a) Quality loss

(b) Energy consumption

Figure 2.17: Comparison of food quality loss and system energy consumption during 48 h under normal defrost and optimization.

Conclusion 2: By utilizing the thermal mass of the air and products inside the cabinet, this optimization scheme forces the compressor to work harder and cool down more prior to the scheduled defrosts, thus guaranteeing the product temperature after the defrost cycles to be within the controlled safe level.

Conclusion 3: To cool down the products in advance, the compressor has to work harder and consumes more energy. A judgement on whether it is worth doing can be based either on the shop owner's priorities (cost/quality) or on a simple economic evaluation (such as in Euro).

2.4.4. Application perspective

This strategy uses the refrigerated foodstuffs inside the cabinet as a thermal storage to minimize the side effect of defrosting to the product quality. The time constant from air temperature to the food temperature $\tau = mCp_{food}/UA$ play a vital role here. If τ

is extremely large, during a defrost cycle, there will be very little temperature change for the product as a whole (except for the surface layer, which may has a relatively larger temperature change), in this case, there is properly nothing worth doing; if τ is extremely small, the product temperature will follow the air temperature so tight that it is impossible to store thermal energy. There exists an certain range for the time constant, such as the case illustrated in our study, where the strategy is beneficial.

2.5. On the Trade-off between Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems

2.5.1. Problem

Many efforts on optimization of cooling systems have been focused on optimizing objective functions such as overall energy consumption, system efficiency, capacity, or wear of the individual components. Significant improvements of system performance under disturbances have been proved, while there has been little emphasis on the quality aspect of foodstuffs inside display cabinets.

A well-designed optimal control scheme, continuously maintaining a commercial refrigeration system at its optimum operation condition, despite changing environmental conditions, will achieve an important performance improvement, both on energy efficiency and food quality reliability.

With only one on/off actuator as in a domestic refrigerator, there is little scope for optimization of the performance. While with the recent advance in automation technology, most of industrial refrigeration plants have a number of actuators, such as electric motors, variable speed fans, which has made the refrigeration system more drivable, and also means it is possible to provide the refrigeration power for a desired temperature by various combinations of compressor speed, evaporator fan and condenser fan speed, temperatures setpoints, number of compressors, or number of condensers etc.

Potentials on optimization: Influence of three setpoints (condensing pressure P_C, evaporating pressure P_E, cabinet temperature T_{cabin}) on total power consumption is shown in Figure 2.18, on food quality is shown in Figure 2.19 (indirectly by T_{cabin}). Figure 2.19 also shows that within a small range (for example $1^{\circ}\mathrm{C} \pm 1$), the relation between quality and temperature can be treated as linear without causing large quality deviation (temperature variation tolerance, recall also Figure 2.6 for the relation between quality decay and temperature). This is the special feature we will use in the optimization.

2.5.2. Solution

We here consider at a time horizon of three days, ambient temperature (T_{amb}) follows a sinusoidal function with a mean value of 20°C, period of 24 h and amplitude of 6°C. The objective is to minimize the energy consumption, subject to maintaining a fixed quality loss, by using those three DOF (compressor speed N_C, condenser fan speed N_{CF},

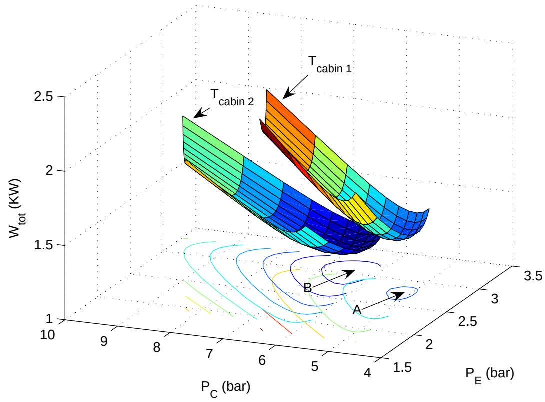


Figure 2.18: Power consumption under different setpoints: surface shows that under two different T_{cabin}, the variation of total power consumption with varying P_C and P_E. Point A is the optimum for T_{cabin1} and point B is the optimum for T_{cabin2}. T_{cabin1} is lower than T_{cabin2}.

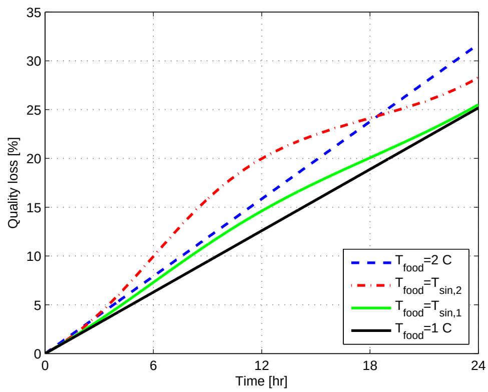


Figure 2.19: Fresh fish products quality loss when stored at different temperatures: 2°C, 1°C and T_{sin}. $T_{sin,1}$ and $T_{sin,2}$ are the sinusoidal function with mean value of 1°C, amplitude of 1°C and 3°C respectively, period is 24 h.

evaporator fan speed N_{EF}). This can be formulated mathematically as

$$\min_{(N_C(t),N_{CF}(t),N_{EF}(t))} J$$

$$\text{where} \qquad J = \int_{t_0}^{t_f} \underbrace{(W_C(t) + W_{CF}(t) + W_{EF}(t))}_{W_{tot}(t)} dt$$

Where W_C, W_{CF} and W_{EF} are power consumption of the compressor, the condenser fan, and the evaporator fan respectively.

The quality loss of the food could be included in the objective function directly, but we choose to limit it by using constraints. The optimization is also subjected to other constraints, such as maximum speed of fans and compressor, minimum and maximum value of evaporator and condenser pressure etc.

Here the food is a fresh cod product. Danish food authorities require it to be kept at a maximum of 2°C. The control engineer may set the temperature setpoint a little lower, for example at 1°C.

> **Case 1** One traditional operation with constant pressures (P_E), (P_C) and constant temperatures ($T_{cabin} = T_{food} = 1\,°\mathrm{C}$)

To get a fair comparison with traditional control, which operates at 1°C, optimization is carried out by considering the following cases:

> **Case 2** T_{cabin} and T_{food} constant at 1°C.
> Two remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption.

> **Case 3** $\overline{T_{food}} = \frac{1}{t_f - t_0} \int_{t_0}^{t_f} T_{food}(t)dt = 1°\mathrm{C}$.
> Three remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption.

> **Case 4** $Q_{food,loss}(t_f) \leq 75.5\,\%$.
> Three remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption. 75.5% is the quality loss at constant temperature of 1°C obtained in Case 1 and 2.

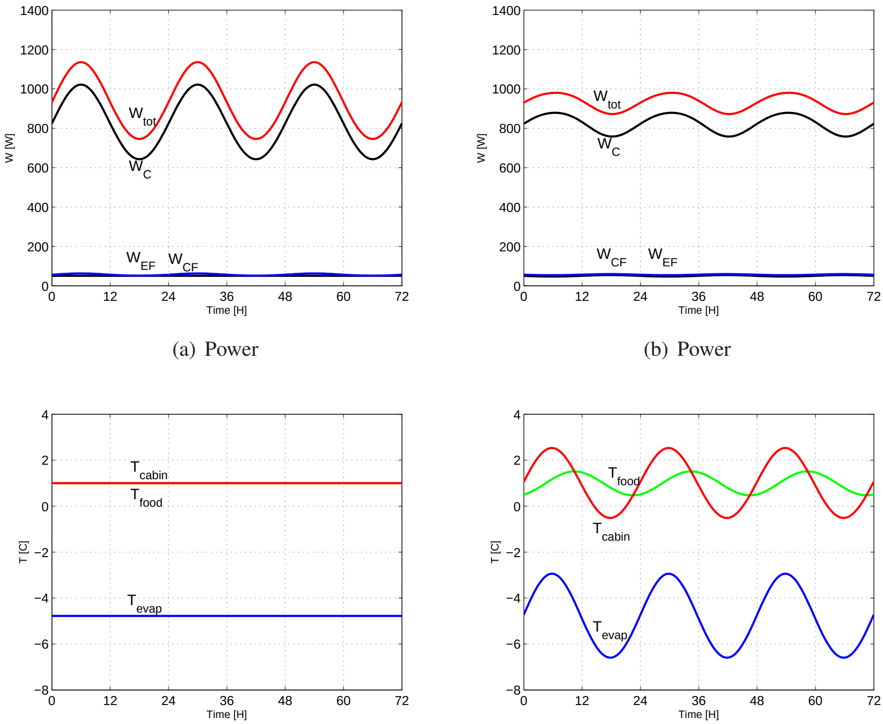
The model is implemented in *gPROMS*® and the optimization is done by dynamic optimization (except for Case 1).

2.5.3. Conclusions

Conclusion 1: Traditional operation where the pressures and air temperature are constant gives an excessive energy consumption. Allowing for varying pressure in the evaporator and condenser reduces the total energy consumption. Varying air temperature (thus food temperature) give extra improvements in terms of energy consumption, at the same time the peak value of the total power consumption is reduced, see Figure 2.20 (figures and detailed data comparison on all the cases can be found from the paper).

Conclusion 2: A lower peak value of total power consumption will further reduce the bill for the supermarket owner, according to the formula for calculating the operating costs.

Conclusion 3: Figure 2.21 plots the Pareto optimal curve between food quality loss and system energy consumption. It shows that reducing quality loss and saving energy is a conflicting objective to the system. An acceptable trade-off between these two goals can be selected by picking a point somewhere along the line. It also shows that Case 1 is far away from optimization; Case 4 is one optimal point, while Case 2 and 3 are near optimal solutions.



(a) Power (b) Power

(c) Low temperature (d) Low temperature

Figure 2.20: Power consumption of system when T_{cabin} (T_{food}) is not allowed (left, Case 2) and allowed (right, Case 4) to vary, under these two cases, food quality loss is equal.

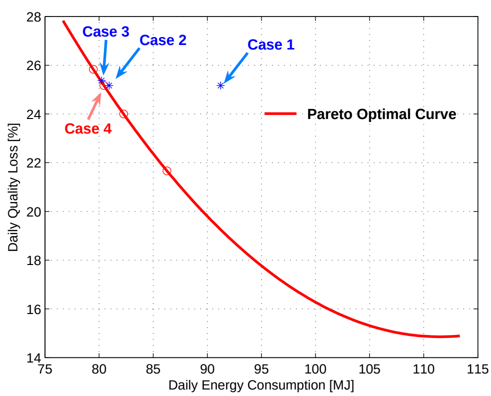


Figure 2.21: Optimization between food quality loss and system energy consumption.

2.5.4. Application perspective

This strategy uses the optimal power combination of compressor, condenser fan and evaporator fan to realize an objective with a weighted system energy and food quality loss, so the precondition is these actuators can be operated continually (or Pulse-width modulation can be applicable) within their physical limitations.

This strategy uses the thermal mass and small range linearity of the non-linear relation between food quality change and temperature to shift the load to a cheaper period (off-peak time with low utility cost) and high efficiency period (low ambient temperature high COP), to realize a trade-off between system energy and food quality. This is not feasible for all the conditions. One of the preconditions is there are sufficiently large thermal mass of foodstuffs inside the cabinet, or UA value from air to food is sufficiently small, which make the thermal energy storage possible; another precondition is the food should be able to tolerate a small temperature variation, for example, this strategy may be not suitable for the case when the food is not packaged, and with large water content.

2.6. Preventing Refrigerated Foodstuffs in Supermarkets from Being Discarded on Hot Days by MPC

2.6.1. Problem

The refrigeration system in a supermarket works all year round. Normally each compressor or a group of compressors has a fixed cooling capacity, which can cope with the most common application for that specific supermarket (here we use a system with single compressor as an example). If one day in summer, the outdoor temperature is extremely high, even when the compressor works at the maximum capacity, it still cannot meet the required cooling demand. In this case, air temperature inside the display cabinet T_{cabin} will rise, and so will the food temperature T_{food}. Since the controller actually controls one air temperature such as the air inlet temperature T_{S3}, so the compressor in this situation will work continually at the saturated condition until T_{S3} back to normal level, as illustrated in Figure 2.22. Depending on the seriousness of situation, sometimes, the stored foodstuffs have to be discarded according to the regulation from the food authorities.

2.6.2. Solution

To deal with the problem stated above, preventing foodstuffs from being discarded by the most energy efficient way, we design MPC as the following, shown in Figure 2.23. Here, due to an unique relation between the saturation temperature and pressure of refrigerant, we use the setpoint of evaporating temperature T_e, condensing temperature T_c and cabinet temperature T_{cabin} as the manipulated inputs, so the total DOF is still three. According to Jakobsen and Skovrup [Jakobsen and Skovrup, 2001], there always exists one optimal temperature difference between T_c and outdoor ambient temperature T_{amb}. In most cases, it is a constant. For simplification, we fix T_c by 10°C higher than T_{amb}, this consumes one DOF. T_{cabin} is one of the controlled outputs here. Therefore we have one DOF left, that is T_e. T_{amb} and T_{store} are measured disturbances, T_{cabin}, T_{food}, T_{S3} and W_C are controlled outputs.

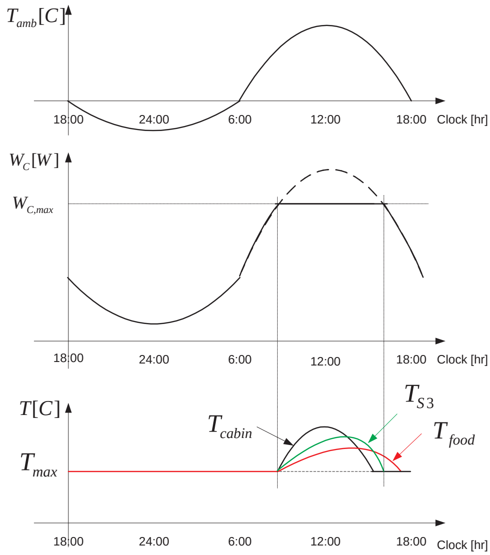


Figure 2.22: Illustration of the problem: saturation happens, T_{cabin} and T_{food} rise, compressor works at saturation until T_{S3} back to normal.

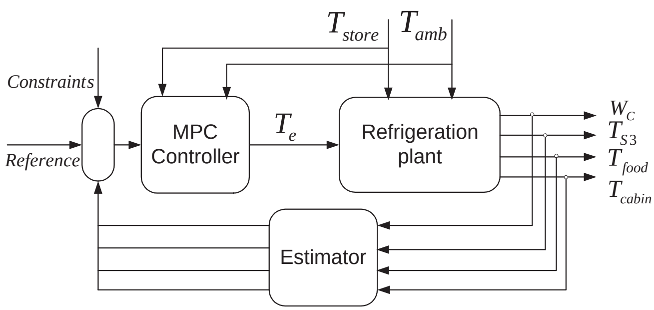


Figure 2.23: MPC controller with observer.

From model equations, we get the following ODEs:

$$\dot{T}_{food} = f_1(T_{food}, T_{cabin})$$
$$\dot{T}_{cabin} = f_2(T_{cabin}, T_{food}, T_{store}, Q_e)$$

Linearization around one steady state equilibrium point, we can get the linear continuous state space model in deviation form as follows:

$$\dot{x} = A_c \cdot x + B_c \cdot u + E_c \cdot d$$
$$y = C_y \cdot x + D_{yu} \cdot u + D_{yd} \cdot d$$

Where

$$
\begin{aligned}
x &= [T_{cabin} - T_{cabin,s}, T_{food} - T_{food,s}]' \\
u &= [T_e - T_{e,s}]' \\
d &= [T_{amb} - T_{amb,s}, T_{store} - T_{store,s}]' \\
y &= [T_{cabin} - T_{cabin,s}, T_{food} - T_{food,s}, T_{S3} - T_{S3,s}, W_C - W_{C,s}]'
\end{aligned}
$$

The above MPC controller is set up in $Matlab^{TM}$ by using MPC toolbox.
Constraints:

$$
\begin{aligned}
u_{jmin}(i) - \varepsilon V^{u}_{jmin}(i) &\le u_j(k+i \mid k) \le u_{jmax}(i) + \varepsilon V^{u}_{jmax}(i) \\
\Delta u_{jmin}(i) - \varepsilon V^{\Delta u}_{jmin}(i) &\le \Delta u_j(k+i \mid k) \le \Delta u_{jmax}(i) + \varepsilon V^{\Delta u}_{jmax}(i) \\
y_{jmin}(i) - \varepsilon V^{y}_{jmin}(i) &\le y_j(k+i+1 \mid k) \le y_{jmax}(i) + \varepsilon V^{y}_{jmax}(i) \\
&\Delta u(k+h \mid k) = 0 \\
&i = 0, \ldots, p-1 \\
&h = m, \ldots, p-1 \\
&\varepsilon \ge 0
\end{aligned}
$$

Where u_{min}, u_{max}, Δu_{min}, Δu_{max}, y_{min}, y_{max} are the lower and upper bound for u, Δu, y respectively, they are relaxed by introduction of the slack variable ε. Normally all the input constraints are hard, such that $V^{u}_{jmin}, V^{\Delta u}_{min}, V^{u}_{max}, V^{\Delta u}_{max} = 0$, while all output constraints are soft, as hard output constraints may cause infeasibility of the optimization problem. In our case, constraint on input T_e is determined from the condition that $P_{e,min} = 2.0\,$bar and $P_e < P_c$, T_c is between 30 and 40°C, so T_e is constrained within a lower and upper bound of -10 and 10°C. The change rate of T_e is selected to be within a lower and upper bound of 2°C.

Cost function: the cost function with soft constraints is formulated as the following form:

$$
\begin{aligned}
\min_{\Delta u(k|k),\ldots,\varepsilon} \{ \sum_{i=1}^{P-1} (\sum_{j=1}^{n_y} |w^{y}_{i+1,j}(y_j(k+i+1 \mid k) - r_j(k+i+1))|^2 \\
+ \sum_{j=1}^{n_u} |w^{\Delta u}_{i,j} \Delta u_j(k+i \mid k)|^2) + \rho_\varepsilon \varepsilon^2 \}
\end{aligned}
$$

Where w^y and $w^{\Delta u}$ are the weighting factor for the output deviations from the references and input changes respectively, the weight ρ_ε on the slack variable ε penalizes the violation of the constraints.

2.6.3. Conclusions

Conclusion 1: This strategy uses the thermal mass of the food and their relative slow temperature change, as well as the significant advantage of the MPC controller, to cool down the food beforehand, preventing the high temperature storage from happening, thus saving them from being discarded, see Figure 2.24, 2.25 and 2.26. From the figure we can see, under normal saturation, if nothing is done, foodstuffs will be stored at a

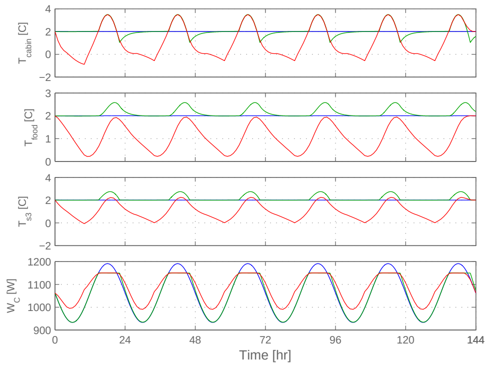


Figure 2.24: Comparison of temperature and power under sufficient capacity (blue), normal saturation (green) and MPC (red).

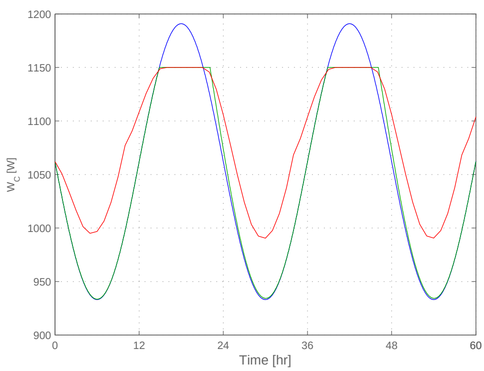


Figure 2.25: Comparison of power W_C under sufficient capacity (blue), normal saturation (green) and MPC (red)-zoomed.

temperature higher than its maximum allowable temperature. According to the relation between food temperature and quality, it is detrimental to the food quality. In the worst case, they have to be discarded, according to the regulation from the food authorities. This will cause a big economic loss to the supermarket owner. MPC, foreseeing this situation, it will force the compressor to use much more power beforehand, comparing with normal saturation, in order to satisfy the constraint on food temperature.

Conclusion 2: The simulation results illustrate the potential of linear MPC in dealing with nonlinearities, handling constraints and saturation, and realizing multiple control objectives.

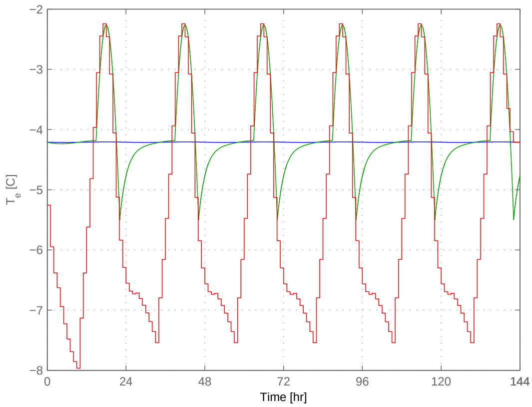


Figure 2.26: Comparison of input T_e under sufficient capacity (blue), normal saturation (green) and MPC (red).

2.6.4. Application perspective

This strategy uses the thermal mass of the food and their relative slow temperature change, as well as the significant advantage of the MPC controller, to cool down the food beforehand, preventing the high temperature storage from happening, thus saving them from being discarded. To implement this strategy, one of the preconditions is there are certain thermal mass of foodstuffs inside the cabinet, or the time constant from air temperature to food temperature is within a proper range, which make the thermal storage possible; another precondition is the capacity of the compressor should be controllable within the certain range.

This strategy uses a 24 h prediction horizon to prevent the future disaster caused by nature from happening. In order to realize this, we need to know the potential risk for extreme conditions at least several hours in advance. One possible solution is to use the weather forecast, which nowadays is easy to get even five days in advance period, another alternative is to include a weather disturbance model in the controller based on the numerical weather prediction (NWP) and local weather measurements. Here the underline assumption is the day to day peak temperature variation is small, which is true for many climates, especially for the countries with long coastline.

Chapter 3

Overall Conclusions and Recommendations

3.1. Overall conclusions

The objective of this Ph.D project is to investigate, further develop and specialize the control methods, which can be applied to a refrigeration system, to deal with the problem characterized by non-linear dynamics subjected to constraints and saturation, and seek a systematic way of optimizing the objective function with two criteria: system energy consumption and refrigerated food quality loss. The optimizing strategy should be realized by the application of model based control on the top of the existing traditional control.

- Modeling: In this project, several models have been developed including heat transfer model, food quality model, frost and defrost model, and energy model. Heat transfer model is used to convert the traditional measured air temperature to the food temperature. Food quality model is used to investigate the food quality loss during different operations. Frost and defrost model is used to investigate the phenomenon of the frost formation and removal, and the most important, its influence to the system performance and its corresponding energy consumption, as well as to the storage quality. Energy model is used to model the energy consumption of the system under different operations. But the purpose of modeling is for control, so all the modeling work follow the same principle: it should be relatively simple but still capture the main dynamics from an input/output point of view.

- Model based multi-objective optimization: Throughout the thesis, all the optimization work were based on the objective function with two criteria: system energy consumption and food quality loss. Optimization strategies were realized by the application of model based control on the top of existing traditional controls. They have the advantage of requiring no or little physical rearrangements to the existing systems.

- Control method: This project solved the saturation problem of refrigeration systems under extreme condition by Model Predictive Control (MPC). It showed successfully the potential of using the advanced control such as MPC dealing with the problem occurred in traditional refrigeration systems, characterized by non-linear dynamics and subjected to constraints and saturation.

- Controller design: In the new defrost-on-demand controller design, it used a feedback loop consisting of on-line model updating and estimation as well as a model based optimization. After the investigation of optimization threshold under the whole working range, it proposed to use the average threshold under dynamic situations as the optimality threshold, by this way, the controller was simplified and the robustness of the controller was improved.

3.2. Recommendations

Recommendations for future work and perspective are given as follows:

- In the frost formation model, we investigated the performance and energy consumption of refrigeration systems which are continually controllable, so the evaporating temperature or pressure can be gradually lower down to meet the cooling requirement due to frost formation. Further investigations on the systems with on-off control and refrigerant charging are needed to apply the new defrost-on-demand control scheme in much wider applications.

- In the paper regarding minimizing the food deterioration as a side effect of defrosting, the strategy is easy to be implemented when the next defrost cycle is known in advance, such as the case in the scheduled defrost. For the refrigeration system with defrost-on-demand scheme, the next defrost cycle is changing with dynamic working conditions. Under this situation, a logic way is to include our new defrost-on-demand controller, such that one control loop generates the next optimal defrost time, another loop optimizes the side effect of defrosting.

- In this project, we have used the fish products as a case study, while the relationship between the food quality change and temperature are quite different from one type of food to another, so modeling of different foodstuffs are required for the control of different display cabinets. When the food are mixed displayed, a choice of the worst case is needed for implementing different strategies.

- Experiments for validating the modeling and control strategies are essential.

- Generalization of ideas and methods: the idea of using the refrigeration foodstuffs as the thermal energy storage to shift the cooling load to a favorable operating condition can be easily generalized in some other areas, such as in the refrigerated transportation, where the container contains normally large quantity of refrigerated foodstuffs. The idea of using MPC to avoid the discarding of the foodstuffs under extreme conditions is also suitable for the long distance transportation. Due to the geographic change, the container may experience a large outdoor temperature variation, one way handling this situation is to equip the container with a compressor with an extra capacity or a group of compressors with variable capacity control, which is normally very expensive; another way is to have an intelligent control, such as by MPC.

Chapter 4

Dynamic Heat Transfer Model of Refrigerated Foodstuffs

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Abstract

Traditional control of commercial refrigeration systems focus on controlling the air temperature inside display cabinets. It is product temperature that directly influences the product quality, so if we want to store the foodstuffs with an optimal quality, we need to know their temperature relation.

This paper discusses the dynamic heat transfer model of foodstuffs inside a display cabinet, one-dimensional dynamic model is developed, an Explicit Finite Difference Method is applied, to handle the unsteady heat transfer problem with phase change, as well as time varying boundary conditions. The influence of different factors such as air velocity, type of food, and food package are investigated, the question such as what kind of food is more sensitive to the surrounding temperature change is answered.

This model can serve as a prerequisite for modeling of food quality changes, thus enabling the possibility of improving the control of supermarket refrigeration systems.

4.1. Introduction

Traditional control of commercial refrigeration systems is typically implemented as a hysteresis function, to maintain a desired air temperature around foodstuffs, not an optimal product temperature. As it is well known, besides the strict legislative control from food authorities regarding safety, customers' perception of food as "good" plays a vital role in the food retail industry. What characterizes "good" food? It could be based on appearance, smell, taste, texture, or nutritious facts, etc., which are influenced mainly by storage temperature.

A well designed optimal control strategy for governing the food temperature can address all these requirements, but it requires some models that are capable of describing the thermal characteristics of food inside the display cabinet. One possible solution is to integrate the food thermal model and quality model into the model of a refrigeration system, to find an optimal temperature profile, which can maintain an optimal food quality, at the same time, take the system energy consumption into consideration. The overall project prospective is depicted as Figure 4.1, where MPC is the abbreviation for Model Predictive Control.

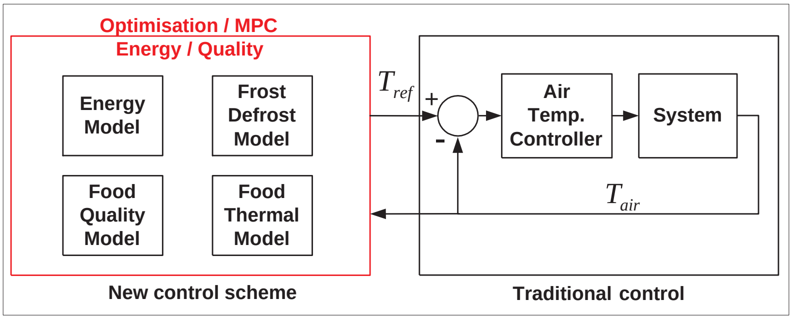


Figure 4.1: Overall project prospective.

As freezing and cold storage are very important operations in the food industry, there has been carried out a lot of research to predict the freezing time or thawing time [Morgan, 1981] [de. Reinick, 1992], main concern is the surface and center temperature profile of the food during the process, which is often a step response.

Our approach at this stage is to construct thermal models for some selected typical foodstuffs in the supermarket. Here fish is selected as the quality of fish products is very sensitive to the temperature change.

This paper is organized as follows: refrigeration of foodstuffs in supermarkets is described in Section 4.2, where a simple introduction to frost and defrost is included, due to its important role in food temperature and quality change. The mathematical model is introduced in Section 4.3, where the thermophysical properties of foodstuffs are discussed. In Section 4.4, we discuss the different methodologies for solving this kind of problems. In Section 4.5, simulation results are presented. Finally some discussions and conclusions are given in Section 4.6.

4.2. Refrigeration of foodstuffs in a supermarket

The display cabinet depicted in Figure 4.2 consists of a food container and an air tunnel, circulating cold air around the food container. An evaporator in the air tunnel cools the passing air which creates a carpet of cold air on the top of food.

The fact that the air carpet is colder than the food and ambient air, will keep the air carpet in place and become denser, enabling the desired effect of heat transfer from the carpet to the container and food. A side effect is that ambient air will infiltrate into the carpet at the load zone. The display cabinet's temperature is normally controlled by a hysteresis controller which opens and closes the inlet valve, to control the flow of

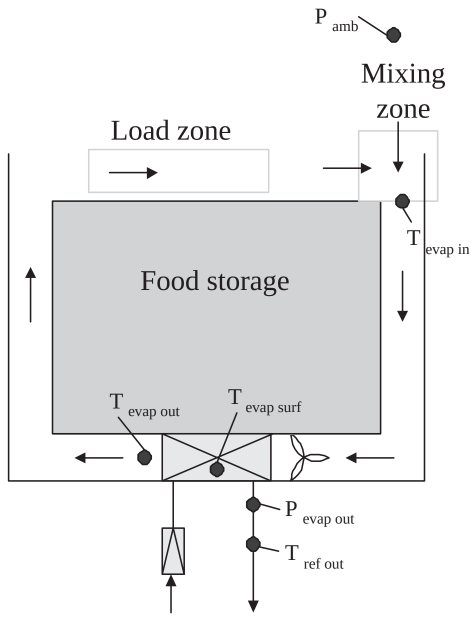
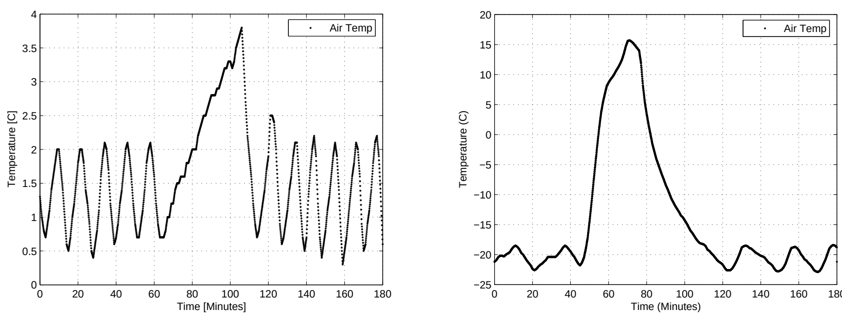


Figure 4.2: A simplified display cabinet in a supermarket.

refrigerant into evaporator, thus keeping the air temperature within the specific bands, see Figure 4.3, where the big change in temperature is caused by a defrost cycle.



(a) Fresh fish products storage

(b) Frozen fish products storage

Figure 4.3: Air temperature profile with one defrost cycle.

4.2.1. Frost formation and defrost cycle

Frost form on evaporator coils as water vapor in air condense and freeze when they contact the coils surface which is below 0°C. This is a well known and undesirable phenomenon. It deteriorates the system performance by decreasing the effective air flow area and increasing the thermal resistance between warm air and cold refrigerant. Currently there are no clear and reliable measures that can prevent frost formation [Shin et al., 2003]. When frost accumulate to a certain level, defrosting must be done to maintain a satisfactory system performance.

Defrost methods vary with refrigeration applications and storage temperature, and initiation and termination of defrost can be controlled by many different parameters, such as a timer or temperature, sometimes up to three times per day. During the defrost cycle, the air temperature inside the cabinet will increase and remain outside the normal temperature range for a period of time.

4.2.2. Legal requirements

In Danish supermarkets, there are legal requirements regarding the storage temperature for different foodstuffs in display cabinets [DSK et al., 2004], here temperature is air temperature:

- Frozen food, the maximum temperature is -18°C.
- Fresh fish and products, the maximum temperature is +2°C.
- Milk, the maximum temperature is +5°C.

In general, the requirement is a temperature below +5°C [Databank, 2007]. There are also some temperature requirements during food processing and transportation.

4.3. Mathematic model

4.3.1. Numerical simulation of unsteady-state conduction

Unsteady state conduction refers to a class of problems in which the temperature of the conduction region varies with time, such as the case in which the boundary condition varies with time such that neither a steady state nor a periodic behavior is ultimately attained, and this is exactly our case.

Governing equation

$$\nabla(k(T)\cdot\nabla T)=\rho\cdot C_p(T)\cdot\frac{\partial T}{\partial t} \tag{1}$$

Where T is temperature, t is time, $k(T)$, ρ, $C_p(T)$ are so called thermophysical properties of foodstuffs: thermal conductivity, density and specific heat capacity. They will be discussed in details later.

Convective boundary condition

$$k(T)\cdot\frac{\partial T}{\partial n}=h\cdot(T-T_\infty) \tag{2}$$

where n is the outward-facing normal vector on the body surface, h is the convective heat transfer coefficients, and T_∞ is the ambient temperature.

Initial condition

$$T=T_0 \tag{3}$$

Where T_0 is a known temperature.

4.3.2. Temperature-dependent thermophysical properties

In many engineering problem, the variation of thermal properties is significant over the temperature range concerned [Jaluria and Torrance, 1986], and must be taken into consideration.

Predictive equation based on the water content φ, actual and initial freezing temperature T_{cr} developed by Fikiin and Fikiin [Fikiin and Fikiin, 1999] is applied in the modeling of fish products.

The water content of the fish varies with different species of fish as the composition is different. Even for the same fish, its constitute values change in the different environments and different seasons. Based on the source [Databank, 2007], the water content of fish varies between 53% (for very fatty fishes) and 83% (for lean fishes). Here cod is selected as one example, with water content of 79.3%.

Specific heat capacity C_p: For the food with critical temperature T_{cr} between -2∼-0.4°C, it can be calculated as follows.

$$C_p(T) = \begin{cases} C_{p,un} = 2.805\varphi + 1.382 & \text{for } T \geq T_{cr} \\ C_{p,fr} = 1.382 - \varphi \cdot A(T) - \varphi \cdot B(T) & \text{for } T < T_{cr} \end{cases} \tag{4}$$

Where subscript fr, un refer to frozen and unfrozen respectively, and

$$A(T) = \frac{2.286}{1 + 0.7138/\ln(T_{cr} - T + 1)} - 2.805$$

$$B(T) = \frac{-264.231(T_{cr} - T + 1)^{-1}}{(\ln(T_{cr} - T + 1) + 0.7138)^2}$$

$$-45 \leq T \leq 45^{\circ}C$$

Thermal conductivity k: It is an intrinsic property of the material, depending on its composition and structure. The empirical correction proposed by Miles [Miles, 1991] can be used with satisfactory precision for engineering investigations.

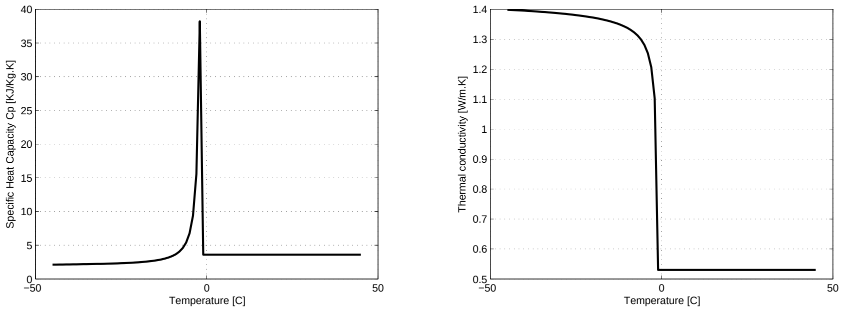
$$k(T) = k_{un}(T) + f \cdot \varphi \cdot (k_i(T) - k_w(T)) \tag{5}$$

where f is a correction factor, for meat and fish, it varies from 0.61 to 0.77. k_{un}, k_i and k_w are the thermal conductivity for the unfrozen product, ice and water respectively.

Density ρ: The density temperature dependence, predetermined mainly by the small difference between water and ice densities, is comparatively weakly expressed. So the average density for frozen and unfrozen fish is used respectively in the model.

4.4. Methodology

The transit heat transfer problems involving melting and solidification are generally refer to as the "phase change" or "moving boundary" problems. Sometimes they are referred as the "Stefan" problems [Ozisik, 1994]. There has been developed several methods and algorithms dealing with such kind of problems.



(a) Specific heat capacity C_p (b) Thermal conductivity k

Figure 4.4: Dependence of thermophysical properties with temperature.

4.4.1. Enthalpy method

It use the mass specific enthalpy, so there is no need to accurately track the phase-change boundary, no need to consider liquid and solid regions separately, see example [Nedjar, 2002].

4.4.2. Improved enthalpy method

It combines the volumetric specific enthalpy and Kirchhoff transformation, so that all the non-linearities, caused by the thermal properties depending on temperature, are introduced in the functional relationship between the volumetric specific enthalpy and Kirchhoff function, see details [Fikiin and Fikiin, 1999].

Both of these methods need much efforts on programming, and are often solved by finite element methods.

4.4.3. Finite difference method

Here Finite Difference Method (FDM) is chosen for simplicity, 1D case is used as an example.

The governing equation (1) can be discretized as:

$$\begin{aligned}(\rho C_p)_i \frac{T_i^{n+1} - T_i^n}{\Delta t} =& \theta \left(k_{i-\frac{1}{2}} \frac{T_{i-1}^{n+1} - T_i^{n+1}}{(\Delta x)^2} + k_{i+\frac{1}{2}} \frac{T_{i+1}^{n+1} - T_i^{n+1}}{(\Delta x)^2} \right) \\ &+ (1-\theta) \left(k_{i-\frac{1}{2}} \frac{T_{i-1}^{n} - T_i^{n}}{(\Delta x)^2} + k_{i+\frac{1}{2}} \frac{T_{i+1}^{n} - T_i^{n}}{(\Delta x)^2} \right)\end{aligned}$$

where constant $\theta (0 \leq \theta \leq 1)$ is the weight factor representing the degree of implicitness, the value of $\theta = 0, 1/2, 1$ corresponds respectively to the explicit, Crank-Nicolson and implicit method. In this paper, the explicit FDM is applied. It is simple but with restrictions on time step and space step, in order to ensure the convergence.

4.4.4. Temperature-dependent properties evaluation

In these transient problems, the temperature variation in the conduction region is small, so the average value of the material properties will be taken at each time interval.

Simplest method evaluating the property is to lag the evaluation by one time step, such as:

$$k^{n+1} = k(T^n) \tag{6}$$

Some more complicate methods include evaluating the property as or at the average value, or use an exploration scheme, for example:

$$k^{n+1} = k^n + \left(\frac{\partial k}{\partial T}\right)^n (T^n - T^{n-1}) \tag{7}$$

4.4.5. Interpolation of air temperature in boundary condition

The air temperature in the real refrigeration system is sampled every minute, it varies with time. To apply it in the dynamic convective boundary condition, we need to interpolate it for each time step, the easiest way is assuming that it is linear within one minute.

4.5. Results

The results are obtained by FDM and based on the following assumptions, parameters are given in Table 4.1.

- A infinite plate of minced fish with the thickness of 20 mm.

- The fish has both sides the same convective boundary condition.

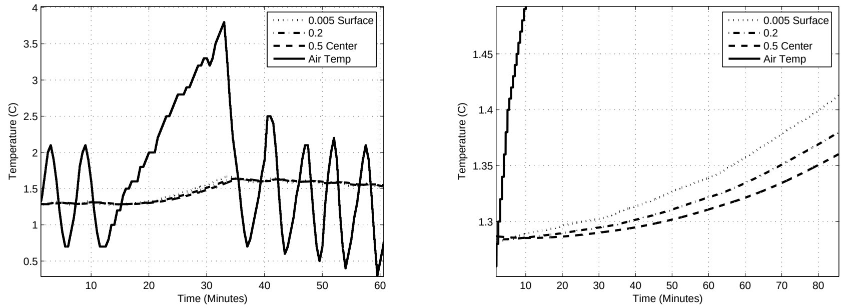
Since it is symmetric, so only half of it needs to be considered.

Table 4.1: Parameters used in the simulation of heat transfer

Specification	Cod	Herring	Unit
Heat transfer coefficient	5	5	$W\,m^{-2}\,K^{-1}$
Relative water content	0.793	0.53	$Kg\,Kg^{-1}$
Thermal conductivity, unfrozen	0.53	0.796	$W\,m^{-1}\,K^{-1}$
Correction factor	0.70	0.70	
Density, unfrozen	1050	930	$Kg\,m^{-3}$
Density, frozen	960	850	$Kg\,m^{-3}$
Initial freezing point	-1	-1	°C
Thermal conductivity ice	2.18	2.18	$W\,m^{-1}\,K^{-1}$
Thermal conductivity water	0.58	0.58	$W\,m^{-1}\,K^{-1}$

4.5.1. Fresh fish products

If we ignore the small change in the thermal properties above the initial freezing point, and assume they are constant, the result is shown in Figure 4.5. In the defrost cycle, the surface temperature increases about 0.4°C, and the maximum temperature difference between the surface and center is less than 0.1°C.

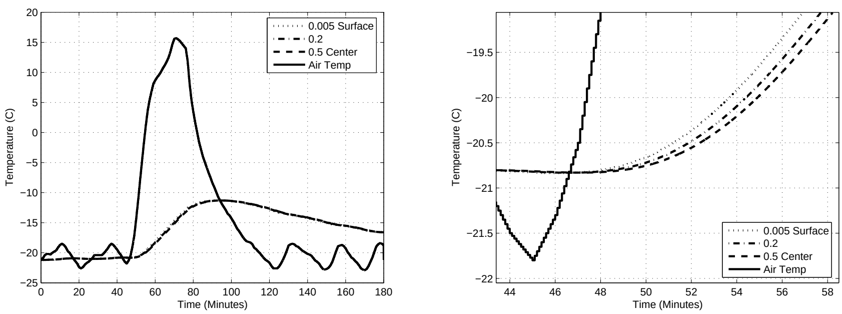


(a) Full profile (b) Partially zoomed

Figure 4.5: Time temperature profile for fresh fish products at different depth.

4.5.2. Frozen fish products

Since it has non linear thermal physical properties, which change with temperature, the evaluation of properties needs to be done at each temperature, here at one time step lag. The result is shown in Figure 4.6. In the defrost cycle, the surface temperature increases 9.9°C, and the maximum temperature difference between the surface and center is around 0.6°C.



(a) Full profile (b) Partially zoomed

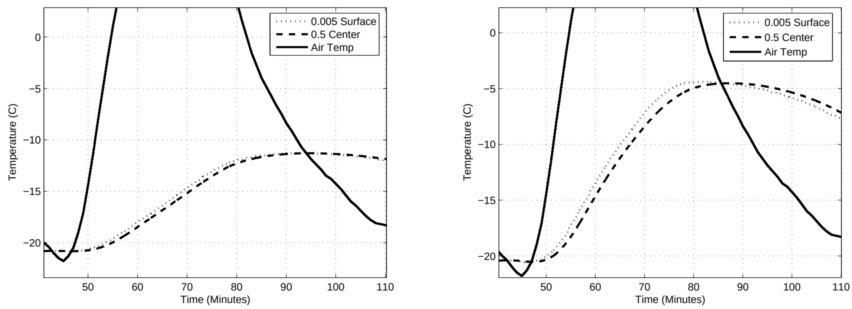
Figure 4.6: Time temperature profile for frozen fish products at different depth.

4.6. Discussion and conclusion

In this section, different factors that influence the heat transfer, such that the storage product temperatures are discussed.

4.6.1. Air velocity inside the display cabinet

The heat transfer coefficient is very often a difficult parameter to obtain for a process. It depends on the shape of the product and the motion of the gas or liquid used for heating or cooling the product. While the measurement of center temperature in a product can provide us with an average heat transfer coefficient for the product investigated [Friis et al., 2004]. In a normal display cabinet, air velocity is around 0.05∼0.2 $\mathrm{m\,s^{-1}}$, heat transfer coefficient h varies within 3∼5 $\mathrm{W\,m^{-2}\,K^{-1}}$. If by disturbance or some other reasons, cold air circulates at a much higher speed, the corresponding h will increase a lot. Here simulation result with h =5 $\mathrm{W\,m^{-2}\,K^{-1}}$ is compared with h =15 $\mathrm{W\,m^{-2}\,K^{-1}}$ in Figure 4.7.



(a) h = 5 $Wm^{-2}K^{-1}$ (b) h = 15 $Wm^{-2}K^{-1}$

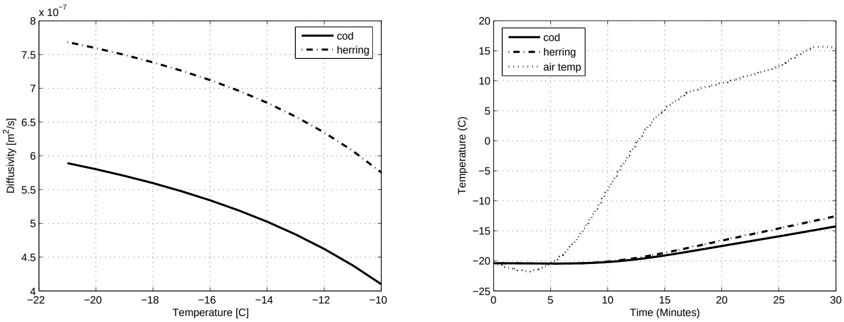
Figure 4.7: Frozen fish temperature profile during defrosting under different heat transfer coefficient.

4.6.2. Packaging of food

In the case of packaging, such as package the fish with a cardboard which has conductivity coefficient k =0.007 $\mathrm{W\,m^{-2}\,K^{-1}}$, the local heat transfer coefficient h should be replaced with overall heat transfer coefficient U, which is normally smaller, and may around 1 $\mathrm{W\,m^{-2}\,K^{-1}}$. This will lead to less temperature change during defrost cycle. General speaking, a proper package can protect foodstuffs from suffering environmental temperature variations.

4.6.3. Type of food

Compare with other type of food, fish has relatively shorter storage life, especially fresh fish, which can only be kept at high quality within $2 \sim 3$ days. So in this paper, we select



(a) Diffusivity (b) Center temperature

Figure 4.8: Time temperature profile for different type of fish products.

fish as one worst case study object. We know that the water content of fish is often used as criteria to categorize the fish into fatty or lean type. So for fatty and lean fish, their thermophysical properties (based on water content) are normally different. Its overall effect is reflected in diffusivity $\alpha = k/(C_p \cdot \rho)$. Figure 4.8 show the diffusivity and time temperature profile for fatty fish herring and lean fish cod.

4.6.4. Biot number B_i

The calculation result shows in some cases, there is no significant temperature difference between different layers, or even between the center and surface layer. Estimation of Biot number, can provide us the information on whether we can treat the model as "lumped" model, or non-isothermal model. In the phase change problem, even k changes with temperature, but we can still by using the correlative equation to predict the range of its value, to estimate B_i.

4.6.5. Conclusion

This model uses simple explicit FDM to deal with the phase change problem, featured with time varying boundary condition, and temperature dependent thermophysical properties. It can be used as one tool, to evaluate the effect of various parameters to the product temperature change.

It can be concluded that the defrost cycle in the refrigeration system influences the product temperature dramatically. For same type of fish, low water content, can lead to higher temperature change, while for different type of fish, we need to consider the overall effect of diffusivity. By simulating the different defrosting scenarios, we can obtain the different product temperature profiles, together with the hereafter developed quality model, we can figure out which defrosting scheme is more optimal for product quality during refrigerated storage.

In later stage, we need also take the overall energy consumption into our control objective. We plan to use the multi-objective optimization to design a new controller, model predictive control could be one of the approaches.

Chapter 5

Quality Model of Foodstuffs in a Refrigerated Display Cabinet

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Abstract

Commercial refrigerating systems need to be defrosted regularly to maintain a satisfactory performance. When defrosting evaporator coils, air temperature inside the display cabinet will increase, and float outside the normal temperature range for a period of time, the question is what happens to the food inside during this period, when we look at their quality factor?

This paper discusses quality modeling of foodstuffs, different scenarios of defrost schemes are simulated, questions such as how the defrost temperature and duration influence the food temperature, thus the food quality, as well as what is the optimal defrost scheme from food quality point of view are answered.

This will serve as a prerequisite for designing an optimal control scheme for commercial refrigeration systems, aiming at optimizing a weighed cost function with both food quality and overall energy consumption of systems.

5.1. Introduction

Quality of food has become a way of profiling the high end supermarkets, which prefer to compete on quality rather than price. Today the refrigeration in supermarkets is controlled based on the regulations from food authorities, keeping the food within a specific temperature limit, but this regulation is only applicable when the refrigeration system is in the normal operational mode.

For normal commercial refrigeration system, frost build-up on evaporator coils is almost unavoidable. It is a function of time and some environmental factors such as air humidity, air velocity, air and fin temperature etc. It is undesired, and if nothing is done, the accumulated frost layer will lead to a dramatic degradation of system efficiency, so defrost has to be performed regularly to maintain a satisfactory system performance. During defrost, air temperature inside the display cabinet around the food will increase, and float outside the normal temperature range for a period of time, depending on defrost schemes and techniques.

There are a lot research on when and how to defrost, while most of them are from the energy point of view [Hoffenbecker et al., 2005] [Payne and ONeal, 1992]. Regarding that how defrost influence food quality, and how to optimize the defrost scheme to minimize the quality decay, few reports exist.

Our approach is by establishing a quality model, to investigate what kind of food is more sensitive to the temperature change, and by simulating the different scenarios of defrost schemes, to find out what kind of defrost scheme is most optimal for food quality.

This paper is organized as follows: refrigeration of foodstuffs in a supermarket is described in Section 5.2, in Section 5.3 we discuss the quality model, finally some discussions and conclusions in Section 5.4.

5.2. Refrigeration of foodstuffs in a supermarket

The display cabinet depicted in Figure 5.1 consists of a food container and an air tunnel, circulating cold air around the food container. An evaporator in the air tunnel cools the passing air which creates a carpet of cold air on top of the food.

The fact that the air carpet is colder than the food and ambient air, will keep the air carpet in place and become denser, enabling the desired effect of heat transfer from the carpet to the container and food. A side effect is that the ambient air will infiltrate into the carpet at the load zone. The display cabinet's temperature is normally controlled by a hysteresis controller which opens and closes the inlet valve, to control the flow of refrigerant into evaporator, thus keeping the air temperature within the specific bands, see Figure 5.2, where the big change in temperature is caused by a defrost cycle.

5.2.1. Frost formation and defrost cycle

Frost forms on evaporator coils as water vapor in the air condense and freeze when they contact the coil surface which is below 0°C. This is a well known and undesirable phenomenon. It deteriorates the system performance by decreasing the effective air flow area and increasing the thermal resistance between warm air and cold refrigerant. Currently there are no clear and reliable measures that can prevent frost formation [Shin et al., 2003]. When frost accumulates to a certain level, defrosting must be done to maintain a satisfactory system performance.

Defrost methods vary with refrigeration applications and storage temperature, and initiation and termination of defrost can be controlled by many different parameters, such as a timer or temperature, sometimes up to 3 cycles per day. During the defrost cycle, air temperature inside the cabinet will increase and remain outside the normal temperature range for a period of time.

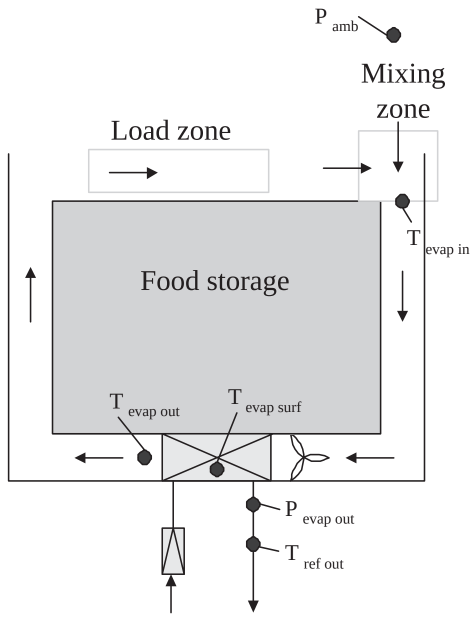
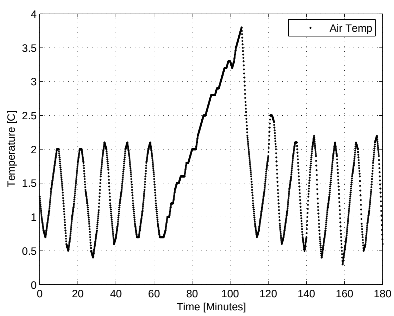
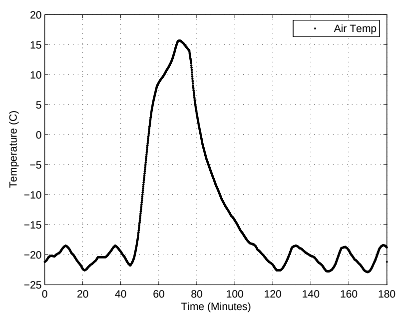


Figure 5.1: A simplified display cabinet in a supermarket.



(a) Fresh fish products storage



(b) Frozen fish products storage

Figure 5.2: Air temperature profile with one defrost cycle.

5.2.2. Legal requirements

In Danish supermarkets, there are legal requirements regarding the storage temperature for different foodstuffs in display cabinets [DSK et al., 2004], here temperature is air temperature:

- Frozen food, the maximum temperature is -18°C.
- Fresh fish and products, the maximum temperature is +2°C.
- Milk, the maximum temperature is +5°C.

In general, the requirement is a temperature below +5°C [Databank, 2007]. There are also some temperature requirements during food processing and transportation.

5.3. Quality model

5.3.1. Background

During food processing and storage a lot of "chemical" reactions occur. In general chemistry "reaction kinetics" are often treated in terms of reaction rates under specified conditions (normally isothermal). In food processing the actual reaction rate is interesting, but only as a means of obtaining the most interesting information: the integral effect, i.e. the accumulated effect after some processing steps or storage periods with varying conditions. Examples of "chemical" reactions:

- Loss of vitamins
- Creation of structure by protein coagulation
- Formation of lactulose due to high temperatures
- Growth of microorganisms
- Texture change (soft, hard, flow, breakage etc.)
- Flavor, color or odor change

The purpose of reaction kinetics in a food process is thus to be able to predict / calculate / estimate the consequences of a given treatment for a food item. The results may be as an absolute number such as a concentration, a fraction remaining or an index of the changes.

5.3.2. Shelf life calculations and quality loss

The shelf life of a product is a very important parameter. If the shelf life is too short, it will be impossible to market the product, and if it is declared too long, consumers may wonder why. The definition of shelf life depends on the limiting factors for the product. It may be microbial spoilage or decolorization as often in chilled storage, or it could be loss of vitamins or color as in frozen storage. These factors may all be observed by objective methods. Not so with the sensory values of the products. Here we are forced to make observations regarding if the product has changed with respect to the fresh product. The investigation is carried out by triangular sensory tests, to establish if the assessors can detect a difference between a product that has not altered (stored at very low temperature) with significance.

Frozen storage (-30∼-18°C): For frozen foods, based on the kind of assessors, it is defined:

- High Quality Life (HQL), when the assessors are trained.
- Practical Storage Life (PSL), when the assessors are untrained (normal consumers).

The relation between HQL and PSL is described by an acceptability factor, which may range from 2 to 10, depending on the product, and may even not be constant for the different temperatures. The experiments are carried out storing the products at different temperatures, and assessing the quality at certain times. As soon as the assessors detect a difference with significance, the quality is deemed to be zero. The product starts with a quality of 100% and end with 0%. We have to stress, that a product of 0% quality, by this definition, is a very good product. The 0% quality is determined by the fact that a set of assessors looking for differences just find it with significance. The normal consumer would not detect the changes.

The time for loss of quality (100%) at temperature T is termed as D_T, the function between D_T and temperature is expressed as 1, where z value is the product temperature sensitivity indicator, D_{ref} is the time for loss of quality at reference temperature T_{ref}.

$$D_T = D_{ref} \cdot \exp\left(-\frac{T - T_{ref}}{z}\right) \tag{1}$$

Calculation of shelf life is based on how much "quality" is consumed during different steps of storage [Andersen and Risum, 1994]. This can be described by a storage profile expressing how long time t, the product has been kept at certain temperature T.

$$Q_{loss}(T,t) = \frac{100\%}{D_T} \cdot t \tag{2}$$

Accumulated quality decay is based on linear additivity and temperature-time profile, such that:

$$Q_{loss,tot} = \sum_i Q_{loss}(T_i, t_i) \tag{3}$$

For frozen products one would normally use -18°C as reference temperature T_{ref}, z value depends on the product.

Chill storage (0°C∼): For chilled food, in contrast to frozen food, the quality of chilled food from start is defined as 100% and 0% when the product is unsuitable for sale. Calculations are based on the same principles as for frozen food. But for chilled food one would normally use 0°C as T_{ref}, z value depends on the product.

Parameters used in the simulation of quality decay are given in Table 5.1.

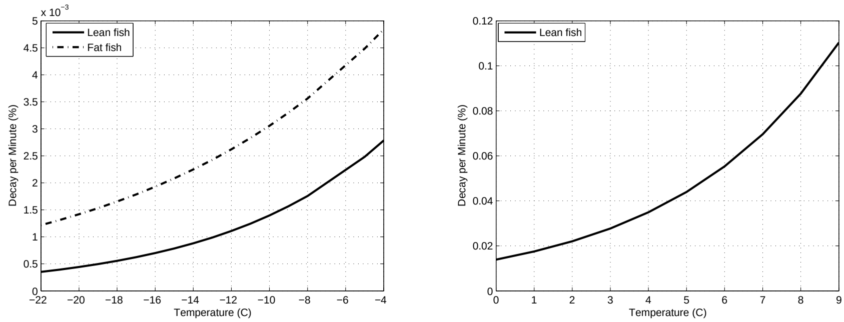
Table 5.1: Parameters used in the simulation of quality decay

Product	D_{ref}	z-value
frozen lean fish	125	20
frozen fat fish	42	30
fresh lean fish	5	10

Quality decay per minute in % for frozen fat, lean fish products and fresh lean fish products are shown in Figure 5.3.

From the figure, we can see that the quality decay is not in linear relation with temperature, high temperature has relatively higher deterioration effect.

By applying the dynamic heat transfer model, we can get the dynamic food temperature profile, therefore we can calculate the real time decay rate and accumulated daily or monthly decay.



(a) Frozen fat and lean fish products

(b) Fresh lean fish products

Figure 5.3: Minute quality decay in % as a function of temperature.

5.4. Discussion and conclusion

In this section, we will simulate the quality decay under different scenarios of defrost schemes, to see what make difference from the quality point of view, and which kind of food is more sensitive to the defrost temperature change.

5.4.1. Different scenarios of defrost schemes

Today the most simple way to defrost is scheduled defrost, which uses a timer to initiate defrost, defrost is normally terminated by a defrost stop temperature while with a maximum defrost time as security. Depending on the storage temperature, store environmental conditions and defrost techniques, defrost frequency and duration are different. For frozen storage, it can be 1$\sim$2 times per day, for chilled food, around 2$\sim$3 times per day.

Scenario 1, one scheme defrosts more frequent and short duration each time, another scheme defrost less frequent with longer duration each time, see Figure 5.4.

Quality loss for these two defrost schemes is shown in Figure 5.5, which corresponding to a daily decay of 27.4% and 27.1% respectively, with a relative difference of 1.1%.

Scenario 2, since the peak value of defrost temperature is one of main factors that influences the quality change, the ideal situation will be a low peak value. We lower the temperature before defrosts as in Figure 5.6, to see what we can gain. Decay for "normal" defrost scheme and manipulated defrost scheme is shown in Figure 5.7, which corresponds to a daily quality loss of 27.4% and 26.0% respectively, with a relative difference of 5.1%.

5.4.2. Type of fish quality sensitivity analysis

Fat fish normally has a shorter shelf life. The International Institute of Refrigeration (IIR) recommends the storage temperature for fat fish such as herring to be -24°C, while for the lean fish such as cod, to be -18°C. Fat fish has normally a higher decay rate comparing with lean fish, recall Figure 5.3a, is that mean that fat fish is more sensitive to the

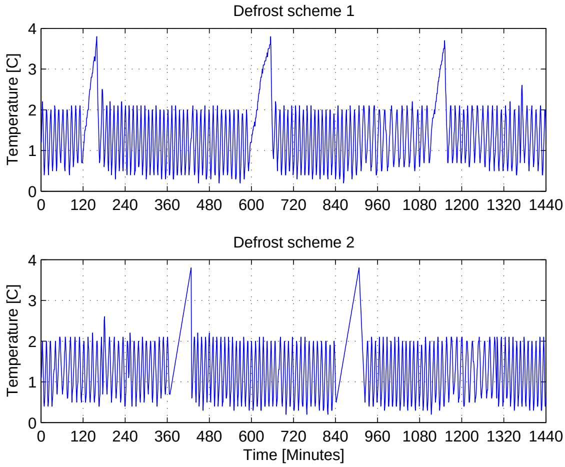
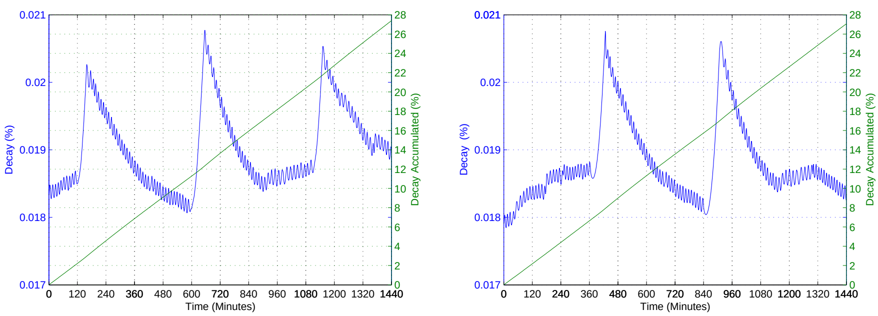


Figure 5.4: Air temperature profile for defrost scheme 1 and 2.



(a) Defrost scheme 1

(b) Defrost scheme 2

Figure 5.5: Fresh fish products quality decay rate and accumulated daily quality decay under defrost scheme 1 and 2.

temperature change? Figure 5.8a shows the fat and lean frozen fish decay rate under the same air temperature profile (accumulated daily decay is 2.2% and 0.7% respectively).

If we zoom the defrost period, to see what is the difference if we defrost comparing if there is no defrost, meaning keep the fish at more or less constant temperature instead. Figure 5.8b shows the decay difference under these two different situations.

For fat fish products, if there is no defrost during this 4 h, decay is 0.37%, with defrost is 0.42%, defrost increases the decay about 13.5%; for lean fish, if there is no defrost during this 4 hours, decay is 0.12%, with defrost is 0.14%, defrost increases the decay about 14.3%.

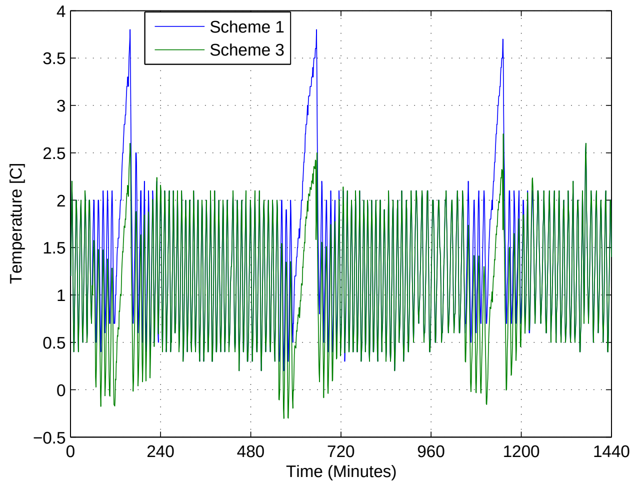
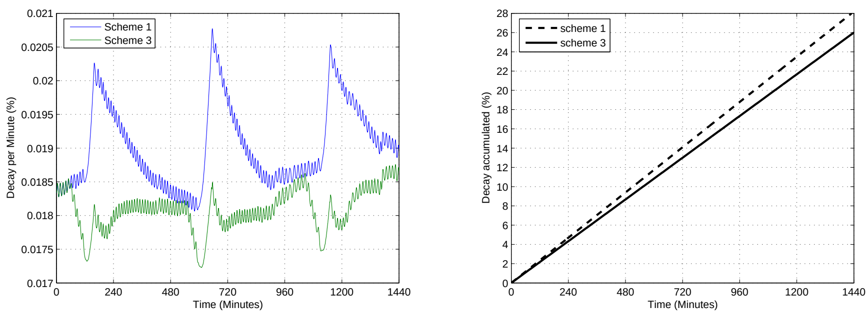


Figure 5.6: Air temperature profile for defrost scheme 1 and 3.



(a) Real time decay

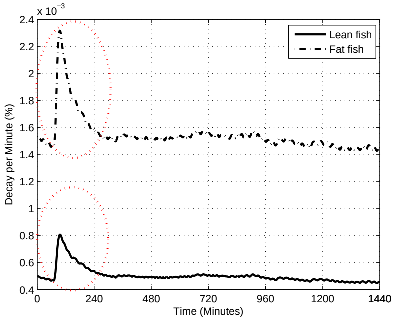
(b) Accumulated daily decay

Figure 5.7: Fresh fish product quality decay under defrost schemes 1 and 3.

5.4.3. Conclusion

By simulating the different defrosting scenarios, we can see that under the same condition, less frequent defrost with longer time duration, and low peak value of defrost temperature lead to less quality decay, and better for food storage. Fat fish has a higher decay rate, but it is not more sensitive to the temperature change, as we assumed.

While when we defrost the evaporator coil for frozen food by an electric heater, we need firstly supply energy to heat up the evaporator coil to melt frost, and after defrosting, supply energy to lower the coil temperature again, in order to bring it back to normal operation. If we lower the temperature before defrosting, system consumes normally more energy. Therefore we need to know what is the cost when we use different defrost schemes, this can be investigated by defrost and energy model.



(a) Fat vs. Lean

(b) Defrost vs. Non-defrost

Figure 5.8: Frozen fish products quality decay and sensitivities to defrosting.

Finally we can combine these two aspects- energy and quality into a weighted overall objective function, and design an optimal control scheme to optimize the overall system performance.

Chapter 6

An Active Defrost Scheme for Balancing Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems

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Abstract

This paper introduces food quality as a new parameter, together with energy, to determine an optimal cooling time between defrost cycles. A new defrost-on-demand scheme is proposed. It uses a feedback loop consisting of on-line model updating and estimation as well as a model based optimization. This scheme automatically adjusts the time interval between defrost cycles with varying operating conditions, continuously seeking an optimal time interval, featuring either an energy optimal time, or a trade-off between energy consumption and food quality loss. This adaptive approach is compared with traditional defrost schemes, found to be able to reduce energy consumption significantly.

6.1. Introduction

Supermarkets are one of the most energy-intensive types of commercial buildings. Refrigeration is the largest component of their electric energy usage, accounting for half or more of the store total. Among others, energy associated with defrosts and anti-sweat heaters in supermarkets may exceed 30% of the total requirement [Howell et al., 1999]. Thus, from an energy point of view, determining how often to defrost is an important issue. From a food quality point of view, it is also important. During defrost, the air temperature inside a display cabinet will normally increase, and so will the food temperature. Depending on the defrost method, the food temperature will usually stay out of the normally controlled range for a period of time, which is harmful to the food quality [Cai et al., 2006c].

Defrost approaches can be classified in two major schemes: one is scheduled defrost, another is defrost-on-demand [Woodley, 1989] [Tassou et al., 2001] [Ciricillo, 1985]. Fahlen [Fahlen, 1996a] studied and compared these two schemes, and found that both of them have some excellent features, but various limitations and drawbacks. Food quality has never been used as an active decision factor in either of these schemes.

Biotechnology provides us the knowledge of food quality change during the refrigerated storage. Research on frost formation enables us to predict the system performance degradation under frosting conditions.

This paper introduces food quality as a new parameter, together with energy, to determine an optimal time between defrost cycles depending on ambient parameters. Based on this, a new defrost-on-demand scheme is proposed.

The paper starts with a simple introduction to supermarket refrigeration systems and the most commonly used defrost methods, defrost schemes, which is in Section 6.2. In Section 6.3, we discuss the problem associated with traditional defrost schemes by using three models, two for energy and one for food quality. In Section 6.4, we propose a new defrost-on-demand control scheme. Simulation results are presented in Section 6.5. Gains by the new defrost-on-demand control scheme is demonstrated in Section 6.6. Discussions and conclusions are given in Section 6.7.

6.2. Simple introduction to refrigeration systems and defrost schemes

6.2.1. Refrigeration of foodstuffs in a supermarket

The display cabinet depicted in Figure 6.1 consists of a food container and an air tunnel. The evaporator inside the air tunnel cools the passing air, which circulates around the food container and creates an air blanket on the top of foodstuffs.

To control properly the temperature inside a cabinet, one or more thermal sensors are required in the system. The number and function of those sensors differ from one application to another. For example in our system, there are three sensors: S_3, S_4, S_5. S_3 and S_4 are used to measure the temperature of air inlet $T_{a,i}$ and outlet $T_{a,o}$ respectively, S_5 is a defrost stop sensor.

To increase sales, any physical obstacles between the products and customers should be avoided. As a consequence, for most display cabinets, one or more forced air blankets (for horizontal display) or air curtains (for vertical display) are the only barrier between the refrigerated foodstuffs and warm ambient air. Due to the disturbance of air flow, a more or less significant amount of warm ambient air is always entrained, which introduces the frost formation, reduces the temperature control capabilities and increases energy consumption.

6.2.2. Requirements on food storage temperature

In supermarkets, there are general requirements regarding the storage temperature for different foodstuffs in display cabinets. For example in Denmark, the temperature according to [Announcement, 2004] and [DSK et al., 2004]:

- Frozen food, the max. temperature is -18°C.

- Fresh fish and fish products, the max. temperature is +2°C.
- Milk, the max. temperature is +5°C.

Usually, the legally constrained temperature is the air temperature, although the surface temperature would be a better measure for food quality.

6.2.3. Frost formation and defrost methods

Frosting is a well known and undesirable phenomenon on evaporator coils. It happens whenever the surface temperature of the evaporator is below 0°C and humid air passes by. Frost decreases the performance of the heat exchanger. In order to maintain a satisfactory performance, defrosting needs to be done regularly.

The most common defrost methods for medium and low temperature applications are:

- Off-cycle defrost: This is the simplest defrost methods: refrigeration is stopped, evaporator fans continue to move room air over the frosted coil surface, which warms and melts the frost.
- Hot/cool gas defrost: During a hot/cool gas defrost, the normal supply of cold refrigerant is stopped. The former involves the circulation the hot gas from the compressor discharge manifold directly to the display cabinet, and the latter utilizes cooler gas from the liquid receiver. The cool or hot gas condenses in the evaporator, releasing heat which melts the ice from the coil.
- Electric defrost: This approach uses the electric heater embedded on the fin surface to supply heat, warm and melt the frost.

However, not all methods can be used in all circumstances. Normally, for medium temperature applications, the cheapest and least energy consuming means of defrost is

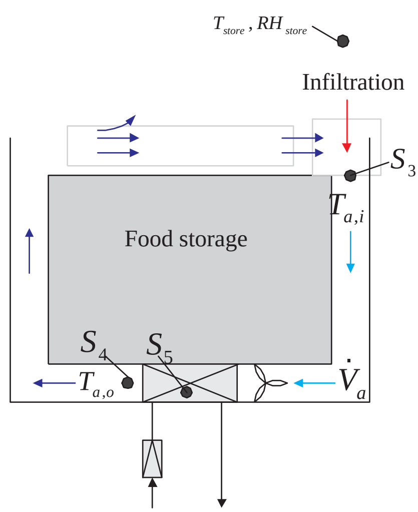

Figure 6.1: A simplified display cabinet in a supermarket.

off-cycle. This method of defrost is widely used in the industry. For low temperature applications, the use of gas defrosting is more energy efficient than electric defrost, but the considerable extra capital costs lead to a long payback period. Consequently, electric defrost is the most commonly used defrost method for low temperature applications.

6.2.4. Defrost schemes

In a simple taxonomy, there are the following two classes of defrost schemes.

- Scheduled defrost: Initiating the defrost cycle by a timer, normally with a fixed number of defrost cycles per day. Defrost is terminated either based on a fixed time or on a stop temperature while with a maximum defrost time as a security.
- Defrost-on-demand: Initiating the defrost cycle only when it is necessary, see [Llewelyn, 1984] [Muller, 1975]. This approach normally uses one parameter to initiate and terminate the defrost process, such parameters could be: air pressure difference across the evaporator, temperature differential between the air and the evaporator surface, fan power sensing etc.

Features and shortcomings of the current defrost schemes are:

- Scheduled defrost is simple and uses a low cost controller, so it is the most commonly used defrost scheme in today's supermarkets, but the time schedule is normally determined based on experience and observation, most cases, based on worst case conditions. It is configured during the commission stage, and can not automatically adapt to the varying shop conditions where the system works at, so the time between two defrost cycles can be either too long or too short.
- Existing defrost-on-demand schemes typically involve the installation of additional sensors to detect frost build-up, and use one parameter to initiate and terminate the defrost cycle. The threshold of this detected parameter is determined mainly to ensure a safe operation, or maintain the degradation in the performance of the system within fixed limits over the whole range of operating conditions. No energy optimality is guaranteed. For example, in [Tassou and Marquand, 1987] a defrost initiation pressure was selected to limit the degradation in performance due to frosting to less than 10% of the steady-state value.
- None of the existing schemes have used food quality as a decision factor.

6.3. Energy and quality modeling

To analyze the energy consumption of the system and quality loss of refrigerated foodstuffs, in this section, we will introduce three models, two for energy consumption and one for quality loss of foodstuffs.

6.3.1. Extra energy to compensate for the reduced performance under frosting

Frosting of the heat exchanger surface affects its thermal performance in the following ways [Chen et al., 2003]:

- It increases the thermal resistance between the fin and airflow, and decreases the cooling capacity of heat exchangers.

- It substantially reduces the airflow through heat exchangers, and increases the air pressure drop through heat exchangers. Depending on the characteristic of the fan, several hours later, the airflow path may be nearly or completely blocked.

Frost build-up is a complex process even on a flat plate. It is affected by many factors, such as air flow velocity, air and plate temperature, air humidity ratio, etc. Most of the research are based on experiments, see example [Sahin, 1994]. Analytical or numerical modeling exists with different limitations, see example [Yang and Lee, 2005]. Frost growth on a real evaporator becomes even more complex, limited modeling existed, see [Yang et al., 2006]. Moreover, the characteristic of the evaporator is the combined characteristic of the evaporator itself as the heat exchanger and fan. When the air flow rate decreases with time due to frost build-up, the overall heat transfer coefficient between air and evaporator will decrease; In order to maintain the same cooling capacity, the temperature drop of the air across the coil must increase, this in turn requires a lower evaporating temperature. Sometimes the drop in the evaporating temperature is significant, and cause an increase in power consumption to the plant [Stoecker, 1957]. All those operations are interdependent. The prediction of the overall system performance under frosting conditions is definitely not trivial.

This paper is not aiming at developing a detailed model to predict the frost formation under varying conditions. Instead it uses energy correlations to calculate how much water is condensed on the surface of evaporator as frost. The purpose of modeling here is for controlling, so the model itself is extremely simple but still captures the main dynamical features seen from an input / output point of view.

Simplifying assumptions:

- Frost is distributed uniformly on the whole heat exchanger surface.

- Density of frost is constant, so only the frost thickness grows with time.

- The heat transfer coefficient of the refrigerant is constant.

Frost growth rate estimation: The nominal cooling demand Q_{nom} for a display cabinet can be calculated as follows [Holm et al., 1996] (without special notification, all the units are standard *SI* units):

$$Q_{nom} = Q_0 \sum_{i=1}^{4} (G_i \cdot X_i \cdot Y_i) \tag{1}$$

Where G_i is the correction factor for the difference between testing conditions and measured actual operating conditions, X_i is the load distribution factor, Y_i is the load reduction factor related to the covering of the display case, if no physical covering $Y_i = 1$. i indicate the load type, 1 for the load from heat conduction, 2 for infiltration, 3 for radiation and 4 for the load from electric equipments, such as light, fan, anti-sweat etc. Q_0 is the standard cooling demand, and can be calculated according to cabinet category and dimension, as follows.

$$Q_0 = F \cdot (A \cdot L_D + B \cdot A_D) \tag{2}$$

Table 6.1: X_i and constants for calculation of standard cooling demand Q_0

Category	A	B	F	X_1	X_2	X_3	X_4
Unit	[W m^{-1}]	[W m^{-2}]	[−]	[−]	[−]	[−]	[−]
M.02.01	70	260	1.000	0.10	0.40	0.30	0.20
M.02.02	70	260	1.353	0.10	0.35	0.30	0.25

where A, B, F are constants related to the different type of display cabinet, as shown in Table. 6.1. L_D is the length of display case, and A_D is the display area.

Where M.02.01 and M.02.02 are island site medium temperature display case, with and without electric defrost respectively.

The correction factor for the infiltration and the infiltration load can be calculated as follows:

$$G_2 = \frac{I_{store,m} - I_{cab,m}}{I_{store,test} - I_{cab,test}} \tag{3}$$

$$Q_{inf} = Q_0 \cdot (G_2 \cdot X_2 \cdot Y_2) \tag{4}$$

Where I are specific enthalpy, which can be calculated from temperature and relative humidity (RH). Subscript $store$, cab represent store and cabinet respectively, $test$ and m refer to test and measurement.

Infiltration is caused by amount of hot humid air from the store entrained in the display cabinet. The load of infiltration can also be calculated in another way:

$$Q_{inf} = \dot{m}_{a,ent}(I_{store,m} - I_{cab,m}) \tag{5}$$

From the above correlation, we can calculate the mass flow rate of air entrained $\dot{m}_{a,ent}$.

When the foodstuffs are properly packed, infiltration is the only source of water that condensates on the cold surface of evaporator and eventually becomes frost. As time goes, it will increase both the thickness and density of frost, as described by:

$$\dot{m}_{fr} = \dot{m}_{a,ent}(x_{store} - x_{cab}) \tag{6}$$

$$\dot{m}_{fr} = \rho_{fr} \cdot \dot{\delta}_{fr} \cdot A_{fr} \tag{7}$$

where x is the specific humidity or humidity ratio of air, based on temperature and RH, $\dot{m}_{fr}$ is the frost mass growth rate. ρ_{fr} and A_{fr} are the frost density, frosting area respectively. $\dot{\delta}_{fr}$ is the frost thickness growth rate. Here we assume the density of frost is constant. A review on frost properties and modeling was given by Iragorry et al. [Iragorry et al., 2004].

Finned tube heat exchanger: Here we use the heat exchanger from Blundell as an example [Blundell, 1977]. It has wavy continuous fins on the circular tube, with a staggered array. Slight modifications have been made on the dimension to meet our cooling demand. The geometry of the heat exchanger is given in Table 6.2.

Figure 6.2 shows a straight continuous fin tube heat exchanger. When frost builds up, the minimum free flow area will decrease accordingly, and finally block the air flow. Comparing with straight fins, wavy fins provide additional surface area and extend the mixing length of air flow. Frosted fin efficiency η_f according to Barrow [Barrow, 1985] can be calculated as follows, here the frost thickness is assumed to be uniform over the

Table 6.2: Finned tube heat exchanger geometry

hydraulic radius: $rh = 0.51\,\text{mm}$
diameter of the circular tube, staggered array: $D = 13\,\text{mm}$
internal diameter of the circular tube: $Di = 12\,\text{mm}$
longitudinal spacing: $SL = 2.54\,\text{cm}$
transverse spacing: $ST = 2.54\,\text{cm}$
fin pitch of wavy continuous fin: $p_f = 2\,\text{mm}$
fin thickness: $t_f = 0.2\,\text{mm}$
fin length: $L = 6.2\,\text{mm}$
thermal conductivity of fin: $k_f = 238\,\text{W}\,\text{m}^{-1}\,\text{K}^{-1}$
free flow area / frontal area: $\eta = 0.448\,\text{m}^2\,\text{m}^{-2}$
heat transfer area / total volume: $\alpha = 878\,\text{m}^2\,\text{m}^{-3}$
fin area / total heat transfer area: $\eta_{fA} = 0.973\,\text{m}^2\,\text{m}^{-2}$
air side area / refrigerant side area: $\eta_A = 15.03\,\text{m}^2\,\text{m}^{-2}$
air side frontal area: $A_{frontal} = 0.29\,\text{m}^2$
depth of heat exchanger: $t_{ex} = 0.03\,\text{m}$
number of rows: $N = 3$

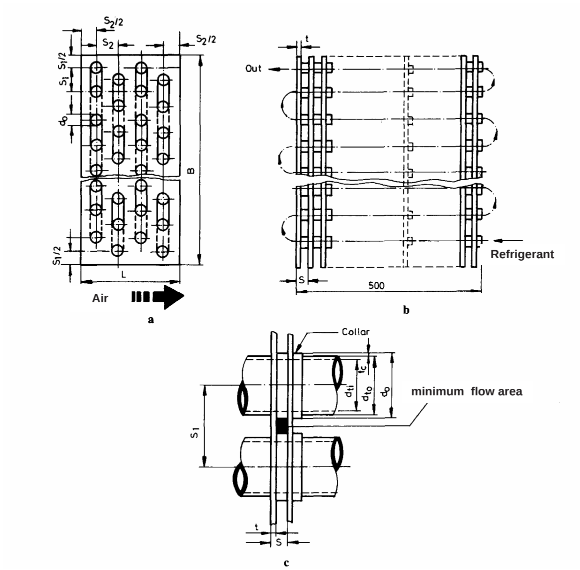

Figure 6.2: A straight continuous fin tube heat exchanger.

tube and the fin, see Figure 6.3. This equation applies to both dry and frosted conditions.

$$\eta_f = \tanh mL/(mL) \tag{8}$$

$$m = \sqrt{\frac{h_a}{k_{fr}\delta_{fr} + k_f t_f/2}} \tag{9}$$

Where L is the effective length of fin, m is a fin parameter, k is used for conductivity, h for heat transfer coefficient, δ and t for thickness, subscript a, fr and f refer to air, frost

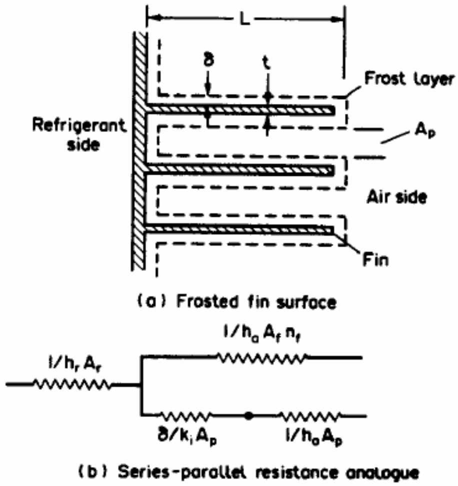

Figure 6.3: Frosted fin and its heat resistance analogue [Barrow, 1985].

and fin.

The overall heat transfer coefficient U based on the total air side area is given by:

$$\frac{1}{U} = \frac{A_a}{h_r A_r} + \left(h_a \left(\frac{A_f}{A_a} \right) \eta_f + \left(\frac{h_a k_{fr}}{k_{fr} + h_a \delta_{fr}} \right) \left(1 - \frac{A_f}{A_a} \right) \right)^{-1} \tag{10}$$

Where A is used for area, subscript r refer to refrigerant.

Evaporator fan: In refrigeration systems, axial fans or centrifugal fans are commonly used. The operating point of the fan installed in a system is established at the intersection of the fan and device curve. To achieve the desired air flow, sometimes combining fans in parallel or series is needed. The efficiency of fan normally varies with its operating point. For simple calculations, it can be assumed to be a constant. The fan curve can be simplified as a linear relation between air flow rate and pressure drop. Figure 6.4 shows the system and fan interaction.

COP (Coefficient Of Performance): The system COP and compressor power W_C can be calculated as follows [Holm et al., 1996].

$$\eta_{c,nom} = Z(C + D \cdot \dot{V}_r) \tag{11}$$

$$COP = \eta_{c,nom}\left(\frac{T_e + 273.15}{T_c - T_e}\right) \tag{12}$$

$$W_C = Q_{nom}/COP \tag{13}$$

Where Z, C, D are constants for different compressors and refrigerant, see Table 6.3. $\dot{V}_r$ is volume flow rate of refrigerant, T_e is the evaporating temperature, T_c is the condensing temperature, assumed to be a constant, at $30°C$. Where $K.02.01$ and $K.02.02$ both relate to a semi-hermetic piston compressor, with refrigerant flow rate less or more than 25 $m^3 h^{-1}$ respectively.

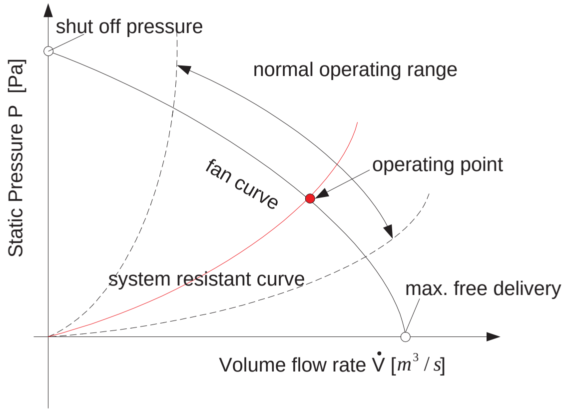

Figure 6.4: Fan and system interaction.

Table 6.3: Constant for calculation of Carnot efficiency for compressor

Category	C [−]	D [h m^{-3}]	Z [−] R22	Z [−] R134a
K.02.01	0.462	0.0031	0.997	1.048
K.02.02	0.540	0.0000	0.997	1.030

Overall calculation procedure: The overall calculation procedure includes two loops. One is the time integration loop for frost growth. Another is an internal iterative loop for finding the operating point of fan. Here we use M.02.02 an island site medium temperature display case for fresh fish products as one example, according to the requirements from authorities, the maximum storage temperature is +2°C. It uses electric defrost.

Procedure:

1. Estimate cooling demand by 2.
2. Estimate the water entrained into the system from infiltration by 1 to 6.
3. Calculate frost thickness growth rate by 7.
4. Estimate the required air flow, select evaporator fan, and simulate the fan curve by a linear relation between air flow rate and pressure drop.

$$\dot{V}_{des} = \frac{Q_{nom,sen}}{\rho_a Cp_a (T_{a,i} - T_{a,o})} \tag{14}$$

$$DP_{fan} = P_0 - \frac{P_0}{\dot{V}_0}\dot{V}_a \tag{15}$$

Where $T_{a,i}$, $T_{a,o}$ are air inlet and outlet temperature, DP_{fan} is the pressure rise of the fan, P_0 is the shut off pressure, $\dot{V}_0$ is the maximum free volume flow rate, selected according to design air volume flow rate $\dot{V}_{des}$, $\dot{V}_a$ is the air volume flow rate, ρ_a Cp_a are density and specific capacity of air respectively, $Q_{nom,sen}$ is the cooling demand sensible part.

5. LOOP: Assume an air flow rate $\dot{V}_a$ inside the cabinet.

6. Calculate the air Reynolds number Re_a.

$$G_a = \frac{\dot{V}_a \rho_a}{A_{free}} \tag{16}$$

$$Re_a = \frac{4rhG_a}{\mu_a} \tag{17}$$

Where rh is hydraulic diameter, μ_a is viscosity of air, G_a is air mass flux rate [$\mathrm{kg\,m^{-2}\,s^{-1}}$], A_{free} is air free flow area.

7. Using the correlation formulae from Blundell [Blundell, 1977], calculate the Colburn number j.

$$j = \frac{1}{10.8Re_a^{0.294}} \tag{18}$$

8. Calculate the air side pressure drop around the evaporator, and the total pressure drop inside the cabinet.

$$F_a = \frac{0.320}{Re_a^{0.336}} \tag{19}$$

$$DP_e = G_a^2 F_a (\frac{A_a}{A_{free}}) \frac{1}{2\rho_a} \tag{20}$$

$$DP_{sys} = DP_e + C_{cab} \cdot \dot{V}_a^2 \tag{21}$$

Where F_a is the dimensionless fanning friction number, DP_e is the pressure drop around evaporator, and DP_{sys} is the total pressure drop inside the cabinet, which includes also the pressure drop from bending of flow direction and varying of flow area. The latter is assumed to be proportional to the square of the flow rate.

9. Calculate the fan pressure rise by 15.

10. Does the pressure rise DP_{fan} match the system pressure drop DP_{sys}? if not, go back to step no. 5 .

11. If yes, by finding the intersection point between fan and system, determine its operating point, the actual air flow rate, and calculate fan power W_{fan}, assuming a constant fan efficiency η_{fan}.

$$W_{fan} = \frac{DP_{fan}\dot{V}_a}{\eta_{fan}} \tag{22}$$

12. Calculate the air side heat transfer coefficient h_a, using G_a, Re_a and j under the actual air flow $\dot{V}_a$.

$$St_a = \frac{j}{Pr^{2/3}} \tag{23}$$

$$h_{a,\infty} = St_a G_a Cp_a \tag{24}$$

$$h_a = h_{a,\infty}(1 - 0.72/N) \tag{25}$$

Where St_a is Stanton number, Pr is Prandtl number, $h_{a,\infty}$ is heat transfer coefficient for exchanger with an infinite number of tube rows [Kays and London, 1998], N is the number of tube rows.

13. Calculate the refrigerant side heat transfer coefficient h_r [Blundell, 1977].

$$\dot{m}_{ref} = \frac{Q_{nom}}{h_{lg,r} \cdot x} \tag{26}$$

$$Re_r = \frac{\rho_r \cdot D_i}{\mu_r} \cdot \frac{\dot{m}_{ref}}{A_{sec}} \tag{27}$$

$$K_f = \frac{h_{lg,r} \cdot x}{L_r g} \tag{28}$$

$$Nu_r = 0.0009 Re_r K_f^{0.5} \tag{29}$$

$$h_r = \frac{Nu_r k_r}{D_i} \tag{30}$$

Where r represents refrigerant, μ_r is the viscosity, $h_{lg,r}$ the specific latent heat of evaporation, x is the dimensionless vapor quality by mass, A_{sec} is the cross section area, $\dot{m}_{ref}$ is the mass flow rate, g is gravity constant, L_r is length of refrigerant path through heat exchanger, K_f is dimensionless load factor, k_r is the conductivity, Re_r and Nu_r are Reynolds and Nusselt number respectively.

14. Calculate fin and overall surface efficiencies by 8 and 9.

15. Calculate the overall heat transfer coefficient U under frosting condition, based on air side area, by 10.

16. Calculate the actual air outlet temperature $T_{a,o}$.

$$C_a = \dot{V}_a \rho_a C p_a \tag{31}$$

$$T_{a,o} = T_{a,i} - \frac{Q_{nom,sen}}{C_a} \tag{32}$$

Where C_a is the air side capacity rate.

17. Calculate the evaporating temperature T_e by the effectiveness ε- NTU method [Incropera et al., 2007], assuming the capacity rate of refrigerant is infinite (no superheat, no phase change).

$$NTU = A_a U / C_a \tag{33}$$

$$\varepsilon = 1 - exp(-NTU) \tag{34}$$

$$T_e = T_{a,i} - \frac{(T_{a,i} - T_{a,o})}{\varepsilon} \tag{35}$$

Where NTU is the Number of Transfer Units, ε is the effectiveness of evaporator.

18. Calculate COP and the corresponding compressor power W_C by 11 to 13.

19. Update frost thickness and the air free flow area A_{free} [Kays and London, 1964], [Ameen, 1993].

$$\delta_{fr} = \delta_{fr} + \int \dot{\delta}_{fr} dt \tag{36}$$

$$A_{free} = A_{free,0} - C\delta_{fr} \tag{37}$$

Where $A_{free,0}$ is the free flow area for clean fin, and C is change rate of free flow area with frost accumulation.

20. Return to step no. 5 for the next time step.

Power and extra energy consumption: In this system, we focus on two power consuming components: compressor and evaporator fan (the power consumption of the condenser fan has no direct relation with frosting, and is ignored here). Extra energy means that if we do not defrost, the efficiency of the system will degrade with frosting, in order to meet the same cooling demand, more power is needed compared with frost free conditions.

$$W_{tot}(t) = W_C(t) + W_{fan}(t) \tag{38}$$

$$W_{extra}(t) = W_{tot}(t) - W_{tot}(0) \tag{39}$$

Where 0 is the frost free time, t is the time for frost growth.

6.3.2. Direct energy use for defrosting

In order to maintain a satisfactory performance of the heat exchanger, a periodic defrost is required to remove frost. When defrosting, the cooling system is shut down, and heat is supplied to the heat exchanger to raise its temperature well above freezing. Niederer [Niederer, 1976] indicated that only 15 to 20% of the heat supplied for defrosting was actually used for removing the frost. The rest of the heat input for defrosting increased the temperature of the heat exchanger and was lost to the surrounding environment.

Typical energy distributions in a defrost cycle are:

- Energy used to warm and melt frost $E_{df,fr}$.
- Energy used to heat the coil of the heat exchanger $E_{df,coil}$.
- Energy used to heat the refrigerant $E_{df,r}$.
- Energy wasted (the defrosting efficiency).

$$m_{fr} = \int_0^t \dot{m}_{fr} dt \tag{40}$$

$$E_{df,fr}(t) = m_{fr}Cp_{fr}(0°C - T_{fr}(t)) + m_{fr}h_{fus} \tag{41}$$

$$E_{df,coil}(t) = m_{coil}Cp_{coil}(2°C - T_{coil}(t)) \tag{42}$$

$$E_{df,tot}(t) = (E_{df,fr}(t) + E_{df,coil}(t))/\eta_{df} \tag{43}$$

Where $E_{df,r}$ is ignored. Theoretically it is a non-flow, constant volume process, but in reality, a defrost cycle will normally not start until the refrigerant is fully evaporated. h_{fus} is the latent heat of fusion for water, T_{coil} is the temperature of the coil, which is assumed to be 2 K higher than the evaporating temperature, T_{fr} is the frost temperature, assumed to be equal to the coil temperature, which is a good assumption at the early stage of frost formation [Hao et al., 2005]. With frost layer increases, the air frost interface temperature will increase gradually, but the coil frost interface temperature will not change much. Here we assume that the defrost is complete when the coil temperature reaches 2°C. In the real case, defrost is terminated either by a preset time interval or a defrost stop temperature, measured by a thermometer on the surface of evaporator, with a maximum defrost time as security. η_{df} is the defrost efficiency, assumed to be 50%.

6.3.3. Food quality loss under defrosting

Food quality decay is determined by its composition factors and many environmental factors, such as temperature, relative humidity, light etc. Of all the environmental factors, temperature is the most important, since it not only strongly affects reaction rates but is also directly imposed to the food externally. The other factors are at least to some extent controlled by food packaging.

The food temperature T_{food} is mainly determined by the cabinet temperature T_{cab}. For simple calculations, we lump the food into one thermal mass, and therefore we do not model a non-uniform temperature distribution, and assume there is only convective heat transfer between air and food. Normally, the surface temperature of food is different with its central layer [Cai et al., 2006a].

$$(mCp)_{food}\frac{dT_{food}}{dt} = UA \cdot (T_{food} - T_{cab}) \tag{44}$$

For chilled food, the quality from start is defined as 100% and 0% when the product is unsuitable for sale. Food quality loss $Q_{food,loss}$ is an accumulated result, it is a function of time and temperature, and can be calculated as follows:

$$Q_{food,loss} = \int_{t_0}^{t_f} 100 \cdot D_{T,ref} \cdot \exp(\frac{T_{food} - T_{ref}}{z})dt \tag{45}$$

Where $D_{T,ref}$, T_{ref}, z are quality parameters. For fresh fish products, such as cod, they are 0.2 day^{-1}, 0°C, and 10°C respectively [Cai et al., 2006c]. UA is the heat transfer coefficient and area from air to products, mCp_{food} is the thermal mass and properties of foodstuffs. Here we assume T_{cab} is the same as $T_{a,i}$, during normal operation, T_{food} and T_{cab} are equal, $UA/(mCp)_{food} = 3.97 \cdot 10^{-4}$.

6.4. New defrost-on-demand controller design

In this section, we will introduce a new defrost-on-demand scheme with a feedback loop based on an estimate of frost layer thickness, provided by a state and parameter estimator, such as an Extended Kalman Filter.

6.4.1. New controller design

To overcome the shortcomings of the current defrost schemes, and realize an objective with a balanced system energy consumption and food quality loss, we propose a new defrost-on-demand control scheme, see Figure 6.5. It uses a feedback loop consisting of an on-line model updating and estimation by an estimator, as well as a model based optimization (see below). z_m is the measured output. It could advantageously use some extra sensors, but this scheme would also work just with the existing sensors in the system. For example, for the refrigeration system in Figure 6.1, if S_3 is the controlled temperature by the normal controller, S_4 could be a good candidate for on-line measurement. The difference between the measurement and prediction is used for on-line updating of model states and parameters. Here we assume that the store temperature T_{store} and relative humidity RH_{store} are measured, such as at every half or at every full hour, depending on the stability of store indoor conditions. $d_{fr,opt,ave}$ is the average optimal frost thickness for defrosting generated by the optimization, but defrost will only be initiated when the estimated frost thickness d_{fr} from the estimator is equal to or larger than $d_{fr,opt,ave}$. This initiating signal is sent to the normal defrost controller, defrosting starts, and it is terminated in the normal way. After the defrost is complete, a reset signal is sent to the estimator, and the process for the next defrost cycle starts again.

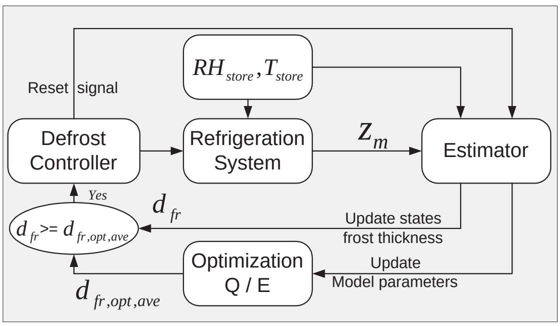

Figure 6.5: On-line new defrost-on-demand control scheme.

6.4.2. Model based optimization

The optimization objective is described as the following:

$$\min_{t_{opt}} E(t) + k \cdot Q_{food,loss}(t) \tag{46}$$

Where $E(t)$ is the system energy consumption, which includes two parts, one is the energy used direct for defrosting, another is the extra energy used for compensating the degraded system efficiency due to frost build-up. $Q_{food,loss}(t)$ is the foodstuff's quality loss, k is a weighing factor based on costs or shop owners' priorities, t_{opt} is the optimal cooling time between defrosting.

From the modeling, we know that the optimal frost thickness for defrosting $d_{fr,opt}$ is a function of store temperature T_{store}, relative humidity RH_{store}, and parameters for cabin, fan, evaporator, compressor, as well as t_{opt}, as described as follows:

$$d_{fr,opt} = f(RH_{store}, T_{store}, P_{cab}, P_{fan}, P_{evap}, P_{comp}, t_{opt}) \tag{47}$$

To determine an optimal frost thickness threshold under dynamic situations is not easy; a simple method is proposed as follows, and shown in Figure 6.6. It is approximated by the average value of optimal frost thickness under the whole working range. Simulation results under the following conditions: [20°C, 25°C] [50%, 60%] showed that by using the average frost thickness threshold to initiate defrost, the maximal energy loss compared with using the true optimal value is less than 4%. The alternative using a detailed model for determining the value given by 46 has from a control point of view a serious lack of robustness, and realistic modeling errors could easily cause larger deviations than 4%. Moreover this 4% in worst-case is insignificant relative to the potential savings demonstrated below.

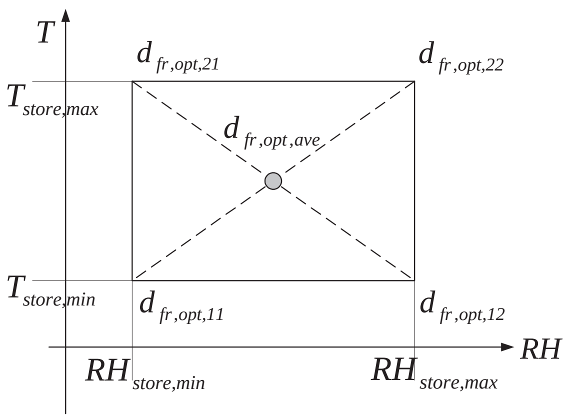

Figure 6.6: The optimal frost thickness threshold for defrosting is approximated by the average frost thickness threshold.

6.4.3. Features and advantages of the new controller

- Adaptivity: The approach suggested uses real time disturbance measurements (store temperature and humidity) for on-line model updating, estimation and optimization, continually seeking an optimal time for defrost under dynamic conditions.

- Optimality: The approach suggested is based on an optimality condition which is a weighted function between system energy consumption and food quality loss, so the resulting closed-loop system will always operate on the Pareto-optimal trade-off curve between system energy consumption and food quality loss.

- Feasibility: The proposed method is a model based control method, and introduction of an estimator can avoid some special sensors, such as fiber optic sensors, differential pressure sensors, or air velocity sensors, etc. Those sensors are normally expensive, and often some reliability and feasibility problems are associated with the complex and unreliable sensing methods. The estimator can infer those values of interest from some available measurements, such as by one or more existing thermal sensors in the system. Thus the new controller can be implemented directly on the top of existing systems, no physical rearrangements or extra components installations are required.

6.5. Simulation results

We are aiming at finding an optimal time between defrost cycles to meet our optimization objective, which is a weighted function between energy consumption and food quality loss, as 46.

Regarding the energy consumption, we need to consider two aspects: extra energy and energy for defrost.

6.5.1. Extra energy

Simulation is carried out for 11 hours under frosting conditions, with a store temperature of 25°C and RH of 55%. Figure 6.7 shows the different operating points of the fan as a function of time, due to frost build-up. When the coil is clean, the fan provides a reasonable high air flow. As pressure drop increases, the air flow is dramatically decreased. After 11 hours, the fan is already working out of its normal operating range. When the air flow rate decreases, the overall heat transfer coefficient between the air and evaporator will decrease. In order to meet the same cooling demand, the temperature drop of the air across the coil must increase. This, in turn requires a lower evaporating temperature, see Figure 6.8. The drop in the evaporating temperature will cause a lower COP and an increased power consumption. Figure 6.9 shows the compressor and fan power consumption for the same cooling demand as a function of time under frosting.

6.5.2. Defrost energy

Figure 6.10 shows the energy used to warm the coil and melt frost as a function of time between defrosting. From the figure we can see that the longer time we wait for initiating the defrost, the more energy is needed both for melting the frost and warming the coil. This is because, on one hand, frost accumulates with time. On the other hand, the coil will become colder when the evaporating temperature goes down, so it need more energy to be warmed up.

6.5.3. Food quality loss

Figure 6.11 shows one example of the daily food quality loss under different defrost frequencies.

6.5.4. Energy vs. Quality

We use one day as an example, and assume we defrost the system 2, 3... up to 6 times, then the cooling time between defrosting will be 12, 8... 4 hours (electric defrost time is ignored), this is also the time that we allow frost to grow and the system performance to degrade. We plot the daily energy consumption and daily food quality loss in Figure 6.12. From the figure, we can conclude that from an energy point of view, we should select an optimal cooling time of 5 hours. But from the food quality point of view, we should defrost at a longer interval, such as 11 hours. This is a conflicting requirement to supermarket owners. It is up to them to make the final decision, based on their priorities on quality, or cost, or a trade-off.

6.6. Potential gains by the new defrost-on-demand control scheme

The above simulation is based on one specific situation, where the store has a constant temperature and relative humidity, which is more or less true for a store with air conditioning systems, while in some European countries, such as Denmark, this is not the case. The indoor environment will normally vary with outdoor condition, staff and customers' activities. The fixed optimal cooling time which is determined off-line and configured at the commissioning phase, as conditions change, may not be the best choice any more.

Figure 6.13 shows the energy optimal cooling time under different store conditions. Generally speaking, a high store temperature and RH gives more load to the system, a faster frost growth, and a quicker performance degradation, which requires more frequent defrost. More precisely, it is the specific enthalpy and humidity ratio that determine the frost formation rate.

From Figure 6.14, focusing on the energy aspect, we can see that if we configure the defrosting of the system at an optimal time interval of 9 hours, according to one initial condition of 20°C, 50% RH, when the store temperature rises up to 25°C, same RH, this 9 hours scheme will lead to a daily energy consumption of 229.8 kJ. Compared with its actual energy optimal point of 101.5 kJ at 6 hours, 55.8% of energy is saved.

6.7. Discussion and conclusion

This paper discussed the problems related to the traditional defrost schemes. Through the analysis on both system energy and food quality, we propose a new way of determining the optimal time between defrost cycles, and a new defrost-on-demand control scheme. It on-line adjusts the cooling time between defrost cycles, according to the varying operation condition, continuously seeking an optimal time interval, featuring either an energy optimal point, or a trade-off between system energy consumption and food quality loss.

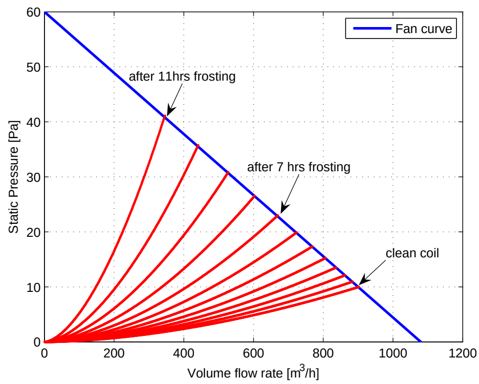

Figure 6.7: Operating point of the fan as a function of time under frost build-up.

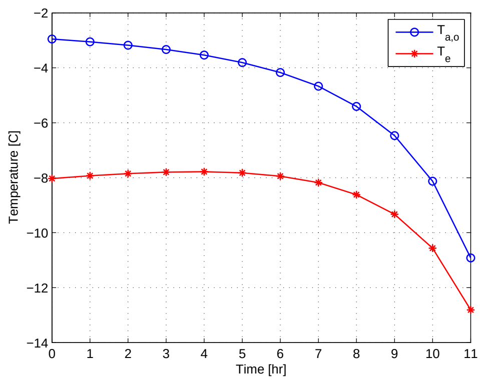

Figure 6.8: Evaporation and air outlet temperature as a function of time under frost build-up.

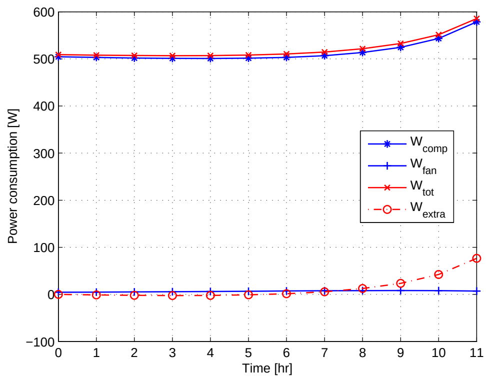

Figure 6.9: Power consumption for compressor, fan, total and extra as a function of time under frost build-up.

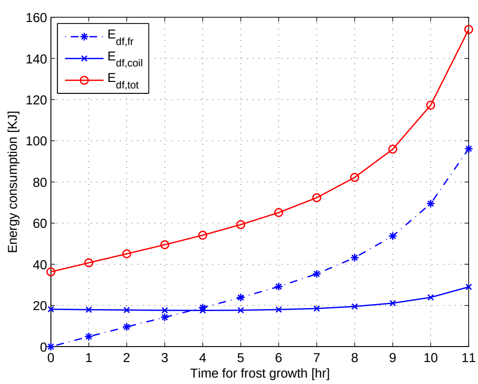

Figure 6.10: Energy consumption for warming and melting frost, warming coil and the total as a function of time under frost build-up.

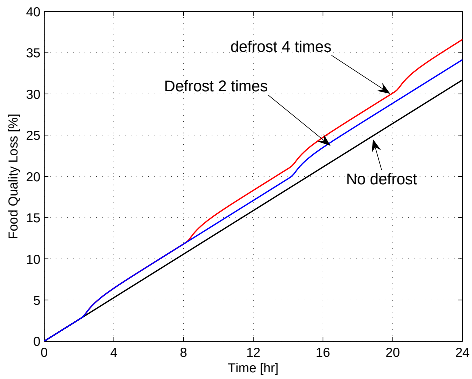

Figure 6.11: One example of food daily quality loss under different defrost frequencies.

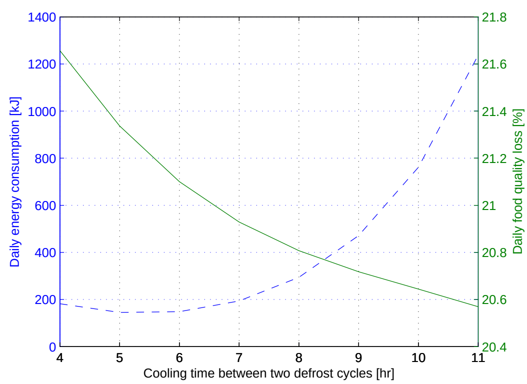

Figure 6.12: Daily energy consumption and food quality loss as a function of cooling time between defrosting.

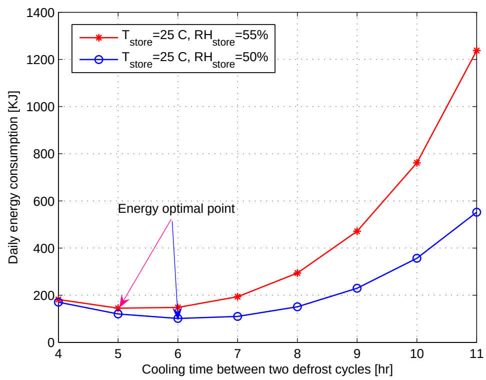

Figure 6.13: Energy optimal time under different store RH.

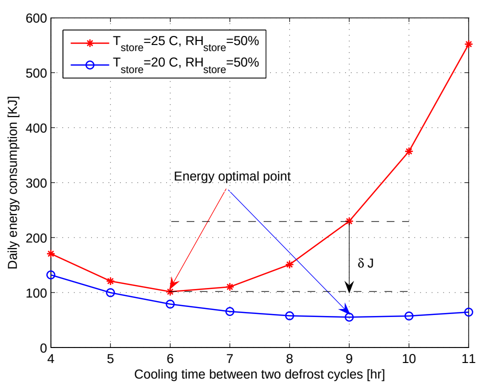

Figure 6.14: Potential gains on energy by the new defrost-on-demand control scheme.

Chapter 7

Minimizing Quality Deteriorations of Refrigerated Foodstuffs as a Side Effect of Defrosting

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Abstract

This paper proposes an optimization scheme for traditional refrigeration systems with hysteresis controllers and scheduled defrosts. It aims at minimizing the side effect of defrost cycles on the storage quality of refrigerated foodstuffs in supermarkets. By utilizing the thermal mass of air and products inside a display cabinet, this optimization scheme forces the compressor to work harder and cool down more prior to the scheduled defrosts, guaranteeing the product temperature after defrost cycles still to be within a controlled safe level.

7.1. Introduction

Frosting of evaporator coils is a well known and undesirable phenomenon. Frosts decrease the performance of heat exchangers, so defrosting need to be done regularly. Nowadays there are basically two defrost control schemes: defrost on demand and scheduled defrost. Due to the shortcoming of defrost on demand related to extra sensor installations, scheduled defrost is still the most commonly used defrost scheme in today's supermarkets.

Commercial refrigeration systems, during normal operations, air temperature inside display cabinets is normally controlled within a specific upper and lower bound by a hysteresis controller. This is sufficient to maintain the product temperature within an ideal level. During defrost, air temperature inside cabinets will rise, and so will the food temperature. Sometimes the temperature is so high that even violates the regulation from food authorities. This higher than normal temperature storage will cause an extra quality loss to food products.

Currently for commercial refrigeration systems, there are no clear and reliable measures that can prevent frost formations, see [Shin et al., 2003], so defrosts have to be done regularly. Therefore for a traditional control, this side effect of defrosts to the storage quality is unavoidable. The only way to compromise is minimizing this side effect by some optimization schemes.

This paper proposes an optimization scheme to minimize the side effect of defrost cycles to the storage quality. By utilizing the thermal mass of air and products, It forces the compressor to work harder and cool down more prior to the scheduled defrosts, thus guaranteing the food temperature after defrost cycles still to be within the controlled safe level.

This paper starts with a simple introduction on supermarket refrigeration systems, including hysteresis controllers, regulations on the storage temperature for different products, frost formations, defrost methods and control schemes, etc., which is in Section 7.2. Modeling and simulations of a hysteresis controller with scheduled defrosts, as well as food quality loss during a refrigerated storage are in Section 7.3. An optimization scheme is proposed in Section 7.4, followed by some discussions and conclusions in Section 7.5.

7.2. Commercial refrigeration system and food storage

7.2.1. Process description

A simplified sketch of the process is shown in Figure 7.1. In the evaporator, there are

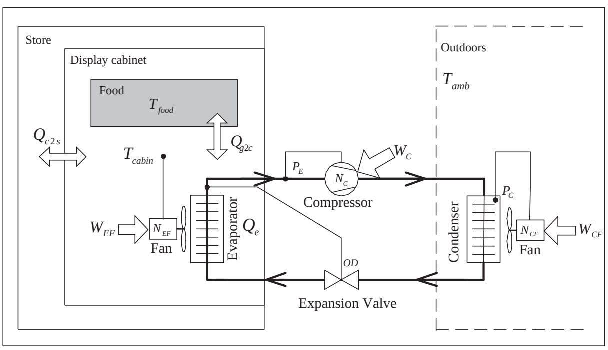


Figure 7.1: Sketch of a simplified supermarket refrigeration system.

heat exchanges between the air inside the display cabinet and cold refrigerant, giving a slightly super-heated vapor to the compressor. After compression the hot vapor is cooled, condensed and slightly sub-cooled in the condenser. This slightly sub-cooled liquid is then expanded through the expansion valve giving a cold two-phase mixture.

Display cabinets are located inside the store. Condensers and condenser fans are located on the roof of the store. Condensation is achieved by heat exchanges with ambient air.

7.2.2. Hysteresis controller

Display cabinet's temperature is normally controlled by a hysteresis controller which opens and closes the inlet valve, to control the flow of refrigerant into the evaporator, thus keeping the air temperature within a specific upper and lower bound. Figure 7.2 shows an air temperature profile for a medium temperature storage in one supermarket, where the big temperature change is caused by a defrost cycle.

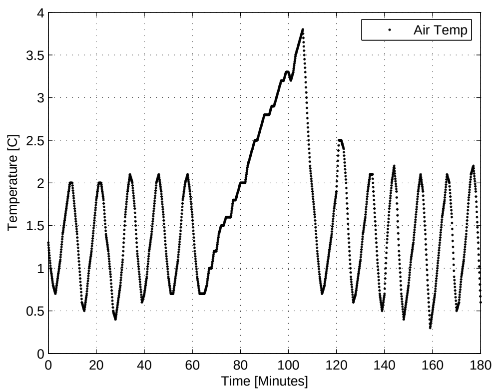


Figure 7.2: Air temperature profile for a medium temperature storage.

7.2.3. Requirements on food storage temperature

In supermarkets, there are general requirements regarding the storage temperature for different foodstuffs in display cabinets. For example in Denmark, the temperature according to [Announcement, 2004] and [DSK et al., 2004]:

- Frozen food, the maximum temperature is -18°C.
- Fresh fish product, the maximum temperature is +2°C.
- Milk, the maximum temperature is +5°C.

The temperature here is the air temperature. In addition, there are also temperature requirements during food processing and transportation.

7.2.4. Frost formation and defrost methods

Frosting of evaporator coils happens whenever the surface temperature of evaporator is below 0°C and humid air pass by. Frosts decrease the performance of the heat exchanger by decreasing the effective air flow area, and increasing the thermal resistance between the warmer air and cold refrigerant inside the evaporator. In order to maintain a satisfactory performance, evaporators need to be defrosted regularly.

The most common defrost methods for medium and low temperature applications are:

- Off-cycle defrost: It is the simplest defrost method. Refrigeration is stopped, evaporator fans continue to move room air over the frosted coil surface, warm and melt frosts.

- Hot/cool gas defrost: The normal supply of cold refrigerant is stopped. The former involves the circulation of the hot gas from the compressor discharge manifold directly to the display cabinet, and the latter utilizes cooler gas from the liquid receiver. The hot or cool gas condenses on evaporators, releases heat to melt the ice from coils.

- Electric defrost: The thermal energy to melt the ice is provided by an electric strip heater which is situated across the face of coils. During defrost, the refrigerant supply to the evaporator is switch off, the electric heater is switched on, and evaporator fans blow the warm air heated by the strip heater through coils, melting the ice from the coils surface.

However, not all methods can be used in all circumstances. Normally, for medium temperature applications, the cheapest and least energy consuming means of defrost is off-cycle. This method of defrost is widely used in industries. For low temperature applications, the use of gas defrosting is more energy efficient than electric defrost, but the considerable extra capital costs lead to a long payback period. Consequently, electric defrost is the most commonly used defrost method for low temperature applications.

7.2.5. Defrost control schemes

Nowadays there are basically two defrost control schemes.

- Defrost on demand: Initiating the defrost cycle only when it is necessary, see [Llewelyn, 1984] and [Muller, 1975]. It normally uses one parameter to initiate and terminate the defrost process, such parameters could be: air pressure difference across the evaporator, temperature differential between the air and the evaporator surface, fan power sensing, etc.

- Scheduled defrost: Initiating the defrost cycle by a timer, normally a fixed number of defrost cycles per day. Defrost is terminated either based on a fixed time or based on a temperature while with a maximum defrost time as a security.

The first scheme typically involves installations of additional sensors to detect frosts build up. However, the harsh environment, where sensors have to operate, create long-term reliability issues, that prevent its widespread adoption. So scheduled defrost is still the most commonly used defrost scheme in today's supermarkets. It is simple and uses a low cost controller.

7.3. Modeling and simulation

7.3.1. Modeling of a hysteresis controller with scheduled defrosts

The main modeling equations for a refrigeration system with a hysteresis controller are given in Table 7.1 (plus some equations correlating refrigerant properties, such as from the saturation temperature to pressure p, from pressure to enthalpy h, etc). There are two states: on and off. When on, compressor works at its full capacity; When off, compressor stops running. The pseudo code for scheduled defrosts is given in Table 7.2. There are two states: normal and defrost. When normal, the controller shifts the state between on and off; When defrost, controller is on state off. Defrost is initiated by a timer, and terminated based on a fixed time. Data for simulation is given in Table 7.3, they are approximated to simulated a real plant.

Table 7.1: Modeling of a refrigeration system with a hysteresis controller

when operating state off
mass flow of refrigerant: $m_{ref} = 0$
cooling capacity: $Q_e = 0$
compressor power: $W = 0$
heat flux condenser: $Q_c = 0$
evaporating temperature: $T_e = T_{cabin}$
condensing temperature: $T_c = T_{amb}$
switch to on, if $T_{cabin} > T_{bound,upper}$
when operating state on
$m_{ref} = V_d \cdot \eta_{vol} \cdot \rho$
$h_{ic} = (1 - f_q)/\eta_{is} \cdot (h_{is} - h_{suc}) + h_{suc}$
$Q_e = m_{ref} \cdot (h_{suc} - h_{oc})$
$W = m_{ref} \cdot (h_{is} - h_{suc})/\eta_{is}$
$Q_c = m_{ref} \cdot (h_{ic} - h_{oc})$
$0 = Q_e - UA_e \cdot (T_{cabin} - T_e)$
$0 = Q_c - UA_c \cdot (T_c - T_{amb})$
switch to off, if $T_{cabin} < T_{bound,lower}$
$Q_{loss} = UA_{s2c} \cdot (T_{store} - T_{cabin})$
$\dot{T}_{food} = (mCp_{food})^{-1} \cdot UA_{c2f} \cdot (T_{cabin} - T_{food})$
$\dot{T}_{cabin} = (mCp_{cabin})^{-1} \cdot \left(-UA_{c2f} \cdot (T_{cabin} - T_{food}) - Q_e + Q_{loss}\right)$

Table 7.2: Modeling of a refrigeration system with scheduled defrost

when normal
timer on
operating state switch between on and off
count up
switch to defrost, when $count = t_{df,ini}$
timer reset
when defrost
operating state = off
count on
switch to normal, when $count = t_{df,duration}$
timer reset

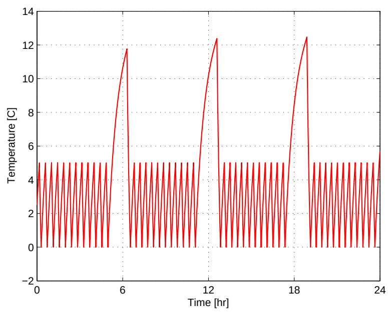
7.3.2. Simulation of a hysteresis controller with scheduled defrosts

Simulation of a hysteresis controller with scheduled defrosts are shown in Figure 7.3a. It simulates a controller which controls the air temperature to be within an upper and lower bound of 5.0°C and 0°C respectively. Defrost is initiated every 5 hours and lasts 1.3 hours. The relation between outdoor ambient temperature T_{amb}, condensing temperature

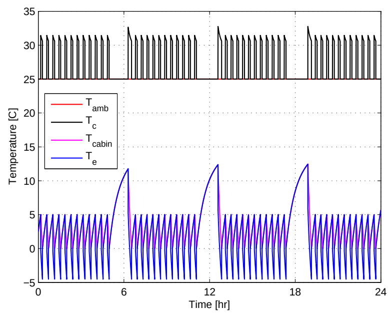
Table 7.3: Data used in the simulation

superheat: $T_{sh} = 3\,\mathrm{k}$
sub-cooling: $T_{sc} = 2\,\mathrm{k}$
volumetric capacity: $V_d = 3.3e-3\,\mathrm{m}^3\,\mathrm{s}^{-1}$
volumetric capacity fraction: $\eta_{vol} = 0.7$
heat loss factor: $f_q = 0.20$
isentropic efficiency: $\eta_{is} = 0.5$
heat transfer coefficient: $UA_{s2c} = 80\,\mathrm{W\,K^{-1}}$
heat transfer coefficient: $UA_{c2f} = 60\,\mathrm{W\,K^{-1}}$
heat transfer coefficient: $UA_c = 1000\,\mathrm{W\,K^{-1}}$
heat transfer coefficient: $UA_e = 1000\,\mathrm{W\,K^{-1}}$
heat capacity: $(mCp)_{cabin} = 300\,\mathrm{kJ\,K^{-1}}$
heat capacity: $(mCp)_{food} = 450\,\mathrm{kJ\,K^{-1}}$
quality parameter: $D_{Tref} = 0.2\,\mathrm{day^{-1}}$
quality parameter: $T_{ref} = 0\,°\mathrm{C}$
quality parameter: $Z = 10\,°\mathrm{C}$

T_c, cabinet air temperature T_{cabin}, and evaporating temperature T_e is shown in Figure 7.3b.



(a) Cabinet temperature



(b) Ambient, condensing, cabinet and evaporating temperature

Figure 7.3: Temperature profile for a medium temperature storage, with three defrost cycles.

The work of compressor is shown in Figure 7.4. After each defrost cycle, the compressor needs to work harder for a period of time, in order to bring the cabinet temperature back to its normal level.

7.3.3. Modeling of quality decay for refrigerated foodstuffs

During food processing and storage, a lot of "chemical" reactions occur. In general, chemistry "reaction kinetics" are often treated in terms of reaction rates under specified conditions. In food processing, the actual reaction rate is interesting, but only as a means of obtaining the most interesting information: the integral effect, i.e. the accumulated effect after some processing steps or storage periods with varying conditions.

Examples of "chemical" reactions are:

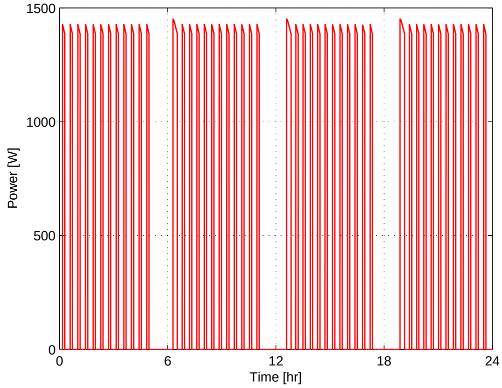


Figure 7.4: Power consumption of compressor with on-off controller and scheduled defrost.

- Loss of vitamins
- Growth of microorganisms
- Enzymatic, non-enzymatic browning
- Changes in color
- Toughness due to oxidation

Food quality decay is determined by its composition and many environmental factors, such as temperature, relative humidity, light etc. Of all the environmental factors, temperature is the most important, since it not only strongly affects reaction rates but is also directly imposed to the food externally. The other factors are at least to some extent controlled by food packaging.

The food temperature T_{food} is determined by the cabinet air temperature T_{cabin}. For simple calculations, we lump the food into one thermal mass, therefore a uniformed temperature, and assume there is only convective heat transfer between air and foodstuffs. Normally, the surface temperature of food is higher than its central layer. See details [Cai et al., 2006a].

$$(mCp)_{food}\frac{dT_{food}}{dt} = -UA_{c2f}(T_{food} - T_{cabin}) \tag{1}$$

Food quality loss $Q_{food,loss}$ can be calculated as follows:

$$Q_{food,loss} = \int_{t_0}^{t_f} 100 \cdot D_{T,ref} \exp(\frac{T_{food} - T_{ref}}{z})dt \tag{2}$$

where $D_{T,ref}$, T_{ref}, Z are quality parameter. For fresh fish product, such as cod, they are 0.2 day^{-1}, 0°C, and 10°C respectively. See details [Cai et al., 2006c].

Quality loss in % per minute for fresh lean fish product, such as cod is shown in Figure 7.5. It shows that the quality decay rate and temperature is not linear related, high temperature causes higher quality decay rate.

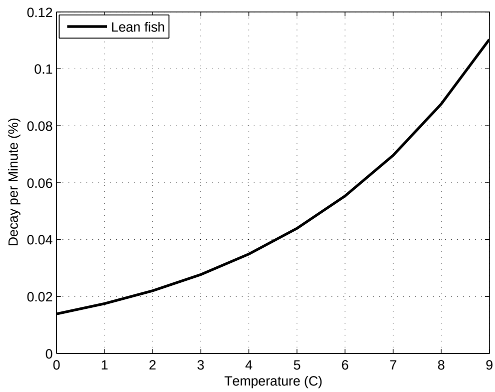
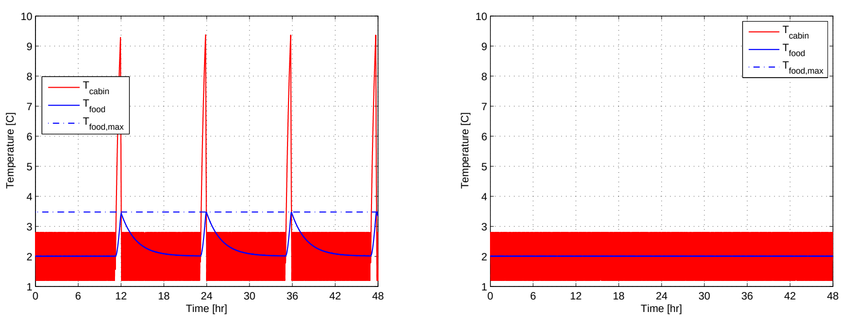


Figure 7.5: Quality decay per minute for fresh lean fish products.

7.3.4. Simulation of quality decay for refrigerated foodstuffs

Here we use one cabinet for storing and selling fresh fish product as one example. Controller controls the cabinet temperature to be within an upper and lower bound of 2.8°C and 1.2°C respectively. We assume fish product is loaded into the display case at 2.0°C. Simulation of cabinet and food temperature with normal defrost cycles is shown in Figure 7.6a. It shows that during and after the defrost cycle, food temperature will rise and stay above 2°C for a period of time. But if no defrost, product will be kept around 2.0°C, as shown in Figure 7.6b. The quality loss of product in 48 h with and without defrosts is shown in Figure 7.7. Defrost cycles cause about 3.8% extra quality loss in 2 days.



(a) Normal defrost

(b) No defrost

Figure 7.6: Air and food temperature profile.

From above simulation, we can see that defrost cycles do cause extra quality loss to the foodstuffs, and even lead the storage temperature to exceed its maximum allowable value. While nowadays, there are no cheap solutions to realize frost free for commercial refrigeration system, so defrosts have to be done in order to maintain a satisfactory performance of system. One way to compromise is optimizing the defrost.

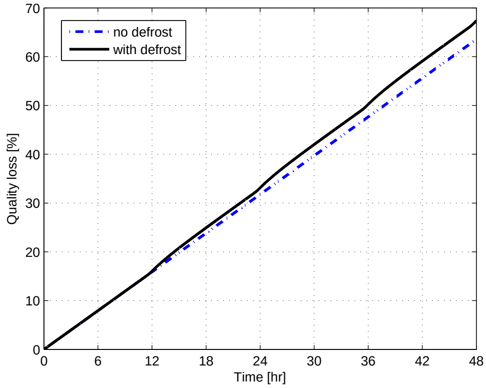


Figure 7.7: Comparison of food quality loss in % during 48 h, with and without defrost.

7.4. Optimization and results

Model of a refrigeration system with a hysteresis controller and scheduled defrosts is implemented in *gPROMS*® [gPROMS, 2007]. Optimization is done by a dynamic optimization, with the following objective function.

$$\min_{(T_{bound,lower}, T_{bound,upper})} J$$

$$\text{where} \quad J = \int_{t_0}^{t_f} W(t)dt$$

$$\text{subject to} \quad T_{food} \leq T_{food,max}$$

Here we limit the maximum temperature after defrost cycles to be 2°C for fresh fish products, and block the first 6 hours after each defrost cycle to be normal setting, and thereafter a new time invariant setting for optimization, result is shown in Figure 7.8.

A comparison of food temperature under normal defrost, and the optimization solution is shown in Figure 7.9, and comparison of quality loss in 48 h is shown in Figure 7.10.

To cool down the product in advance, in order to prevent the food temperature after defrost cycles from violating the regulation, compressor has to work harder than normal case. A comparison of energy consumption in 48 h under normal defrost, and the optimization solution is shown in Figure 7.11.

As shown in figures, in 2 days, for the optimization solution, energy consumption is 76.2 MJ, quality loss is 56.8%. For normal case, energy consumption is 71.0 MJ and quality loss is 67.3%.

7.5. Discussion and conclusion

This paper deals with the problem caused by defrost cycles in supermarket refrigeration systems. Traditional hysteresis controller with a fixed upper and lower bound ensures

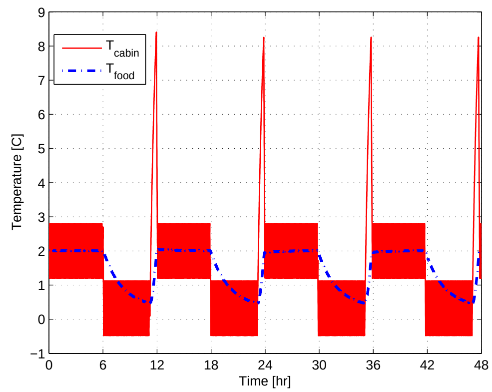


Figure 7.8: Optimization: food temperature is never allowed to exceed its maximum allowable value.

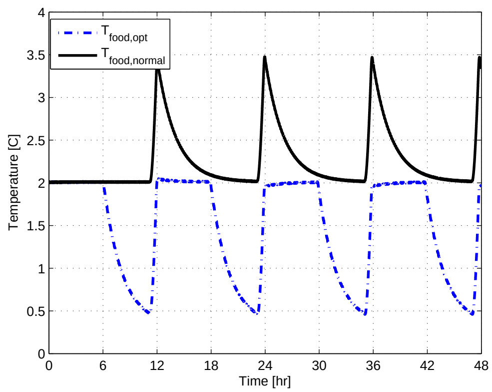


Figure 7.9: Comparison of food temperature under normal defrost and optimization.

the foodstuffs to be kept within a regulated temperature, but only in normal operations. During defrost, the high air temperature will raise the product temperature to a much higher level, and cause an extra loss to its storage quality. To solve this problem, one way is to modify the system, realizing frost free, so no defrost is needed. Another way is to modify the control of the system, minimizing the risk of defrosts. We focus on the latter.

By utilizing the thermal mass of the air and product inside cabinets, this optimization scheme forces compressor to work harder and cool down more prior to the scheduled defrosts, thus guaranteeing the product temperature after defrost cycles to be within the controlled level.

How to deal with the case related to defrost on demand, where the next defrost cycle is difficult to predict, and how to cope with the situation related to the dynamic loading of products, where the thermal capacity varies, require further investigations.

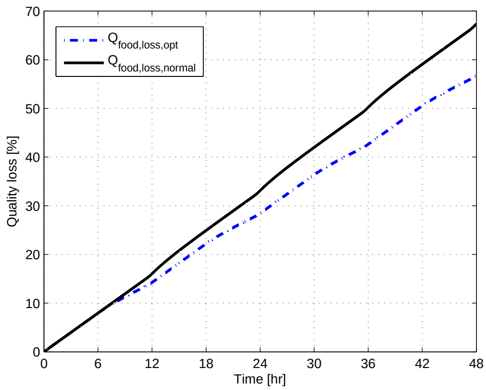


Figure 7.10: Comparison of food quality loss during 48 h under normal defrost and optimization.

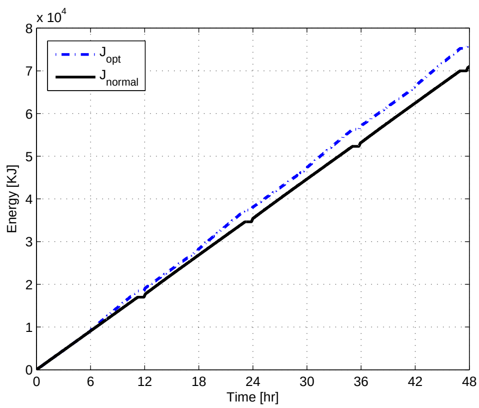


Figure 7.11: Comparison of system energy consumption during 48 h under normal defrost and optimization.

Chapter 8

On the Trade-off between Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems

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Abstract

This paper studies the trade-off between energy consumption and food quality loss, at varying ambient conditions, in supermarket refrigeration systems. Compared with the traditional operation with pressure control, a large potential for energy savings without extra loss of food quality is demonstrated. We also show that by utilizing the relatively slow dynamics of the food temperature, compared with the air temperature, we are able to further lower both the energy consumption and the peak value of power requirement. The Pareto optimal curve is found by off-line optimization.

8.1. Introduction

Increasing energy costs and consumer awareness on food products safety and quality aspects impose a big challenge to food industries, and especially to supermarkets, which have direct contacts with consumers. A well-designed optimal control scheme, continuously maintaining a commercial refrigeration system at its optimum operation condition, despite changing environmental conditions, will achieve an important performance improvement, both on energy efficiency and food quality reliability.

Many efforts on optimization of cooling systems have been focused on optimizing objective functions such as overall energy consumption, system efficiency, capacity, or wear of the individual components, see [Leducqa et al., 2006], [Swensson, 1994], [Jakobsen and Rasmussen, 1998], [Jakobsen et al., 2001], and [Larsen and Thybo, 2004]. They have proved significant improvements of system performance under disturbances, while there has been little emphasis on the quality aspect of foodstuffs inside display cabinets.

This paper discusses a dynamic optimization of commercial refrigeration systems, featuring a balanced system energy consumption and food quality loss. A former developed quality model of food provides a tool for monitoring the quality loss during the whole process, see [Cai et al., 2006c].

The paper is organized as follows: Operation and modeling of a refrigeration systems is presented in Section 8.2. In Section 8.3 we introduce the problem formulation used for optimization. Different optimization schemes and results are presented in Section 8.4. Finally some discussions and conclusions follow in Section 8.5 and Section 8.6.

8.2. Process description

A simplified sketch of the process is shown in Figure 8.1. In the evaporator there is

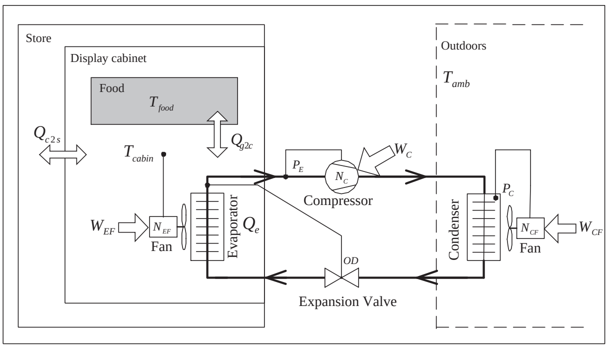


Figure 8.1: Sketch of a simplified supermarket refrigeration system.

heat exchange between the air inside the display cabinet and the cold refrigerant, giving a slightly super-heated vapor to the compressor. After compression the hot vapor is cooled, condensed and slightly sub-cooled in the condenser. This slightly sub-cooled liquid is then expanded through the expansion valve giving a cold two-phase mixture.

The display cabinet is located inside a store and we assume that the store has a constant temperature. The condenser and fans are located at the roof of the store. Condensation is achieved by heat exchange with ambient air.

8.2.1. Degree Of Freedom (DOF) analysis

There are five DOF (input) in a general simple refrigeration system. Four of these can be recognized in Figure 8.1 as the compressor speed (N_C), condenser fan speed (N_{CF}), evaporator fan speed (N_{EF}) and opening degree (OD) of the expansion valve. The fifth is related to the active charge in the system [Jensen and Skogestad, 2007].

Two of the inputs are already used for control or are otherwise constrained:

- Constant super-heating ($\Delta T_{sup} = 3\,\mathrm{K}$): This is controlled by adjusting the opening degree (OD) of the expansion valve.

- Constant sub-cooling ($\Delta T_{sub} = 2\,\mathrm{K}$): We assume that the condenser is designed to give a constant degree of sub-cooling, which by design consumes the degree of freedom related to active charge, see [Jensen and Skogestad, 2007].

So we are left with three unconstrained DOF that should be used to optimize the operation. These are:

1. Compressor speed N_C
2. Condenser fan speed N_{CF}
3. Evaporator fan speed N_{EF}

These inputs are controlling three variables:

1. Evaporating pressure P_E
2. Condensing pressure P_C
3. Cabinet temperature T_{cabin}

However, the setpoints for these three variables may be used as manipulated inputs in our study so the number of DOF is still three.

8.2.2. Mathematical model

The model equations are given in Table 8.1, see [Larsen, 2005] for the modeling of refrigeration systems. We assume that the refrigerator has fast dynamics compared with the display cabinet and food, so for the condenser, evaporator, valve and compressor we have assumed steady-state. For the display cabinet and food we use a dynamic model, as this is where the slow and important (for economics) dynamics will be. The food is lumped into one mass, and the air inside the cabinet together with walls are lumped into one mass. The main point is that there are two heat capacities in series. For the case with constant display cabinet temperature we will also have constant food temperature. There are then no dynamics and we may use steady-state optimization.

Some data for the simulations are given in Table 8.2, see [Larsen, 2005] for further data.

8.2.3. Influence of setpoints on energy consumption

As stated above, this system has three setpoints that may be manipulated: P_C, P_E and T_{cabin}. In Figure 8.2, surface shows that under two different cabinet temperatures, the variation of energy consumption with varying P_C and P_E. Point A is the optimum for cabinet temperature T_{cabin1} and point B is the optimum for T_{cabin2}. T_{cabin1} is lower than T_{cabin2}, so the energy consumption is higher in point A than in point B.

8.2.4. Influence of setpoints on food quality

Food quality decay is determined by its composition factors and many environmental factors, such as temperature, relative humidity, light etc. Of all the environmental factors, temperature is the most important, since it not only strongly affects reaction rates

Table 8.1: Model equations

Compressor
$W_C = \frac{m_{ref}\cdot(h_{is}(P_e,P_c)-h_{oe}(P_e))}{\eta_{is}}$
$h_{ic} = \frac{1-f_q}{\eta_{is}}\cdot(h_{is}(P_e,P_c)-h_{oe}(P_e))+h_{oe}(P_e)$
$m_{ref} = N_C\cdot V_d\cdot\eta_{vol}\cdot\rho_{ref}(P_e)$
Condenser
$W_{CF} = K_{1,CF}\cdot(N_{CF})^3$
$m_{air,C} = K_{2,CF}\cdot N_{CF}$
$T_{aoc} = T_c+(T_{amb}-T_c)\cdot\exp\left(-(\alpha_C\cdot m_{air,C}^{m_C})/(m_{air,C}\cdot Cp_{air})\right)$
$0 = m_{ref}\cdot(h_{ic}(P_e,P_c)-h_{oc}(P_c))-m_{air,C}\cdot Cp_{air}\cdot(T_{aoc}-T_{amb})$
Evaporator
$W_{EF} = K_{1,EF}\cdot(N_{EF})^3$
$m_{air,E} = K_{2,EF}\cdot N_{EF}$
$T_{aoe} = T_e+(T_{cabin}-T_e)\cdot\exp\left(-(\alpha_E\cdot m_{air,E}^{m_E})/(m_{air,E}\cdot Cp_{air})\right)$
$0 = Q_e-m_{air,E}\cdot Cp_{air}\cdot(T_{cabin}-T_{aoe})$
Display cabinet
$Q_{c2f} = UA_{c2f}\cdot(T_{cabin}-T_{food})$
$Q_{s2c} = UA_{s2c}\cdot(T_{store}-T_{cabin})$
$\frac{dT_{food}}{dt} = (mCp_{food})^{-1}\cdot Q_{c2f}$
$\frac{dT_{cabin}}{dt} = (mCp_{cabin})^{-1}\cdot(-Q_{c2f}-Q_E+Q_{s2c})$
$Q_{food,loss} = \int_{t_0}^{t_f}100\cdot D_{T,ref}\exp(\frac{T_{food}-T_{ref}}{z})dt$

Table 8.2: Some data used in the simulation

Refrigerator
min. pressure: $P_{E,min} = 2.0\,\text{bar}$
max. pressure: $P_{C,max} = 11.0\,\text{bar}$
max. fan speed: $N_{EF,max} = 60\,\text{s}^{-1}$
max. fan speed: $N_{CF,max} = 60\,\text{s}^{-1}$
max. compressor speed: $N_{C,max} = 60\,\text{s}^{-1}$
Display cabinet[a]
heat transfer: $UA_{s2c} = 160\,\text{W}\,\text{K}^{-1}$
heat capacity: $mCp_{cabin} = 10\,\text{kJ}\,\text{K}^{-1}$
Food
heat transfer: $UA_{c2f} = 20.0\,\text{W}\,\text{K}^{-1}$
heat capacity: $mCp_{food} = 756\,\text{kJ}\,\text{K}^{-1}$
quality parameter: $D_{T,ref} = 0.2\,\text{day}^{-1}$;
quality parameter: $T_{ref} = 0\,°\text{C}$
quality parameter: $Z = 10\,°\text{C}$

[a] Combined values for the air inside the cabinet, walls etc.

but is also directly imposed to the food externally. The other factors are at least to some extent controlled by food packaging.

Here we focus on the temperature influence to food quality Q_{food}. The only setpoint directly influencing food temperature (and thus food quality) is T_{cabin}. Figure 8.3 shows the daily quality loss for chilled cod product under four cases: T_{food} of 2°C, 1°C and T_{sin}. $T_{sin,1}$ and $T_{sin,2}$ are the sinusoidal function with mean value of 1°C, amplitude of 1°C and 3°C respectively, period is 24 h. Note that the quality loss is higher with higher temperature, but there is only minor extra loss over 24 h by using a sinusoidal temperature with small amplitude. A sinusoidal with large amplitude has a larger influence on quality due to the non-linearity of the quality function, it will not be considered here.

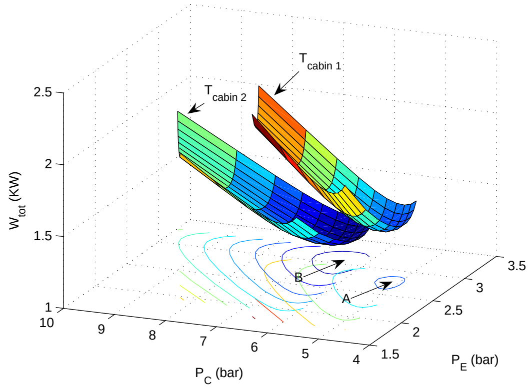


Figure 8.2: Energy consumption under different setpoints.

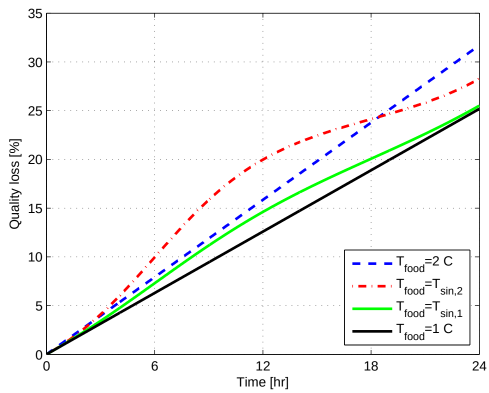


Figure 8.3: Fresh fish quality loss when stored at different temperatures.

8.3. Problem formulation

We here consider at a time horizon of three days, ambient temperature (T_{amb}) follows a sinusoidal function with a mean value of 20°C, period of 24 h and amplitude of 6°C. This is a normal temperature profile in Denmark during summer, see [DMI, 2007].

The objective is to minimize the energy consumption, subject to maintaining a fixed quality loss, by using those three DOF. This can be formulated mathematically as

$$\min_{(N_C(t), N_{CF}(t), N_{EF}(t))} J \tag{1}$$

$$\text{where} \qquad J = \int_{t_0}^{t_f} \underbrace{(W_C(t) + W_{CF}(t) + W_{EF}(t))}_{W_{tot}(t)} dt \tag{2}$$

The quality loss of the food could be included in the objective function directly, but we choose to limit it by using constraints. The optimization is also subjected to other physical and mechanical constraints, such as maximum speed of fans and compressor, minimum and maximum value of evaporator and condenser pressure etc.

In this paper, the food is a fresh cod product. Danish food authorities require it to be kept at a maximum of 2°C. The control engineer will normally set the temperature setpoint a little lower, for example at 1°C.

Case 1 Traditional operation with constant pressures (P_E), (P_C) and constant temperatures ($T_{cabin} = T_{food} = 1\,^{\circ}\mathrm{C}$)

There are usually large variations in the ambient temperature during the year so in traditional operation it is necessary to be conservative when choosing the setpoint for condenser pressure. To reduce this conservativeness it is common to use one value for summer and one for winter. We will here assume that the summer setting is used.

To get a fair comparison with traditional control, which operates at 1°C, we will illustrate our optimization by considering the following cases:

Case 2 T_{cabin} and T_{food} constant at 1°C.
Two remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption in 2.

Case 3 $\overline{T_{food}} = \frac{1}{t_f - t_0}\int_{t_0}^{t_f} T_{food}(t)dt$ = 1°C.
Three remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption in 2.

Case 4 $Q_{food,loss}(t_f) \leq 75.5\,\%$.
Three remaining unconstrained degrees of freedom as functions of time are used for minimizing the energy consumption in 2. 75.5% is the quality loss at constant temperature of 1°C obtained in cases 1 and 2.

8.4. Optimization

8.4.1. Dynamic optimization

The model is implemented in *gPROMS*® [gPROMS, 2007] and the optimization is done by dynamic optimization (except for Case 1). For the Case 2, we have used piecewise linear manipulated variables with a discretisation every hour. For the cases with varying cabinet temperature (Case 3 and 4), we have used sinusoidal functions $u = u_0 + A \cdot \sin(\pi \cdot t/24 + \phi)$, where u_0 is the nominal input, A is the amplitude of the input, t is the time and ϕ is the phase shift of the input.

Using a sinusoidal function has several advantages:

- There are much fewer variables to optimize on, only three for each input, compared with three parameters for each time interval for discrete dynamic optimization

- There are no end-effects.

In all cases we find that the phase shift is very small.

8.4.2. Optimization results

Table 8.3 compares the four cases in terms of the overall cost J, end quality loss, maximum total power ($W_{tot,\max}$) and maximum compressor power ($W_{C,\max}$). The two latter variables might be important if there are restrictions on the maximum compressor power or on the total electric power consumption.

Some key variables, including speed and energy consumption for compressor and fans as well as temperatures, are plotted for each case in Figure 8.5 and 8.6.

Table 8.3: Traditional operation and optimal operation for three different constraints

	Case 1[a]	Case 2[b]	Case 3[c]	Case 4[d]
J [MJ]	273.7	242.8	240.7	241.4
$Q_{food,loss}(t_f)$ [%]	75.5	75.5	76.1	75.5
$W_{C,\max}$ [W]	955	1022	836	879
$W_{tot,\max}$ [W]	1233	1136	946	981

[a] Traditional operation; $T_{cabin} = 1\,°\mathrm{C}$, $P_E = 2.4\,\mathrm{bar}$ and $P_C = 8.0\,\mathrm{bar}$
[b] $\overline{T_{cabin}} = 1.0\,°\mathrm{C}$
[c] $\overline{T_{food}} = 1.0\,°\mathrm{C}$
[d] $Q_{food,loss}(t_f) \leq 75.5\,\%$

For Case 1 (traditional operation) the total energy consumption over three days is 273.7 MJ. Note that the condenser temperature (and pressure) is not changing with time.

If we keep $T_{cabin} = T_{food}$ constant at 1°C, but allow the pressures (and temperatures) in the condenser and evaporator to change with time (Case 2), we may reduce the total energy consumption by 11.3% to 242.8 MJ. Figure 8.5 (right) shows that the evaporator temperature is constant, because we still control the cabinet temperature, while the condenser temperature varies with ambient temperature. The quality is the same as in Case 1 as shown in Figure 8.5 (left), because of the constant cabinet temperature. The power variations are larger, but nevertheless, the maximum total power ($W_{tot,\max}$) is reduced by 7.9% to 1,136 W.

Next, we also allow the cabinet temperature to vary, but add a constraint on the average food temperatures $\overline{T}_{food} = 1.0\,°\mathrm{C}$ (Case 3). This reduces the total energy consumption with another 0.9%, while the food quality loss is slightly higher. Note Figure 8.6 (left) shows that the evaporator, cabinet and food temperature is varying a lot.

Finally, in Case 4 we do not care about the average food temperature, but instead restrict the quality loss. With $Q_{food,loss}(t_f) \leq 75.5\,\%$, which is the same end quality we obtained for Case 1, we save 11.8% energy compared with Case 1, but use slightly more than for Case 3 (0.29%). Note Figure 8.6 (right) shows that the amplitude for food, cabinet and evaporator temperature are slightly reduced compared to Case 3.

An important conclusion is that most of the benefit in terms of energy savings is obtained by letting the setpoint for P_E and P_C vary (Case 2). The extra savings by changing also the cabinet temperature T_{cabin} (Case 3 and 4) are small. However, the peak value for compressor power and total system power is significantly decreased for Case 3 and 4. This is also very important, because a lower compressor capacity means a lower investment cost, and a lower peak value of total power consumption will further reduce the bill for supermarket owner, according to the following formula:

$$C_{op} = \int_{month}^{year} (P_{el}(t) \cdot E_{el}(t) + \max(P_{el}(t)) \cdot E_{el,dem}(t))dt \qquad (3)$$

Where C_{op} is the operating cost, E_{el} is the electricity rate, P_{el} is the electric power, $E_{el,dem}$ is the electricity demand charge, $\max(P_{el}(t))$ is the maximum electric power during one month.

8.4.3. Trade-off between energy consumption and food quality loss

Figure 8.4 plots the Pareto optimal curve between food quality loss and energy consumption. It shows that reducing quality loss and saving energy is a conflicting objective to a system. An acceptable tradeoff between these two goals can be selected by picking a point somewhere along the line. It also shows that Case 1 is far away from optimization; Case 4 is one optimal point, while Case 2 and 3 are near optimal solutions.

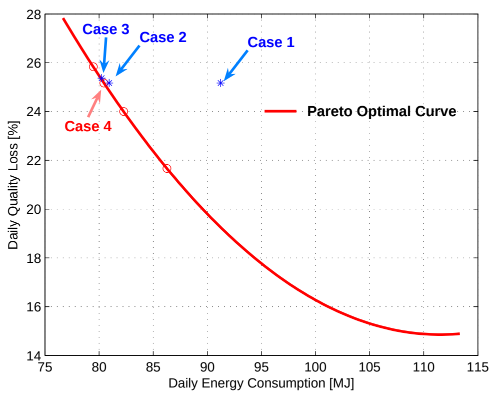


Figure 8.4: Optimization between food quality loss and system energy consumption.

8.5. Discussion

Having oscillations in the pressures will impose stress and cause wear on the equipment. This might not be desirable in many cases, but in this study the oscillations are with a period of one day, so this should not be an issue.

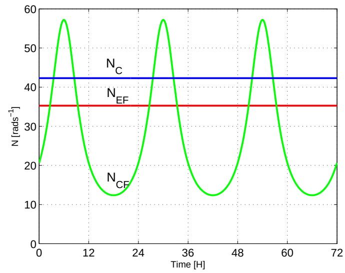
Experiments on the influence of fluctuating temperatures on food quality were reviewed by Ulrich [Ulrich, 1981], where marginal reduction in final quality due to fluctuations was reported. In our case, food temperature is only slowly varying, and with an amplitude of less than 1°C. Thus, this will not pose any negative influence on food quality.

8.6. Conclusion

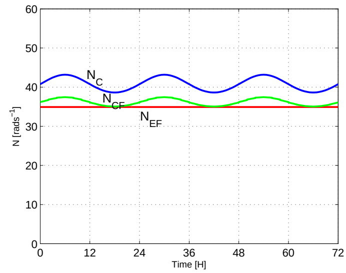
We have shown that traditional operation where the pressures are constant gives excessive energy consumption. Allowing for varying pressure in the evaporator and condenser reduces the total energy consumption by more than 11%. Varying food temperature gives only minor extra improvements in terms of energy consumption, but the peak value of the total power consumption is reduced with an additional 14% for the same food quality loss.

Reducing quality loss and saving energy is a conflicting objective. Our optimization result will help the engineer to select an acceptable tradeoff between these two goals by picking a point somewhere along the Pareto front line.

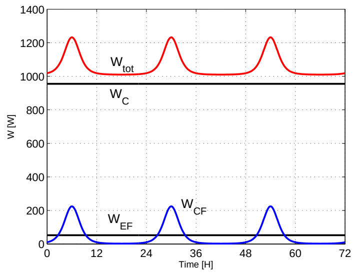
This paper investigates the potential of finding a balancing point between quality and energy consumption, by open-loop dynamic optimizations. It uses the sinusoid ambience temperature as one example. In real life, weather patterns are not exactly a sinusoidal function, but real weather conditions can be easily obtained in advance from forecast. Practical implementation, including selecting controlled variables and using closed-loop feedback control, will be the theme of future research.



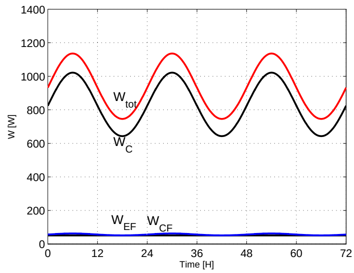
(a) Power



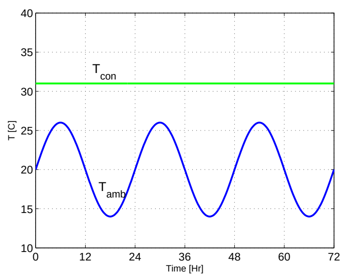
(b) Power



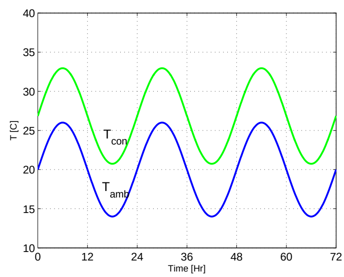
(c) Power



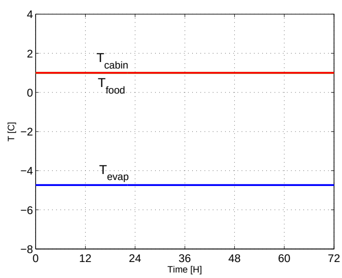
(d) Power



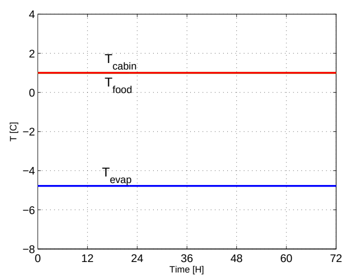
(e) High temperature



(f) High temperature

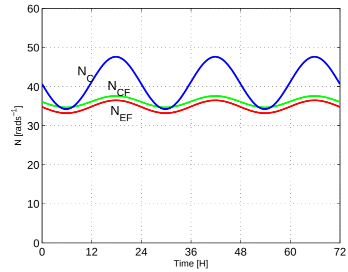


(g) Low temperature

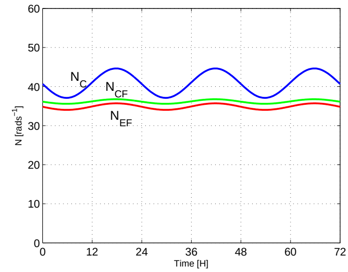


(h) Low temperature

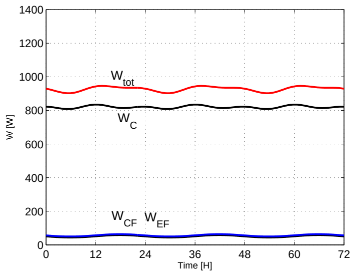
Figure 8.5: Traditional control Case 1 (left) and optimal operation Case 2 (right).



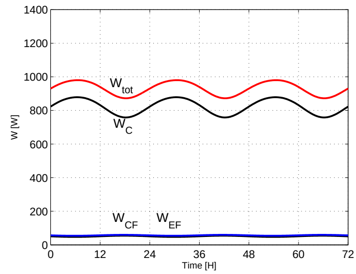
(a) Speed



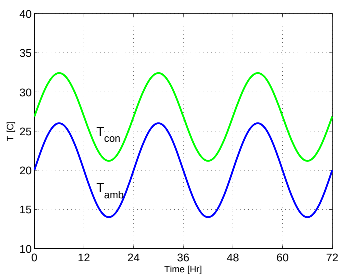
(b) Speed



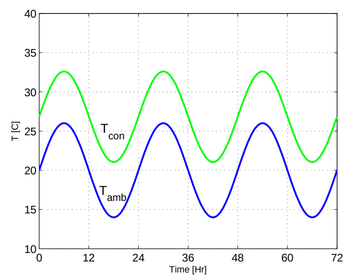
(c) Power



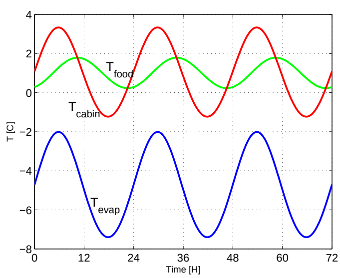
(d) Power



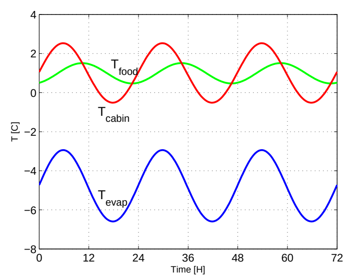
(e) High temperature



(f) High temperature



(g) Low temperature



(h) Low temperature

Figure 8.6: Optimal operation Case 3 (left) and Case 4 (right)

Chapter 9

Preventing Refrigerated Foodstuffs in Supermarkets from Being Discarded on Hot Days by MPC

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Abstract

This paper presents an optimization strategy for supermarket refrigeration systems. It deals with one special condition when the extremely high outdoor temperature causes the compressor to saturate, and work at its maximum capacity. In a traditional control, refrigerated foodstuffs inside display cabinets will suffer from a consequential higher temperature storage, which is detrimental to the food quality, and in worst cases they have to be discarded according to the regulation from authorities. This will cause a big economic loss to the shop owner. By utilizing the thermal mass of foodstuffs and their relative slow temperature change, Model Predictive Control (MPC), foreseing this situation, it will use more compressor power to cool down the foodstuffs in advance, preventing the high temperature storage from happening, thus saving them from being discarded.

9.1. Introduction

Increasing energy costs and customer awareness on food safety and quality aspects impose a big challenge to the food industries, especially to supermarkets, which have a direct contact with consumers. A well-designed optimal control scheme, continuously maintaining a commercial refrigeration system at its optimum operation condition, despite changing environmental conditions, will achieve an important performance improvement, both on energy efficiency and food quality reliability.

Many efforts on optimization of cooling systems have been focused on optimizing objective functions such as overall energy consumption, system efficiency, capacity, or wear of the individual components, see [Leducqa et al., 2006], [Swensson, 1994], [Jakobsen and Rasmussen, 1998], [Jakobsen et al., 2001]. They have proved significant improvements of system performance under disturbances, while there has been little emphasis on the quality aspect of foodstuffs inside display cabinets.

This paper discusses an optimization strategy for commercial refrigeration systems, focusing on one special condition when the extremely high ambient temperature causes the compressor to saturate, and work at its maximum capacity. In such a case, if nothing is done, the accumulated detrimental effect of high temperature storage on food may cause them to be discarded. This optimization strategy will cool the foodstuffs in advance, and prevent it from happening.

The paper is organized as follows: the refrigeration process is described in Section 9.2. Problem analysis and expected solution from MPC is illustrated in Section 9.3. MPC basic is introduced in Section 9.4. MPC formulation of our problem is presented in Section 9.5, and followed by some simulation results, which is in Section 9.6. Finally some discussions and conclusions are given in Section 9.7.

9.2. Process description

A simplified sketch of the process is shown in Figure 9.1. In the evaporator there is heat

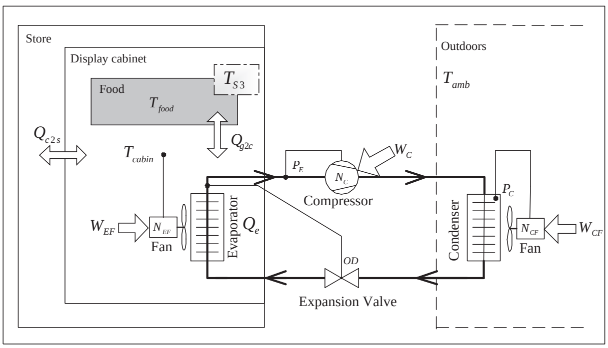

Figure 9.1: Sketch of a simplified supermarket refrigeration system.

exchange between the air inside the display cabinet and the cold refrigerant, giving a slightly super-heated vapor to the compressor. After the compression the hot vapor is cooled, condensed and slightly sub-cooled in the condenser. This slightly sub-cooled liquid is then expanded through the expansion valve giving a cold two-phase mixture.

The display cabinet is located inside a store. Condenser and condenser fans are located on the roof of the store. Condensation is achieved by the heat exchange with ambient air.

9.2.1. Mathematical model

Larsen [Larsen, 2005] provided a general introduction for modeling and parameter identifications of cooling systems. As illustrated in Figure 9.1, the supermarket refrigeration system consists of a cooling system and the display cabinet. The dynamic of the cooling system is much faster than the dynamics of the foodstuffs inside the display cabinet. Therefore we model the cooling system statically. The main modeling equations are given as follows:

$$W_C = \frac{m_{ref}}{\eta_{is}} \cdot (h_{is}(P_e, P_c) - h_{oe}(P_e)) \tag{1a}$$

$$m_{ref} = N_C \cdot V_d \cdot \eta_{vol} \cdot \rho_{ref}(P_e) \tag{1b}$$

$$Q_c = UA_c \cdot (T_c - T_{amb}) \tag{1c}$$

$$Q_e = UA_e \cdot (T_{cabin} - T_e) \tag{1d}$$

Where W is for power, m for mass flow rate, h for enthalpy, ρ for density, P for pressure, T for temperature, N for speed, Q for heat capacity, η for efficiency, UA for heat transfer coefficient, subscript C for compressor, c for condenser, e for evaporator, ref for refrigerant, amb for ambient, is for isentropic, oe for evaporator outlet, $cabin$ for cabinet, vol for volumetric.

The heat transfer coefficient are determined by the speed and parameters of the evaporator and condenser fan, as follows:

$$UA_c = \varphi_1(N_{CF}, \alpha_c, m_c, K_{CF}) \tag{2a}$$

$$UA_e = \varphi_2(N_{EF}, \alpha_e, m_e, K_{EF}) \tag{2b}$$

Table 9.1: Data for the simulation

Compressor
volumetric capacity: $V_d = 53.86\,\mathrm{cm}^3$
volumetric capacity fraction: $\eta_{vol} = 0.7$
heat loss factor: $f_q = 0.20$
isentropic efficiency: $\eta_{is} = 0.5$
Evaporator
heat transfer constant: $\alpha_e = 1,170$
mass flow constant: $K_{EF} = 0.02\,\mathrm{kg}$
heat transfer exponent: $m_e = 0.50$
fan speed: $N_{EF} = 40\,\mathrm{s}^{-1}$
minimum pressure: $P_{e,min} = 2.0\,\mathrm{bar}$
Condenser
heat transfer constant: $\alpha_c = 1,170$
mass flow constant: $K_{CF} = 0.02\,\mathrm{kg}$
heat transfer exponent: $m_c = 0.50$
fan maximum speed: $N_{CF,max} = 60\,\mathrm{s}^{-1}$
maximum pressure: $P_{c,max} = 11.0\,\mathrm{bar}$
Display cabinet
parameter: $\alpha = 0.3$
heat transfer coefficient: $UA_{s2c} = 160\,\mathrm{W\,K}^{-1}$
heat capacity: $mCp_{cabin} = 50\,\mathrm{kJ\,K}^{-1}$
heat transfer coefficient: $UA_{c2f} = 15.0\,\mathrm{W}\,K^{-1}$
heat capacity: $mCp_{food} = 400\,\mathrm{kJ\,K}^{-1}$

For the display cabinet and foodstuffs we use a dynamic model, as this is where the slow and important dynamics will be. Foodstuffs are lumped into one mass, and the air inside the cabinet together with walls are lumped into one mass, here we assume that there is only convective heat transfer between the foodstuffs and air. The modeling equations are given as follows:

$$\dot{T}_{food} = (mCp_{food})^{-1} \cdot Q_{c2f} \tag{3a}$$
$$\dot{T}_{cabin} = (mCp_{cabin})^{-1} \cdot (-Q_{c2f} - Q_e + Q_{load}) \tag{3b}$$

Where

$$Q_{c2f} = UA_{c2f} \cdot (T_{cabin} - T_{food}) \tag{4a}$$
$$Q_{load} = UA_{s2c} \cdot (T_{store} - T_{cabin}) \tag{4b}$$

Here we have to notice that for simplifying modeling, we assume that the air inside cabinets have an uniformed temperature. In a real refrigeration system, air temperature has a non-uniformed space distribution. Air after the evaporator (measured by one temperature sensor $S4$) have much lower temperature than air return back to the evaporator (measured by $S3$), this is mainly due to heat loads from infiltrations, radiations, heat conduction and convection, etc. A real controller will use either one or two these measured temperatures. Here we assume the controller will use T_{S3}, as illustrated in Figure 9.1, it can be estimated as follows.

$$T_{S3} = \alpha \cdot T_{cabin} + (1 - \alpha) \cdot T_{food} \tag{5}$$

When air and foodstuffs have the same temperature, T_{S3} will have the same temperature as them as well, but when air and food temperature is different, T_{S3} will be at a temperature in between, α can be approximated by heat transfer between two fluids, where one of them has isothermal temperature.

[Larsen, 2005] identified the parameters for the cooling system, they are given in Table 9.1. Data for thermal capacity and heat transfer coefficient inside the display cabinet are approximated to simulate a real plant.

9.2.2. Requirements on food storage temperature

In supermarkets, there are general requirements regarding the storage temperature for different foodstuffs in display cabinets. For example in Denmark, the temperature according to [Announcement, 2004] and [DSK et al., 2004]:

- Frozen food, the maximum temperature is -18°C.
- Fresh fish product, the maximum temperature is +2°C.
- Milk, the maximum temperature is +5°C.

The temperature here is the air temperature. In addition, there are also temperature requirements during food processing and transportation.

9.3. Problem analysis

9.3.1. What is the problem?

The refrigeration system in a supermarket works all year round. Normally each compressor or a group of compressors have a fixed cooling capacity, which can cope with the most common applications for that specific supermarket (here we use a system with single compressor as one example). If one day in summer, the outdoor temperature is extremely high, even when the compressor works at the maximum capacity, it still can not meet the required cooling demand. In this case, air temperature inside the display cabinet T_{cabin} will rise, and so will the food temperature T_{food}. Since the controller actually controls T_{S3}, so the compressor in this situation will work continually at the saturated condition until T_{S3} back to normal level, as illustrated in Figure 9.2. Depending on the seriousness of situation, sometimes, the stored foodstuffs have to be discarded according to the regulation from food authorities.

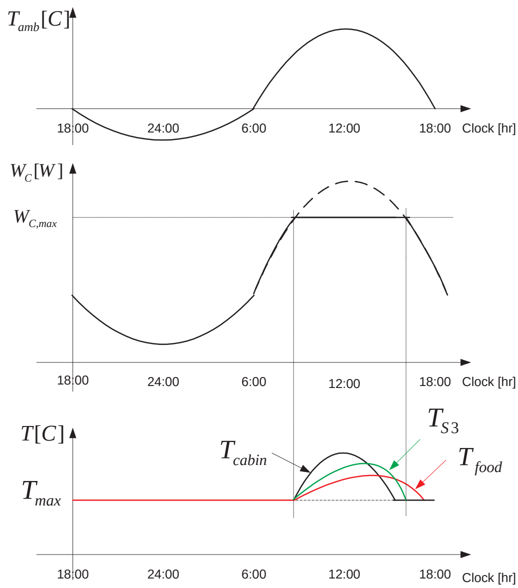

Figure 9.2: Illustration of the problem: saturation happens, T_{cabin} and T_{food} rise, compressor works at saturation until T_{S3} back to normal.

9.3.2. Why Model Predictive Control (MPC)?

In this case, we need to look for the future disturbance, that is the weather condition. Handle constraints, both from inputs (such as the mechanical limitation of components) and outputs (such as the required storage temperature). Work with nonlinearities caused by saturation. The properties of this problem determine that MPC to be one of the most suitable approaches.

MPC as one candidate, has several technical advantages, for examples, explicit process models allow control of difficult dynamics, such as: dead-time (time delay); inverse response; interactions (multivariate); nonlinearity. Optimization of future plant behavior handles, such as: feedforward from measured or estimated disturbances; feedforward from setpoint changes and desired future trajectory; feedback. Handling of input and output constraints, see [Maciejowski, 2002].

Based on the features of MPC controller, we expect the controller to take measures beforehand, when it can foresee the potential problem, in order to meet the constraints on both inputs and outputs.

9.4. Model Predictive Control

9.4.1. MPC principle and basic idea

MPC or receding horizon control (RHC) is a form of control in which the current control action is obtained by solving on line, at each sampling instant, a finite horizon open loop optimal control problem, using the current state of the plant as the initial state; the optimization yields an optimal control sequence and the first control in the sequence is applied to the plant. This is its main difference from the conventional control which uses a pre-computed law. The basic idea of MPC is illustrated in Figure 9.3.

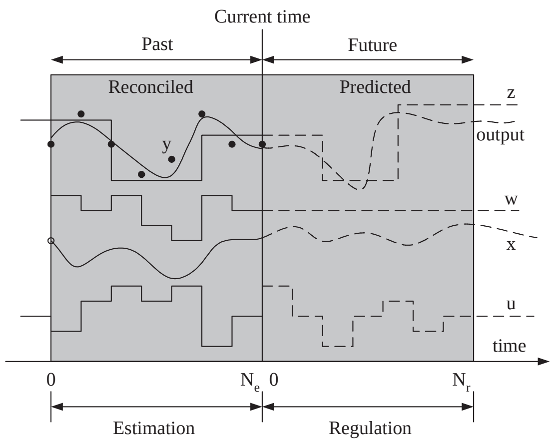

Figure 9.3: MPC basic idea- regulation and estimation problem.

9.5. Problem formulation by MPC

9.5.1. Degree Of Freedom (DOF) analysis

There are four degrees of freedom (input) in a simple refrigeration system. They can be recognized in Figure 9.1 as the compressor speed N_C, condenser fan speed N_{CF}, evaporator fan speed N_{EF} and opening degree (OD) of the expansion valve.

Here we assume a constant super-heat ($\Delta T_{sh} = 3\,^{\circ}$C), it is controlled by adjusting the opening degree of the expansion valve. This will consume one DOF.

So we are left with three unconstrained degrees of freedom that should be used to optimize the operation. They are:

1. Compressor speed N_C

2. Condenser fan speed N_{CF}

3. Evaporator fan speed N_{EF}

These inputs are controlling three variables:

1. Evaporating pressure P_e
2. Condensing pressure P_c
3. Cabinet temperature T_{cabin}

However, the setpoints for these three variables may be used as manipulated inputs in our study, so the number of degrees of freedom is still three.

9.5.2. MPC controller

To deal with the problem stated above, preventing foodstuffs from being discarded by the most energy efficient way, we design MPC as follows, and shown in Figure 9.4. Here, due to an unique relation between the saturation temperature and pressure of refrigerant, we use the setpoint of evaporating temperature T_e, condensing temperature T_c and cabinet temperature T_{cabin} as the manipulated inputs, so the total DOF is still three. According to Jakobsen and Skovrup [Jakobsen and Skovrup, 2001], there always exists one optimal temperature difference between T_c and outdoor ambient temperature T_{amb}. In most cases, it is a constant. For simplification, we fix T_c by 10°C higher than T_{amb}, this consumes one DOF. T_{cabin} is one of the controlled outputs here. Therefore we have one DOF left, that is T_e. T_{amb} and T_{store} are measured disturbances, T_{cabin}, T_{food}, T_{S3} and W_C are controlled outputs.

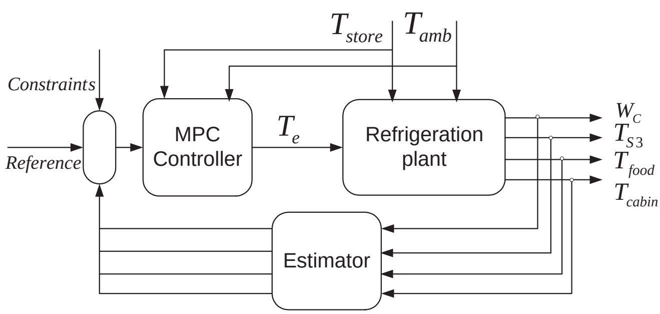

Figure 9.4: MPC controller with observer.

From 3 and 4, we get the following ODEs:

$$\dot{T}_{food} = f_1(T_{food}, T_{cabin}) \tag{6a}$$
$$\dot{T}_{cabin} = f_2(T_{cabin}, T_{food}, T_{store}, Q_e) \tag{6b}$$

Where:

$$Q_e = f_3(T_e, T_{cabin}) \tag{7}$$

Equation 7 is derived from 1d and 2b, where we assume that evaporator fan has a constant speed.

Together with initial conditions, we can rewrite 6 as the following:

$$\dot{T}_{food} = f_1(T_{food}, T_{cabin}) \tag{8a}$$
$$\dot{T}_{cabin} = f_4(T_{cabin}, T_{food}, T_{store}, T_e) \tag{8b}$$
$$T_{cabin}(t_0) = T_{cabin,i} \tag{8c}$$
$$T_{food}(t_0) = T_{food,i} \tag{8d}$$

The controlled outputs of the system are:

$$T_{food} = g_1(T_{food}, T_{cabin}) \tag{9a}$$
$$T_{cabin} = g_2(T_{food}, T_{cabin}, T_{store}, T_e) \tag{9b}$$
$$T_{S3} = g_3(T_{food}, T_{cabin}) \tag{9c}$$
$$W_c = g_4(T_e, T_{amb}, T_{store}) \tag{9d}$$

Equation 9d is non linear, derived from 1a, 1b, 1c, 2a, plus some equations correlating refrigerant properties, such as from the saturation temperature to pressure p, from pressure to enthalpy h, etc.

Linearization around one steady state equilibrium point, we can get the linear continuous state space model in deviation form as follows:

$$\dot{x} = A_c \cdot x + B_c \cdot u + E_c \cdot d \tag{10a}$$
$$y = C_y \cdot x + D_{yu} \cdot u + D_{yd} \cdot d \tag{10b}$$

Where

$$\begin{aligned}
x &= [T_{cabin} - T_{cabin,s}, T_{food} - T_{food,s}]' \\
u &= [T_e - T_{e,s}]' \\
d &= [T_{amb} - T_{amb,s}, T_{store} - T_{store,s}]' \\
y &= [T_{cabin} - T_{cabin,s}, T_{food} - T_{food,s}, T_{S3} - T_{S3,s}, W_C - W_{C,s}]'
\end{aligned}$$

9.5.3. Problem setup by using MPC toolbox in $Matlab^{TM}$

The above MPC controller is set up in $Matlab^{TM}$ by using MPC toolbox.
Constraints:

$$\begin{aligned}
u_{jmin}(i) - \varepsilon V^{u}_{jmin}(i) \leq & u_j(k+i \mid k) \leq u_{jmax}(i) + \varepsilon V^{u}_{jmax}(i) \\
\Delta u_{jmin}(i) - \varepsilon V^{\Delta u}_{jmin}(i) \leq & \Delta u_j(k+i \mid k) \leq \Delta u_{jmax}(i) + \varepsilon V^{\Delta u}_{jmax}(i) \\
y_{jmin}(i) - \varepsilon V^{y}_{jmin}(i) \leq & y_j(k+i+1 \mid k) \leq y_{jmax}(i) + \varepsilon V^{y}_{jmax}(i) \\
& \Delta u(k+h \mid k) = 0 \\
& i = 0, \ldots, p-1 \\
& h = m, \ldots, p-1 \\
& \varepsilon \geq 0
\end{aligned}$$

Where u_{min}, u_{max}, Δu_{min}, Δu_{max}, y_{min}, y_{max} are the lower and upper bound for u, Δu, y respectively, they are relaxed by introduction of the slack variable ε. Normally all the input constraints are hard, such that $V^{u}_{jmin}, V^{\Delta u}_{min}, V^{u}_{max}, V^{\Delta u}_{max} = 0$, while all output constraint constraints are soft, as hard output constraints may cause infeasibility of the optimization problem. In our case, constraint on input T_e is determined from the condition that $P_{e,min} = 2.0\,\text{bar}$ and $P_e < P_c$, T_c is between 30 and 40°C, so T_e is constrained within a lower and upper bound of -10 and 10°C. The change rate of T_e is selected to be within a lower and upper bound of 2°C. Constraints on outputs will be discussed in details later.

Cost function: the cost function with soft constraints is formulated as the following form:

$$\min_{\Delta u(k|k),\ldots,\varepsilon} \{ \sum_{i=1}^{P-1} (\sum_{j=1}^{n_y} |w^y_{i+1,j}(y_j(k+i+1 \mid k) - r_j(k+i+1))|^2 + \sum_{j=1}^{n_u} |w^{\Delta u}_{i,j} \Delta u_j(k+i \mid k)|^2) + \rho_\varepsilon \varepsilon^2 \}$$

Where w^y and $w^{\Delta u}$ are the weighting factor for the output deviations from the references and input changes respectively, the weight ρ_ε on the slack variable ε penalizes the violation of the constraints.

9.6. Simulation results

Here we use one case to illustrate the basic principle. Foods here are fresh fish products with a recommended maximum storage temperature of 2°C. We assume ambient temperature T_{amb} fluctuates during day and night as a sinusoidal function, with a nominal value of 25°C, amplitude of 5°C, period of 24 h. Furthermore, we assume that store has a constant temperature of 20°C, weather forecast is reachable 24 h in advance. There will be two scenarios:

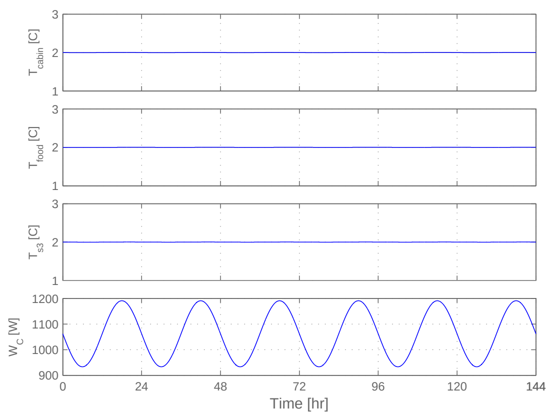

Figure 9.5: Compressor has a sufficient capacity to maintain T_{cabin}, T_{food} and T_{S3} at their setpoint of 2°C.

The compressor has a sufficient capacity: if the compressor has a sufficient capacity, it will be capable of maintaining the cabinet, food and $S3$ temperature at their setpoint,

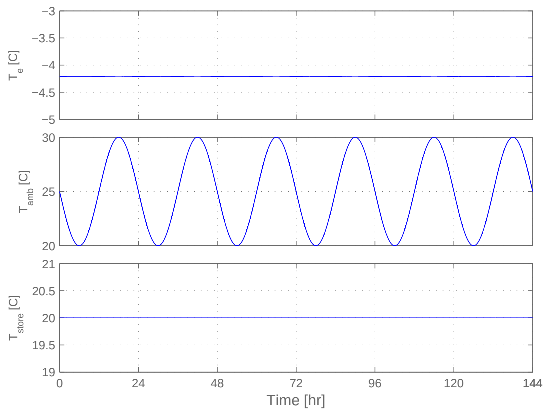

Figure 9.6: The input of controller T_e and disturbances from T_{amb} and T_{store}, when compressor has a sufficient capacity.

for example 2°C, no matter how the ambient condition changes. In this case, the compressor works hard, when T_{amb} is high. The simulated outputs are shown in Figure 9.5. The input for the controller T_e and disturbances from T_{amb} and T_{store} are shown in Figure 9.6.

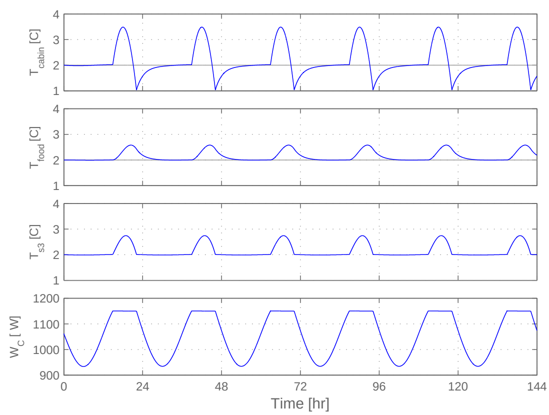

Figure 9.7: Saturation happens, T_{cabin} and T_{food} rise, compressor works at saturation until T_{S3} back to normal level.

The compressor has a limited capacity: for example that the compressor has a maximum capacity of 1,150 W. It is sufficient for most of cases, but not for the case when T_{amb} is higher than 29°C. Under this situation, compressor will work in a saturated condition, at its maximum power. It is not enough to maintain the required cabinet and food

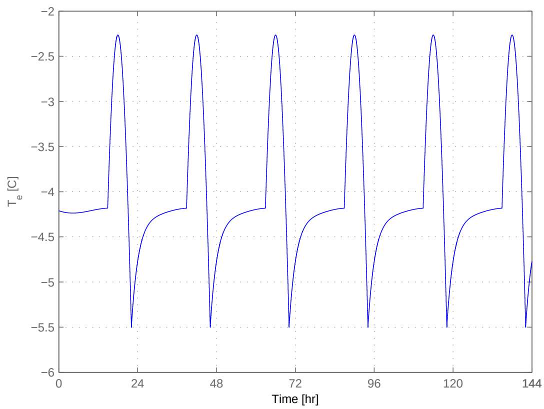

Figure 9.8: Input of controller T_e, when saturation happens.

temperature, both of them will increase accordingly. Compressor will work continually as saturated condition until T_{S3} back to normal setpoint. The simulated outputs are shown in Figure 9.7. Input T_e is shown in Figure 9.8, disturbances from T_{amb} and T_{store} are the same as in Figure 9.6 for all the cases.

As we can see from Figure 9.7, if nothing is done, foodstuffs will be stored at a temperature higher than its maximum allowable temperature. According to the relation between food temperature and quality, it is detrimental to the food quality, see [Cai et al., 2006c]. In the worst case, they have to be discarded, according to the regulation from food authorities. This will cause a big economic loss to the supermarket owner.

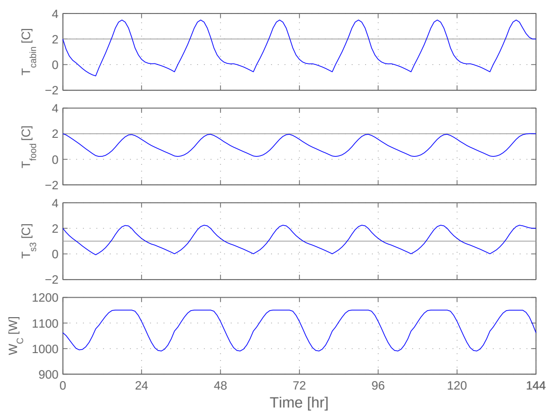

Figure 9.9: With MPC, compressor works harder beforehand to prevent T_{food} from exceeding its maximum value of 2°C.

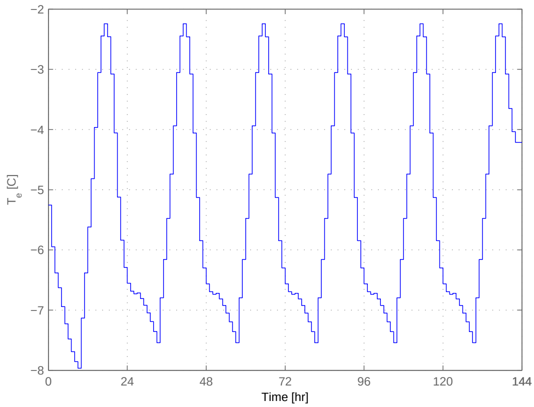

Figure 9.10: Input of controller T_e under MPC.

Optimization strategy by MPC: the strategy is to use the thermal mass of the food and their relative slow temperature change, as well as the significant advantage of MPC controller, to cool down the food beforehand, preventing the high temperature storage from happening. In this case, we have the constraint on the output W_C with a upper bound 1,150 W, at the same time, we have also the constraint on the food temperature T_{food}, with a upper bound of 2°C and a lower bound of 0°C. The reason that we set a lower bound for the food temperature is that we do not want the fresh fish to be frozen. We use the sampling time of 1 h, prediction horizon of 24 h, and controlled horizon of 12 h. References for outputs T_{cabin}, T_{food}, T_{S3} are set as their steady state values, the reference for W_C is set as 0 W, by this way, system will try to find the most energy efficient way. Weight w^y for outputs T_{cabin}, T_{food}, T_{S3}, W_C are set to be 1000, 0, 0, 1 respectively, and weight $w^{\Delta u}$ for ΔT_e is 1000. The simulated outputs for this case are shown in Figure 9.9. Input is shown in Figure 9.10.

Comparison: A comparison of the MPC optimization strategy with the cases under sufficient capacity and normal saturation is shown in Figure 9.11 and Figure 9.12. From figures, we can see that MPC forces the compressor to use much more power beforehand (red), comparing with normal saturation (green), in order to satisfy the constraint on food temperature. The inputs of the controller T_e under these three cases are shown in Figure 9.13.

9.7. Discussion and conclusion

This paper using one example, discussed the problem related with the traditional control, when the high ambient temperature causes the compressor to saturate. The accumulated effect of high temperature storage on foodstuffs will cause an extra quality loss. In the worst cases, they have to be discarded according to the regulation. To solve this problem, MPC will by utilizing the thermal mass of refrigerated foodstuffs and their relative slow temperature changes, cool down them more beforehand, preventing the high temperature storage from happening, thus saving them from being discarded.

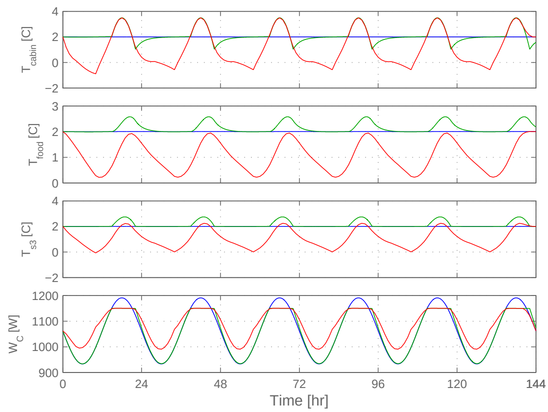

Figure 9.11: Comparison of temperature and power under sufficient capacity (blue), normal saturation (green) and MPC (red).

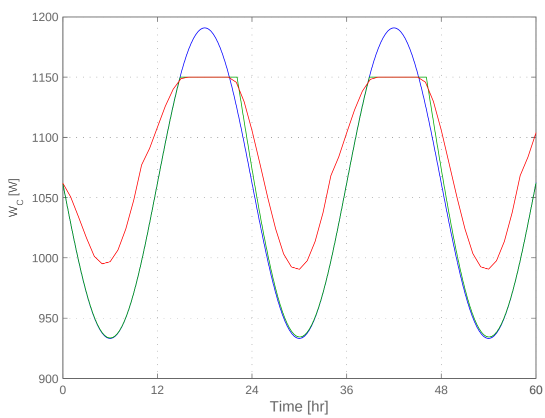

Figure 9.12: Comparison of power W_C under sufficient capacity (blue), normal saturation (green) and MPC (red)-zoomed.

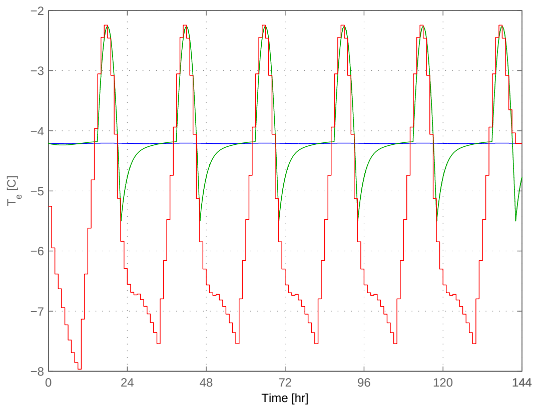

Figure 9.13: Comparison of input T_e under sufficient capacity (blue), normal saturation (green) and MPC (red).

Bibliography

[Allard and Heinzen, 1988] Allard, J. and Heinzen, R. (1988). Adaptive defrost. *IEEE Transactions on Industry Applications*, 24(1).

[Altwies and Reindl, 1999] Altwies, J. E. and Reindl, D. T. (1999). Passive thermal energy storage in refrigerated warehouses. *20th Int. Congress of Refrigeration, IIR / IIF*.

[Ameen, 1993] Ameen, F. R. (1993). Study of frosting of heat pump evaporators. *ASHRAE transactions research*.

[Andersen and Risum, 1994] Andersen, P. E. and Risum, J. (1994). Introduction to the foodstuff, bind 5, preservation (in danish). (12).

[Announcement, 2004] Announcement (2004). *Announcement 1271 concerning hygiene of foodstuffs by Danish authorities at 13, Dec. 2004, http://www.foedevarestyrelsen.dk*.

[Aparicio-Cuesta and Garcia-Moreno, 1988] Aparicio-Cuesta, M. P. and Garcia-Moreno, C. (1988). Quality of frozen cauliflower during storage. *J. Food Science*, 53(2):491– 493.

[Ashby et al., 1979] Ashby, B. H., Bennett, A. H., Bailey, W. A., Moleeratanond, W., and Kramer, A. (1979). Energy savings and quality deterioration from holding frozen foods at two daily temperature levels. *Transactions of the ASAE*, pages 938– 943.

[Aubourg et al., 2005] Aubourg, S. P., Pineiro, C., Gallardo, J. M., and Barros-Velazquez, J. (2005). Changes in texture and structure of cod and haddock fillets during frozen storage. *Food Chemistry*, 90:445– 452.

[Badii and Howell, 2002] Badii, F. F. and Howell, N. K. (2002). Changes in texture and structure of cod and haddock fillets during frozen storage. *Food hydrocolloids*, 16:313– 319.

[Barrow, 1985] Barrow, H. (1985). A note on frosting of heat pump evaporator surfaces. *Heat Recovery Systems*, 5(3):195– 201.

[Baxter, 2003] Baxter, V. D. (2003). Iea annex 26: Advanced supermarket refrigeration/heat recovery systems, final report volume 1 executive summary.

[Bhattacharyya et al., 2005] Bhattacharyya, S., Mukhopadhyay, S., Kumar, A., Khurana, R. K., and Sarkar, J. (2005). Optimization of a co/sub 2/-c/sub 3/h/sub 8/ cascade system for refrigeration and heating. *Int. J. Refrigeration*, 28(8):1284– 1292.

[Blundell, 1977] Blundell, C. J. (1977). Optimizing heat exchangers for air-to-air space-heating heat pumps in the united kingdom. *Energy Research*, 1:69– 94.

[Boggs et al., 1960] Boggs, M. M., Dietrich, W. C., Nutting, M., Olson, R. L., Lindquist, F. E., Bohart, G. S., Neumann, M. J., and Morris, M. J. (1960). Time temperature tolerance of frozen foods xxi: Frozen peas. *Food Technology*, 14:181.

[Braun, 2003] Braun, J. E. (2003). Load control using building thermal mass. *J. Solar Energy Engineering*, 125:292– 301.

[Cai et al., 2008a] Cai, J., Jensen, J. B., Skogestad, S., and Stoustrup, J. (2008a). On the trade-off between energy consumption and food quality loss in supermarket refrigeration systems. Seattle, Washington, USA. 2008 American Control Conference.

[Cai et al., 2006a] Cai, J., Risum, J., and Thybo, C. (2006a). Dynamic heat transfer model of refrigerated foodstuffs. Purdue, USA. 11th Int. Refrigeration and Air Condition Conference.

[Cai et al., 2006b] Cai, J., Risum, J., and Thybo, C. (2006b). Dynamic heat transfer model of refrigerated foodstuffs. Odense, Denmark. Danske Koeledage.

[Cai et al., 2006c] Cai, J., Risum, J., and Thybo, C. (2006c). Quality model of foodstuffs in a refrigerated display cabinet. Purdue, USA. 11th Int. Refrigeration and Air Condition Conference.

[Cai et al., 2006d] Cai, J., Risum, J., and Thybo, C. (2006d). Quality model of foodstuffs in a refrigerated display cabinet. Odense, Denmark. Danske Koeledage.

[Cai and Stoustrup, 2008] Cai, J. and Stoustrup, J. (2008). Minimizing quality deteriorations of refrigerated foodstuffs as a side effect of defrosting. Seattle, Washington, USA. 2008 American Control Conference.

[Cai et al., 2008b] Cai, J., Stoustrup, J., and Joergensen, J. B. (2008b). Preventing refrigerated foodstuffs in supermarkets from being discarded on hot days by mpc. Seoul, Korea. 17th IFAC World Congress.

[Cai et al., 2008c] Cai, J., Stoustrup, J., and Rasmussen, B. D. (2008c). An active defrost scheme for balancing energy consumption and food quality loss in supermarket refrigeration system. *Int. J. Refrigeration (submitted)*.

[Cai et al., 2008d] Cai, J., Stoustrup, J., and Rasmussen, B. D. (2008d). An active defrost scheme with a balanced energy consumption and food quality loss in supermarket refrigeration systems. Seoul, Korea. 17th IFAC World Congress.

[Chen et al., 2000] Chen, H., Thomas, L., and Besant, R. W. (2000). Modeling frost characteristics on heat exchanger fins, part i: Numerical modeling. *ASHRAE Transactions*, 106(2):358– 367.

[Chen et al., 2003] Chen, H., Thomas, L., and Besant, R. W. (2003). Fan supplied heat exchanger fin performance under frosting conditions. *Int. J. Refrigeration*, 26(1):140– 149.

[Cho and Norden, 1982] Cho, C. H. and Norden, N. (1982). Computer optimization of refrigeration systems in a textile plant: a case history. *Automatica*, 18(6):675– 683.

[Ciricillo, 1985] Ciricillo, S. F. (1985). Heat pump de-icing /controlling for energy conservation and costs. In Proceedings of the Clima 2000. Congress on Heating, ventilation and Air Conditioning.

[CMBC, 2004] CMBC (2004). http://www.cmbc.dk.

[Dalgaard, 2005] Dalgaard, P. (2005). http://www.dfu.min.dk/micro/sssp/home/home.aspx.

[Danfoss, 2006] Danfoss (2006). http://www.danfoss.com.

[DanishStandards, 2007] DanishStandards (2007). http://www.ds.dk.

[Databank, 2007] Databank (2007). *http://www.foodcomp.dk, The Danish Food Composition Databank*.

[Davey and Pham, 2006] Davey, L. M. and Pham, Q. T. (2006). A multi-layered two-dimensional finite element model to calculate dynamic product heat load and weight loss during beef chilling. *Int. J. Refrigeration*, 29:876– 888.

[de. Reinick, 1992] de. Reinick, A. C. R. (1992). Estimation of average freezing and thawing temperature versus time with infinite slab correlations. *Computers in Industry*, 19(3):297– 305.

[DMI, 2007] DMI (2007). http://www.dmi.dk.

[DSK et al., 2004] DSK, Group, D. S., and COOP (2004). *Hygiene and Self Control, Regulation for Supermarkets*.

[Esbjerg and Bruun, 2003] Esbjerg, L. and Bruun, P. (2003). Legislation, standardization, bottlenecks and market trends in relation to safe and high quality food systems and networks in denmark. SAFEACC WP 3, Assignment 2, www.globalfoodnetwork.org.

[Fahlen, 1996a] Fahlen, P. (1996a). Frosting and defrost of air coils - results from laboratory testing.

[Fahlen, 1996b] Fahlen, P. (1996b). Frosting and defrost of air coils -literature survey.

[Fikiin and Fikiin, 1999] Fikiin, K. A. and Fikiin, A. G. (1999). Predictive equation for thermophysical properties and enthalpy during cooling and freezing of food material. *J. Food Engineering*, 40:1– 6.

[Friis et al., 2004] Friis, A., Jensen, B. B. B., and Risum, J. (2004). *Food Process Engineering, An Introduction*. Biocentrum-DTU, first edition.

[Gordon et al., 1997] Gordon, J. M., Ng, K. C., and Chua, H. T. (1997). Optimizing chiller operation based on finite-time thermodynamics: universal modeling and experimental confirmation. *Int. J. Refrigeration*, 20(3):191– 200.

[Gormley et al., 2002] Gormley, R., Walshe, T., Hussey, K., and Butler, F. (2002). The effect of fluctuating vs. constant frozen storage temperature regimes on some quality parameters of selected food products. *Lebensm.-Wiss. u.-Technol*, 35:190– 200.

[Gortner et al., 1948] Gortner, W. A., Fenton, F., Volz, F. E., and Gleim, E. (1948). Effect of fluctuating storage temperatures on quality of frozen foods. *Industrial Engineering and Chemistry*, 40(8):1423– 1426.

[gPROMS, 2007] gPROMS (2007). http://www.psenterprise.com/gproms/index.html.

[Hao et al., 2005] Hao, Y. L., Iragorry, J., and Tao, Y. X. (2005). Frost-air interface characterization under natural convection. *J. Heat Transfer*, 127(10):1174– 1180.

[Henze and Krarti, 1998] Henze, G. P. and Krarti, M. (1998). Ice storage system controls for the reduction operating costs and energy. *J. Solar Energy Engineering*, 120(4):275– 281.

[Henze et al., 1997] Henze, G. P., Krarti, M., and Brandemuehl, M. J. (1997). A simulation environment for the analysis of ice storage controls. *Int. J. HVAC&R Res.*, 3:128– 148.

[Hoffenbecker et al., 2005] Hoffenbecker, N., Klein, S. A., and Reindl, D. T. (2005). Hot gas defrost model development and validation. *Int. J. Refrigeration*, 28(4):605– 615.

[Holm et al., 1996] Holm, H. V., Danig, P. O., and Rasmussen, B. D. (1996). Energy saving for remote refrigeration and frozen equipments in trading and service sector, calculations of norm energy consumption, report nr. 3 (danish).

[Howell et al., 1999] Howell, R. H., Rosario, L., Riiska, D., and Bondoc, M. (1999). Potential saving on display case energy with reduced supermarket relative humidity. Sydney, Australia. 20th Int. Congress of Refrigeration, IIR/IIF.

[Hustrulid and Winter, 1943] Hustrulid, A. and Winter, J. D. (1943). The effect of fluctuating storage temperature on frozen fruits and vegetables. *Agricultural Engineering*, 24(12):416.

[Incropera et al., 2007] Incropera, F. P., DeWitt, D. P., Bergman, T. L., and Lavine, A. S. (2007). *Fundamentals of Heat and Mass Transfer*. ISBN: 978-0-471-45728-2. John Wiley & Sons, 6th edition.

[Iragorry et al., 2004] Iragorry, J., Tao, Y. X., and Jia, S. (2004). Review article: A critical review of properties and models for frost formation analysis. *Int. J. Refrigeration*, 10(4):393– 420.

[Jakobsen and Rasmussen, 1998] Jakobsen, A. and Rasmussen, B. D. (1998). Energy-optimal speed control of fans and compressors in a refrigeration system. *Eurotherm 62*, pages 317– 323.

[Jakobsen et al., 2001] Jakobsen, A., Rasmussen, B. D., Skovrup, M. J., and Fredsted, J. (2001). Development of energy optimal capacity control in refrigeration systems. Purdue, USA. In proc.: Int. refrigeration conference.

[Jakobsen and Skovrup, 2001] Jakobsen, A. and Skovrup, M. J. (2001). Proposal of energy optimal control of condensing pressure, case report from eso project (in danish).

[Jaluria and Torrance, 1986] Jaluria, Y. and Torrance, K. E. (1986). *Computational Heat Transfer, Series in Computational Methods in Mechanics and Thermal Sciences*. Hemisphere Pub. Corp., Washington, D.C.

[Jensen and Skogestad, 2007] Jensen, J. B. and Skogestad, S. (2007). Optimal operation of simple refrigeration cycles. Part I: Degrees of freedom and optimality of sub-cooling. *Comput. Chem. Eng.*

[Kays and London, 1964] Kays, W. M. and London, A. L. (1964). *Compact Heat Exchangers*. New York: McGraw Hill Book Company, 2nd edition.

[Kays and London, 1998] Kays, W. M. and London, A. L. (1998). *Compact Heat Exchangers*. ISBN: 978-1575240602. Krieger, 3rd edition.

[Kristensen et al., 2001] Kristensen, K., Juhl, H. J., and Østergaard, P. (2001). Customer satisfaction: some results for european retailing. *Total Quality Management*, 12(7):890– 897.

[Larsen, 2005] Larsen, L. F. S. (2005). *Model based control of refrigeration system*. PhD thesis, Department of Control Engineering, Aalborg University, Denmark.

[Larsen and Thybo, 2004] Larsen, L. F. S. and Thybo, C. (2004). Potential energy saving in refrigeration system using optimal setpoint. Taipei, Taiwan. In proc.: IEEE Conference on control applications.

[Leducqa et al., 2006] Leducqa, D., Guilparta, J., and Trystramb, G. (2006). Non-linear predictive control of a vapour compression cycle. *Int. J. Refrigeration*, 29:761– 772.

[Lehman, 2006] Lehman, III, R. D. (2006). The state of energy consumption legislation for commercial reach in refrigerators and freezers and other refrigerated products. Purdue, USA. 11 Int. refrigeration and Air Conditioning conference6.

[Llewelyn, 1984] Llewelyn, D. S. (1984). A significant advance in defrost control. *Int. J. Refrigeration*, 7(5):334– 335.

[Lorentzen, 1987] Lorentzen, G. (1987). Refrigeration throughout the world. *Int. J. Refrigeration*, 10(1):6– 13.

[Maciejowski, 2002] Maciejowski, J. M. (2002). *Predictive control with constraints*. Pearson Education Limited.

[Martinez-Frias and Aceves, 1999] Martinez-Frias, J. and Aceves, S. M. (1999). Effects of evaporator frosting on the performance of an air-to-air heat pump. *J. energy resources technology*, 121(1):60– 65.

[MBE, 2007] MBE, R. H. (2007). Refrigeration and food safety. Beijing, China. Int. Congress of Refrigeration.

[Miles, 1991] Miles, C. (1991). *The Thermophysical Properties of Frozen Food. In W. Bald (Ed.), Food Freezing, Today and Tomorrow*. London: Springer-Verlag, 45-65.

[Morgan, 1981] Morgan, K. (1981). A numerical analysis of freezing and melting with convection. *Computer Methods in Applied Mechanics and Engineering*, 28(3):275–284.

[Muller, 1975] Muller, E. D. (1975). A new concept for defrosting refrigeration plants. *Kalte*, 28(2):52– 54.

[Nedjar, 2002] Nedjar, B. (2002). An enthalpy-based finite element method for non-linear heat problems involving phase change. *Computers and Structures*, 80(1):9–21.

[Niederer, 1976] Niederer, D. H. (1976). Frosting and defrosting effects on coil heat transfer. *ASHRAE Trans*, 82(1):467– 473.

[Opara and Mazaud, 2001] Opara, L. U. and Mazaud, F. (2001). Food traceability from field to plate. *Outlook on Agriculture*, 30(1):239– 247.

[Ozisik, 1994] Ozisik, M. N. (1994). *Finite Difference Methods in Heat Transfer*. Boca Raton:CRC.

[Payne and ONeal, 1992] Payne, V. and ONeal, D. L. (1992). Examination of alternate defrost strategies for an air-source heat pump: Multi-stage defrost. recent research in heat pump design, analysis, and application. *American Society of Mechanical Engineers, Advanced Energy Systems Division (Publication) AES*, (28):71– 77.

[Pham, 2005] Pham, Q. T. (2005). Modelling heat and mass transfer in frozen foods: a review. *Int. J. Refrigeration*, 29:876– 888.

[Powell, 2002] Powell, P. (2002). In search of energy efficiency. *Air Conditioning, Heating and Refrigeration News*, 217(10):17.

[Sahin, 1994] Sahin, A. Z. (1994). An experimental study on the initiation and growth of frost formation on a horizontal plate. *Expert Heat Transfer*, 7(2):101–119.

[Shiba and Bejan, 2001] Shiba, T. and Bejan, A. (2001). Thermodynamic optimization of geometric structure in the counterflow heat exchanger for an environmental control system. *Energy*, 26(5):493– 512.

[Shin et al., 2003] Shin, J., Tikhonov, A. V., and Kim, C. (2003). Experimental study on frost structure on surface with different hydrophilicity: Density and thermal conductivity. *J. Heat Transfer*, 25:84– 94.

[Stoecker, 1957] Stoecker, W. F. (1957). How frost formation on coils affects refrigeration systems. *Refrigerating Engineering*, 65(2):69– 94.

[Swensson, 1994] Swensson, M. C. (1994). *Studies on on-line optimizing control, with application to a heat pump*. PhD thesis, Department of Refrigeration Engineering, NTNU, Norway.

[Tassou et al., 2001] Tassou, S. A., Datta, D., and Marriott, D. (2001). Frost formation and defrost control parameters for open multideck refrigerated food display cabinets. *Proceedings of the Institution of Mechanical Engineers, Part A: J. Power and Energy*, 215(2):213– 222.

[Tassou and Marquand, 1987] Tassou, S. A. and Marquand, C. J. (1987). Effects of evaporator frosting and defrosting on the performance of air-to-water heat pumps. *Applied Energy*, 28:19– 33.

[Thybo and Izadi-Zamanabadi, 2004] Thybo, C. and Izadi-Zamanabadi, R. (2004). Development of fault detection and diagnosis schemes for industrial refrigeration systems - lesson learned. *CCA*.

[Toukis et al., 1991] Toukis, P. S., Fu, B., and Labuza, T. P. (1991). Time temperature indicators. *Food Technology (U.S.)*, 45:70– 82.

[Trienekens, 2006] Trienekens, J. (2006). Impacts of quality standards on food chains, comparison of three regions. Iama Symposium.

[Trienekens et al., 2005] Trienekens, J., van Plaggenhoef, W., Boschma, S., Willems, S., and Esjberg, L. (2005). Research agenda on safe and high quality international food chains. EU Concerted Action Safe, ACC publication 2005 and www.globalfoodnetwork.org, pages 26.

[Ulrich, 1981] Ulrich, R. (1981). Variations de temperature et qualite des produits surgeles. *Review Generale due Froid*, 71:371– 389.

[UNEP, 2006] UNEP (2006). 2006 report of the refrigeration, air conditioning and heat pumps technical options committee.

[Unnevehr, 2000] Unnevehr, L. J. (2000). Food safety issues and fresh food product exports from ldcs. *Agricultural Economics*, 23(3):231– 240.

[Usta and Ileri, 1999] Usta, N. and Ileri, A. (1999). Computerized economic optimization of refrigeration system design. *Energy Conversion and Management*, 40:1089–1109.

[Walker and Baxter, 2003] Walker, D. H. and Baxter, V. D. (2003). Analysis of advanced, low-charge refrigeration for supermarkets. *ASHRAE Winter Meetings CD, Technical and Symposium Papers*, pages 275– 281.

[Wong and James, 1988] Wong, A. K. H. and James, R. W. (1988). Capacity control of a refrigeration system using a variable speed compressor. *Building Services Engineering Research and Technology*, 9(2):63– 68.

[Wong and James, 1989] Wong, A. K. H. and James, R. W. (1989). Influence of control systems on the performance of refrigeration systems. *Australian Refrigeration, Air Conditioning and Heating*, 43(5).

[Woodley, 1989] Woodley, C. B. C. (1989). Saving on the defrost. *Air Conditioning Refrigeration News*, 62.

[Woodroof and Shelor, 1947] Woodroof, J. G. and Shelor, E. (1947). Effect of freezing storage on strawberries, blackberries, raspberries, and peaches. *Food Freezing*, (8):206– 209.

[Yang and Lee, 2005] Yang, D. K. and Lee, K. S. (2005). Modeling of frosting behavior on a cold plate. *Int. J. Refrigeration*, 28(3):1396– 1402.

[Yang et al., 2006] Yang, D. K., Lee, K. S., and Song, S. (2006). Modeling for predicting frosting behavior of a fin tube heat exchanger. *Int. J. Heat and Mass Transfer*, 49:1472– 1479.

[Zhou et al., 2005] Zhou, G., Krarti, M., and Henze, G. P. (2005). Parametric analysis of active and passive building thermal storage utilization. *J. Solar Energy Engineering*, 127(37).