CHALMERS
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Conditions for Design and Control of Refrigeration Systems in Fish Processing Plants

Requirements in the fish processing industry, conditions for efficient refrigeration and an energy optimisation case study using model predictive control

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Master's thesis in Sustainable Energy Systems

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Department of Civil and Environmental Engineering BOMX02-16-43
Department of Signals and Systems EX034/2016
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Master's thesis

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Abstract

Energy efficiency is today becoming more and more important due to increasing energy prices and a higher awareness of climate changes. This project focus is on the fish processing industry as it is energy intensive and has large potential for improvements. The study focus is on improving the efficiency of a specific plant where the refrigeration system was designed around 1999 and is fairly typical in the industry. The study also evaluates if thermal energy storage can be used in combination with model predictive control to take advantage of the variations in electricity prices.

The existing system is evaluated through study visits to the site and interviews with its personnel, as well as visits and comparison to other refrigeration plants in Sweden. A thorough literature study is also done to gain knowledge about refrigeration technology, control and fish processing. After this the requirements of the industry are specified and possible technological solutions are evaluated. Finally a redesigned system is created and evaluated.

The redesigned system yields savings of 40% compared to the reference plant by utilising a thermal energy storage and model predictive control to run the compressors optimally as well as constructing a cascade system instead of the former ammonia system to improve the efficiency of the compressors. The study does however not show any large savings through the use of medium size thermal energy storage to counter the variations in electricity prices.

Keywords: Energy efficiency, Thermal energy storage, Model predictive control, Refrigeration, Cascade, Compressors, Fish processing, Refrigerated storage, Freezing, Energy system

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Adam Björk and Christoffer Schou Kongstad, Gothenburg, June 2016

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Nomenclature

TES	Thermal energy storage	
MPC	Model predictive control	
PCM	Phase Change Material	
R744	Carbon Dioxide	
R717	Ammonia	
Re	Reynold number	
COP	Coefficient of Performance	
$\dot{m}$	Mass flow	$[\frac{\mathrm{kg}}{\mathrm{s}}]$
$\dot{Q}$	Heat flow	$[\frac{\mathrm{J}}{\mathrm{s}}]$
$\dot{q}$	Specific heat flow	$[\frac{\mathrm{J}}{\mathrm{s\,kg}}]$
$\dot{V}$	Volume flow	$[\frac{\mathrm{m^3}}{\mathrm{s}}]$
η	Efficiency	[-]
λ	Thermal conductivity	$[\frac{\mathrm{W}}{\mathrm{m^2\,K}}]$
ν	Kinematic viscosity	$[\frac{\mathrm{m^2}}{\mathrm{s}}]$
Θ	Weight	[-]
A	Area	$[\mathrm{m^2}]$
c_p	Specific heat capacity	$[\frac{\mathrm{J}}{\mathrm{K\,kg}}]$
D	Diameter	[m]
E	Electricity	[W]
H	Enthalpy	[J]
h	Specific enthalpy	$[\frac{\mathrm{J}}{\mathrm{kg}}]$
k	Friction coefficient	[-]
L	Length	[m]
m	Mass	[kg]
P	Power	[W]
Q	Heat	[J]
q	Specific heat	$[\frac{\mathrm{J}}{\mathrm{kg}}]$
T	Temperature in Kelvin	[K]
t	Temperature	[°C]
U	Heat transfer coefficient	$[\frac{\mathrm{W}}{\mathrm{m^2\,K}}]$
u	Control signal	[-]
V	Volume	$[\mathrm{m^3}]$
v	Velocity	$[\frac{\mathrm{m}}{\mathrm{s}}]$
$w_{compressor}$	Compressor work per kg fluid	$[\frac{\mathrm{kJ}}{\mathrm{kg}}]$
w_{pump}	Pump work per kg fluid	$[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

1

Introduction

Energy efficiency is today becoming more and more important. This can for example be seen in the goal of a 20% decrease in Sweden's energy intensity until 2020 set by the European parliament in 2009 [1]. Another such goal is the European parliament's directive of a 9% reduction in energy end use until 2016 compared to the average energy end use of 2001-2005. This shows that energy efficiency is on the political agenda. Concurrently *Bixia*'s projections of future Swedish electricity prices indicate an increased energy cost for energy intensive industries [2]. In addition to this, intermittent energy sources will increase due to goals by the *Swedish government* [3] which in turn will increase the fluctuations in electricity availability and hence also in price.

The fish processing industry is in the aspect of energy efficiency very interesting due to high energy usage. According to the Norwegian state organisation *Enova* the Norwegian fish industry used about 1.1 TW h energy during 2007 [4]. The energy use pattern is varied as fish is frozen quickly with high refrigeration load during relatively short times and then stored at very low temperatures for extended periods. This means that if there was an opportunity to use cheap energy at the high loads, large savings could be made. This could be achieved by using more electricity when it is cheap and store the energy in a low temperature thermal energy storage, and discharge it and thus reduce the electricity use when the price is higher. In addition, the thermal energy storage can be used to increase the efficiencies of the compressors. The load can be balanced so that the compressors can run at full load more or less constantly. A previous report by *Mendoza-Serrano and Chmielewski* have indicated that the implementation of model predictive control on a system with thermal energy storage can reduce the energy use with up to 39% [5].

When most fish processing plants in the Nordic region were built, prior to year 2000, energy was cheap and little attention was given to the energy use [6]. This can still be seen today as energy usage differs a lot from plant to plant, with older plants having large specific energy use, i.e. units energy per product processed. According to a report by *Enova*, the difference in energy use between an average plant, and an effective plant is almost 80% [4]. Therefore there is potential for improved energy efficiency through implementation of modern refrigeration technology such as carbon dioxide and ammonia cascade system.

1.1 Overview of a typical fish refrigeration plant

A fish refrigeration plant is often organised as in Figure 1.1. The fish is first unloaded from the fishing vessel and then transported into the factory. Thereafter the fish is washed in filtered sea water and graded, i.e. sorted into different size groups. After this the fish, which is sold fresh, is packed in boxes with ice and shipped out.

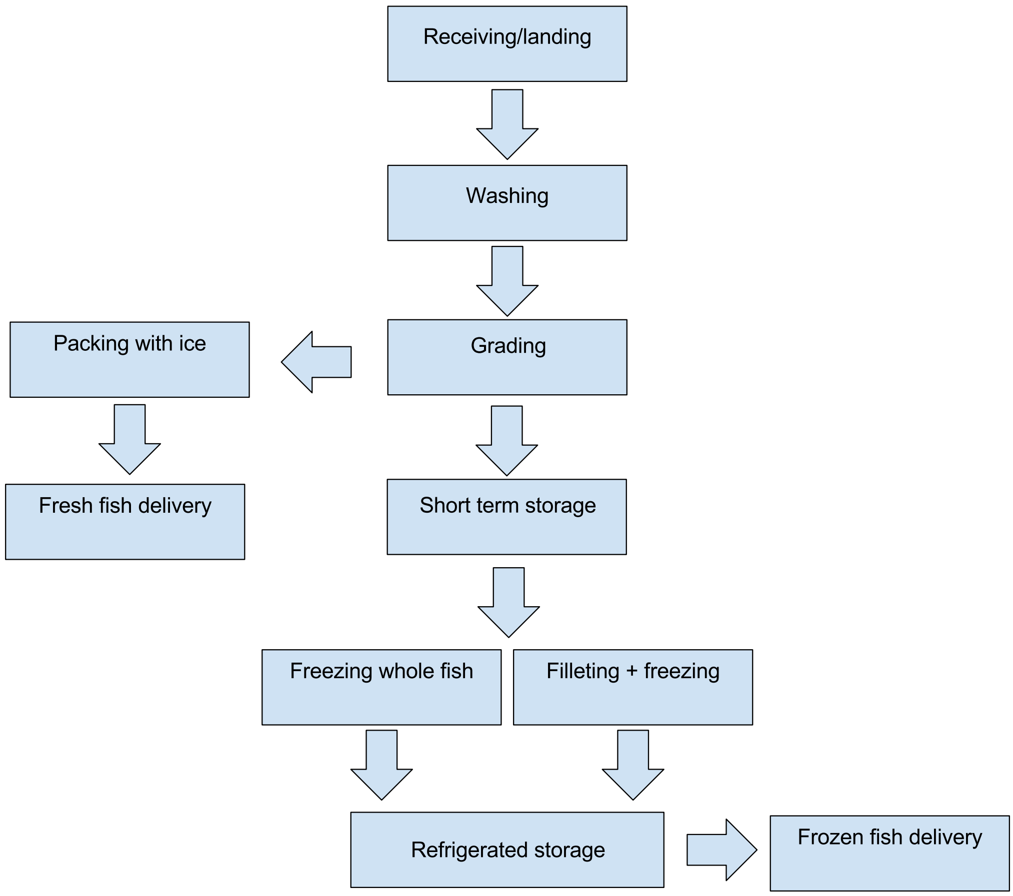

Figure 1.1: Flowchart of the processes in a typical fish refrigeration plant

The fish to be frozen is initially handled in the same way as mentioned above, but with some additional steps after the grading. As the fish often arrive in very large loads on-board fishing vessels, and usually only once or twice per day, the sorting capacity has to be much higher than the freezing capacity. This means that the fish has to be stored between the sorting and freezing in a short term storage. After this temporary storage the fish is transported to the freezing area where it is quickly frozen. The frozen fish is then transported into a refrigerated storage it is stored until delivery. In addition to freezing whole fish there is also the possibility to fillet the fish before freezing, which adds value but increases complexity of the fish handling.

1.2 Purpose and objective

The purpose of this thesis is to investigate how a fish refrigeration plant works, what requirement it has, and how improvements can be made using modern control and refrigeration technology. This thesis could be used by stakeholders considering construction of a new factory, or retrofit of existing, to increase the energy and cost efficiency of the refrigeration processes. It can also contribute to the knowledge base on how large refrigeration load facilities are affecting its local and regional energy systems. The report should give insight into what parameters, type of refrigeration machine, temperatures in the refrigeration process, and control methods will lead to an increase in the energy efficiency of a fish refrigeration plant.

The following main objectives regarding fish refrigeration plants will be investigated in this report:

- Determine the requirements of processes commonly included in a fish refrigeration plant.
- Investigate how local thermal storage can be used to optimise the energy and cost efficiency of the refrigeration system using predictive control.
- Determine the refrigeration system's components and design that should be used to minimize the energy use.

1.3 Scope

The investigation of requirements on a fish refrigeration plant is mainly based on conditions at a specific site in Sweden. This means that some aspects are only applicable for these conditions. However, conclusions and results of this study can easily be applied in other climate zones. As the topic is large and the time is limited, initially only higher system levels will be considered. In cases where data such as weather and energy prices are used, they are not measured but rather gathered from external sources.

1.4 Methodology and structure of the report

The method used in this thesis is based on a common three step engineering practice for product development: requirement formation, analysis and finally design. The design was also evaluated and analysed.

First the requirements of the fish processing industry were gathered and formulated in Chapter 2. The requirements were based on study visits at existing sites, interviews with experts in the area and a review of existing literature in the area of foodstuff processing. Second, another set of study visits were made on facilities using state-of-the-art refrigeration systems and reports on innovative refrigeration- and control technology were reviewed and summarised in Chapter 3 and 4.

In order to evaluate and combine the conditions and technologies mentioned above, the requirements had to be quantified. To get reasonable requirements, input regarding commercial aspects was taken from an existing fish refrigeration plant outside Gothenburg. The commercial aspects were then combined with legislation regarding food production and working environment mentioned in Chapter 2 to form a specific requirement document for the case.

With starting point in the requirement specification a refrigeration system was designed in Chapter 5. The different refrigeration components and system setup were chosen based on how well they fulfilled the requirements. A system controller was also designed utilising model predictive control

In Chapter 6, a simplified version of the designed system was modelled in `MATLAB`. The system was then simulated and optimised by minimising the losses and utilising the ability to thermally store energy. In addition, different scenarios were simulated by changing parameters and objectives in order to investigate how they affected the system design.

Finally a sensitivity analysis of the designed system, models and controllers was performed in the respective sections to investigate how simplifications and assumptions could have affected the validity of the conclusions.

2

Requirements of fish refrigeration plants

In this section the main requirements for the fish refrigeration industry are summarised. First primary requirements, i.e. the conditions for fish processing are investigated. Thereafter the secondary requirements concerning the working environment and sustainability aspects such as energy and water use are summarised.

2.1 Conditions for handling fish

When designing a new fish processing plant, the first step should be to determine the requirements of the system. These are mainly based on clients' request, legal obligations and physical constraints regarding the type of fish processed and its quality. This section is mainly based on the work of *Michael Hall* [7] and *L A Granata et. al.* [8].

Fish and shellfish are very perishable foodstuffs and thus the quality quickly deteriorates if they are stored under improper conditions. This can cause undesirable enzymatic action, microbial growth and finally lead to an unhealthy product with unwanted flavor and texture. The quality of the fish product is dependent on the initial quality, which cannot be exceeded, the preserving method and duration from the moment of death. The lifetime of the product can be extended by processes such as drying, salting, chilling and deep freeze. The latter method is good at preserving the texture and flavors over a long time and is therefore the focus of this section.

2.1.1 Freezing process

Freezing is the process of lowering the temperature of a material beneath its freezing point and hence phase shifting it from liquid to solid form. Fish contains approximately 75 mass-% water but dissolved salts lowers the freezing point beneath 0 °C. A typical interval of fish freezing temperatures is between -1 and -2 °C. The fish temperature will remain more or less constant at the freezing point until most of the cellular water is frozen. This will occur in the so called critical zone which is the temperature range between -1 and -5 °C. As more water is phase shifted into ice the remaining water's salt concentration is increased until all water is frozen. However, the freezing point is continuously lowered as the salt concentration is increased, and even at -25 °C there is still about 5 % water left in liquid form [7].

Table 2.1: Percentage of frozen water in fish as a function of temperature [8]

Temperature [°C]	Frozen water [%]
-0.9	0
-1.1	32
-2.2	61
-3.3	76
-4.4	83
-5.5	86
-7.8	89

The time spent in the critical zone affects the quality of the post-thawed fish and therefore it is common to group frozen fish based on the time spent in the critical zone. A fast freezing results in the creation of numerous and small ice crystals. If on the other hand the freezing process is slow, the crystals formed will be few and large. This might cause muscle cell wall disintegration resulting in liquid loss and texture alterations of the thawed fish product.

- Sharp freezing: time in critical freezing zone is greater than 2h. The fish is left on trays/shelf in a room with air temperature of -18 degrees.
- Quick freezing: time spent in critical zone is below 2h. The freezing can be done by both indirect freezing (plate freezers) or direct freezing (submerged in or sprayed by a suitable cooling liquid). The latter can lead to a high NaCl uptake.
- Ultra-rapid freezing: fish is submerged in or sprayed on by liquid nitrogen or liquid carbon dioxide.

2.1.2 Refrigerated storage

The legal required temperature when storing frozen fish is −18 °C according to regulations enacted by the European parliament *Europaparlamentets och rådets förordning* [9]. A Swedish frozen food interest group, *Föreningen Fryst och Kyld Mat* [10], instead recommends temperature between −20 °C and −25 °C. This temperature span creates a temperature reserve of 2 °C to 7 °C which is necessary to ensure the product quality after the loading of trucks and normal handling in non refrigerated areas post departure from the refrigerated storage. The method by which these temperatures should be validated and what type of temperature is regarded is not specified. However, the legal requirements state that the refrigeration storage facility should be able to keep a product at maximum −18 °C. This suggests that the air temperature should be below a dry bulb temperature of −18 °C. The impact of storage temperature on high-quality storage time for some types of seafood can be seen in Table 2.2.

Table 2.2: High-quality shelf life in months at three different temperatures for various frozen seafood [8]

Seafood	−18 °C	−25 °C	−28.9 °C
Cod	3-5	6-8	8-10
Haddock	3-5	6-8	8-10
Fatty fish	2-3	3-5	6
Lobster, crab	2		
Shrimp	6		
Clams	3-4		

The dry bulb temperature will fluctuate slightly due to changes in the refrigeration load and imperfections in the control system. Assuming a large mass of frozen product in the storage room the thermal inertia will inhibit large variation in product temperature, even when the air temperature fluctuates. According to industry guidelines the quality of an EU pallet of frozen fish product remains good for up to 30 min in air temperature of 15 °C to 25 °C [10]. This allows for relaxed air temperature requirements for short periods to avoid large over-dimensioning of the refrigeration system. This relaxation allows for increases in air temperature when defrosting and infiltration from open doors during the summer.

2.1.3 Cold storage

For short term storage before the fish is frozen, there are no strict legal limits on time or temperature. The requirement from Livsmedelsverket [11] is that the foodstuff should be frozen as quickly as possible. The recommendation that could be found was from *Föreningen Fryst och Kyld Mat* [10] where it was recommended that fresh fish was kept at ice during transport, hence holding a temperature around 0 °C. In the book *Seafood Industry : Species, Products, Processing, and Safety* [8] it is recommended that fish that is stored 24 hours after capture, without freezing, should be kept at a maximum of 4.4 °C. Storing the fish in iced water for up to 24 hours should therefore be sufficient.

2.1.4 Air purity

There is a large difference between fish processing plants with freezing processes only and plants which include fresh- and hot fish processing. In the case without refining, other than freezing, the low temperatures decreases the microbiological threat and air purity demands mainly concerns protection from outdoor pollution and leakage of refrigerant fluid. In cases where there is refining at higher temperatures the air purity is a main consideration. These are often handled using the hazard analysis critical control point (HACCP) management systems, but this is outside of the scope of this text.

2.2 Working environment

Using a simple categorisation buildings can be divided into two types. The first one is buildings which has a main purpose hosting people, e.g. office buildings. The second type is buildings with the main purpose of encapsulating a process, for example buildings with a manufacturing industry. The latter type is the case for a fish refrigeration plant. This means that there will be some considerations for the working environment but the focus will be on creating a good environment for the process. Some working environment considerations, based on the work of Enno Abel, [13] are presented below. The considerations are in some cases not binding for work in industries and can be met through working clothes rather than system design. The aspects which are especially relevant for the system design in this thesis are presented in 5.1.3.

2.2.1 Thermal climate

The indoor factors affecting a person's perception of its thermal climate is dry bulb temperature, surface temperature, humidity and movement of air. In addition to this the metabolism and clothing of people has en obvious effect on perceived temperature. These are measured in units met and clo respectively.

To simplify the concept of thermal climate a combination of radiation from surfaces and dry bulb temperature creates the operative temperature which more reflects the perceived temperature than the dry bulb temperature. The equation for the operative temperature is:

$$t_{op} = \frac{\lambda_{ic} \cdot t_a + \lambda_r \cdot t_r}{\lambda_{ic} + \lambda_r}$$

where λ_{ic} is the coefficient of surface heat transfer for convection between the skin and air, λ_r is the coefficient of surface heat transfer for radiation between surrounding surfaces and the skin, t_a is air dry bulb temperature and t_r is the mean radiant temperature in [K] derived from

$$(t_r + 273)^4 = \sum_i (T_{r,i}^4 \cdot F_i)$$

where F_i is the angle factor for each surface.

People are commonly not annoyed by air movements slower than $0.20\ \frac{\mathrm{m}}{\mathrm{s}}$, so if a comfortable operative temperature is reached, along with satisfactory level of humidity, a good thermal climate is reached. The air velocity is a standard for e.g. office buildings, and a higher air velocity can be acceptable for some working places such as in industrial buildings. The perceived climate is very individual, so there will always be some people in a group being dissatisfied. Predicted mean vote (PMV) is a questionnaire tool to assess how people perceive the thermal climate. A common goal is to adjust the operative temperature and air movement so that the PMV results in a 10 % predicted percentage dissatisfied (PPD) index. This means that 90 % is satisfied with the thermal climate. The criteria above are measured in the occupied zone, i.e. the parts people normally use.

In refrigerated storage the product requirement of a low temperature is the main concern and people will have to adapt through use of protective clothing and by adjusting the time spent in the facility. The Swedish work environment organisation *Arbetsmiljöverket* [14] have legal requirements, *Arbetarskyddsstyrelsens författningssamling (AFS)*, for short term work e.g. loading and delivery in facilities with temperature below 16 °C. They state that all personal shall use protective clothing including gloves. Furthermore it states that other protective equipment should be used if necessary.

For other low temperature facilities such as freezing room, where people spend more time, there are further requirements. The air movement caused by temperature differences should be kept at a low level to avoid bothersome draught. This is achieved by insulation and necessary air tightness around doors, gates and between the freezing room, refrigerated storage and the building's outside respectively. The transport opening between the refrigerated storage and freezing room should be arranged so that unnecessary draught is avoided and an airlock should be used if deemed necessary. Refrigeration and ventilation equipment should be arranged so that annoying draught is avoided in the occupied zone.

The freezing room in a fish refrigeration plant, and other production facilities, are allowed to keep an air temperature below 16 °C if the temperature requirement by *Livsmedelsverket* [11] on the activity is a lower temperature. However, the air temperature should be as high and close to 16 °C as is practically possible. Furthermore there is a requirement on a relative humidity below 60 %, if not special circumstances apply. A room with a temperature of at least 20 °C and seating, for work pauses, should be available in proximity to the freezing room. If proper thermal protective clothing is used over eight hours working time is allowed in freezing rooms as calculated by *Swedish Standard Organisation* [12]. The same standard have a requirement of around 30 min to recuperate after work in a freezing room.

2.2.2 Air quality

Air quality is the degree to which the air is free from pollutants that can harm the occupants of a room. There are two types of pollutants: particulate- and gaseous pollutant. Technically it is relatively simple to filter out particulate matter but much more difficult filtering gases. *Arbetsmiljöverket* legal requirements on facilities handling gases are summarised below [15]. For facilities with higher temperature processes the air quality with regard to bacterial growth is an important consideration. This is however not included in this study.

All owners of activities where gaseous substance are handled are obligated to do a risk assessment. This should be repeated every time new equipment are implemented or if the operating conditions are changed. With starting point in these assessments appropriate measures should be adopted to ensure a secure and safe facility. The ventilation system must be designed to enable oxygen volume concentrations over 20 % and below 22 %.

In connection to the place where gases are handled there should be equipment installed for measurement of gas concentrations, wind direction and equipment for setting of the gas- and fire alarm.

Swedish Standard Institute's (SIS) documents SS-EN 378-1, -2, -3, -4 present further security and environmental requirements for cooling and refrigeration systems. A synopsis of *Samon safe monitor* [16], [17] summary of the above mentioned document regarding ammonia and carbon dioxide that are important for fish refrigeration plants are presented below.

Ammonia

A requirement of detection system for ammonia gases is included in engine rooms and other rooms where people can be harmed of gas leaks for system exceeding 50 kg ammonia [16].

Some guideline levels for the detection system for different ammonia concentrations are listed below.

- Non-emergency C-alarm (pre-alarm) directed to maintenance personal at 50-300 PPM.
- Emergency B-alarm (operation alarm) to maintenance personal, flashing light and possibly siren at 500-1000 PPM.
- Emergency A-alarm has the same procedure as B-alarm adding alarm signal to emergency services and stopping of the refrigeration system completely at concentrations over 3000 PPM.
- The detection system and alarms should have a battery back-up to sustain function during 60 minutes in case of power outage.

Carbon dioxide

Carbon dioxide is non-toxic but it can be dangerous since it forces oxygen away. Carbon dioxide may either form a layer beneath the air, due to its relatively high density, or mix with the air. If a layer is formed and it reaches the breathing area it can cause asphyxiation. In case it mixes with the air the oxygen concentration is decreased and this may eventually also cause asphyxiation. For systems with more than 25 kg carbon dioxide a detection system must be installed. The guidelines for detection and alarm system for refrigeration system using carbon dioxide are presented below.

- Non-emergency C-alarm (pre-alarm) directed to maintenance personal is not suitable since up to 1000 PPM is acceptable air quality.

- Emergency B-alarm (operation alarm) to maintenance personal, flashing light and possibly siren at 2000 PPM.
- Emergency A-alarm has the same procedure as B-alarm adding a complete stop of the refrigeration system at concentrations exceeding 5000 PPM.
- The detection system and alarms should have a battery back-up to sustain function during 60 minutes in case of power outage.

3

An introduction to refrigeration and control

In this chapter a brief overview of the most common type of refrigeration system and the concept of control is introduced. The chapter is mainly aimed at people lacking the knowledge in one or both of the disciplines to better understand the following chapters.

3.1 An introduction to refrigeration

A refrigeration system consist of a number of components and the most important of these in terms of energy efficiency are described in this section. The section about vapour compression cycle is based on basic thermodynamic literature [18] and the sections on different system components and consideration is mainly based on *ASHRAE refrigeration* [19].

3.1.1 Vapour-compression cycle

A basic compression cycle consists of four stages and can be seen in Figure 3.1. In the first stage the working fluid, e.g. ammonia, in high pressure and liquid form travels from the condenser through an expansion valve. The pressure is lower after the valve and thus the ammonia temperature is decreased. In the second stage the ammonia is transferred to the evaporator. Its temperature is higher than the ammonia and heat is thus transferred from the evaporator walls to the ammonia. The heat transferred raises the ammonia's temperature to its boiling point and exceeds the heat of vapourisation. The ammonia thus phase changes to gas form and the evaporator is cooled. The ammonia is then transferred to the next stage where its pressure is increased through a compressor by adding work. The pressure increase leads to a temperature raise. In the fourth stage the high pressure, high temperature ammonia is phase changed back to liquid in a condenser. Heat is transferred from the hot ammonia to the lower temperature condenser walls. Hence, heat is transferred from the environment of the evaporator to that of the condenser. The efficiency of a vapour compression refrigeration system is often measured by a

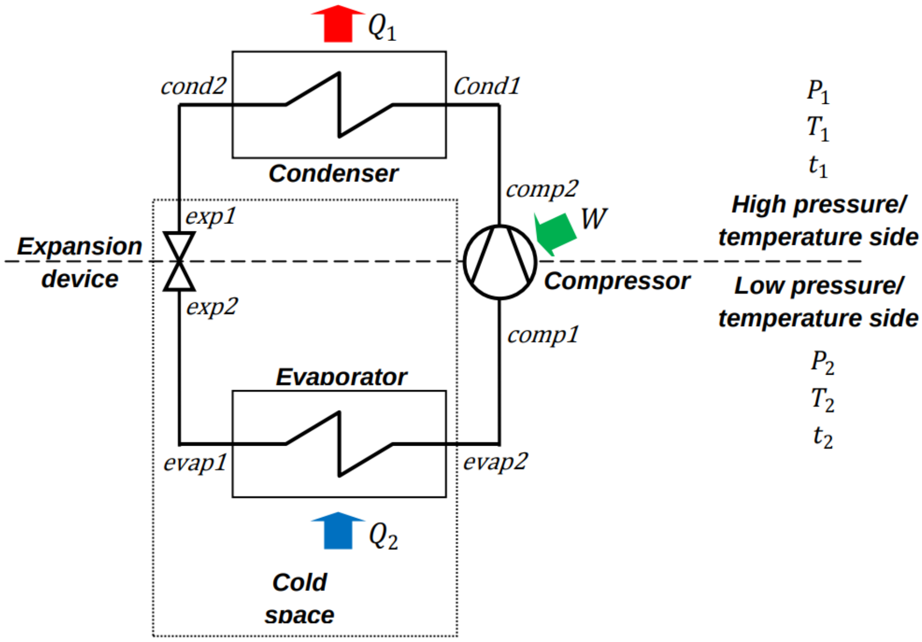

Figure 3.1: Principal sketch of a vapour compression cycle [20]

coefficient of performance (COP). The COP is calculated by dividing refrigeration effect Q_2 by the compressor work W.

3.1.2 Compressors

The helical rotary screw compressor is the most common choice of compressor in refrigeration systems. It works with a constant volume displacement design. The refrigerant fluid is trapped in the suction chamber with volume V_s at the beginning of the screw. The two screws rotate so that the screw bodies occupy the suction volume. This forces the refrigeration fluid along the screw thread and traps it in a smaller volume chamber.

The refrigerant is successively pushed along the screw until it occupies the discharge chamber, with volume V_d, at the far end of the screw.

$$V_i = \frac{V_s}{V_d}$$

V_i can either be fixed or variable. In the latter case V_d is regulated by a slide- or bypass valve. For compressors with fixed V_d, V_i is a parameter to consider when selecting compressor size. A rule of thumb is that the most efficient function is achieved at $CR = V_i^k$ for a certain V_i, given a compression ratio (CR). The k-value varies with pressure and temperature and hence it differs between the suction- and discharge chamber. However, for single stage compression the CR is smaller and the k-value varies less. Therefor the k-value can be approximated at suction conditions.

For multistage compression an average or weighted average value between suction- and discharge condition can be used to decrease the discrepancies [21]. CR, i.e. the suction pressure p_s over the discharge pressure p_d, is dependent on the choice of evaporator- and condenser temperature. It is common for the evaporator temperature to stay constant during the year but changing outdoor temperature often alters the condensing temperature. Hence, the condensing pressure and CR are varying. To work around this issue and select a compressor size for efficient function, CR can be calculated using the yearly average condenser pressure. Compressors with variable V_i control makes for easier size selection since the internal volume ratio can be matched with the variable external pressure ratio.

The reciprocating compressor has a function similar to the internal combustion engine. The cycle consist of two stages. First the gaseous refrigerant fluid enters the cylinder, the inlet closes and the piston slides into the cylinder. Thus the volume that the refrigerant occupies decreases and its temperature and pressure increases. Second the outlet opens and the compressed refrigerant is released.

3.1.3 Condensers

The condenser is the component where the refrigerant phase changes from gas to liquid and heat is rejected. Heat of condensation is released as the hot refrigerant fluid enters the cold tubes. The amount of heat removed by the evaporator, i.e. the refrigeration load, is directly proportional to the amount of heat rejected by the condenser. The amount of heat removed in the condenser can be quantified using enthalpies:

$$Q = UA\Delta_{T_{lm}} \tag{3.1}$$

where U is a constant heat transfer coefficient, dependent on fluids and material in the condenser, A is a constant area of the contact surface between the fluids and

$$\Delta_{T_{lm}} = \frac{(T_{H_{\text{In}}} - T_{C_{\text{Out}}}) - (T_{H_{\text{Out}}} - T_{C_{\text{In}}})}{\log((T_{H_{\text{In}}} - T_{C_{\text{Out}}})/(T_{H_{\text{Out}}} - T_{C_{\text{In}}}))} \tag{3.2}$$

At ideal conditions the only varying parameter after design will be $\Delta_{T_{lm}}$. The suction temperature, i.e. the temperature before the compressor, is often constant in refrigeration systems. This means that also the suction pressure can be considered constant. The temperatures changing are thus those in the condenser, regarding both its sides. The temperature on the shell side is dependent on the condenser pressure, i.e. the refrigerant's pressure after the compressor. The compressor work increases with increasing pressure ratio, i.e. condenser pressure over suction pressure, and since the inlet pressure is constant, only the outlet pressure can be varied. The compressor power stands for a large portion of the electricity consumed in a refrigerant system and thus should be minimized in an energy efficient system, hence the condenser pressure should be minimized.

Assuming constant U and A values as in Equation 3.1 and no losses in the cycle, we have:

$$\dot{Q}_{\text{Evaporator}} = \dot{m} \cdot (c_p \cdot (T_{C_{\text{In}}} - T_{C_{\text{Out}}}) + \Delta h_{\text{Evaporation}})$$

We now know all parameters except $T_{C_{\text{In}}}$, $T_{C_{\text{Out}}}$, $T_{H_{\text{In}}}$ and $T_{H_{\text{Out}}}$. As seen in the previous paragraph the condenser pressure should be minimized, meaning that the condenser temperature inlet $T_{H_{\text{In}}}$ is automatically minimized. The $T_{H_{\text{Out}}}$ is directly proportional to the evaporation inlet temperature and thus constant. This leaves us with the temperatures on the cold side of the condenser, i.e. $T_{C_{\text{In}}}$ and $T_{C_{\text{Out}}}$. These temperatures will be constrained by the ambient temperature and by type of condenser being used. The ambient temperature can be outdoor air temperature or cooling water temperature.

The different type of condensers are often grouped by the method that is used for transferring the heat away from the refrigerant but also based on which medium on the shell side. The three most common types of condensers are the air cooled condenser, the evaporative condenser and water cooled condenser. The air cooled condenser uses ambient air to cool the refrigerant. In the evaporative condenser a mix of water and air is sprayed onto the tubes simultaneously. Due to the nature of the evaporative condenser the shell side temperature is low when compared to a air cooled condenser. The $T_{H_{\text{In}}}$ in the latter is the dry bulb temperature while it in an evaporative condenser is dew point temperature. The water cooled condenser uses water on the shell side. One advantage of this type of condenser compared to an air cooled condenser is a higher heat transfer rate. This is due to an inherent relatively high density and heat transfer capacity of water. The temperature $T_{H_{\text{In}}}$ is the temperature of the water inlet and will vary dependent on what water source is used. The water temperature often fluctuates less than air and can in some cases be very low. In locations with non-proximity to a lake, a river or the sea a cooling tower can be used.

$T_{C_{\mathrm{In}}}$ is thus dependent on type of condenser and the geographical location, but the heat transfer can be increased in all cases. To increase the amount of heat that can be transferred between the ambient air or water and the tubes, the cooling medium velocity is increased by a fan or pump. This adds convection, in addition to conduction, and the overall heat transfer is increased. However, there is a trade-off between an increased heat transfer and increased cooling medium velocity. The increase in heat transfer results in a lower condenser tube side temperature $T_{H_{\mathrm{In}}}$ and hence a decreased compressor power. On the other hand, an increased cooling medium velocity results in an increase of fan- or pump power. At some point an increase in velocity will cause the fan- or pump power increase to surpass the decrease in compressor work and the total energy efficiency of the cycle will thus decrease.

The sizing of the condenser is based on the maximum refrigeration load condition. This occurs when the compressor works at full load and the condenser's ambient temperature is at design maximum. The condenser functions efficiently when the condensing surface is kept free from non-condensables and liquid refrigerant.

3.1.4 Evaporator

The evaporator is a heat exchanger (HX) where the refrigeration fluid has a saturation temperature that is lower than the temperature of the surroundings. The refrigeration fluid thus evaporates in the HX and heat from the surroundings is transferred to the refrigeration fluid. Heat equalling the specific heat of vaporisation plus the temperature drop can be extracted from the shell side for each mass unit of refrigeration fluid passing through the tubes.

The evaporator can be used for different purposes and thus have different design. When used for refrigeration of air, e.g. in a refrigeration storage facility, a fan blows air onto the the cold tubes. The temperature difference forces heat to flow from the air into the refrigeration fluid inside the tubes. Similarly to the condenser case in Section 3.1.3 the air velocity can be used to increase the convection part of the heat transfer and hence increase the overall heat transfer capacity.

3.1.5 Superheat

There are two types of superheat in a refrigeration system, namely evaporator and compressor superheat. Both concern the temperature difference between the saturation temperature and actual measured temperature. The refrigerant liquid boils in the evaporator and all refrigerant turn to vapor. However, this is the ideal case and in reality some refrigerant remains in liquid phase. This is troublesome since liquid damages the compressor. Therefore the refrigerant is superheated and complete vaporization is reached. Thus, the evaporator superheat is the difference between the saturation temperature and the measured temperature after the evaporator. The compressor superheat is instead measured just before the compressor.

3.1.6 Sub-cooling

Sub-cooling is the lowering of refrigerant temperature below the saturated temperature after the condenser. It is advantageous to sub-cool to avoid flashing, i.e. formation of gas in the liquid line. This ensures that all enthalpy of evaporation can be used in the evaporator and a higher efficiency is reached. In addition poor sub-cooling prevents the thermal expansion- or electric expansion valves to adequately control the refrigerant flow to the evaporator.

3.2 An introduction to control

In order to keep a system at a desired point of operation, control systems are typically required. A typical feedback system can be seen in Figure 3.2 below and consists of three main components. First the desired reference value is compared to the current measured value, generating an error value. This error value is then sent into the controller, which in turn calculates a control signal that is calculated differently depending on the control algorithm used. After this the control signal is sent into the system, or in this case the system model. When the system receives the control signal the error is reduced and a new system value is generated. This value is then measured by sensors, filtered and sent back to the controller.

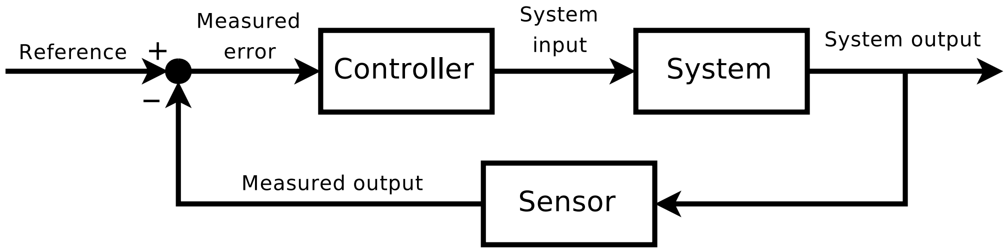

Figure 3.2: Figure of a typical feedback system [22]

3.2.1 Control methods

To control a system many different methods can be used, but most of them can be divided into two different groups, traditional control and model based control. In this case only two control methods will be used, one traditional control method and one model based, these methods will be explained below.

PID

The traditional control methods are widely used in the industry since they can be implemented easily and do not require a model to function. These controllers are tuned by using trial and error methods and can be re-tuned quickly if the system changes.

The most common of these methods is the PID method. PID is an acronym of the words proportional, integral and derivative which is the different parts of the controller given by:

$$u(t) = K_P \cdot e(t) + K_I \cdot \int_0^t e(\tau)\,\mathrm{d}\tau + K_D \cdot \frac{\mathrm{d}}{\mathrm{d}t} e(t), \tag{3.3}$$

where $K_{P,I,D}$ are tuning constants, e is the error and τ is the integration variable.

The first part is the proportional part and basically provides a quick response to any changes in the system. The second part is the integral part which is a bit slower than the proportional, but has the ability to completely remove steady state errors. Lastly the derivative part helps increasing the stability of the system as it reacts quickly if the error increases, this can however be devastating if the sensors give inconsistent values.

In this method, the parts have different purpose and can hence be combined in different ways depending on the desired behaviour, forming P,PI,PD and PID controllers. These factors makes the controller very versatile.

Model predictive control

In contrast to traditional control methods, the model based methods use a description of the real system to calculate control outputs. This enables them to predict how the system will behave before sending the signal, and hence generate smoother and more consistent behaviour.

The control method which will be used in this thesis is called Model Predictive Control, or MPC. This method is based on the concept of a receding horizon which means that for every step the optimal sequence of control signals is calculated for a finite future. For every timestep, the horizon moves one step ahead, creating a receding horizon. An example of a MPC controller in one timestep can be seen in Figure 3.3 below. In the figure the MPC calculates the optimal control outputs to reach the reference trajectory with the given conditions.

The traditional way of implementing MPC is using quadratic programming(QP) to formulate a convex problem which can be solved quickly in every iteration. An example of such a problem is the following:

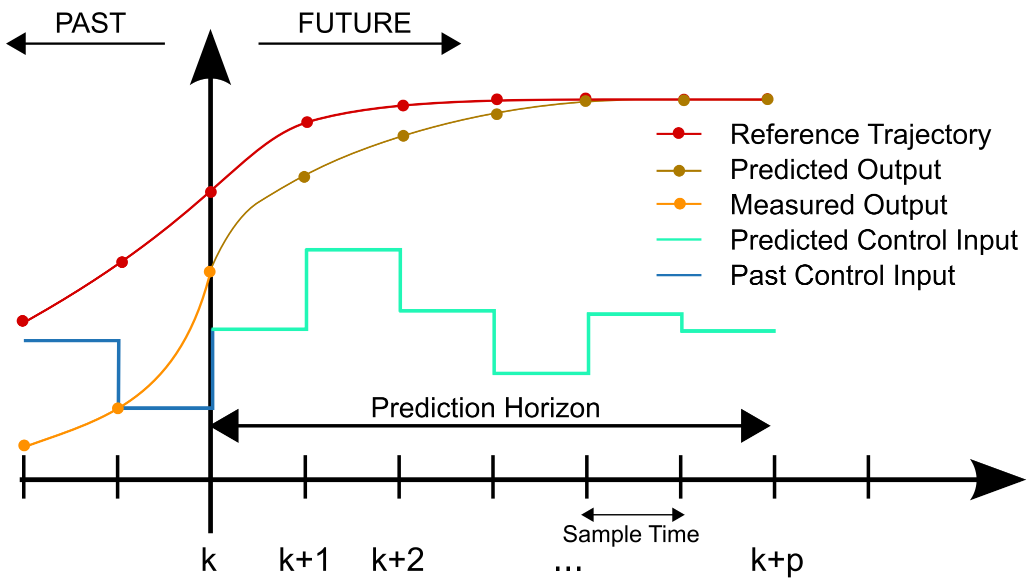

Figure 3.3: Figure of a typical MPC controller [23]

$$
\begin{aligned}
\underset{z}{\text{minimize}} \quad & \frac{1}{2} z^T H z + f^T z \\
\text{subject to} \quad & A_{eq} \mathrm{z} = b_{eq} \\
& A_{in} \mathrm{z} \leq b_{in}
\end{aligned}
\tag{3.4}
$$

Here z is a vector of future states and control signals, H is related to the cost on each state, A_{eq} and b_{eq} define equality constraints and A_{in} and b_{in} define inequality constraints.

An advantage of using MPC is that it is easy to implement constraints in the control system. Instead of adding ad-hoc solutions constraining the output signal itself, the constraints can be implemented in the optimisation algorithm as can be seen in Equation 3.4. This means that when the controller is generating future control signals, it takes into account what the constraint of the system is, and ensures that the reference is reached within these limits. Another advantage is that known future data can be implemented easily in the algorithm and used to increase accuracy. For example one could use future weather reports in order to control the temperature of the house.

The reason that MPC has not had more impact on the industry is in part that accurate modelling is required which might be time consuming if the system is complex or if it changes regularily. The other reason is that a substantial amount of computing power is required in order to optimise a complex system for a long horizon in every iteration. Because of this, MPC is primarily used in slow systems and where small improvements in efficiency are very valuable.

3.2.2 Control hierachies

Some systems can be better controlled by using a hierarchic system of several different controllers. This can for example be a petroleum plant that has one MPC which controls each section and sends reference values to PIDs at pumps and valves. By doing this a MPC can drastically improve the system efficiency while not having to model every detail. This will also decrease computation requirements for the MPC.

4

Technical systems, components and considerations

In this chapter carbon and ammonia systems are described and evaluated. After this, some common processes for fish refrigeration plants are summarised. Then the impact of infiltration on energy usage in refrigerated storage facilities are investigated. Finally the concept of thermal energy storage is briefly discussed.

4.1 Refrigeration systems

In this section two different refrigeration fluids will be discussed, Ammonia(R717) and Carbon Dioxide(R744). Different advantages and weaknesses of each type of system will be discussed and finally a cascade system where the two refrigerants are combined will be elucidated.

4.1.1 Carbon dioxide systems

R744 has been an economically viable option for some refrigeration systems, generally with evaporation temperatures around $-40\,°C$ and below, since the early 2000's. The main advantage with R744 as a refrigerant is an increased COP due to its physical properties. R744 has a high saturation pressure so even at evaporation temperatures as low as $-50\,°C$ the suction pressure is 6.8 bar. This results in a low pressure ratio and in addition a high gas density and thus large compressor capacity. The material properties for R744, R717 and chlorodifluoromethane (R22) are listed in Table 4.1.

The following sections are based on *Design criteria for CO2 evaporators* by Roland Handschuh [24] and *CO2 systems - Introduction to refrigerant and design manual for CO2 system* by Danish Technological Institute [25].

A negative aspect of using R744 is the low enthalpy of evaporation, resulting in a higher mass flow, which in turn increases the pressure losses in evaporators. A high pressure loss usually gives a higher saturation temperature, low COP and causes malfunction. However, this is not a great problem for R744 because its weak pressure-temperature dependency, seen in Table 4.1. Even though the pressure loss

Table 4.1: Material properties of R744, R22 and R717 at $-40\,°C$ [24]

Refrigerant	R744	R22	R717
Vapour pressure [bar]	10	1	0.7
Enthalpy of evaporation [kJ/kg]	322	243	1387
Density of gas [kg/m3]	26.24	4.85	0.64
Gas volume flow for 10 kW [m3/h]	6	41	47
Mass flow for 10kW $[\frac{kg}{h}]$	157.4	198.9	30
Pressure temperature dependency $\frac{dp}{dT}$ $[\frac{bar}{K}]$	0.37	0.05	0.04

over the evaporator becomes greater, compared to e.g. R717, the temperature drop is smaller so a low saturation temperature is reached and the driving forces are sustained.

The weak pressure-temperature dependency affects the design and dimensioning of the suction pipes as well. A larger diameter increases the investment costs and a smaller diameter increases the pressure drop and thus the running costs. In the case of R744 small pipe diameters can be used without reducing capacity or COP.

The high pressures which are required for a R744 system as can be seen in Table 4.2 does however make the system complicated to design since all the components need to be designed for this pressure.

4.1.2 Ammonia systems

According to *The Linde Group* [26] R717 has been used in the industry since the early 1930s. It is a very efficient refrigerant due to its low boiling point. As can be seen in Table 4.2 the vapour pressure is only 0.72 bar at $-40\,°C$ making the systems fairly simple to construct. In Table 4.1 it can seen that the enthalpy of evaporation is also very high making it possible to keep very low mass flows compared to R744. On the other hand the volume flows of R717 are very high, requiring large pipe dimensions. Another aspect to consider when using R717 is that it is considered toxic for humans. It does however have a zero global warming and ozone depletion potential making it environmentally friendly.

One major problem of using R717 is that the compression ratio gets very large at relatively low temperature lifts. As can be seen in Table 4.2 the compression ratios for R744 are substantially smaller even though the pressure is very large. For a system with condensation temperature of $20\,°C$ and evaporation temperature of $-40\,°C$ the ratio for R717 is 11.9 while it is only 5.7 for R744 making the lift more than twice as large. As compression ratios are directly related to the COP of the system, this is a very important aspect to consider.

Table 4.2: Vapour pressures of R744 and for R717

Temperatures [°C]	R744 [bar]	R717 [bar]
-40	10.05	0.72
-20	19.7	1.90
0	34.85	4.29
20	57.29	8.57

4.1.3 Cascade systems

R744 has very advantageous properties in refrigeration systems, but lacks the ability to operate efficiently in a sub critical state at higher temperatures. This means that if R744 is to be used, it is often necessary to combine it with other refrigerants in a cascade system.

One example of this is a system where R717 is first used to lower the temperature from the outside temperature with an air cooled condenser. The cold ammonia is then used as the high temperature side in a R744 system. This means that the carbon dioxide system only has to operate at a constant temperature which is independent from the outside temperature and hence can be much more efficient. Using a heat cascade does of course induce losses in the system, as more components are used and pressure losses increase, but this can be outweighed by the gains of decreased compression ratio when using R744.

In addition to this, two different temperature flows are created and can be used in different processes. For example a high temperature R717 flow at $-10\,°C$ can be used for cooling while a low temperature R744 flow at $-40\,°C$ can be used for freezing.

4.2 Fish refrigeration processes

In order to fulfill a fish refrigeration plant's requirements, discussed in Chapter 2, several different technical components and systems were investigated. Some of these are presented and evaluated below.

4.2.1 Cooling tanks

In order to reach the requirements for the short term cold storage a common solution is to store the fish in large tanks filled with cold sea water. This water is cooled constantly and kept just above the critical zone, explained in Chapter 2. In this way the fish is cooled very quickly due to the high thermal conductivity of water and can easily be transported onwards with pumps.

4.2.2 Freezing machines

There are several different methods of freezing fish. What method to choose depends on how refined the fish is, if it is a whole fish or a fillet. It also depends on how quickly the fish needs to be cooled, some methods are drastically quicker than others.

Air blaster

The air blast freezer is a very versatile freezing machine since it can handle all different shapes of fish. The freezer freezes the fish by blowing very cold air on it until the product reaches the desired temperature. One of the most common types is the gyro freezer which freezes the fish while it is being transported on a circling conveyor belt. This maximises the exposure time and enables large amounts of fish to be frozen quickly, defined as in less than two hours. One version of this freezer can be seen in Figure 4.1.

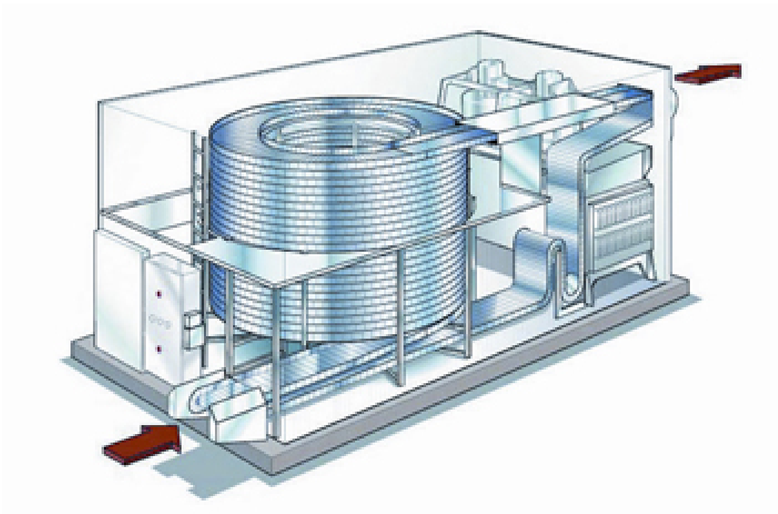

Figure 4.1: A figure of a typical gyro freezer. The air flows through the spiral to quickly freeze the fish. Picture taken from Albrechts Machinery [27]

Another option is to simply place the fish in front of a large cooling fan and letting it freeze in large amounts at a time, this is however fairly slow and typical freezing time are from ten to twenty hours.

Plate freezer

The plate freezer functions by utilising as direct contact as possible between the refrigeration fluid and the fish. The fluid is circulated through a metal plate which is then placed directly against the fish. By doing this the freezing can be done very quickly while minimising losses. One problem is that a good contact surface between the fish and the wall is required, otherwise air might be trapped between the wall and the fish and hence increase freezing time. If the machine is used properly with sufficient contact area the freezing time is defined as quick.

Alternative techniques

Other methods can also be used, for example the fish can be completely submerged in a very cold fluid like CO_2 or N_2 which cools it ultra-rapidly. This is usually used for tuna which otherwise takes a very long time to freeze.

4.2.3 Refrigerated storage

As the fishing season does not last all year, and fish hauls may vary a lot in size, it is almost always necessary for fish refrigeration plants to store fish in refrigerated areas. The scale and temperatures in with this is done may however vary. As can be seen in Table 2.2 the fish may be stored for several months in $-18\,°C$ but in the more general case, a lower temperature is required.

As the fish is already frozen when it reaches the storage, the main requirement is to cool the fish down to the desired storage temperature, keep the fish at that level and facilitate easy delivery to customers. This means that the most important considerations are usually minimising transmission losses and infiltration which is described more thoroughly in Section 4.3.

A typical scale of a storage is between 4000 - 8000 tonnes which means that the storage volume and wall surface areas is very large and thus subjected to large transmission losses.

4.2.4 Ice production

When the fish is delivered fresh, large amounts of ice is required to keep it cool until delivery. This is usually produced in ice machines, which utilises the central refrigeration system to freeze fresh water. A typical amount of ice produced in one day is around 5000 to 8000 kg.

4.3 Infiltration

Infiltration is the movement of air between outside and inside of a building through the building envelope. The air movement can be intentional, e.g. through vents, or unintentional through cracks and open gates. The building envelope in refrigerated storage facilities is often constructed to be air tight and therefore the focus in this section will be on air infiltration through open gates.

The main problem with infiltration in a refrigerated storage is condensation of warm humid air on the evaporators resulting in higher energy use. The effect on the inside air purity are in some cases important but this is not included in this report.

4.3.1 Humidity and defrosting

The most direct impact of humid air on the technical system is an increased amount of ice on the evaporator tubes. Frost formation occurs when the dew point is below the surface temperature of the evaporator tubes and this temperature is below 0 °C. Ice on tubes increase the air flow resistance and hence the pressure drop over the evaporator increases. Ice has a low heat transfer coefficient compared to the tube material and therefore it decreases the refrigeration effect of the evaporator.

Defrosting of the evaporators is affecting the air temperature and will cause relatively high temperature during short periods. Defrosting requires heat which will not contribute to a production increase and is therefore considered an energy loss. In addition to the energy loss due to defrosting, the refrigeration of humid air also increases the refrigeration effect during normal operation.

The difference between refrigeration of humid and dry air can be quantified by a simple example below. The refrigeration effect is calculated for refrigeration of air from 20 °C to −30 °C with a relative humidity (RH) of 0 and 60 respectively.

$$\dot{q}_{\mathrm{Refrigeration}_{\mathrm{dry}}} = 1\,\frac{\mathrm{kg}}{\mathrm{s}} \cdot 1\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}}(20\,^{\circ}\mathrm{C} - -30\,^{\circ}\mathrm{C}) = 50\,\frac{\mathrm{kJ}}{\mathrm{kg\,s}} \tag{4.1}$$

$$\begin{aligned} \dot{q}_{\mathrm{Refrigeration}_{\mathrm{humid}}} = {} & 1\,\frac{\mathrm{kg}}{\mathrm{s}} \cdot 1\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}}(20\,^{\circ}\mathrm{C} - -30\,^{\circ}\mathrm{C}) + \\ & 0.008\,\frac{\mathrm{kg}}{\mathrm{s}} \cdot (2.0\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}} \cdot (20\,^{\circ}\mathrm{C} - 12\,^{\circ}\mathrm{C}) + 2250\,\frac{\mathrm{kJ}}{\mathrm{kg}}) + \\ & 0.008\,\frac{\mathrm{kg}}{\mathrm{s}} \cdot (4.2\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}} \cdot (12\,^{\circ}\mathrm{C} - 0\,^{\circ}\mathrm{C}) + 333\,\frac{\mathrm{kJ}}{\mathrm{kg}}) + \\ & 0.008\,\frac{\mathrm{kg}}{\mathrm{s}} \cdot (2.1\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}} \cdot (0\,^{\circ}\mathrm{C} - -30\,^{\circ}\mathrm{C}) \simeq 71.7\,\frac{\mathrm{kJ}}{\mathrm{kg\,s}} \end{aligned} \tag{4.2}$$

Some sources suggest that as much as 30% of the total energy usage in refrigeration system is due to defrosting. In many cases only about 20 % of defrost energy is used to melt the ice. [28]

4.3.2 Heat gain by infiltration

To calculate the heat gains due to infiltration an ASHRAE [19] equation modified by Cleland et al [29] is used. First the entalphy is calculated for the incoming air where ΔT is the difference between the outside and inside temperature.

$$h_{\mathrm{Infiltration}} = (c_{p,\mathrm{Air}} + \frac{c_{p,\mathrm{Water}} \cdot \frac{m_{\mathrm{Water}}}{V_{\mathrm{Air}}}}{\rho_{\mathrm{Air}}}) \cdot \Delta T$$

Then the average air flow through the door is calculated as:

$$\dot{V} = \dot{V}_{AT} \cdot (1 - D_t) + \dot{V}_f \cdot D_f \cdot D_t \cdot (1 - E) + \dot{V}_{tr} \quad (4.3)$$

In the equation many different factors are taken into account. $\dot{V}_{AT}$ is the air flow when the door is closed and D_t is the fraction of the total time in which the door is open. $\dot{V}_f$ is the fully developed air flow when the door is open, D_f a door flow factor to account for the time when the flow is not fully developed and E which takes leakage protection factors like air curtains into account. $\dot{V}_{tr}$ is the air flow due to fork lift traffic. The computed air flow is then used to calculate the required cooling effect from:

$$\dot{q} = \dot{V} \cdot \rho_{\text{RefrigeratiedAir}} \cdot (h_{\text{InfiltrationAir}} - h_{\text{RefrigeratedAir}}). \quad (4.4)$$

4.4 Thermal energy storage

Thermal energy storage (TES) is a process where a material is either heated or cooled and then contained in an insulated container for later use. In this report the focus is on low temperature TES, which usually is achieved by using phase change of the storage media.

4.4.1 Latent and sensible heat

There are two different ways to store energy in a TES; the first and most important is latent energy storage. Latent heat is the energy differences between the different aggregated states. The most common way to utilise this is to freeze and thaw a material. The frozen material only have slightly larger density and thus requires a relatively small volume. Another method of storing latent heat is to use the condensation and evaporation of a material. This requires much more space as the gas form usually has a much lower density than the liquid and must be contained in expensive pressure containers. By using latent heat large quantities of energy can be stored with low temperature differences.

The other way to store energy is through sensible heat. This means that the material is cooled or heated without changing aggregated state. The sensible heat is much smaller than the latent and hence requires larger reserves in order to store a sufficient amount of energy. For water the sensible heat is around $4.2\,\frac{\text{kJ}}{\text{kg}}$ and the heat of evaporation is $2260\,\frac{\text{kJ}}{\text{kg}}$ which shows that is is much more efficient to use latent heat.

4.4.2 Charge and discharge

One of the most important aspects of a TES is the ability to charge and discharge it. If a heat exchanger is used in the TES, it will not be able to transfer a lot of heat when the PCM is in gas form as the thermal conductivity of gas is typically very low. If the PCM is instead in frozen form the discharge effect will quickly decrease as the ice close to the heat exchanger is melted, creating a heat transfer barrier of water around the heat transferring fins.

Another option which can be used to avoid these effects is to not use a heat exchanger but instead store the energy in liquid form and evaporate it in the cold consumers individually. In this way the heat transfer will be mainly from liquid which has great thermal conductive properties.

5

Case study: introduction, requirements and system design

The fish refrigeration plant, that is the object of this case study, is located on the west cost of Sweden, close to the city of Gothenburg. The company running the plant has its own fishing fleet and the catch is delivered by ships directly to the facility, unloaded using two automatic unloading stations. Herring and mackerel are the only fish species being processed and the current limitation for freezing is 200 ton per day. In addition to freezing there is a capacity for ice production of 25 tons per day. The ice is used to cool the fresh catch on the ships before arrival to the facility but also for cooling of fresh fish deliveries. Before the fish is frozen, it is stored in buffer tanks with $-1\,°C$ water. There are four buffer tanks with an aggregated storage capacity of 240 metric tons of fish. After freezing, the fish is stored in a refrigerated storage at $-29\,°C$, with a capacity of 4000 tons.

5.1 Requirement specification

The requirements presented in this section are based on applicable parts of Section 2. The commercial aspects are mainly based on the interviews with employees of the plant. These are combined with the authors' interpretations of the regulation regarding foodstuff production, storage and working environment in Sweden.

5.1.1 Temperature

The fish temperature requirements before, during and after the freezing process is decided based on customers' requested quality of the post-thawed fish together with the time between freezing and usage. Herring is often eaten pickled, i.e. without being heated, and very high quality is therefore required. The short term storage pre-freezing is very important for the quality and therefore the fish should be stored at the lowest possible temperature without reaching the critical zone. When stored at the temperature specified in Table 5.1 it does not reach the critical zone of $-1\,°C$ and does not exceed the requirement of $4.4\,°C$ for short term storage.

Freezing the fish the core should always reach $-18\,°C$ in less than 2 h to ensure quality. If the fish core reaches lower temperatures, this is not a problem as long as the mean temperature of the fish does not exceed the long term storage temperature.

For long term storage, as can be seen in Table 2.2, a temperature of −28.9 °C is superior to the legal requirement of −18 °C and it also gives a large temperature reserve when the fish is transported. Since a fish refrigeration plant is the first step in the production process, and fish is exported to countries all around the Baltic, long shelf life and large temperature reserves are very important. Lower temperatures can be tolerated if they are deemed more efficient at a certain time.

Table 5.1: The required temperatures in the fish refrigeration plant with regard to desired shelf life and preserved quality.

Process	Required Temperature[°C]
Cold Storage	0 (-1,+4.4)
Freezing	-18 (-2,+0)
Refrigerated Storage	-29 (-6,+4)

In Table 5.1 the required temperatures in each step of the process is specified, with temperature intervals given as 1(-2,+1)°C which means that the temperature can fluctuate between −1 °C and 2 °C, but the aim should be to keep it at 1 °C.

These required temperatures will ensure quality throughout the freezing process with minimal damage to the fish muscle. If the requirements are upheld the shelf life of the fish should be approximately eight months and the quality should be very good.

5.1.2 Product flow

The product mass flow of frozen fish is currently set to maximally 200 $\frac{\mathrm{t}}{\mathrm{d}}$. This is largely limited by fishing quotas and the available fishing fleet, making it hard to affect in this project. The flow within the factory with the temperatures from Section 5.1.1 can be seen in Figure 5.1.

The storage capacities are dimensioned with respect to the length of the fishing season, the product flow and the frequency of deliveries. The inflow, outflow and storage capacities can be seen in Table 5.2.

Table 5.2: The maximum processing capacities and available storage in the different parts of the factory.

Process	Inflow [$\frac{\mathrm{t}}{\mathrm{h}}$]	Outflow [$\frac{\mathrm{t}}{\mathrm{h}}$]	Storage capacity [t]
Sorting	50	50	0
Cold Storage	50	8.33	240
Freezing	8.33	8.33	16
Refrigerated Storage	8.33	100	4000

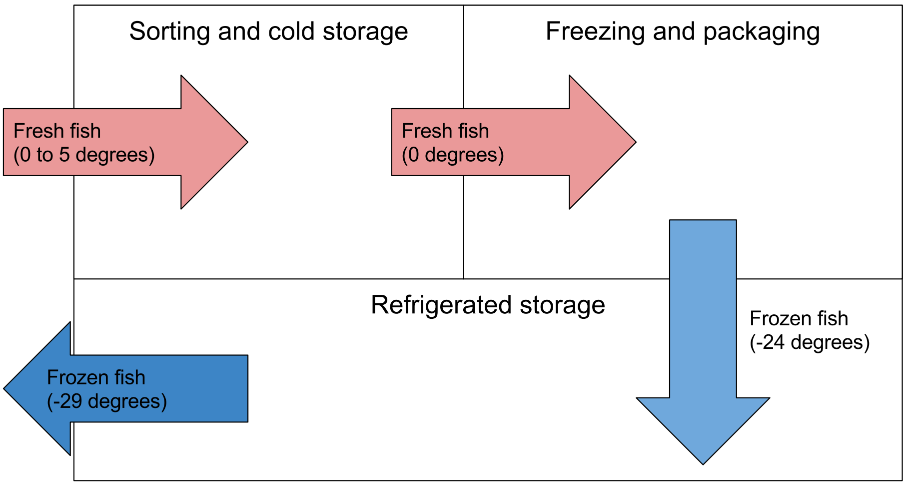

Figure 5.1: A illustration of the overall floor plan of the plant

5.1.3 Working environment

As described in Section 2.2 the regulation of the working environment in a fish refrigeration plant is more concerned with protective equipment and the planning of the work than it is with the design of refrigeration plant, other than normal consideration for security and safety in industries. Draft around doors is avoided by air-tight seals. The air-tightness is greatly dependent on the construction quality and thus the air leakage from the building is assumed to be negligible in the model.

5.2 Technical system design

In order to create a model of the case study fish refrigeration plant, input data of the refrigeration system is required. The initial refrigeration concept design is the basis for the various component models later constructed for the simulations. Even though the concept design holds necessary input parameters to create a model, the simulation of the model also gives important input back to the system design. The simulation of the model gives information of refrigeration loads over time, which is vital for the dimensioning of the different components. The concept part of the system design, i.e. choosing refrigerants, type of system and temperatures, has therefore taken place before the modelling and simulations and the dimensioning afterwards. The system design includes choice of components, refrigerants, temperatures and pressures but excludes more detailed system design, such as pipe material, diffuser type and motor sizes. However, these components' effect on the system are still regarded through approximations of pipe pressure losses, motor efficiencies etc.

5.2.1 Working temperatures

In order to create large enough driving forces and sufficiently low temperatures to reach the rapid freezing demanded in Section 5.1.1, a refrigerant evaporation temper-

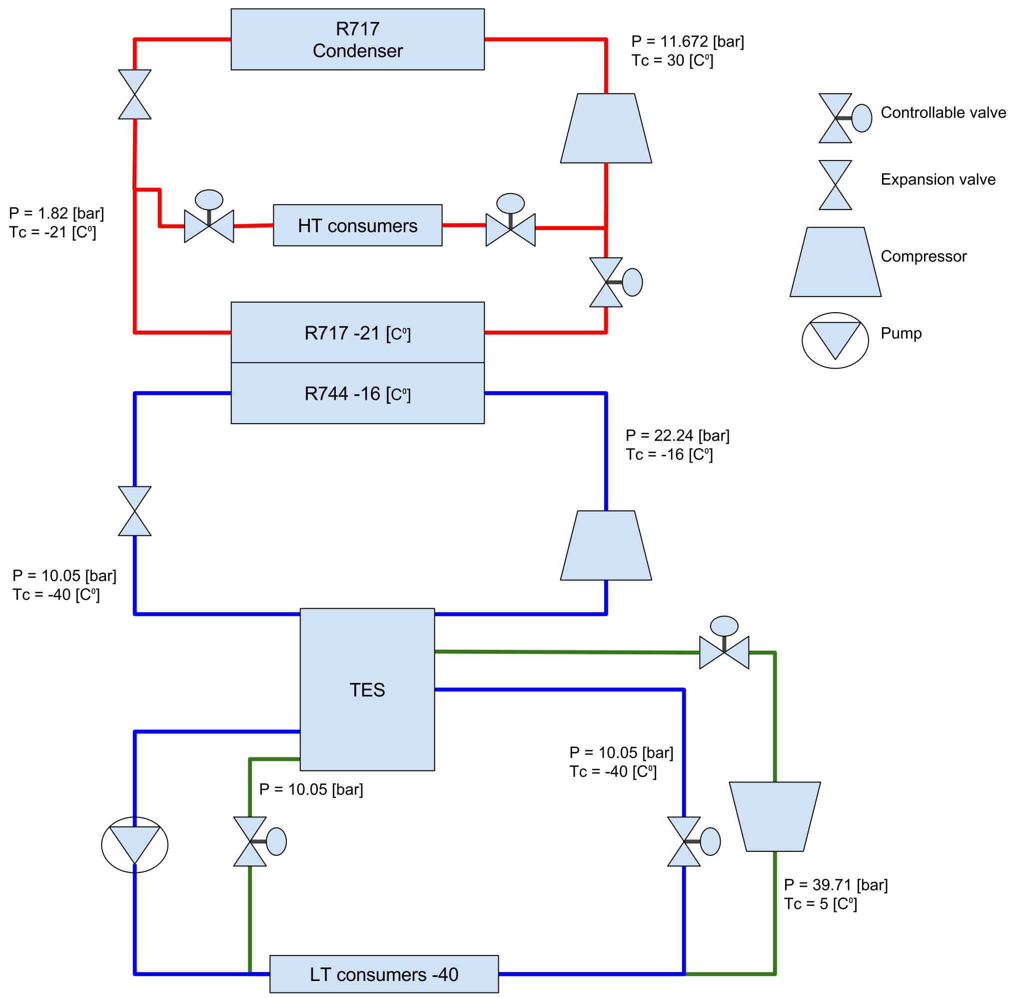

Figure 5.2: A principal sketch of the cascade refrigeration system

ature of −40 °C is required. For evaporation temperatures above −30 °C ammonia (R717) can be used, but for temperatures of −40 °C and below a carbon dioxide (R744) system design can reach higher system coefficient of performance. According to a study the efficiency gain by implementing a R744 cascade system instead of two-staged R717 system are 12.3 % and 37.2 % for evaporation temperature of −29 °C and −40 °C respectively [31]. As can be seen in Figure 5.2 the chosen system design is a R744/R717 cascade system. R744 is working between −40 °C and −16 °C on the low temperature side and R717 has a temperature range between −21 °C and 30 °C. The cascade heat exchanger temperatures, i.e. R744 condensing temperature −16 °C and R717 evaporation temperature −21 °C was chosen using a multi linear regression analysis by *H.M. Getu* [32]. This analysis does not take into account that the R717 cycle includes evaporators in addition to the cascade heat exchanger. It is thus likely that the temperature choice is non optimal. The optimal temperature is probably slightly higher so that the R717 cycle temperature lift decreases. In addition to being more efficient, cascade systems have lower investment costs. Due to the superior physical qualities of R744 at low temperatures, investment cost for pumps, pipes as well as the cost for installation is lower when compared to conventional two stage R717 systems [33].

Position	Refrigerant	Temperature °C	Pressure bar
Condensation	R717	30	10.03
Evaporation	R717	-21	1.83
Condensation	R744	-16	22.24
Evaporation	R744	-40	10.05

Table 5.3: Temperature design conditions for the refrigeration system

The low condensing temperature is due to the choice to use an evaporative condenser. The condenser pressure and thus the condensing temperature is variable and is the same as the value of the outdoor dew point temperature plus the minimum temperature difference in the condenser. In the next section, 5.2.2, the condenser pressure's effect on the compressor is evaluated.

5.2.2 Components

An important part of the system design is the refrigeration system component choices. The main components are listed in Table 5.4. As mentioned in the previous section, the condensing pressure varies depending on the outdoor dew point temperature.The condensing pressure, or the discharge pressure, is altered through the use of a variable compression ratio screw compressor. Another motivation to use screw compressor over a reciprocating compressor, is the former's advantage regarding the isentropic efficiency. Screw compressors have an average isentropic efficiency of 7 % -unit higher than their counterparts [34]. An outdoor temperature sensor sends information to the system controller, which in turn controls the compressor's slide valve, altering the compression ratio and thus the discharge pressure. The choice of method for controlling the compression ratio stood between variable frequency drive (VFD) and slide valve. Since the compressors will run on full load the choice became slide valve, as VFD has a 4 % penalty at these conditions [35].

Refrigerant	Function	Type
R717	Compression	Variable compression ratio screw compressor
R717	Expansion	Electronic expansion valve
R717	Condensation	Evaporative condenser
R717/R744	Evaporation/condensation	Semi-welded plate heat exchanger
R744	Compression	Screw compressor
R744	Expansion	Electronic expansion valve
R744	Evaporation	Plate freezers & air cooler

Table 5.4: Main components in the refrigeration system

5.2.3 Sizing and sequencing of the compressors

The number of compressors is based on load profiles obtained from simulation results. For the R717 cycle the maximum instant refrigeration load during a normal year is 1250 kW with a mass flow of 1.15 $\frac{kg}{s}$. In order to choose the compressor sizes to work optimally, they were selected to match the most frequently required refrigeration effects that can be seen in Figure 5.3. To reach this effect five compressors with sizes 550 kW, 300 kW, 200 kW, 150 kW and 50 kW are combined.

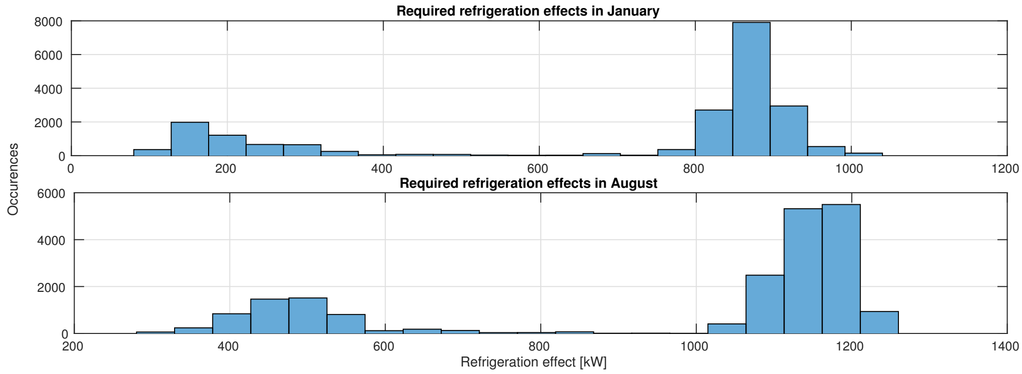

Figure 5.3: Histogram of the required refrigeration effects for the R717 cycle in January and August 2015.

The same dimensioning approach is used for the compressors in the R744 cycle. For the R744 cycle the maximum instant refrigeration load during a normal year is 820 kW with a mass flow of 3.3 $\frac{kg}{s}$. To reach this effect five compressors with sizes 350 kW, 200 kW, 100 kW, 100 kW and 70 kW are combined.

The refrigeration demand varies over time and to achieve the correct load the compressors are sequenced. This means that they are turned on and off so that the sum of the active compressors' refrigeration effect equals the demand. The load duration curve shows a very large variation of the combined compressor effect target. To avoid the necessity to run some compressors on part load at lower efficiencies, and still reach the balance between refrigeration- and compressor effect, a thermal energy storage (TES) unit is included in the system design. The unit handles the overshooting effect from compressors by charging the TES and compensates a refrigeration effect shortage by discharging the TES.

5.2.4 Evaporators

The choice of type and the dimensioning of the evaporators of the system is based on the required refrigeration effect, temperatures, pressures and refrigeration medium. The evaporators' refrigeration effects and performance can be found in Table 5.5.

Table 5.5: The table shows the different evaporator's quantity, location, performance and refrigeration effect.

Component	Units	Location	Performance	Refrigeration effect [kW]
Plate freezers	16	LT cycle	75 min / batch	960
Storage air coolers	3	LT cycle	50 $\frac{m^3}{s}$	300
Ice-machine	1	HT cycle	8000 $\frac{kg}{d}$ ice	30
Cooling tanks	4	HT cycle	Cooling 240 $\frac{m^3 water}{d}$	410

5.2.5 Defrosting of the evaporators

Defrosting of the evaporation coil in the frozen storage is achieved through the use of hot gas defrosting. The condensing pressure of the hot gas is set to 39.705 bar giving condensation temperature of 5 °C. The compression are achieved using a screw compressor designated for defrosting. The maximum aggregated R744 flow for defrosting of both the plate freezers and storage evaporators are 0.6 $\frac{kg}{s}$ resulting in a heat flow of around 200 kW.

The defrost in the frozen storage should start when the ice layer on the tubes are 2 mm and the running time is 10 min. The rate of ice layer creation is approximated by the assumption that all moisture entering the storage ends upon the evaporator coils. According to a defrost study [36] the combination of this ice layer thickness, defrost condensing temperature and running time results in approximately 43 % defrost efficiency, i.e. the share of total heat added used to melt the ice. Around 27 % of the added heat is dispersed into the room and the rest is used to heat the evaporator pipes or remains as non evaporated refrigerant. The defrosting of the plate freezers uses the same defrost efficiency as the storage.

5.2.6 Thermal Energy Storage

The thermal energy storage is a 100 m^3 receiver where condensed R744 can be stored at the evaporation pressure. The discharge of R744 liquid is done by a pump which has a low energy consumption relative to the compressor.

The discharge is a limiting factor for many other TES solutions using liquid-solid phase shift materials. The chosen TES setup enables a high capacity of both charge and discharge. The maximum charge capacity is limited by the compressor's mass flow capacity and constrained by the mass flow of R717. The heat of condensation for the R744 must be absorbed by the R717. Since compressors constitutes a substantial part of a refrigeration system's investment cost, and also the largest part of its energy use, the size of the compressor is not increased just to increase the TES charge capacity. The discharge on the other hand is limited by the pump's capacity, which is a less expensive unit. The pump's energy use is small relative to the compressor and it can thus be economically viable to invest in a larger pump to increase the discharge capacity.

5.2.7 Fans and pumps

The compressors are the most important components regarding the energy efficiency of the system due to their large loads. The pumps and fans still have considerable energy usage and to optimise the whole system they are an important part. The storage air coolers consists of three fan units, one in each storage room, with three fans each. The individual fans will be cycled on and off to handle large variations in load, but there is capacity for variable fan speed for smaller variations as well. The same sequencing will be used for the condensing fans. The pumps' pressure build will vary with mass flow and so will the efficiency.

Table 5.6: Design values of electricity effect and performance for pumps and fans of the system

Component	Units	Location	Performance	Electricity effect kW
Condenser fans	4	HT cycle	60 $\frac{m^3}{s}$	30
Storage fans	3	LT cycle	40 $\frac{m^3}{s}$	20
R744 pumps	1	LT cycle	3.14 $\frac{l}{s}$	10

5.2.8 Climate shell

The climate shell is not designed in this project, but taken as an estimate of the original system. The estimated insulation and heat transfer parameters are found in Table 5.7. As described in Section 5.1.3 the air tightness of the building is assumed to be very high. This is mainly due to lack of information since no tests of air tightness in the actual building have been made.

Table 5.7: The u-values and thicknesses of the factory walls

Layer	U value $[\frac{W}{m^2 K}]$
Storage room walls and ceiling	0.18
Freezing room walls and ceiling	0.4
Floors	0.2

5.2.9 Economics of the R744/R717 cascade system

In this section some economic impacts of choosing a cascade R744/R717 system over a traditional two-stage R717 setup are presented. The economic focus of the thesis has been on how the running cost of the designed system can be minimised. The impact of the other costs have only briefly been analysed. This means that the system design could be changed if investment costs were to be thoroughly analysed. The sections on investment costs and maintenance costs are general concerns of R744/R717 cascade systems and not of the designed system.

Running costs

The economic impact of using an R744/R717 cascade system are greatly dependent on the temperature lift and load pattern. Therefore it is difficult to draw a general conclusion of whether or not the running costs are decreased by implementing a cascade system. For the case study the running cost reduction is approximately 40%. The running cost reductions for different scenarios can be found in Section 7.2.

Component costs

The cost difference of investing in components for an R744/R717 cascade system compared to a traditional R717 two staged system vary with the components. For example, the R744 heat exchangers used as evaporators and condensers can be designed to have smaller surface areas and thus be cheaper. The smaller area is due to higher heat transfer coefficients as a result of the high pressure and density. A set of components that might become more expensive are the pipes. The tube wall thickness needs to be increased to sustain the higher pressures. However, the increase in density lowers the volumetric flow and reduces the pipe diameter and thus reduces the cost increase [37]. The higher refrigerant density also have implications on the pump work and R744 pumps are generally smaller and less expensive than R717 pumps [38].

For compressors it is the opposite way around. The R744 compressor's specific cost are about 20 % higher than its R717 counterpart [37]. However, the designed system allows for a decrease of the R744 compressor size. Normally the compressors have to be dimensioned to manage the system's maximum load. When thermal energy storage and model predictive control are used, the maximum load can be satisfied as long as the discharge effect is large enough. The R744 compressors total cost are therefore lower for the designed system than for a typical cascade system.

Installation costs

Installation costs comprise a substantial part of the overall costs for all refrigeration systems. For R744/R717 the installation cost are about 40%-50% of the total investment costs. This is slightly higher than a traditional R717 system.

Refrigerant costs stands for about 5% of the installations costs. R717 is cheap compared to other refrigerants but R744 is even cheaper [37]. So the refrigerant cost is decreased by choosing an R744/R717 system [39].

Maintenance cost

The maintenance cost for an R744/R717 cascade system is generally higher than for an R717 only system. This is mainly due to higher pressures increasing the wear on seals. Mixing of R744 and R717 and water contamination has to be avoided by all means and therefore the cascade heat exchanger should be maintained regularly. The overall maintenance costs are in some studies approximated to 2 % yearly of the initial investment cost.

6

Case study: modelling

To make the modelling of the system manageable, the fish refrigeration plant was divided into several different parts, which in turn were divided into two groups. The first group were the consumers of refrigeration effect, i.e. the cooling tanks, freezing room, refrigeration storage, the TES and the ice machine. The other were the producers of refrigeration effect, which were the compressors, pumps and the fans. These systems were modelled in order to simulate the system behaviour and analyse the performance with the new system design and control. In some cases where more information was needed on how vapour compression cycles were modelled *Dynamic modeling and Model Predictive Control of a vapor compression system* [41], was used for inspiration.

6.1 Cooling Tanks

The area where the fish is placed after being delivered from the fishing vessel, i.e. the cooling tanks, was a bit different to the other rooms since the temperature is much higher. The water in the tanks was pumped in from the ocean and then cooled down to $-1\,°\mathrm{C}$. The amount of heat that needs to be removed from the water to reach the desired temperature was calculated as:

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta T \tag{6.1}$$

where ΔT is calculated as:

$$\Delta T = T_{\mathrm{OceanWater}} - T_{\mathrm{CoolingTankWater}}.$$

In addition to cooling the water, the received fish also needed to be cooled down to the values specified in Table 5.1. This was calculated using the same equation as above. To simplify the calculations in this model, all the liquid and fish in the cooling tank was assumed to be cooled down continuously. It was also assumed that the fish that leaves the cooling tank was always at the desired temperature.

6.2 Refrigerated Storage

The refrigerated storage was modelled as a box with roof, ceiling and four walls. Two of the walls faces the outside and the two remaining walls face the freezing room and the cooling tank room, respectively. The heat balance of the room was divided into three categories which combined were used to calculate the refrigeration load. The categories were transmission losses, infiltration losses and cooling requirement of the product.

For each time step the transmission losses were modelled as a constant heat loss through the walls dependant of the outside temperature. Each wall was divided into several sections with varying heat transfer coefficients and the losses through one wall was calculated as:

$$\begin{aligned} \Delta T &= T_{\text{AirStorage}} - T_{\text{AirOutside}} \\ \dot{Q}_{\text{Transmission}} &= \lambda_{\text{Wall}} \cdot A_{\text{Wall}} \cdot \Delta T \end{aligned} \tag{6.2}$$

Where $\dot{Q}$ is the heat loss, λ the surface heat transfer coefficient and A the area of the wall.

The infiltration losses were modelled as described in Section 4.3.2. Finally the cooling requirement was modelled by taking the difference of the fish temperature entering into the refrigerated storage and air temperature in the storage:

$$\begin{aligned} \Delta T &= T_{\text{InStorage}} - T_{\text{OutStorage}} \\ \dot{Q}_{\text{Product}} &= \dot{m}_{\text{Product}} \cdot cp_{\text{Product}} \cdot \Delta T. \end{aligned}$$

The resulting temperature in the cold storage was finally calculated by doing an energy balance based on the losses and the supplied cooling effect as follow:

$$\begin{aligned} \Delta T &= \frac{\sum Q_{\text{Dispersed}} - \sum Q_{\text{Losses}}}{m \cdot c_p} \\ T(i+1) &= T(i) + \Delta T \end{aligned} \tag{6.3}$$

6.3 Freezing Room

The freezing room was modelled as a box with two outer walls and two inner walls facing the cold storage and the cooling tank room. This room did not have an active climate control and was only affected by the heat absorbed into the plate freezers, heat gain by workers, ventilation and lighting as well as transmission and infiltration losses.

To calculate the room temperature, the transmission losses and infiltration was calculated as in Equations 6.2 and 4.4. The cooling from the plate freezers was calculated as the heat absorbed in the plate freezers in Section 6.3. The heat gain from workers and lighting was a constant contribution and the required effect to heat ventilation air was calculated as in Equation 6.1. The temperature dynamics was modelled as in Equation 6.3.

Plate freezers

The required refrigeration effect from the plate freezers was calculated as a function of the mass flow of fish in the process and the temperature difference between the fish, plate freezers and the air of the freezing room.

To estimate the required heat that had to be removed from the fish at each time instance, several different processes had to be modelled. First, the fish enters the freezer in an unfrozen state and leaves it in a frozen state, so the freezing must be modelled. Secondly, the temperature at each point must be modelled to ensure that the quality requirements were fulfilled and lastly the heat absorbed from the surrounding air must be calculated.

The heat removed at each time step in the freezer was calculated as:

$$\dot{Q} = \lambda \cdot A \cdot \Delta T \tag{6.4}$$

where λ is the surface heat transfer coefficient

$$\lambda = \begin{cases} \lambda_{\text{Fish/Water}} & \text{for } Q > 0 \\ \lambda_{\text{Fish/Ice}} & \text{for } Q \leq 0 \end{cases},$$

A is the area of the steel plates that transfer heat from the fish to the refrigerant and Q is the current heat in the slice.

The calculated effect $\dot{Q}$ was then used to calculate the state and temperature of the fish block. First the block has to be frozen which means that the heat of fusion need to be removed. This was done by first setting the current heat of the slice to:

$$Q_{\text{Initial}} = \Delta H_{H_2O,\text{Fusion}}.$$

Then the heat balance was calcuated by summing the removed heat in Equation 6.4:

$$Q = Q_{\text{Initial}} - \sum_i \dot{Q}.$$

The block was assumed to be thawed until all the latent heat of evaporation has been removed, after that it was considered frozen and λ was changed to correspond to the new state. When the block was frozen, the temperature was calculated using an equation similar to 6.3.

To be able to do this within reasonable computation time, the fish block was divided into a grid of smaller blocks. The block of fish in the plate freezer was divided into two halves since the freezer has two plates with refrigeration fluid. The half block was then divided into small slices as can be seen in Figure 6.1. At each time instance, the heat $\dot{Q}$ out from each block and the new resulting temperature was calculated and used to estimate the energy requirement of the plate freezers.

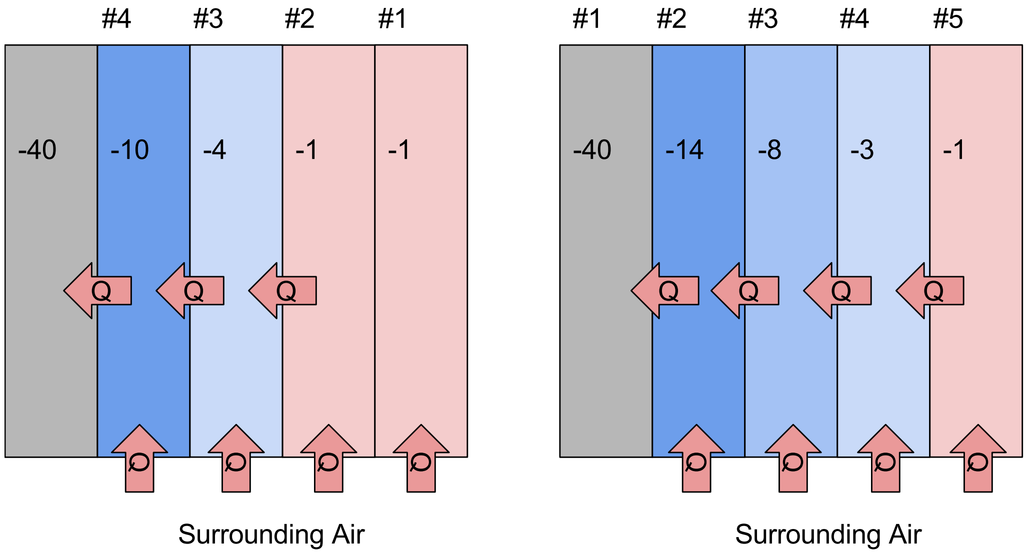

Figure 6.1: A figure showing the block grid in the plate freezer. The figure shows the temperature at two different time instances. The gray area is the plate freezer wall with assumed to have the same temperature as the liquid refrigerant, the blue area the frozen fish and the red area the thawed fish.

In this method, each slice has a current state which was either frozen or thawed. This was calculated by using the mass and heat of fusion of the slice and the total energy removed. The temperatures could also be considered to be an average of the entire slice. In this model it was assumed that the water and fish mix enters the plate freezers at a temperature just above its freezing point. This means that as soon as heat was removed from the mix, it started to freeze.

To calculate the heat that was absorbed from the surrounding air, a very simple estimate was used. The heat was estimated to primarily enter from the uninsulated top area of the slices and the other areas were considered well insulated. Then Equation 6.4 was used to calculate the absorbed heat.

The data for heat absorbed into the air and required heat removal at each time step was then simulated and made to a lookup table which could be used efficiently during system simulations.

Evaluation of the model

The hardest parameter to decide for the plate freezer model was the value for the thermal conductivity. Initial tests with the values for fish gave very large errors compared to the freezing time which had been given. This might be in part because the values that were used first were for pure muscle and not for whole fish, and in part because of inpurities of the water, such as air bubbles. Because of this the value had to be estimated by estimating the amount of air in a typical mixture.

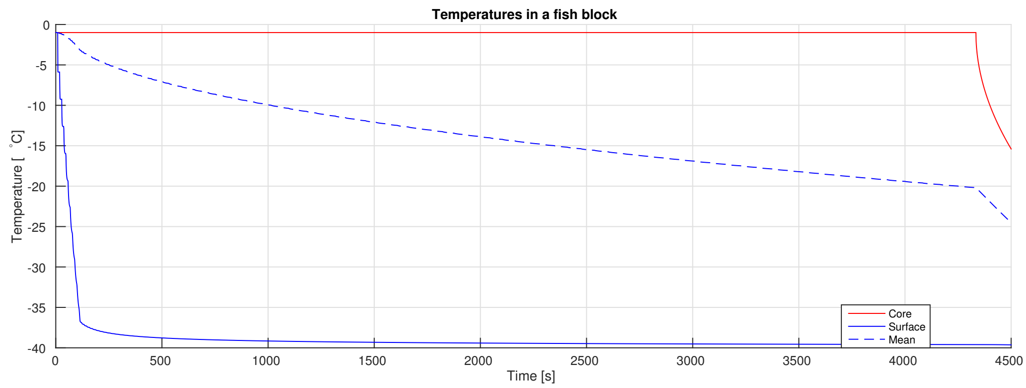

Figure 6.2: The core, mean and surface temperature of a block of fish which was frozen in a plate freezer

The model for the plate freezer was simulated for a period of 1.25 h to see how the temperature changed over the usual freezing time. The result can be seen in Figure 6.2

As can be seen in the figure, the surface reaches very low temperatures almost instantly while the core does not freeze until the end of the simulation time. This result was deemed reasonable because of the extreme temperature difference between the initial water temperature and the refrigeration fluid.

In order to determine how sensitive the model was to changes in simulation parameters it was simulated for a selection of different number of slices. The initial change of final temperature was rather large as can be seen in Figure 6.3, but as it settled around 100 slices this number was chosen as a compromise. Simulation of more than 100 slices required a substantially larger amount of time. This is due to the fact that very thin slices will cool down very quickly when simulated for a "long" period. As the simulation was discrete additional iterations were required at each time step with smaller slices to make the model stable.

As the pipe dimensions of the plate freezers were set by the producers, the maximum amount of heat that could be removed from one plate freezer was limited to 320 kW to avoid large pressure losses. The required heat removal from one plate freezer can be seen in Figure 6.4. The amount of heat to be removed was substantially higher in the beginning since the temperature gradient to the cooling plate was very large. As the difference decreases over time the requirement drops down to a more constant value.

When Figure 6.4 was combined to simulate a continuous cycle where a new plate freezer was started every 475 seconds, it created a cyclic pattern where energy use varied dramatically over time and hence it would be very advantageous to be able to average out this load with the help of a TES.

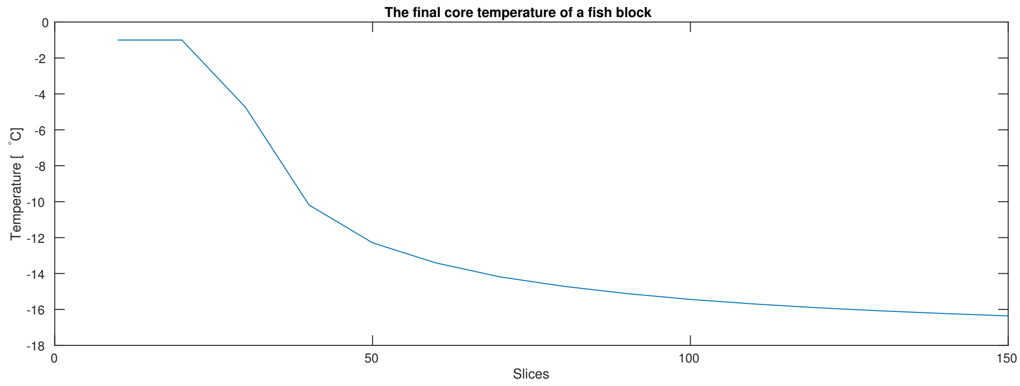

Figure 6.3: The final core temperature of the fish block dependant on how many slices the block was divided into in the calculations.

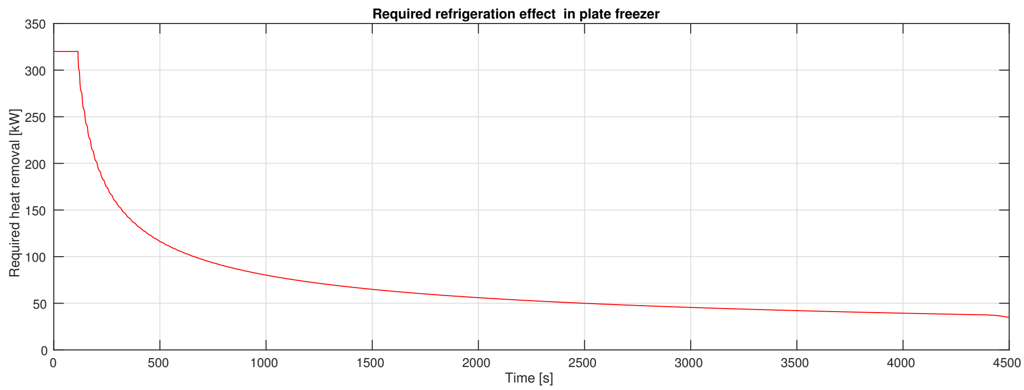

Figure 6.4: The amount of heat removed from a fish block by a plate freezer.

6.4 Ice machine

The ice machine has similar characteristics as the cooling tanks since it was basically about cooling water. The heat that needed to be removed in this machine was calculated using Equation 6.1.

6.5 Thermal energy storage

The thermal energy storage(TES) was modelled as an energy storage fulfilling the equation:

$$E_{t,\text{TES}} = E_{t-1,\text{TES}} + \Delta E \tag{6.5}$$

where E_{TES} is the stored energy and ΔE is the charge or discharge of energy from the TES.

The maximum storage $E_{\mathrm{TES,MAX}}$ was calculated by setting a certain maximum volume $V_{\mathrm{TES,MAX}}$ of gas in the storage tank when the TES has no liquid R744:

$$E_{\mathrm{TES,MAX}} = V_{\mathrm{TES,MAX}} \cdot \Delta h_{\mathrm{R744,Evaporation}} \cdot \rho_{\mathrm{R744}}. \tag{6.6}$$

6.6 Heat exchangers and fans

In order to simulate the heat flow from the refrigerated air into the refrigeration system, as well as the dispersion of heat from the condenser, the heat exchangers and fans had to be modelled. As the physical properties of a fan was inherently nonlinear, a linearised version also had to be created for use in the system controller.

First the temperature difference ΔT was set to 5 °C in in order to have large enough driving forces. After this the design mass flow $\dot{m}_{air}$ was calculated as:

$$\dot{m}_{\mathrm{Design,Air}} = \frac{\dot{Q}_{\mathrm{Storage,Max}}}{\Delta T \cdot c_{p,\mathrm{air}}}$$

The volumetric design airflow $\dot{V}$ was then calculated by multiplying $\dot{m}$ with the density. The pressure drop over the heat exchanger was then calculated as:

$$\Delta p_{\mathrm{Design,Drop}} = k_{\mathrm{Laminar}} \cdot \dot{V}_{\mathrm{Air}} + k_{\mathrm{Turbulent}} \cdot \dot{V}_{\mathrm{Air}}^2,$$

where k is the friction coefficient. After this the design fan power was calculated as:

$$P_{\mathrm{Design,Fan}} = \Delta p_{\mathrm{Design,Drop}} \cdot \dot{V}_{\mathrm{Design,Air}}$$

The control signal u was then inserted in the equations below where RPM_{Design} was a preset value.

$$\begin{aligned} RPM_{\mathrm{Actual}} &= u \cdot RPM_{\mathrm{Design}} \\ P_{\mathrm{Fan}} &= P_{\mathrm{Design,Fan}} \cdot \frac{RPM}{RPM_{\mathrm{Design}}} \end{aligned} \tag{6.7}$$

For the linear model, the fan power was linearised around the points where the pump is operating under normal conditions.

6.7 Pipes and pumps

The model contains two pumps, one in the R744 circuit used for discharge of the TES and another one providing the low temperatures consumers with R717. Both pumps have two models each: one linearised version which the MPC uses and another one with non-linear calculations used when simulating.

For the non-linearised case the system's pressure drops in pipes and heat exchangers were calculated in a separate function. The pump affinity laws were then used to calculate the motor RPM, pressure increase and volumetric flow as in the fan case, see Section 6.6. For the linearised case the pressure drop was linearised at the design value. The pressure drops were calculated for gas and liquid separately but using the same procedure. The pipe loss was calculated using:

$$\Delta p_{\text{Drop}} = k \cdot L \cdot \rho_{\text{Fluid}} \cdot v^2/(D/2) \tag{6.8}$$

where L is the pipe length, D the pipe diameter, v the fluid velocity and p the pressure.

In order to calculate the friction coefficient k it has to be decided whether the flow was laminar or turbulent. This was done by examining the Reynold's number (Re). For Re below 2300 it was assumed that the flow was laminar and above 2300 the flow was turbulent. The Reynold's number was calculated using the kinematic viscosity ν at temperature $-40\,°\text{C}$ and pressure $10.05\,\text{bar}$ for liquid and gaseous R744 respectively. For R717 the calculations were done with the kinematic viscosity at $-21\,°\text{C}$ and $1.83\,\text{bar}$.

$$Re = v \cdot D_{\text{Pipes}}/\nu \tag{6.9}$$

The friction coefficient for turbulent flow was calculated using Blasius correlation:

$$k = 0.3164 \cdot Re^{(-0.25)} \tag{6.10}$$

and the friction coefficient for laminar flow was calculated using:

$$k = 64/Re \tag{6.11}$$

The pipe length was a preset value and based upon the case plant design. The pressure drop across the heat exchangers were estimated by assuming an area/length ratio. This was given by the preset areas using the same pipe diameter as was used to transport liquid between the different components.

6.8 Compressors

The different compressors vary slightly in design but were modelled using approximately the same equation. First the pressure drops were calculated in a similar way to in the pumps, see Section 6.7. For the linear model the design mass flow value was used instead of the actual mass flow to simulate a worst case scenario. Next the pressures and temperatures for the high pressure side of the compressor were calculated using table values. For the R744 compressor these values stayed constant and were design parameters. As the R717 compressor's discharge pressure was affected by the outside air temperature, the compressor efficiency will vary. Implementing this variation would cause the model to be non-linear which cannot be solved in reasonable time with MPC. Therefore the varying efficiency was excluded from the compressor model and instead the constant maximum efficiency was lowered.

In order to reach higher efficiency in the system, an evaporative condenser was used and hence the dew-point temperature was used for the outside temperature. The dew-point temperature was estimated using the relative humidity ϕ and dry bulb temperature:

$$T_{\text{Dewpoint}} = 243.04 \cdot \frac{(\log(\frac{\phi}{100}) + \frac{(17.625 \cdot T_{\text{DryBulb}})}{(243.04 + T_{\text{DryBulb}})})}{(17.625 - \log(\frac{\phi}{100}) - (\frac{(17.625 \cdot T_{\text{DryBulb}})}{(243.04 + T_{\text{DryBulb}})}))} \tag{6.12}$$

When the pressures p and high pressure temperature T_1 was determined the isentropic and real temperature had to be calculated to estimate the work and losses:

$$T_{2,\text{Isentropic}} = T_1 \cdot (p_2/p_1)^{((k-1)/k)}$$
$$T_{2,\text{Real}} = \frac{T_{2,\text{Isentropic}} - T_1}{0.8} + T_1$$

The k-value is the specific heat ratio and this value was iterated using tabulated values since it depends on the inlet and outlet temperature with the latter varying, as described in the previous section. Thus the outlet temperature T_2 was calculated. Using the known temperatures the compressor work was calculated as:

$$w_{\text{Compressor}} = c_p \cdot (T_{2,\text{Real}} - T_1). \tag{6.13}$$

The compressors had different efficiencies depending on what load they were running on. Compressors in the model were constantly running on full load. This was a good approximation since there were several compressors which could be coordinated and the TES handled the difference in required and produced effect, see Section 5.2. This resulted in a constant efficiency and the electricity consumption was calculated as:

$$E_{\text{Compressor}} = w_{\text{Compressor}} \cdot \dot{m}_{\text{Refrigerant}} \cdot \eta_{\text{Motor}}. \tag{6.14}$$

In the case where component control was needed, the mass flow of refrigerant was calculated using a similar equation to 6.7.

6.9 Validation of model

As an example to see how the model behaved, it was simulated for the two first weeks of January. The result can be seen in Figure 6.5.

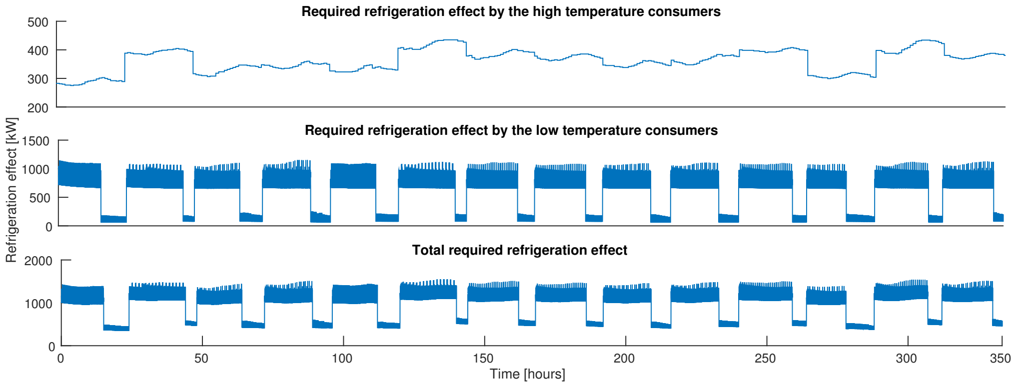

Figure 6.5: The required refrigeration effect by the different parts of the system.

As can be seen in the figure, the total required refrigeration effect is around 1200 kW. The high temperature consumers, i.e. cooling tanks and ice machines, vary with ocean temperature as their main consumer is cooling ocean water. They also vary with the current product flow as this determines how much water needs to be cooled down. The low temperature consumers on the other hand, is mainly dependant on the product flow. The oscillations are due to the changing requirements of the plate freezers.

7

Case study: control, simulation and evaluation

In this chapter a control system is designed using model predictive control (MPC). The designed system is then simulated with this controller and the results are evaluated.

7.1 Control design

The control system of the refrigeration system is divided into several different levels to decrease the computational power that is required both in simulations and in real life applications. This means that a top level controller sends reference signals to lower level controllers which in turn send control signals to the components. The control hierarchy of system and component controllers can be seen in Figure 7.1.

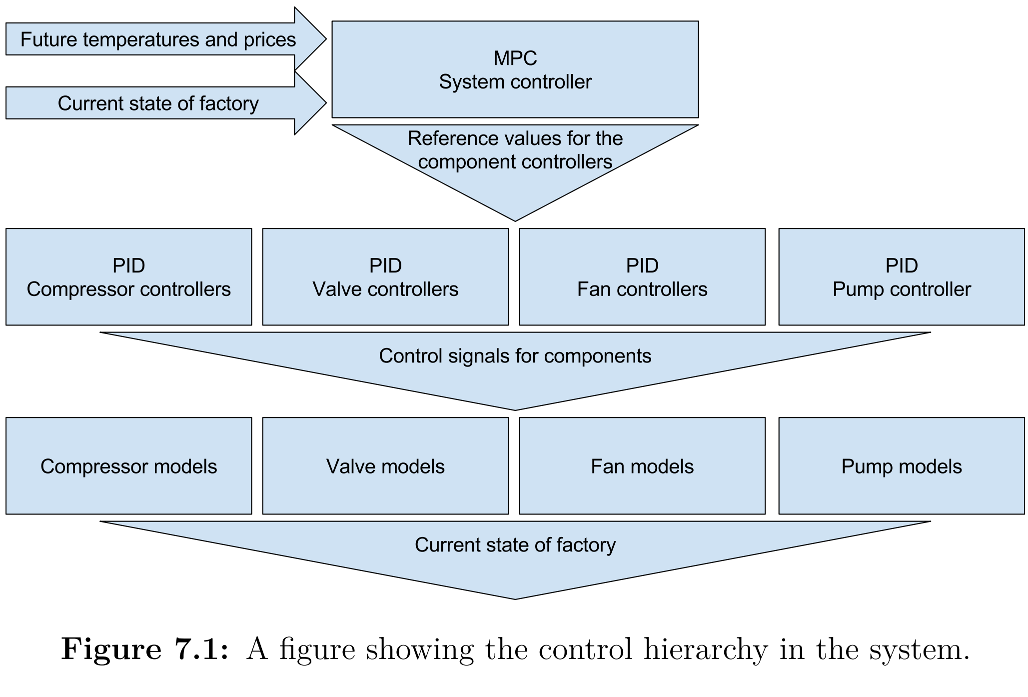

Figure 7.1: A figure showing the control hierarchy in the system.

7.1.1 System control

In order to control the thermal energy storage (TES), to ensure that the storage temperature is below the limits and to optimise the economical efficiency, an MPC is created as a top level controller for the system. The MPC utilises the dynamics described in Chapter 6 but with some parts linearised. The optimisation variables are the effects $\dot{Q}$, temperatures T and the stored energy in the TES Q_{TES}. By controlling these variables, the system controller is controlling all the components in the system.

The system controller has knowledge of future defrosting requirements, product flows, electricity prices and weather forecasts and uses these to minimise the running cost.

Minimisation objectives

The problem is formulated on an Economical Model Predictive Control (EMPC) form which is a variant of MPC where the classic quadratic objective function is exchanged with the total expenditure over the prediction horizon as in text *EMPC and the impact of forecasting* [40]. That is, the objective is to minimise the cost of buying the electricity needed by the refrigeration system. There are however also secondary objectives which are explained below.

In order to efficiently formulate an MPC objective, the problem needs to be convex. This is mainly because convex problems are very quick to minimise. To achieve this, the models were linearised around the operating points and some simplifications, as is mentioned in Section 6, were made. In order to be able to prioritise the different objectives, the different parts of the minimisation problem are all multiplied with a weighting factor Θ.

The four different parts, Γ, of the minimisation problem is formulated as follows:

First the total energy cost is minimised by multiplying the used energy E with the prices at each time instance. $E_{n,sum}$ is calculated in Equation 7.1.1 as a combination of expressions that are functions of affine expressions like Equation 7.6. As this affine expression is then multiplied with two constant parameters, the expression is affine and hence convex.

$$\Gamma_{i,\text{Price}} = \Theta_{\text{Price}} \cdot \text{Prices}_i \cdot E_{i,\text{Sum}} \tag{7.1}$$

where

$$E_{i,\text{Sum}} = E_{i,\text{Charge}} + E_{i,\text{R717}} + E_{i,} + E_{i,\text{StorageFan}} + E_{i,\text{CondensorFan}} + E_{i,\text{Defrost}}.$$

is the sum of all the electricity used in the system.

In addition it is also necessary to put a penalty on changes from the desired temperature as the temperature is allowed to increase when defrosting or when electricity is particularly expensive and decrease when electricity is cheap. The temperature does however need to stay around the desired temperature on average as the quality of the fish will otherwise deteriorate. This is done in Equation 7.2.

$$\Gamma_{i,Ts} = |Ts_{i,\text{Desired}} - Ts_{i,\text{Actual}}| \tag{7.2}$$

As the generated optimal mass flows calculated by the MPC algorithm are then executed using PID controllers, the values might occasionally miss the optimal value and hence fall outside the MPC constraints. To avoid this problem the constraints on both storage temperature and TES balance were relaxed as can be seen in Equations 7.9 and 7.10. To keep the constraint breaches to a minimum, the relaxation variables α were minimised in Equation 7.3.

$$\Gamma_{i,\alpha} = \Theta_{\alpha,\text{Ts}} \cdot \alpha_{i,\text{Ts}}^2 + \Theta_{\alpha,\text{TES}} \cdot \alpha_{i,\text{TES}}^2 \tag{7.3}$$

As the compressors run most efficiently at a set mass flow, it is desirable to minimise the changes in compressor mass flows and hence minimise the amount of different levels at which the compressors operate. This is done in Equation 7.4.

$$\Gamma_{i,\text{Comp}} = \Theta_{\text{Comp}} \cdot |\dot{m}_{i-1,\text{Charge}} - \dot{m}_{i,\text{Charge}}| \tag{7.4}$$

As all weights Θ are positive and the absolute value of any affine expression is convex, all the objectives are convex.

All the Equations 7.1 to 7.4 are then combined to one minimisation function

$$\text{Minimise} \quad \sum_{i=1}^{ph} (\Gamma_{i,\text{Price}} + \Gamma_{i,\text{Ts}} + \Gamma_{i,\alpha}) + \sum_{i=2}^{ph} \Gamma_{i,\text{Comp}} \tag{7.5}$$

where ph is the prediction horizon.

Constraints

The minimisation in Equation 7.5 is subject to a number of constraints as can be seen in the equations below. The constraints are mostly based on the functions from the modelling in Section 6 and calculated as functions of the mass flows.

The next set of constraints are to set the mass flows in relation to each other and to the required cooling effects in the system. First the mass flow from the R744 compressors, $\dot{m}_{Charge}$, is set to fulfill the cooling effect requirement from the TES, $\dot{Q}_{Charge}$:

$$\dot{m}_{i,\text{Charge}} = \frac{\dot{Q}_{i,\text{Charge}}}{(1 - x_{i,\text{R744}}) \cdot \Delta h_{\text{R744}}(-40°, 10.05\,\text{bar})}, \tag{7.6}$$

where Δh is the heat of evaporation at different conditions and x is the vapour ratio of the liquid vapour mix as some of the liquid will evaporate before the evaporators due to losses in the system.

After this the mass flow of R717 into the cascade heat exchanger is set to be the same as the difference between the cooling effect requirements from the high temperature consumers which is calculated in Section 6 and the total cooling effect of the R717 system.

$$\dot{m}_{i,\mathrm{R717,HX}} = \frac{\dot{Q}_{i,\mathrm{R717}} - \dot{Q}_{i,\mathrm{HtConsumers}}}{(1 - x_{i,\mathrm{R717}}) \cdot \Delta h_{\mathrm{R717}}(-21^\circ, 1.66\,\mathrm{bar})} \tag{7.7}$$

Next the cooling effect from the equation above is set to match the cooling effect from the other side of the cascade heat exchanger.

$$\dot{m}_{i,\mathrm{R717,HX,Liquid}} = \frac{\dot{m}_{i,\mathrm{Charge,Liquid}}}{\frac{(1-x_{i,\mathrm{R717}})\cdot\Delta h_{\mathrm{R717}}(-21^\circ,1.66\,\mathrm{bar})}{(1-x_{i,\mathrm{R744}})\cdot\Delta h_{\mathrm{R744}}(-16^\circ,20\,\mathrm{bar})}} \tag{7.8}$$

In the last step of the mass balance for the cascade heat exchanger, the total mass flow of R717 is constrained to correspond to the mass flows through the heat exchanger and the high temperature consumers.

$$\dot{m}_{i,\mathrm{R717}} = \frac{\dot{Q}_{i,\mathrm{HX}} + \dot{Q}_{\mathrm{i,HtConsumers}}}{(1 - x_{i,\mathrm{R717}}) \cdot \Delta h_{\mathrm{R717}}(-21^\circ, 1.66\,\mathrm{bar})}$$

In the next part of the mass balances the mass flow from the TES is set to match the requirements from the low temperature consumers.

$$\dot{m}_{i,\mathrm{Discharge}} = \frac{\dot{Q}_{\mathrm{i,Discharge}}}{(1 - x_{i,\mathrm{R744}}) \cdot \Delta h_{\mathrm{R744}}(-40^\circ, 10.05\,\mathrm{bar})}$$

And finally the mass flow in the defrost system is set to correspond to the requirements from the defrost consumers which is pre-calculated.

$$\dot{m}_{i,\mathrm{Defrost}} = \frac{\dot{Q}_{i,\mathrm{PlateFreezerDefrost}} + \dot{Q}_{i,\mathrm{StorageDefrost}}}{(1 - x_{i,\mathrm{R744}}) \cdot \Delta h_{\mathrm{R744}}(5^\circ, 39.71\,\mathrm{bar})}$$

After this the constraints for the different heat flows $\dot{Q}$ were defined as can be seen below:

$$\begin{aligned}
&\dot{Q}_{i,\mathrm{Required,FreezingRoom}} + \dot{Q}_{i,\mathrm{Storage}} \geq \dot{Q}_{i,\mathrm{Discharge}} \geq \dot{Q}_{\mathrm{Discharge,MAX}} \\
&\dot{Q}_{i,\mathrm{Required,HtConsumers}} \geq \dot{Q}_{i,\mathrm{R717}} \geq \dot{Q}_{\mathrm{R717,MAX}} \\
&0 \geq \dot{Q}_{i,\mathrm{Charge}} \geq \dot{Q}_{\mathrm{Charge,MAX}} \\
&0 \geq \dot{Q}_{\mathrm{i,Storage}} \geq \dot{Q}_{\mathrm{Storage,MAX}}
\end{aligned}$$

This means that all the cooling effect are constrained within their physical limits which are set by compressors and pumps. All the effect can however vary within their constraints, which for example means that the storage can cool down more than what is actually required to keep the temperature and hence better be able to counteract temperature increases from defrosting.

The next constraints are on the amount of heat stored in the TES. The limits are in reality on storage space for gas, and total amount of R744 in the system, but in this equation it is simplified as stored energy. The changes in heat are based on the heat transport to the TES from the low temperature consumers and defrosting and in the heat transported from the TES by the R744 system. The constraints in this equation were relaxed with α as was explained in the beginning of this section.

$$\begin{aligned} &Q_{i,\text{TES}} = Q_{i-1,\text{TES}} + \dot{Q}_{i,\text{Charge}} - \dot{Q}_{i,\text{Discharge}} - \dot{Q}_{i,\text{Defrost}} \\ &Q_{i,\text{TES}} \geq Q_{\text{TES,min}} + \alpha_{i,\text{TES}} \\ &Q_{i,\text{TES}} \leq Q_{\text{TES,max}} + \alpha_{i,\text{TES}} \end{aligned} \tag{7.9}$$

The last constraints are based on the temperature calculations from Section 6.2 and both the removed heat from the storage, and the temperature is calculated in order to make sure that the temperature is within limits.

$$\begin{aligned} &\Delta Ts_i = \frac{\dot{Q}^-_{i,\text{Storage}} + \dot{Q}^-_{i,\text{Defrost}} - \dot{Q}^-_{i,\text{Required,Storage}}}{m \cdot c_p} \\ &Ts_i = Ts_{i-1} + \Delta Ts_i \end{aligned} \tag{7.10}$$

These equations together create reference values for the compressors, fans and pumps for a set prediction horizon ph.

7.1.2 Component control

As the system controller can only run intermittently due to computational limits and only simulates linearised models, it will not control the individual component but only generate reference values for the component controllers. The individual components such as the valves that control the flow of refrigerant to the rooms will instead be controlled by PIDs that try to maintain the value set by the MPC. These PIDs are tuned using the Ziegler-Nichols method and little time is spent on making them perform optimally. The main purpose of these controllers in the simulations is to add uncertainty into the model as the components will not always reach the values set by the system controller and thus making the simulations more realistic. This makes it possible to determine if the system is relatively robust against minor mistakes by low level controllers.

7.2 Simulation

The models and control algorithms above are simulated using external data and the system parameters decided in the system design section. The simulations were carried out for the first two weeks in January and August in the year 2015. This was done to be able to compare summer weeks to winter weeks since the energy requirements as well as electricity prices will vary with temperature and season.

In all of the simulations the same key simulation parameters were used as can be seen in Table 7.1.

Table 7.1: Key simulation parameters used in every scenario

Parameter	Value
Time step	60 s
Prediction Horizon	11 to 35 h
MPC execution interval	1800 s
Wpr	1
Wts	1
WrTes	100
WrTs	100
Wcomp	1.2

7.2.1 Algorithm

The simulations were made in `Matlab` using the `CVX` package, which is used to solve convex optimisation problems. The advantage of using `CVX` is that changes can be made very easily and it is easy to change parameters and constraints. One downside is however that the program can be significantly slower than formulating the problem on the standard form and using for example `quadprog`.

The solver that was used was the `MOSEK` solver which can be utilised with an academic license. This solver showed to be significantly faster and more robust than the other included options. The optimisation problem had approximately 20 000 constraints and 35 000 scalar variables for a horizon of 24 hours and took 0.73 seconds to solve on a `core i5, 16Gb RAM` computer for each iteration.

7.2.2 Data

The data used in the simulation was taken from external sources and then modified for different simulations. To simulate knowledge of future weather, data from *SMHI* for 2015 was used [42]. This data was then only used for the current prediction horizon, assuming perfect weather forecasts.

The Electricity prices were taken from *Nordpool* [43], also here for 2015 to correlate with the temperatures. As the prices are published at 13:00 every day for the following day, the knowledge of future electricity prices will vary between 11 and 35 hours. As the MPC need this information to calculate the optimal reference signals, the prediction horizon will also vary. At 12:00 the prediction horizon will be 12 hours as no information is available after 23:59 the same day. At 13:00 however, the MPC will have a prediction horizon of 35 hours as electricity prices for the next day has been released.

As the prices were taken from *Nordpool* they do not include the profits of the distributors, the electricity certificate fees and the monthly fee which is usually required.

7.2.3 Simulation of old system

The original system with only an R717 refrigeration cycle and no MPC or TES was simulated to be able to benchmark the designed system. It was simulated for two weeks in January and the temperature in the storage and system COP is plotted in Figure 7.2. The average COP for the January weeks was 1.1, and for August 0.7

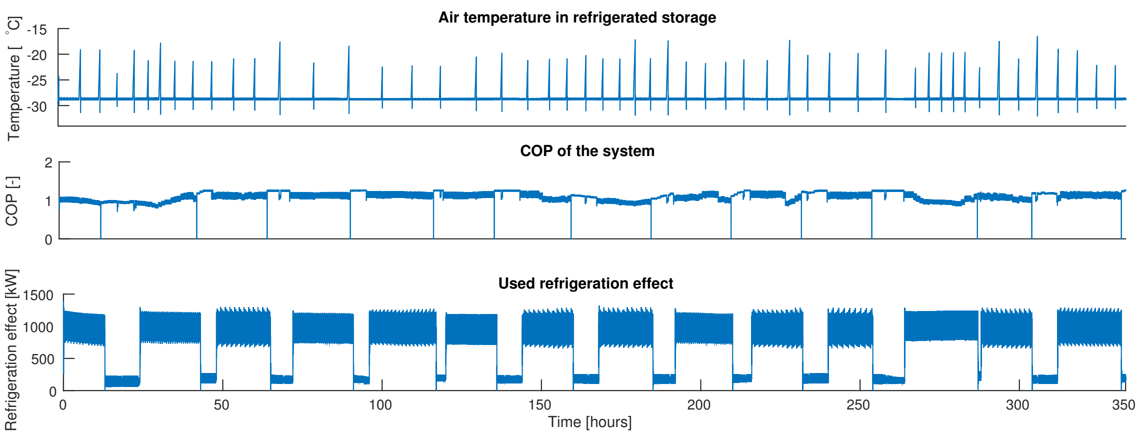

Figure 7.2: The temperature, COP and used refrigeration effect of the old refrigeration system.

As can be seen, the COP is low and the temperature in the refrigerated storage fluctuates severely when defrosting. To counter this rapid temperature change, the fans need to work on full power to remove the heat induced by the defrosting.

7.2.4 Simulation of designed system

The model was first simulated using the parameters set in the system design Section 5.2 and a TES size of 100 m^3. The simulation was done for the first two weeks in January 2015. The most interesting results from this simulation was the temperature in the refrigerated storage, the stored energy in the TES and the corresponding electricity prices which can all be seen in Figure 7.3.

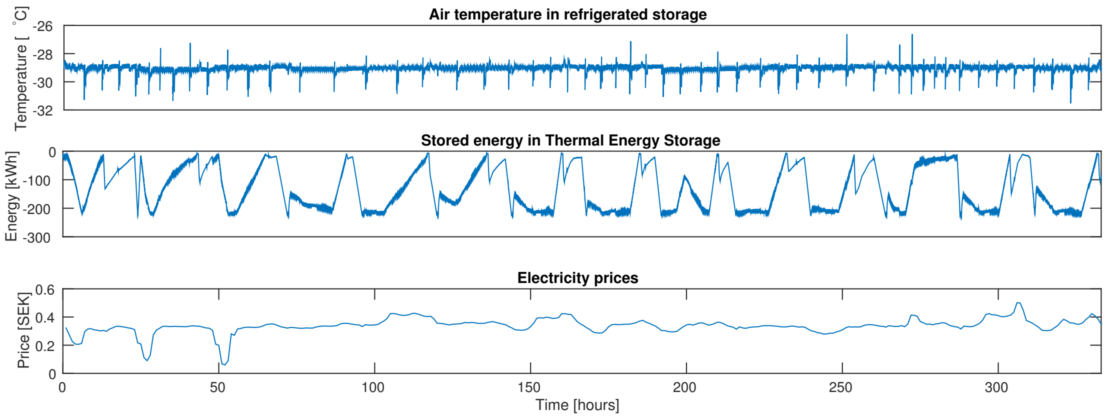

Figure 7.3: The temperature and stored effect in the TES of the designed system and the corresponding electricity prices.

As can be seen in the figure, the TES is charged when electricity prices are low and is then used when they increase again. It is also used to absorb the small oscillations which occur due to changing energy demand by the consumers. The spikes in air temperature are from the heat dispersion during defrosting, and these are also sometimes countered by a rapid temperature decrease just before to keep the average temperature around −29 °C. If compared to the old case, the MPC keeps the temperature much more steady with a maximum temperature of under −26 °C instead of the −18 °C which can be seen in Figure 7.2 for the old system.

The produced refrigeration effect of the different parts of the cascade system are shown in Figure 7.4. As can be seen both the R717 and the R744 charging system remain rather stable at two different levels due to the TES system; there are some variations but these are rather small. The TES discharge effect does however vary a lot as the requirements of the freezing process are rapidly changing.

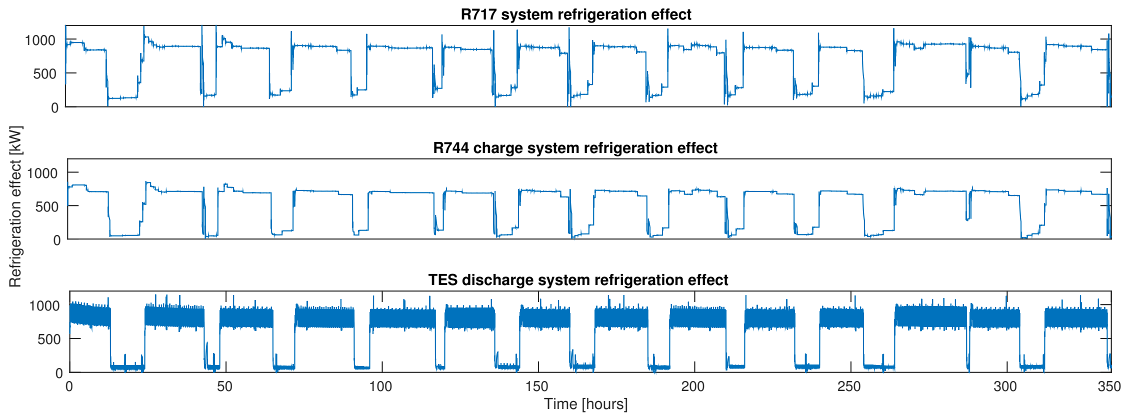

Figure 7.4: The refrigeration effect in the different parts of the cascade system.

The designed system reaches a COP of 0.84 in August and 1.52 in January which is due to the adjustable condenser temperature. In the summer the condenser temperature is around 20 °C and in the winter it is locked to a minimum of 6 °C to avoid ice building up on the condenser. The COP of this system is hence higher than for the old system, yielding a significant energy use reduction of 40%.

7.2.5 Scenario 1 - Projected energy prices

In order to analyse the effect of future electricity prices on the designed system, several simulations were done by changing two parameters. These parameters are a price variation factor (VF) and a price increase factor (PI). How the parameters were used to change the electricity price can be seen in in 7.11 and 7.12 respectively. The PI-values are based on future energy predictions by *Bixia* [2] and the VF-values are altering between 1 and 2. The PI-values represent year 2019, 2020 and 2030 respectively.

$$\begin{aligned} Prices_{i,\text{Variation}} &= Prices_{i,\text{Old}} - \frac{\sum_{i=0}^{t} Prices_{i,\text{Old}}}{t} \\ Prices_{i,\text{Amplified}} &= VF \cdot Prices_{i,\text{Variation}} \\ Prices_{i,\text{New}} &= Prices_{i,\text{Amplified}} + \frac{\sum_{i=0}^{t} Prices_{i,\text{Old,}}}{t} \end{aligned} \tag{7.11}$$

$$Prices_{i,\text{New}} = Prices_{i,\text{Old}} + PI \cdot \frac{\sum_{i=0}^{t} Prices_{i,\text{Old,}}}{t} \tag{7.12}$$

In Table 7.2 a comparison between the old system and the improved system using different electricity prices can be seen. Looking at the projected prices for year 2030 using the large variation factor, the running cost savings from implementing the improved system is 3801 SEK/d on average during the January simulation. By using the average daily savings from January and August for half a year respectively, the average daily savings for a whole year is approximated. Assuming that the systems runs for 365 days per year the cost savings would be about $1.5\,\frac{\text{MSEK}}{\text{y}}$. The same number with today's electricity prices is circa $0.7\,\frac{\text{MSEK}}{\text{y}}$.

The cost savings are largely due to an improved system coefficient of performance. This can be understood when the comparing the running cost for the old- and improved system between the different price cases. The daily cost reduction is around 60 % for the cases with price increases of 5 % or 62 % and 1 or 2 in variation. Cost reductions from using the TES to run the compressors during times of cheaper electricity prices and shut them off during high price periods can probably not be fully facilitated. The probable reason for this is TES size which needs to be very large in order to save enough energy when prices are low. The impact of TES size is evaluated in Section 7.2.7.

Table 7.2: Increased prices and price variation's effect on running cost for two weeks in January 2015

Price increase [%]	Variation factor [-]	Running cost [$\frac{SEK}{d}$] Designed system	Running cost [$\frac{SEK}{d}$] Old system
0	1	3312	5501
5	1	3436	5704
38	1	4240	7043
62	1	4835	8017
5	2	4338	7226
38	2	5146	8565
62	2	5738	9539

7.2.6 Scenario 2 - Different Thermal Energy Storage capacities

In order to determine what TES storage capacities are required to minimise the cost, several simulations were run under the same conditions with different maximum capacities. These different simulations were then compared with regard to average daily running cost in Table 7.3.

Table 7.3: Running cost of the refrigeration system at different TES capacities and maximum charge effects in two weeks in January and August 2015.

Capacity[m^3]	Running cost (Jan) [$\frac{SEK}{d}$]	Running cost (Aug) [$\frac{SEK}{d}$]
6	3435	1717
14	3328	1658
50	3324	1655
100	3312	1650
1000	3225	1607
3000	3170	1574

As can be seen in Table 7.3 above, the savings of choosing the largest TES instead of the smallest was approximately 8% with today's prices, corresponding to around 74 500 SEK in savings per year. More interestingly, the saving between 6 m^3 and 14 m^3 capacity were 3% or 30 300 SEK. This shows that the saving yielded with an initial increase of 8 m^3 were almost the same as an additional increase of 2986 m^3. As investment cost and complexity increases with size, a capacity choice around 10 m^3 should be appropriate. It is however possible that the largest TES capacities would yield more savings if the system was redesigned with increased compressor effects. This would increase the amount of energy that could be charged when prices were low.

By comparing the system behaviour of three different capacities 10, 100 and 1000 m^3 in Figure 7.5 it could also be seen that the usage of the TES varies when the capacity is changed. At the smaller capacity, the TES is mostly used to counteract the inherent variations of the refrigeration plant. When the capacity increases the TES is in addition used to store energy when it is cheap, and spend it when it is expensive.

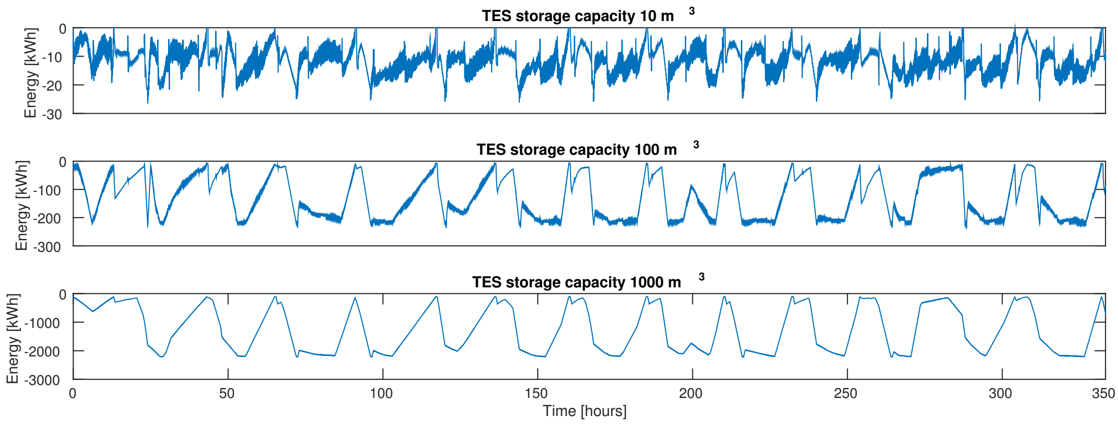

Figure 7.5: The behaviour of the TES at different maximum capacities.

7.2.7 Scenario 3 - Different Thermal Energy Storage capacities and projected energy prices

In Scenario 2 different TES capacities were tested for economical efficiency with respect to the energy prices of 2015. But as was stated earlier in this report these prices are expected to increase in the coming years, and to simulate how the system would behave in such a scenario, different TES capacaties were also evaluated for the expected prices of 2030 which can be seen in Section 7.2.5. This is summarised in Table 7.4.

Table 7.4: Running cost of the refrigeration system at different TES capacities in two weeks in January with projected electricity prices for 2030.

Capacity[m^3]	Running cost [$\frac{SEK}{d}$]
6	5890
9	5779
14	5764
50	5755
100	5738
500	5650

It can be seen in Table 7.4 that some savings can be achieved by increasing the TES size up to 500 m^3 as saving compared to the smallest size are 4%, this is however not enough to warrant the increased investment. There are however large relative gains from increasing the size from 6 m^3 to 9 m^3, so a minimum TES size of 9 m^3 is recommended in this type of system to be able to counter the rapid changes in the energy requirement.

7.2.8 Scenario 4 - Changed temperatures in refrigerated storage

In this scenario the model was simulated for six different storage temperatures. The simulation was done to evaluate the energy use impact of higher storage temperatures. A higher temperature requires a trade-off between product shelf life and energy use. The higher temperature decreases the driving forces and thus the transmission losses. The shelf life dependency of temperature is found in Section 2.1

The results from a two week simulation can be seen in Table 7.5. The running costs increase slightly with a lower storage temperature. The energy use difference of 2 °C, between -24 and -26, are just about 138 kW h. But with decreasing temperatures the energy use difference increase as can be seen when comparing the −30 °C and −32 °C cases. Here a 2 °C decrease of storage temperature corresponds to around 362 kW h in energy use. The energy savings by increasing the storage temperature from −32 °C to −24 °C is 5.8 %for the August period. With the 2015's electricity prices this is a cost reduction of merely 200 $\frac{\text{SEK}}{\text{d}}$. In case of large increases in the electricity prices a reduction in temperature could potentially be economic viable in the future. As of today, it is likely that the up to 50 % decrease of shelf life cost more in terms of lost customers, than the electricity cost savings by increasing the temperature.

Table 7.5: Evaluation of storage temperature on running cost and electricity use for two weeks in January

Temperature [°C]	Energy use [$\frac{\text{kJ}}{\text{d}}$]	Running cost [$\frac{\text{SEK}}{\text{d}}$]
-24	9612	3207
-26	9750	3254
-28	9852	3288
-30	9996	3338
-32	10207	3409

8

Conclusion and further work

8.1 Conclusions and discussion

The results of the study show that very large savings are possible from implementing cascade systems with MPC control in refrigeration plants of the type studied compared with conventional R717 systems. The savings are around 40% with an average thermal energy storage size. This is mainly due to the better system COP that is achieved through running the compressors at a stable level with the model predictive control, and to the higher efficiency of R744 at low temperatures.

The ability to store energy at low electricity prices and use it when prices are high does also yield savings, but these are rather small. It might be possible to gain additional savings by further increasing the capacity of the thermal energy storage but the problem is that with the solution presented in this thesis it would be rather impractical and expensive to construct a gas storage tank of that size. There are however alternative solutions such as using a freeze-thaw cycle with salt water, which could store this amount of energy fairly cheaply. These solutions do however introduce other problems with charge and discharge and are not covered in this thesis.

When optimising running costs, the investment cost needs to be taken into consideration. The major costs in this solution is the refrigeration system with compressors and thermal energy storage as the largest parts. There are of course also many smaller components like pumps, valves and piping, but these should be similar in all refrigeration solutions and are not particular expensive in this case. The possible higher investment cost of a cascade R717/R744 are not likely to outweigh the savings that are possible due to higher efficiency for this application.

Developing control systems do also require some time and effort; the system need to be thoroughly modelled and computers are needed to compute control signals. This cost could be somewhere between 400 000 and 700 000 SEK depending on the system complexity [44]. A control system is however always required and that means that a part of the cost will always be there, even though the implementation of model predictive control does of course increase it. The gains from model predictive control in this study was mainly due to the load balancing of compressors. This can be done in other ways but model predictive control is definitely a good option.

8.2 Further Work

The study gives an overview of the different processes in fish refrigeration systems and the use of model predictive control and should in the next step include more detailed studies. The work has gone reasonably well even though the extent of the study should perhaps have been more limited from the beginning. Since so many factors were included, a lot of simplifications were necessary, and even though all the assumptions were founded in facts, some errors might still have been introduced.

It was rather surprising that the ability to store energy with regard to future energy prices did not have a larger economic impact, but as the total energy storage capacity relative to the energy use of the system was rather small it is logical. Further studies could for example investigate if the thermal energy storage capacity could be increased more than in this thesis using other storage methods, and how this affects the cost efficiency of the system. An investigation of the effect on economic savings by varying the discharge size of the thermal energy storage would also be interesting.

A more thorough study into the application of model predictive control on refrigeration plants might show savings that were not discovered in this case. For example it is possible that applying model predictive control on compressors can be very efficient, and this could perhaps also be applied on valves and pumps. In this study, simple PID-regulators were used to control the components and this might have introduced unreasonably inaccurate control and hence yielding smaller savings than would actually be possible. As the available computing power were limited, large time steps were used in the simulations. This is a factor which might have decreased the performance of the PID-regulators considerably.

The use of `CVX` to solve the convex optimisation problem might not be the fastest solution. Some tests done by the authors indicated that reformulating the problem to standard form and solving it with e.g. `quadprog` in `Matlab` could be significantly faster. This would enable longer simulation periods and give more opportunities to test different solutions; in this thesis the analysis was very limited by the time it took to simulate each change before being able to evaluate it.

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