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Kristina Norne Widell

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— towards an optimal operation of compressors and air fans

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Abstract

Fish is one of Norway's main exports, and can be shipped fresh, frozen or dried. This thesis examines the freezing of fish in batch tunnels and ways to increase the energy efficiency of this process. A fish freezing plant on the west coast of Norway was used as a baseline case and measurements were made of the freezing system. Different aspects of this system were simulated, mainly using MATLAB.

The focus was on the compressors and the freezing tunnels of an industrial refrigeration system. The compressors and the freezing tunnel fans are the largest consumers of electricity, but they are often not operated at the highest efficiency. An analysis of the compressor operation showed that it was far from optimal, with several compressors often operating at part-load simultaneously. These were screw compressors regulated by slide valves, which provide easy capacity control, but also have low energy efficiency. The refrigeration system had several different sized compressors, and the results showed that it was possible to run the system with only one compressor at part-load operation. The total coefficient of performance was improved by as much as 29% for a low production period. A further analysis showed that installing a variable speed drive on one compressor would also improve energy efficiency and make capacity regulation straightforward.

The freezing system included five batch freezing tunnels, each of which had a freezing capacity of more than 100 tonnes of pelagic fish. A typical freezing period lasted typically 20 h and decreased the fish temperature to -18°C or below. The main task was to develop a computer program that could simulate the freezing process and the refrigeration system and locate opportunities for improvement. The air velocities inside the freezing tunnel varied with location, which were pinpointed using the computational fluid dynamics software program Airpak. These velocities were used in freezing time calculations. It was shown that a guide blade installed in the air flow at a critical location improved the air velocity distribution compared with no guide blade. Without the guide blade, the freezing times of the products

were between 16 h and 32 h, but with a guide blade they were between 17 h and 21 h, a span of only 4 h. These freezing times were calculated with a modified Plank's equation.

A numerical model was programmed in MATLAB and it was used to simulate the temperatures of the products. The model was a two-dimensional finite difference approximation of the heat conduction equation. The simulation results were compared to measured temperatures, to validate the model. The measured temperatures were also used for validation of another simulation program, programmed in Modelica.

The final stage of this research involved testing different alternatives for reduced fan operation. The program for the product model was extended with models for calculating the energy consumption of the air fans and the compressors and using Airpak-simulated velocities. The air fan speed was reduced to 83%, 67%, 50% and 33% of full air fan speed. This was tested at 5 different points during the freezing period, to see how the freezing times were affected. Full air speed during the freezing period resulted in a total freezing time of 20 h. A reduction in air fan speed to 33% after 8 h resulted in an increase in total freezing time of 10 hours (47% longer) but reduced energy consumption to 73.8% of the baseline case. An alternative with only 4 h longer freezing time resulted in an energy consumption of 80.5% of the baseline case. It was assumed that the fans had variable speed drives. The effect of reduced air inlet temperature was also tested and the results show that this can reduce freezing times. The effect on the total energy consumption was not large and also depends on the rest of the refrigeration system.

Issues raised by this thesis are relevant for future research. It is suggested that the main simulation program is expanded by incorporating more detailed models of the refrigeration system. Dynamic operation of the air fans is also a possibility, for example to gradually reduce fan speed with decreasing product heat load.

Acknowledgements

Firstly I want to thank my supervisor Trygve M. Eikevik for his guidance and support during my work and for the opportunity to study this very interesting subject. I also wish to thank my other colleagues at NTNU and SINTEF for advice and inspiration, you made my work less solitary. I want to thank Ole Kristian Solheim and Jon Arve Engebakken, whose master's theses have provided important background information for my work. I would also like to thank Glenn Halnes and Arve Saltkjel at Norway Pelagic for their help and friendly answers to all my questions.

I am also grateful to my family and friends, thank you for your support and for pleasant lunches, dinners, hikes, holidays, journeys and much more.

A special thanks to Armin for your positive and encouraging support in discussing my work and seeing the larger perspective. Last, but most important, thanks to my wonderful children, Heidi and Simon, I'm so proud of you.

Contents

List of papers

This thesis is based on 6 papers, listed below. They are referred to in the text by name and year or by Roman numerals. The papers can be found in appendices I-VI.

I. K. N. Widell and T. Eikevik. *Reducing power load in multi-compressor refrigeration systems by limiting part-load operation.* In: IIR Gustav Lorentzen Conference. Copenhagen, Denmark, 2008.

II. K. N. Widell and T. Eikevik. *Reducing power consumption in multi-compressor refrigeration systems.* International Journal of Refrigeration 33 (1). 2010.

III. K. N. Widell and F. Frydenlund. *Air velocity field in an air blast freezing tunnel.* In: Deutche Kälteverein (DKV) Tagung. Berlin, Germany, 2009.

IV. H. T. Walnum, T. Andresen and K. N. Widell. *Verification of a Modelica-based dynamic simulation model for batch freezing tunnels.* In: International Congress of Refrigeration. Prague, Czech Republic, 2011.

V. K. N. Widell and T. Eikevik. *Numerical and experimental analysis of food products in an air blast freezing tunnel.* In: International Congress of Refrigeration. Prague, Czech Republic, 2011.

VI. K. N. Widell and T. Eikevik. *The effect of reduced air fan speed on freezing time and energy consumption in a freezing tunnel.* Manuscript submitted to IIR Gustav Lorentzen Conference. Delft, Netherlands, 2012.

The author's contribution

The PhD candidate (Kristina N. Widell) has been the main author of five of the articles included in this thesis and the co-author of one. Measurement data were collected by the author at a fish processing plant in Måløy (Norway) and analysed in papers I, IV and V. Different simulation functions and programs have been developed in MATLAB by the candidate and used in papers I – III and V – VI. The candidate performed the analysis and interpreted the results of papers I – III and V – VI.

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Nomenclature

Roman Symbols

A	area $[m^2]$
c	specific heat capacity $[J/kgK]$
d	thickness $[m]$
h	enthalpy $[kW/kg]$
h	heat transfer coefficient $[W/mK]$
k	thermal conductivity [W/(mK)]
L	latent heat $[J/kg]$
m	mass $[kg]$
p	pressure $[bar]$
V	volume $[m^3]$
$\dot{Q}$	heat flow or load $[kW]$
q''	heat flow $[kW/m^2]$
t	time $[s]$
U	overall heat transfer coefficient [W/(mK)]
v	velocity $[m/s]$
$\dot{W}$	work $[kW]$

Greek Symbols

ΔT	temperature difference [°C]
η	isentropic efficiency [-]
ρ	density $[kg/m^3]$
σ	Stefan-Boltzmann constant = 5.67 W/m^2K^4

ε	emissivity [-]

Subscripts

a	air
c	convection
$comp$	compressor
$evap$	evaporation
f	frozen or freezing
i	initial
if	initial freezing
nat	natural
out	outlet mass-average condition
$oval$	oval surfaces
p	packaging
$plan$	planar surfaces
$prod$	product
r	radiation
s	ideal, isentropic process
s	surface
sur	surroundings
u	unfrozen

Abbreviations

CFC	Chlorofluorocarbon
COP	Coefficient of performance
GWP	Global warming potential
$HCFC$	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
ODP	Ozone depletion potential

Chapter 1

Introduction

1.1 Thesis overview

This thesis is written as a collection of articles. It includes one journal paper, four conference papers (all of which were subject to review) and one manuscript sent to a conference. Paper I was published at the 2008 Gustav Lorentzen conference in Copenhagen. Paper II was published in the International Journal of Refrigeration in 2010. Paper III was published at DKV Tagung 2009 (the annual meeting of the German Refrigeration and Air-conditioning Association). Papers IV and V were published at the International Congress of Refrigeration in Prague 2011. Paper VI was sent to Gustav Lorentzen conference that will take place in Delft 2012.

The background for the research and status of knowledge are provided in Chapter 1. Industrial refrigeration and food freezing modelling are discussed in Chapter 2 and an overview OF methods for increasing energy efficiency is given in Chapter 3. Summaries of the papers are provided in Chapter 4. Results and discussion, conclusion and suggestions for further work are given in chapters 5-7.

1.2 Background

Refrigeration is one of the best possible ways of preserving food as close to its original state. However, large volumes of food, 15% to 50%, are estimated to be lost (Singh, 2011) and improvements to the food cold chain and better use of refrigeration can help to reduce these numbers. When the product temperature is decreased, microbiology activity and chemical and physical changes are slowed down and shelf life is extended.

Table 1.1: *Energy consumption in kWh/tonnes for the pelagic fish industry in Norway (Enova, 2003) and (Enova, 2009).*

year	**average**	**lowest**
2002	218	163
2008	195	118

Global energy consumption is constantly growing and ways to meet both present and future demands are needed. Increasing energy efficiency is often less expensive compared to developing new energy production plants. Industry plays an important role in this, since it represents a significant part of the total energy consumption, and many processes are less efficient than they could be. There are at least 100 000 refrigeration plants in Norway (household refrigerators and freezers excluded) which consume at least 6 TWh per year (Røsvik et al., 2008). In the fish industry, refrigeration is used for cooling, freezing and drying of fish and fish products. Table 1.1 shows that energy consumption per kilogram of processed fish varies significantly between different fish processing plants. The numbers also indicate improvements from 2002 to 2008, both for the best plant and for the majority of the plants. Electricity prices are expected to continue to increase, and the industry has to continue working on improving processes and increasing energy efficiency.

The barriers for making a system more energy efficient relate to economy and uncertainty. Extra capital costs could delay or prevent the installation of new equipment, such as a variable speed drive or a better control system. In addition, the amount of savings from different investments or the length of the payback time may be uncertain. Delaying or stopping production during a hectic season is also not desirable. The industry needs clear guidelines about which changes can be made and how much can be gained. Policymakers also need to be aware of the need for change and the resulting gains, so that appropriate laws can be made or enforced. Research in this field can illuminate the relationship between operational alternatives and associated benefits.

The opportunities for energy savings in the seafood industry are many. Refrigeration systems are the main electrical consumer, and are used for chilling, ice production, cold storage and freezing of fish. Possible improvements include better dimensioning of the system (currently, the design cooling load is often higher than normal operating cooling load), better system regulation of the components, and more uniform use of energy over a 24-

hour period. Energy savings can also be had by integrating refrigeration systems with heating systems. (Gjøvåg, 2004) (Aprea et al., 2007)

A refrigeration system has to be thoroughly analysed before the possibilities of electrical energy reduction can be found. An analysis can contain measurements at an existing plant, calculations and simulations. The freezing time is often used as a measure of the process, but this alone does not provide much helpful information. A transient analysis is necessary since the process changes with time. Compressors are started and stopped during the cooling and freezing process, temperatures of refrigerant, condenser water and evaporator media also change, and with them thermophysical properties. There are many different methods for predicting cooling and freezing processes, with the most appropriate approaches dependent on the process and which results are of interest.

1.3 Status of knowledge

It is difficult to describe the status of knowledge for such a broad concept as industrial refrigeration for food freezing. A few articles describing the recent research in this field are summarized here.

Food cold chains include production, transport and retailers. Several papers addressing these topics have recently been published. James and James (2011a) describes a project that focuses on improving energy efficiency of food cold-chains in the UK. The first challenge was to find adequate data, something that is probably a problem in most countries. The energy-saving potential for the top ten food refrigeration processes was least 20%, and for some processes was up to 50%. Simple changes in construction and operation are suggested as an important beginning. The authors also mention the problem of food poisoning caused by high temperatures, and that refrigeration can be an important way to prevent this problem. However, energy usage for refrigeration would thus increase, increasing the need for efficiency improvements.

A comprehensive review of emerging technologies (all alternatives to the vapour compression cycle) within the field of food refrigeration was conducted by Tassou et al. (2010). Sorption systems could be used when waste heat is available (at temperatures between 50-90°C). Existing systems are mainly for cooling, but freezing systems have also been studied. Air cycle refrigeration could be an alternative when low temperatures (-50 to -100°C) are needed. Superchilling (Magnussen et al., 2008) is when the products are cooled to 1-2°C below their initial freezing point, which can be an alternative to chilling. This type of processing extends shelf life and no extra ice is needed during transportation.

Even though ammonia has long been used as a refrigerant, it is currently the subject of a considerable amount of research. The synthetic refrigerants CFCs, HCFCs and HFCs have been used for many years, because they were seen as safer than natural refrigerants, such as ammonia, carbon dioxide and hydrocarbons. After the discovery that synthetic fluorocarbon-based refrigerants affect the ozone layer and contribute to global warming, a number of them are being phased out (Kuijpers, 2009). As a result, natural refrigerants are more popular again and their advantages are being rediscovered. Ammonia is already used in many industrial refrigeration plants in Norway. Research in this field focuses on decreasing the refrigerant charge, new components, secondary systems and blends (Ammonia Conference, 2009).

The rapid development of computers and computer software has made simulation and modelling more common in studies of energy efficiency and refrigeration. Ever more complex models can now be solved using a common desktop computer. Experiments are still important in scientific research, but simulation and modelling are becoming more and more common. Measurements can be costly and disturb production, which is to be avoided. While many mathematical models for heat and mass transfer have been developed over the years, unsolved problems remain. Good examples of this are the modelling of heat and mass transfer in stacks of produce, turbulence, and multiphase transport in porous media (Nicolaï and Pham, 2006). Care should be used in adapting existing models to new situations, however, because many factors affect their reliability.

The type of compressor that is most often installed nowadays is the screw compressor. It is more reliable and requires less service than the reciprocating compressor (Pearson, 2008c). Screw compressors may have a slide valve for capacity regulation, but variable speed drives are more energy efficient.

1.4 Aims of the study

The main objective of this thesis was to analyse and develop methods to increase energy efficiency in the fish processing industry. Many initiatives can be taken to improve efficiency in the industry, but more complicated methods need to be investigated. As a result, the focus was on the main electricity consumers in the refrigeration system, compressors and freezing tunnel fans. Screw compressors are widely used in this industry because they can work across large pressure differences and slide valves make capacity reduction simple. However, slide valve regulation is not energy efficient, which makes it a clear candidate for further investigation.

An essential part of this work has been to develop a computer simulation program for evaluating energy consumption in blast freezing tunnels. The computer program enables different alternative operations to be tested without disturbing production. A fish processing plant was selected for measurements and validation of the simulations. The processing plant had an ammonia-based refrigeration system, which is common in these types of systems. NTNU and SINTEF scientists have also focused their research on systems that use natural refrigerants.

Chapter 2

Fundamentals

2.1 Industrial refrigeration

The term industrial refrigeration is used to describe large refrigeration systems used in the food, chemical and processing industries. Industrial refrigeration normally makes use of a vapour compression process, much like air-conditioning systems and domestic refrigerators. These systems have both similarities and differences. Industrial refrigeration systems are typically built for special requirements, often have parallel components and are often extended by adding one or more components. Domestic refrigerators and air-conditioning systems are more often standardized, hermetic systems. Industrial refrigeration typically operates between 15°C and -70°C, but when temperatures are below -70°C, the process is called cryogenic freezing. (Stoecker, 1998, chap. 1)

This section first describes vapour compression systems, followed by a subsection on refrigerants and concludes with a subsection about food chilling and freezing.

2.1.1 Vapour compression systems

An example of a simple vapour compression system can be seen in Figure 2.1. $\dot{Q}_{evap}$ is heat from the refrigerated area that is absorbed in the evaporator. The compressor uses work ($\dot{W}_{comp}$) to increase the refrigerant pressure from p_1 to p_2. The difference between an ideal and an actual process is visualized in the log p-h-diagram by point 2s (ideal) and 2 (actual). The relation for the isentropic compressor efficiency η_{comp} is shown in Equation 2.1 (Moran and Shapiro, 2004, chap. 10).

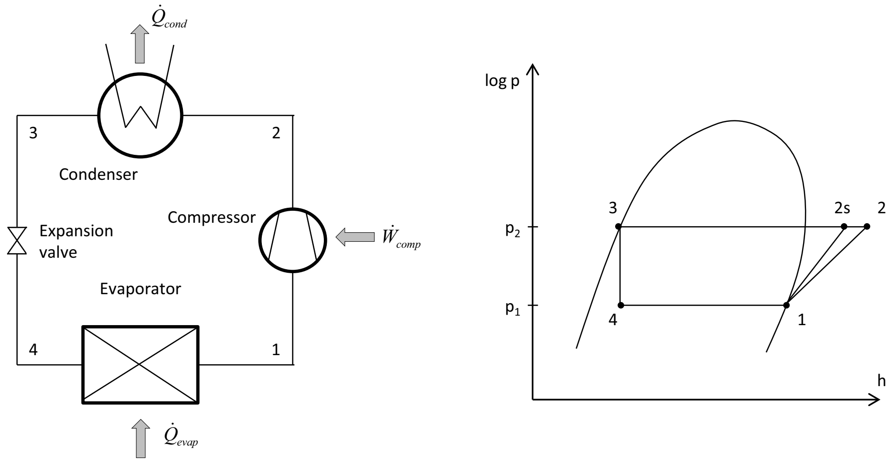

Figure 2.1: *A simple vapour compression system.*

$$\eta_{comp} = \frac{\dot{W}_{comp,s}}{\dot{W}_{comp}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)} \tag{2.1}$$

The isentropic efficiency depends on volume and pressure ratios of the compressor. Examples of isentropic efficiencies can be seen in Table 2.1. Heat is removed from the system in the condenser ($\dot{Q}_{cond}$). The refrigerant pressure is reduced from p_2 to p_1 in the expansion valve, between state 3 and 4.

The coefficient of performance (COP) is calculated with Equation 2.2 and it depends on refrigerant type, pressure levels and isentropic efficiency.

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} = \frac{(h_1 - h_4)}{(h_2 - h_1)} \tag{2.2}$$

Table 2.1: *Examples of isentropic efficiencies for an ammonia single screw compressor (ASHRAE, 2008, chap. 37).*

	Volume ratio		
Pressure ratio	**2.6**	**3.5**	**4.9**
3	81%	—	—
5	80%	77%	66%
7	70%	75%	72%
9	64%	69%	72%
11	—	64%	70%

High COP is achieved by having a small temperature difference between evaporator and condenser. Higher evaporation temperatures are also good for the COP, but are often limited by product and process requirements. However, by increasing the heat exchanger area and improving heat transfer coefficients, it is possible to achieve lower temperature differences between the refrigerant and air temperature. Condenser temperatures should also be kept as low as possible. Seawater-cooled condensers have lower fluid temperatures and better heat transfer properties than air-cooled condensers. However, they are also subject to fouling problems and need to be cleaned more often. If the condensing temperature is decreased by 2°C, the COP will improve by 4.4% and if the evaporation temperature is increased by 2°C, the COP will improve by 5.5% for an 20°C/-40°C ammonia system. From this we see that condensing temperature does not affect the COP as much as evaporating temperature, but due to ambient conditions, it can vary more than the evaporating temperature.

The simple system shown in Figure 2.1 can be extended in industrial refrigeration systems with several compressors, evaporators with liquid separators and condensers. The additional equipment that will be needed includes operating and controlling system, liquid pumps, defrosting systems, etc. Industrial refrigeration systems often consist of several compressors working in parallel between two or more pressure levels. The most common types of compressors in these systems are reciprocating and screw compressors. Screw compressors can be operated in a single stage with pressure ratios above 20:1 (ASHRAE, 2008, chap. 37). The energy efficiency of a screw compressor can be improved by using an economizer as shown in Figure 2.2. This compressor has an extra suction port at a middle pressure for flash gas from a liquid separator, the economizer. Only parts of the refrigerant have to be compressed from the lowest pressure, which results in a lower compressor power load. This arrangement is particularly good for large pressure ratio systems.

A cascade system with ammonia on the high-temperature side and carbon dioxide on the low-temperature side can be more efficient and more cost-effective than a low-temperature system that relies only on ammonia (Pearson, 2008c). Several papers describe cascade systems with NH_3/CO_2. Getu and Bansal (2008) conducted an analysis of the effect of subcooling, superheating, condensing and evaporating temperatures and isentropic efficiency on COP and mass flow ratio. They also provided models for finding optimal system parameters. Rezayan and Behbahaninia (2011) optimized parameters for a cascade system and used the annual costs as the objective function. This resulted in reduction in total costs of 9.34% compared with

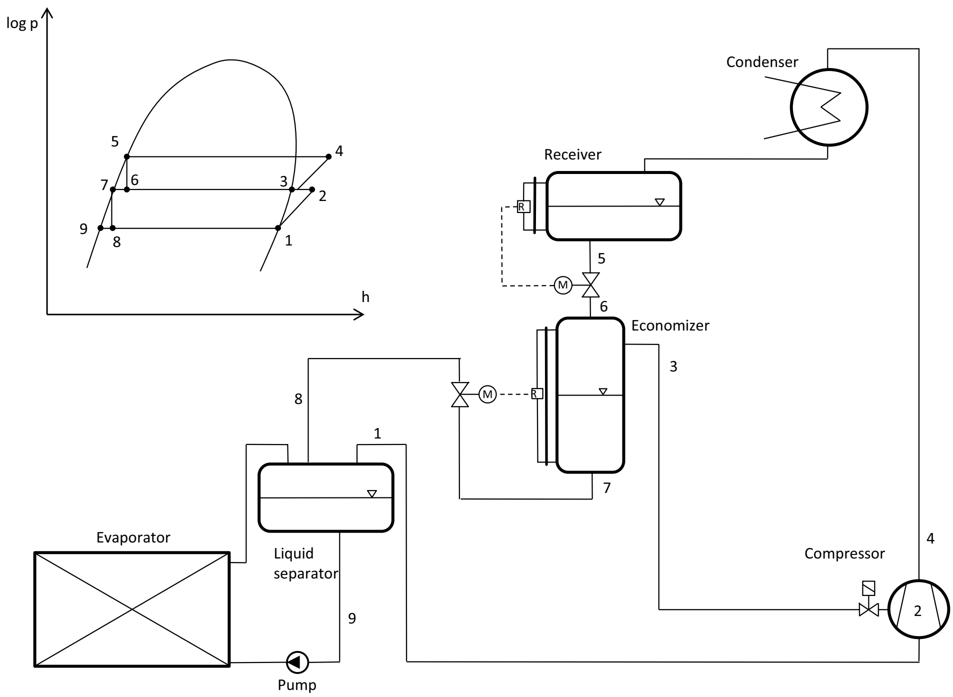

Figure 2.2: *A vapour compression system with economizer.*

Table 2.2: *Refrigerant properties (Pearson, 2008a), (Coolpack, 2001).*

		R717	R744	R404a	R22
Molar mass	[g/mol]	17	44	98	86
Normal boiling point	[°C]	-33.3	-93.7	-46.6	-40.8
Critical temperature	[°C]	132.3	31.1	72.1	96.2
Critical pressure	[bar abs]	113.3	73.8	37.4	49.9
Global warming potential		0	1	1810	3900
Ozone depletion potential		no	no	no	yes
Specific refrigerating effect (at -38/20 °C)	$[kJ/kg]$	1116	181	116	165
Pressure ratio (at -38/20 °C)		10.7	5.3	7.5	7.9

a base case. Bingming et al. (2009) built a test rig and made experimental investigations on how different parameters (evaporating temperatures, condenser temperatures, and degrees of superheating and subcooling) affected the system performance. They show that the COP of a cascade system is always higher than in a single-stage NH_3-system, and is better than a two-stage NH_3-system for evaporation temperatures below -40°C.

2.1.2 Refrigerants

The refrigeration system size and temperature levels are most important in determining which refrigerant is the most suitable. A list of refrigerant properties can be found in Table 2.2, which contains a comparison of four refrigerants for large refrigeration systems: Ammonia (R717), carbon dioxide (R744), R404a — HFC blend of R143a/R125/R134a and R22 — the HCFC chlorodifluoromethane. This last is characterized as having ozone depletion potential (ODP) and is being phased out. R404a is not characterized as having ODP, but has large global warming potential (GWP). Carbon dioxide was used in the beginning of refrigeration, but abandoned when synthetic refrigerants[1] refrigerants were introduced, around 1940. It has been reintroduced over the last two decades as a promising refrigerant.

Ammonia is a natural refrigerant that has been used in vapour compression systems for more than one hundred years, and is widely used in the Norwegian fish processing industry. In 1993, Tokle et al. investigated the use of different refrigerants in Norway. At that time, there were 2510 refrigeration plants in the fish processing and freezing industry, of which

[1]synthetic refrigerants: CFCs, HCFCs and HFCs

520 (21%) were using ammonia. The rest of the plants used CFCs (52%) or HCFSs (27%). However, the large plants used ammonia as the refrigerant and the total mass of ammonia was 1550 tonnes, which represented 96% of the total of all refrigerant being used in the industry. The number of industrial refrigeration plants with ammonia has most likely increased since the 1993 survey, because of the phase-out of synthetic refrigerants. Ammonia has no global warming potential (GWP), its atmospheric lifetime is very short, and it does not affect the ozone layer. It is therefore a good substitute for synthetic refrigerants (although it cannot be directly substituted into existing synthetic refrigeration systems).

A common concern with ammonia as a refrigerant is the safety. The toxicity of ammonia makes it best suited to industrial plants, where operating personnel are trained and extra precautions can be taken. These include the use of gas detection equipment, emergency ventilation etc. With proper maintenance, strict safety regulations and trained operating staff, accidents can be avoided and leakage can be kept to a minimum. If untrained people are exposed to leaking ammonia, panic can spread, even at low ammonia concentrations. This must be considered when the system is designed. However, the distinct smell is also a advantage, because it makes people react and leaks can quickly be prevented (Lindborg, 2009). Ammonia is also explosive, but at concentrations higher than what is likely to occur, because it is much lighter than air and will diffuse quickly (Lorentzen, 1988).

Aside from the safety issues[2], ammonia is very well suited as a refrigerant. It has very high latent heat and the refrigeration effect per unit mass flow is the highest of all refrigerants used in traditional vapour compressor systems. The critical temperature is also high, which makes it suitable for air-cooled systems. Because ammonia has low molar mass it has much higher particle velocity than all other refrigerants. A consequence of this is that the system can tolerate higher gas velocities inside pipes and other equipment, without excessive losses. Small pipe sizes can be used, but precautions have to be taken so that the expansion valve is not blocked. (Pearson, 2008b)

When the evaporation temperature is below -33.4°C, the pressure in the system is sub-atmospheric, which can allow air to leak into the system. Air inside an ammonia system can be removed by purgers, if not, it will accumulate in the condenser and the cooling effect will decrease. The air contains water vapor and the water will have a negative effect the system efficiency. The water will accumulate on the low pressure side of the system, where a

[2]Many experts believe that the dangers with ammonia are overrated. They argue that synthetic refrigerants can have many of the same safety issues as ammonia

water filter can be installed. It is important to avoid copper components because ammonia and water will corrode copper, zinc and their alloys. For example, ammonia and water will destroy the copper windings of the electric motor in a hermetic compressor. Oil from the compressors is another contaminant, which can be removed with an oil separator located after the compressors and an extra oil drain valve in the low parts of the system. (Pearson, 2008b), (Lorentzen, 1988)

Ammonia is mostly used in large systems, but research is now being conducted on its use in smaller systems, too. The lack of suitable components is often the issue for these systems. Pearson (2008b) provides an example of how domestic refrigerators and air conditioners could be developed to use ammonia as the refrigerant. This requires the use of small charges and hermetic systems completely free of water. Safety regulations would also have to be more strongly enforced for these applications. Palm (2008) describes a prototype of a small-scale heat pump with ammonia. The results show that it is possible to design such a system, but the limiting factor is finding the proper components for such small charges. The main problems were oil return and heat transfer in the evaporators. Oil return can be solved in several ways; an oil separator can be installed after the compressors and valves can drain oil from the evaporators. The type of oil is also important and the design of the system can also be optimized for oil return. Different alternatives of evaporators were tested.

Carbon dioxide is infrequently used in industrial refrigeration systems, but it has potential as an alternative to synthetic refrigerants, mainly because it is harmless to the environment. The possibility of having a transcritical process with gliding temperatures on the heat output side is an advantage. Lorentzen (1995) states that specific power consumption can be reduced by up to 40% by using this system compared to a conventional process.

Ammonia has limited use at very low temperatures, since it becomes sub-athmospheric below -33.3°C. This problem can be avoided by using a cascade system with ammonia and carbon dioxide, as described in Section 2.1.1, where evaporation temperatures of -50°C can be used. Lower evaporation temperatures provide faster food freezing times and lower product temperatures, both of which are beneficial for food quality. A two-stage CO_2 system can also be an alternative, as described by Visser (2010). This system was transcritical and replaced several systems mainly for cooling, but also for heating. It was estimated that the specific electrical energy use would be reduced by 75.5%.

The pressure ratio between 20°C and -38°C is 5.3 for CO_2-systems and

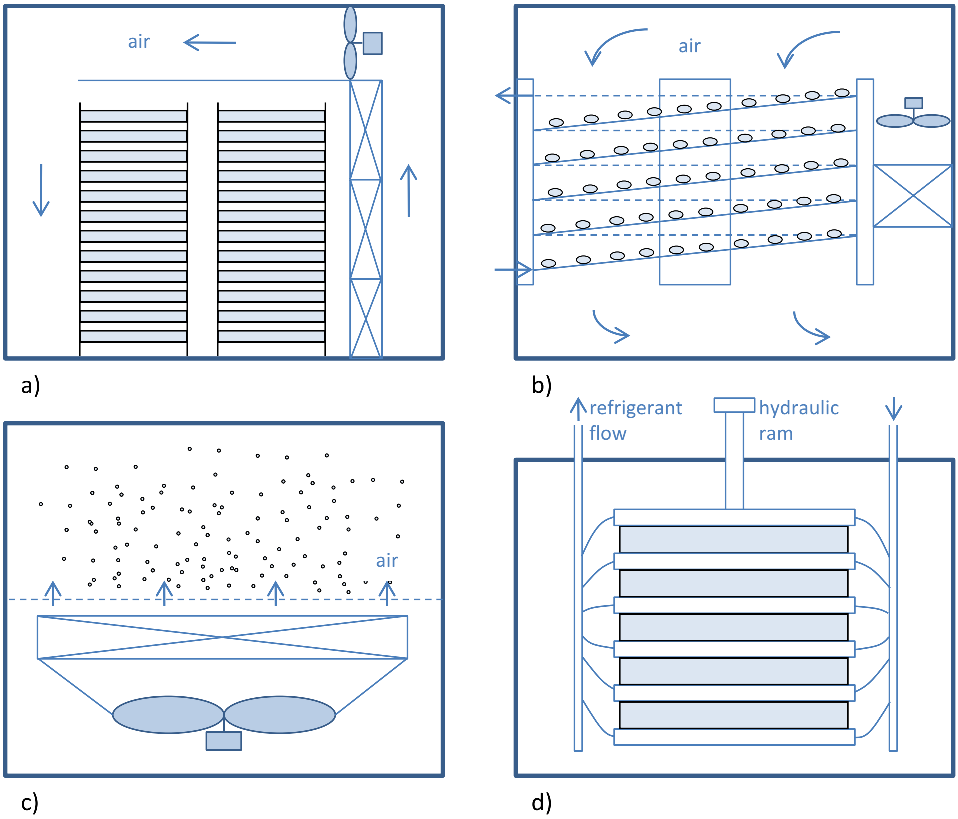

Figure 2.3: *Different kinds of food freezing equipment: a) blast freezer b) spiral belt freezer c) fluidized bed freezer d) plate freezer (Valentas et al., 1997, chap. 3).*

10.7 for NH_3-systems. Despite working across far higher pressure ratios, ammonia compressors can work in one stage, where two stages are necessary for carbon dioxide compressors. However, Pearson (2006) suggests that costs are still lower for the two-stage carbon dioxide system. High system pressures are sometimes mentioned as an disadvantage of CO_2, but this need not be so. The equipment for these systems is under development and high pressures can even be used as an advantage. For example, high pressures give high volumetric capacity, which allows small component sizes to be used. (Nekså, 2002)

2.1.3 Food chilling and freezing

Refrigeration includes both chilling and freezing and is one of the best ways of preserving food as close to its original state as possible. When the product temperature is decreased, microbiology activity and chemical and physical changes are slowed and shelf life is extended. Just how much these processes are retarded depends on the product and the temperatures. (Zhou et al., 2010). Many fruits and vegetables must be stored above freezing temperatures to avoid damage. Examples are bananas (13-15°C), lemons (10-13°C), cucumber (10-12°C), mangos (13°C), tomatoes (8-13°C) and potatoes (4-15°C) (ASHRAE, 2006, chap. 11). However, chilling of these products is still important to prevent them from becoming overripe.

Superchilling is a technology for preserving the fresh quality of the products but at as low temperatures as possible. The products are partially frozen, typically at a temperature 1-2°C below the initial freezing point. At this temperature, the microbial activity is reduced comparing with conventional chilling methods. A thin shell of frozen product will be built up during superchilling. This layer functions as stored refrigerating capacity afterwards and the shelf life is extended (Magnussen et al., 2008). Extra ice is not necessary with superchilling, which results in less energy demand when the products are transported (Hemmingsen, 2002)

Frozen food products are mainly stored between -20°C and -30°C. The lower temperature levels are necessary for fish and ice cream. There are several different food freezing methods, which can be basically divided into two groups: those that use air as the cooling media, and those that use other media (liquids or contact).

Air is often used because it is easily available, even though it has poor heat transfer properties. Air blast fans increase heat transfer, but they also use electricity and add heat to the freezing tunnel. Large products are often frozen in air blast freezing tunnels, while smaller products are frozen in belt freezers or fluidized bed freezers, as is shown in Figure 2.3. If the product is unwrapped, a disadvantage is mass reduction by evaporation from the surface. Packaging, on the other hand, results in poorer heat transfer and longer freezing times. There is a risk of uneven air distribution and uneven freezing times in a blast freezer, which can partly be avoided if the products are continuously loaded and unloaded. Proper tunnel design and loading is essential.

Unpacked, single products can be frozen in impingement freezers at high production rates. In these, air jets (20-40 m/s) are directed at the products, which provides higher heat transfer than in blast freezers (Salvadori and Mascheroni, 2002). It is possible to independently adjust velocity and temperature inside the freezer, for optimum use of cooling capacity (Kaale et al., 2011).

The second group of freezers does not use air as cooling media. A common type is the plate freezer, where the refrigerant is circulated inside plates and the products are pressed between the plates. The products are frozen into equally shaped boxes, which is beneficial for stacking and transport. These freezers are energy efficient because no fan energy is needed and higher evaporation temperatures can be used than in air blast freezers. Liquid as a cooling media is used in cryogenic freezers. The products are sprayed with, for example, liquid carbon dioxide resulting in high temperature differences and high production rates. A refrigeration plant is not needed, but the cooling media is expensive, and large quantities are necessary (Shaikh and Prabhu, 2007). All types of freezers should be insulated from surrounding (warmer) areas to prevent heat leakage into the freezer.

Spoiled food will have low or no commercial value and can cause illness, which makes high food quality essential. However, it is difficult to measure quality directly and operational energy costs often get the most attention. One way of representing product quality is by relating it to storage temperatures and times. The concept of a cold chain is used for the sequence of distribution and storage of products where temperature is maintained below a certain value, depending on the product.

2.2 Modelling food freezing

A combination of measurements and simulations is appropriate when evaluating industrial refrigeration plants. The simulation program is based on heat transfer models and product and system assumptions, which are described in this section.

2.2.1 Thermal properties of food

Before heat transfer calculations can be made, the thermal properties for the food products have to be determined. These are not found explicitly in tables, but have to be measured or predicted based on assumed chemical composition and temperature. The contents of different food components will vary from item to item, which will influence thermal properties. However, prediction methods are still reliable, and could also be better than some of the poorer measurement methods (Valentas et al., 1997, chap. 3). Composition data can be found in literature, for example in ASHRAE (2006,

Table 2.3: *Typical values of heat transfer coefficients calculated with equations 2.4–2.7 (Valentas et al., 1997, chap. 3).*

		Input	W/mK
h_r	T_s/T_{sur} :	-31 / -37°C	$6.3 \cdot 10^{-4}$
$h_{c,nat}$	ΔT :	10 K	4.1
$h_{c,plan}$	v:	2 m/s	12.7
$h_{c,plan}$	v:	6 m/s	30.6
$h_{c,oval}$	v:	2 m/s	19.0
$h_{c,oval}$	v:	6 m/s	36.6

chap. 9). Models for calculating thermophysical properties such as density, specific heat, enthalpy, thermal conductivity and thermal diffusivity are also included.

Food products always contains water, which influences thermal properties. Most of the properties change around the freezing point, due to the phase change of water. Since food contains salts and other components that suppresses the freezing point, the initial freezing point is always below 0°C. Fish and meat have an initial freezing point around -2°C (ASHRAE, 2006, chap. 9). When freezing has started, the rest of the unfrozen water in the product will have higher concentration of solutes, which will further suppress the freezing. Latent heat will be removed from the object over a temperature range. Graphs for thermophysical properties for Atlantic mackerel can be found in Figure 2.4. The thermal conductivity for frozen food is higher than for unfrozen; for Atlantic mackerel it is 0.43 W/mK at 5°C and 1.37 W/mK at -25°C. As a consequence of this, thawing is a slower process than freezing.

2.2.2 Heat transfer coefficients

Heat transfer at the surface of a product in an air blast freezer can be by convection, radiation, evaporation or combinations of these. An overall heat transfer coefficient (U) is calculated based on the effects of the different parts, as can be seen in Equation 2.3. Typical values are given in Table 2.3.

$$\frac{1}{U} = \frac{1}{h_c + h_r + h_{evap}} + \sum \frac{d_p}{k_p} + \frac{d_a}{k_a} = \frac{\Delta T \cdot A}{q''} \tag{2.3}$$

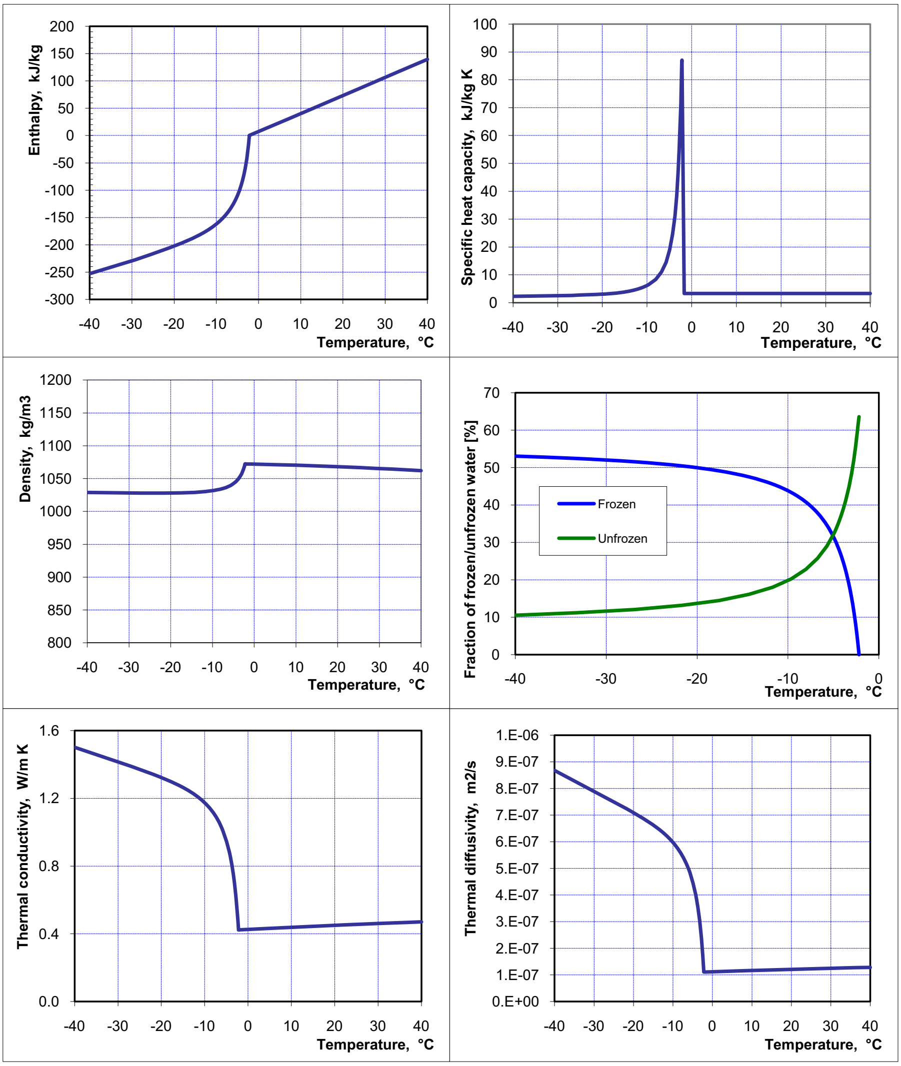

Figure 2.4: *Thermophysical properties of Atlantic mackerel. (ASHRAE, 2006, chap. 9)*

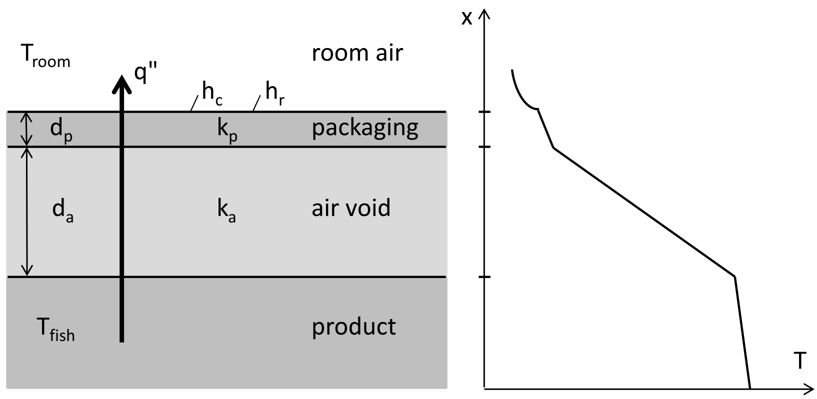

Figure 2.5: *Product heat transfer and temperature variation.*

The evaporation coefficient, h_{evap}, is important for unwrapped products, but can be omitted when the products are wrapped. The radiation coefficient, h_r, can be calculated from Equation 2.4 (Incropera and DeWitt, 2002, chap. 1). Radiation is most significant when the objects have large temperature differences, such as between products and evaporators.

$$h_r = \varepsilon \cdot \sigma \cdot (T_s + T_{sur}) \cdot \left(T_s^2 + T_{sur}^2\right) \tag{2.4}$$

Many different models and data for convective heat transfer coefficient can be found in literature. They depend on the heat transfer medium and the product geometry. Valentas et al. (1997, chap. 3) have set up three approximations for h_c. For natural convection and air velocity below 0.4 m/s:

$$h_{c,nat} = 2.3 \cdot (T_s - T_a)^{0.25} \tag{2.5}$$

For forced convection, air velocity above 1 m/s, and objects with planar surfaces:

$$h_{c,plan} = 7.3 \cdot v_a^{0.8} \tag{2.6}$$

For forced convection and oval objects:

$$h_{c,oval} = 12.5 \cdot v_a^{0.6} \tag{2.7}$$

In fluidized bed and belt freezers, air velocities are generally higher and product sizes smaller, which gives higher h_c than in tunnel freezers. Valentas et al. (1997, chap. 3) give typical values of 120-200 W/m^2K. In impingement freezers, air jets with very high velocities are directed at single products, which can give convective heat transfer coefficients of 100-400 W/m^2K (Salvadori and Mascheroni, 2002).

If the product is wrapped, which is typical inside blast freezing tunnels, the packaging material and the air void between this and the product will reduce heat transfer. In Equation 2.3, k_p and k_a are conductivity for packaging and for the air void, respectively. The thickness is represented by d. If the air void is thicker and natural convection is present within the air layer, this increases the heat transfer and reduces the insulating effect of the packaging. Average values should be used if the packaging thickness and number of layers varies.

Heat transfer in plate freezers depends on the contact between the products and the plates. h_c is typically 200-500 W/m^2K when contact is good. Poor plate contact can occur if there is not enough pressure from the plates, if spacers are inserted between the plates, or when the product or packaging is uneven. Typical values are then 50-100 W/m^2K. (Valentas et al., 1997, chap. 3)

2.2.3 Modelling freezing time

There are several different methods that can be used to model freezing processes. These can be divided into two types: analytical and numerical methods. Plank's equation (Equation 2.8) is an analytical model that gives the freezing time for certain product properties and temperatures (see Valentas et al. (1997, chap. 3) for more details about the variables). This equation is easy to apply, but it only gives the time from start of freezing until the latent heat has been removed, and sensible heat is not considered.

$$t_f = \frac{\rho L}{(T_{if} - T_a)} \left[\frac{R}{h} + \frac{R^2}{2k_f} \right] \tag{2.8}$$

Valentas et al. (1997, chap. 3) instead recommends a modified Plank's equation, which includes sensible heat. This equation was used when calculating the freezing times in Paper III.

$$t_f = \frac{1}{E} \left[\frac{\Delta H_1}{\Delta T_1} + \frac{\Delta H_2}{\Delta T_2} \right] \left[\frac{R}{h} + \frac{R^2}{2k_f} \right] \tag{2.9}$$

Instead of using analytical models like Plank's equation, numerical methods can be used. These are more accurate and more flexible, but also require more information and longer calculation times. Numerical methods can be divided into temperature methods and enthalpy methods. Because these methods have limitations, Pham (1985) developed a new combined method. This method is stable and simple to implement and does not require much CPU time. It does not require the iterations needed by enthalpy methods and the risk of underestimating the latent heat (which can happen with temperature methods) is low.

The product can be analysed with finite differences or finite elements. Finite difference methods are more practical to use when the products are simple and have a regular shape. Finite element methods are more suited when the shape is irregular (Wang and Sun, 2003).

Bonacina and Comini (1973) have described a two-dimensional, finite difference, unsteady state approximation for the heat conduction equation. The procedure is to calculate first in the x-direction and then in the y-direction. The temperatures from the first calculations are used in the second. The method is based on the three time level Lees scheme (Cleland, 1990, chap. 4), but is extended to two dimensions. The boundary condition describing the heat transfer on the surfaces can be included in the conduction equation by setting the temperature outside of the product to the air temperature and the outer conduction equation to include the convective heat transfer. The method is also described in Papers V and VI.

Freezing times calculated with both the modified Plank's equation (Eq. 2.9) and the numerical method described here were compared for a product box of mackerel. Some input values and the results are given in Table 2.4. Both methods gave a freezing time of close to 16 hours.

Table 2.4: *Comparing freezing times calculated from Plank's equation and a numerical model.*

Input data	
$w_{prod} \times l_{prod}$	1.09 × 0.89 m
h_{prod}	0.10 m
c_u	3.33 kJ/kgK
c_f	2.23 kJ/kgK
L	212 kJ/kg
T_i	0°C
T_{fin}	-18 °C
T_a	-30°C
ρ	1030.00 kg/m^3
k_f	1.40 W/mK
v_{air}	5 m/s
k_a	0.0223 W/mK
k_p	0.1800 W/mK
d_a	0.0010 m
d_p	0.0010 m

Results	**Numerical model**	**Plank's equation**	
Calculating time	3.78	0	minutes
Freezing time	16.04	15.79	hours

Chapter 3

Increasing energy efficiency

Energy efficiency can be increased for most industrial refrigeration systems, but the amount of improvement depends on the total size, the age of the components, the control and operation of the system, and the product load. This chapter presents areas where energy savings can be gained, including system design and upgrades, maintenance, compressors, freezing tunnels, integration with heating system and cold thermal storage.

3.1 System design and upgrades

A common issue with industrial refrigeration systems are that they are often working outside of the conditions for which they were designed, which is less energy efficient. Therefore, when designing a system, it is vital to try to predict the "normal" future operating conditions, to ensure that the system works efficiently. Pressure levels in suction and discharge lines affect compressor performance and energy consumption. The pressure levels are influenced by surrounding conditions. The required food temperature and heat transfer conditions will affect the evaporator temperature and the cooling air or water will affect the condenser temperature. Lower differences between these temperatures result in higher COP. Some measures can be taken to ensure low temperature differences, described in the following text.

Water-cooled condensers are more effective than air-cooled condensers because of higher heat transfer coefficients. At fish freezing plants it is often natural to use seawater-cooled condensers. The cooling water temperature is less affected by annual temperature fluctuations if the water inlet is deep than if it is close to the sea surface.

The evaporators in large freezing tunnels are mainly of the liquid recirculation type. These have wetted surfaces and therefore have good heat transfer properties (Pearson, 2008c).

Free cooling means using surroundings, for example ambient air, as a part of the process. This can be especially useful when cooling or freezing processed and heated food. James and James (2011b) have analysed how ambient cooling as part of two refrigeration processes affects total energy consumption. Five and ten minutes of ambient cooling (at 20°C) of fried hash browns reduced the heat load by 31% and 44%, respectively. It also resulted in 60 kg less ice formation on the evaporator per hour. Without the ambient cooling, the freezer had problems reducing the temperature to the desired temperature. In the second process, pie cooled in ambient air (10°C) for 30 min removed 50% of the total heat. Ambient cooling made the process somewhat slower, but the authors believed that it would not affect bacterial growth and was therefore safe.

A system can be upgraded to increase energy efficiency by installing variable speed drives on the compressor and fan motors, described later in this chapter. Other upgrading initiatives could be to improve the control system or to utilize the heat from the oil cooling system.

3.2 Maintenance

Apart from the fact that basic design is important for energy efficiency, proper maintenance is also essential. Efficiency depends on the pressure differences over the compressor, which is affected by the condenser and evaporator performance, which means that these heat exchangers must be supervised and maintained. Normally, evaporators have to be defrosted, which commonly is done using hot refrigeration gas or electricity. The advantages of hot gas defrost is that it is more energy efficient and it helps remove oil from the heat exchanger. If the ice is not completely removed, it will be partly melted and frozen again, giving the system extra heat load. The ice can also build up and damage the equipment (Koster, 2009). Evaporators should not be defrosted too often either, because it adds heat to the evaporator and extra energy is necessary. An optimization of the defrost scheme is necessary for good energy efficiency. Only 1/3 of the evaporators in a system should be defrosted at the same time (Pearson, 2008a).

Condensers are often more exposed to fouling than evaporators. The local environment around the water outlet of seawater-cooled condensers is beneficial for biological growth. Sessile invertebrates such as mussels, barnacles and sea anemones have to be removed regularly to ensure a high water mass flow.

If the suction pressure is below atmospheric pressure, air with water vapour will leak into the refrigeration system. Water will tend to accumulate on the suction side of the compressor, while air will accumulate on the pressure side. Because of the extra water pressure, the evaporator temperature will decrease. If this temperature is maintained at a given temperature, the compressor has to work at a lower suction pressure, which demands more power (Ficker, 2009). Another risk with water in an ammonia system is corrosion, since ammonia and water will corrode copper, zinc and their alloys. These metals should be avoided in ammonia systems.

Halnes (2008) was the refrigeration system operator at the ammonia refrigeration system where measurements for this PhD work were made. He observed that energy consumption decreased after a water separator was installed. Water content should be less than 1 % (normally 0.6-0.7 %) but it was above 2 % before the installation.

Air inside a refrigeration system will accumulate in the condenser and increase the condenser pressure (Welch and Wright, 2008). This will lead to higher energy usage, but also more wear on mechanical parts because of the higher temperature. The oxygen could increase corrosion of pipes and vessels (Ficker, 2009).

Oil is necessary in the compressors for reducing friction and increased sealing between moving and static parts. Compressors should be equipped with oil separators to ensure oil return, but some of the oil will always follow the refrigerant. If the oil is not miscible, such as mineral oil and ammonia, it can cause problems. An oil film can build up inside the evaporator and reduce the heat exchange. Because of lower density oil will also accumulate in the low parts of the system, and oil traps can be installed here for removal. (Koster, 2009), (Pearson, 2008a)

3.3 Compressor

There are several different types of compressors. For industrial ammonia systems, reciprocating and screw compressors are the most common. Reciprocating compressors can have more energy efficient capacity regulation and operation (at low cooling capacities), but require more service (Pearson, 2008c). Screw compressors are more reliable and can also work across larger pressure differences. Screw compressors are also more common when larger cooling capacities are required. When only one screw compressor is necessary, compared with two or more reciprocating compressors, the investment

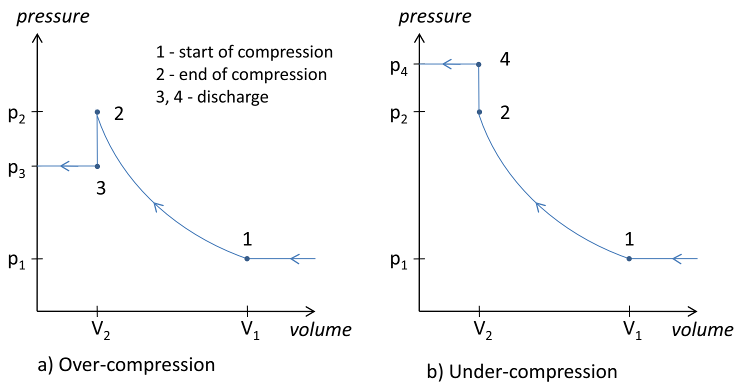

Figure 3.1: *Pressure-volume diagram for a compressor with over- and under-compression. (Stoecker, 1998, chap. 5)*

costs will also be lower.

A screw compressor has a built-in volume ratio, which does not necessarily correspond to the system pressure ratio. There will be nonproductive work and losses if the pressure ratio between the discharge line and the suction line is different from the built-in ratio. Figure 3.1 shows over-compression and under-compression, where in both cases the losses come from unrestrained expansion of the gas. Between 2 and 3, a smaller amount of gas will flow rapidly from the compressor discharge port to the discharge line. This results in larger losses than between 2 and 4, where the gas rushes into the compressor from the discharge line. (Stoecker, 1998, chap. 5)

Due to pressure losses in the suction line, the refrigerant will be superheated when entering the compressor. This is lost refrigeration effect and the suction line pressure drop should therefore be minimized. Regular servicing of the compressors is necessary, to clean and exchange dirty and worn parts. Oil levels also need to be supervised. Too little oil in the compressor will lead to increased wear and higher friction.

3.3.1 Capacity control

The most common way of varying the capacity of a screw compressor is with a slide valve. This is illustrated in Figure 3.2. When the slide valve is opened, some of the gas inside the compressor will be vented back to the suction port. This provides smooth regulation down to 10% of full capacity. It is easy, but the energy efficiency is reduced, because of friction in the gas and a reduction in built-in volume ratio (Stoecker, 1998, chap. 5). The

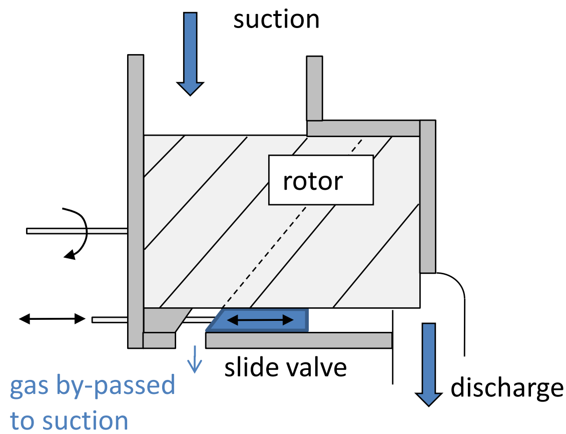

Figure 3.2: *Compressor with slide valve in operation. The dash line indicates where compression starts at part load operation.*

relation between the slide position and the capacity is not linear, so a small opening results in a larger capacity drop, as can be seen in Figure 3.3. The benefits of an economizer will also be lost when capacity is reduced below about 80%. Papers I and II evaluated measured data from a compressor system that had slide valve regulation. The analysis showed that more than one compressor often were at part-load operation during low production. A better solution is to have compressors at full load or off and only one at part-load. An even better way of controlling the capacity is with a variable speed drive, which can change the speed of the compressor motor. This can be done with a two-speed motor or with a frequency converter. A frequency

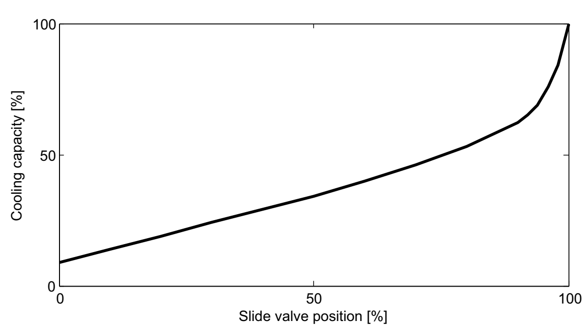

Figure 3.3: *Theoretical curve showing cooling capacity as a function of compressor slide valve position.*

Table 3.1: *Freezing times of a box of Atlantic mackerel calculated using a modified Plank's equation (Valentas et al., 1997, chap. 3).*

		Cardboard and air layer	**Plastic and vacuum**
Air velocity	[m/s]	5	5
Initial temperature	[°C]	0	0
Final temperature	[°C]	-18	-18
Overall heat transf.	[W/mK]	10.2	25.1
Freezing time	[h]	15.5	7.7

converter is normally operated between 30 and 60 Hz. This suggests that cooling capacity also can be increased above the design value (for 50 Hz), which could be useful at peak loads (if the compressors are built for this).

A third, and not so common, way of capacity control is by volume ratio modification. This approach works like the slide valve, but can also modify the discharge port volume. This solution is more complicated to control and requires the maintenance of more parts.

3.4 Freezing tunnels

Many different kind of products can be frozen in freezing tunnels, but the most common are products of larger size or products packed inside boxes. The packaging will reduce the heat transfer and extend freezing times. When packed inside a cardboard box, the main heat resistance will be in the air between the packing and the product. This is also discussed in Paper III. The reason for using packaging is that it allows for easier handling and because of customer demand. An alternative to the cardboard box is to vacuum pack the products, which will result in higher heat transfer properties, but will also require new equipment on the production line. As an example, two different packaging are used when calculating[1] freezing times of a box of mackerel ($0.33 \times 0.50 \times 0.2$ m). The results can be seen in Table 3.1. The freezing time for the cardboard box is double the freezing time for the plastic vacuum packaged box.

[1] calculating with the modified Plank's equation described in section 2.2

Table 3.2: *Percentage component heat loads for batch air-blast freezer (Valentas et al., 1997, chap. 3).*

product	50–80 %
fans	10–40 %
pull-down	< 10 %
defrost	< 5 %
other	< 5 %

3.4.1 Reducing heat loads

Energy saving measures of an existing system can begin with reducing heat loads, which will reduce the cooling demand. The different contributors to the heat load have to be surveyed to determine where most savings can be had. In an air-blast batch freezing tunnel, heat loads are typically distributed as is shown in Table 3.2.

Other heat loads include insulation ingress, air infiltration, equipment, etc. *Pull-down* heat load is present if the tunnel has been off. Heat loads from people, trucks and lightning should be excluded during the freezing period.

The product heat load depends on the temperatures at start and end, along with the product mass and thermophysical properties (Valentas et al., 1997, chap. 3):

$$\dot{Q}_{prod} = \frac{m_{prod}}{t_{prod}} \cdot [c_u \cdot (T_i - T_{if}) + L + c_f \cdot (T_{if} - T_{out})] \tag{3.1}$$

It is best if the products have not been unnecessarily heated before they are put into the freezer. This might happen if loading is inefficient. A low final temperature is best for good food quality. It should be noted that there are temperature differences both within a product and between products.

In addition to requiring electricity, the fans will add heat to the refrigeration system. Installing the fan motor outside of the freezing tunnel would decrease the heat load, but only to a limited extent, since most of the heat would still be released inside the tunnel (Valentas et al., 1997, chap. 3).

Thicker and better insulation will prevent heat ingress through the walls, floor and ceiling of the tunnel. However, losses in efficiency from this heat transfer are small and it is more important to have a complete vapour barrier. Vapour inside the insulation will increase the heat transfer and could also damage the insulation (when it freezes). The largest heat leaks often occurs

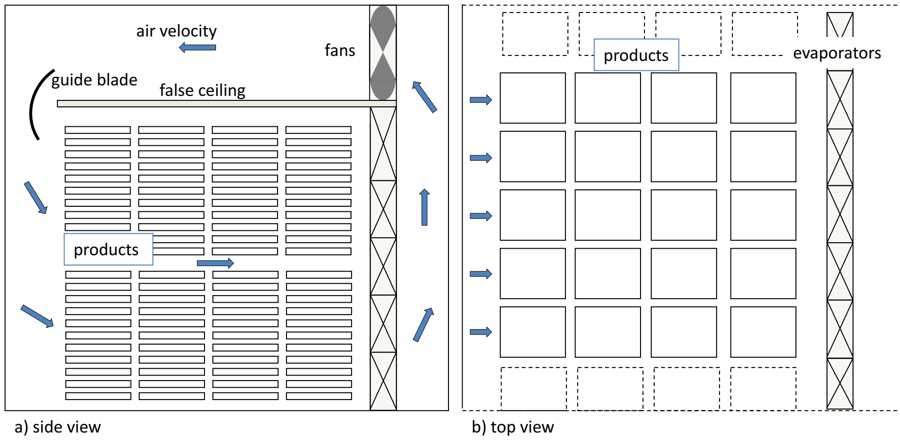

Figure 3.4: *Airflow in a freezing tunnel with food products.*

by the doors, so these should be properly constructed. If moist air leaks into the tunnel, this will lead to more frost on the evaporators.

Batch freezing tunnels for fish, especially pelagic fish, often have very varied operation over a year. In some periods, the tunnels might be running for many days, with loading and unloading every day. Some defrosting may be needed during these periods. Other periods may have more scattered production days, where the tunnels are turned off when they are not in use. The need for defrosting would be lower, but the additional cooling of the tunnel (walls, floor and ceiling) and shelves results in an extra heat load (called pull-down).

3.4.2 Air velocity field

A uniform air velocity field inside a freezing tunnel provides uniform freezing times for products, but it is complicated to achieve. Products with uniform freezing times will have similar temperatures when they are transferred to storage facilities, which gives predictability. If products are transferred too early, after-freezing in the cold store is necessary and this will be slower than in the freezing tunnel. Longer freezing times ensure the correct temperatures, but will have higher energy consumption.

Figure 3.4 shows the structure of a freezing tunnel where the air flows across a false ceiling, through the product racks and finally through the evaporators. The air velocity field can be improved by construction design changes. The general idea is to force the air to flow across the products,

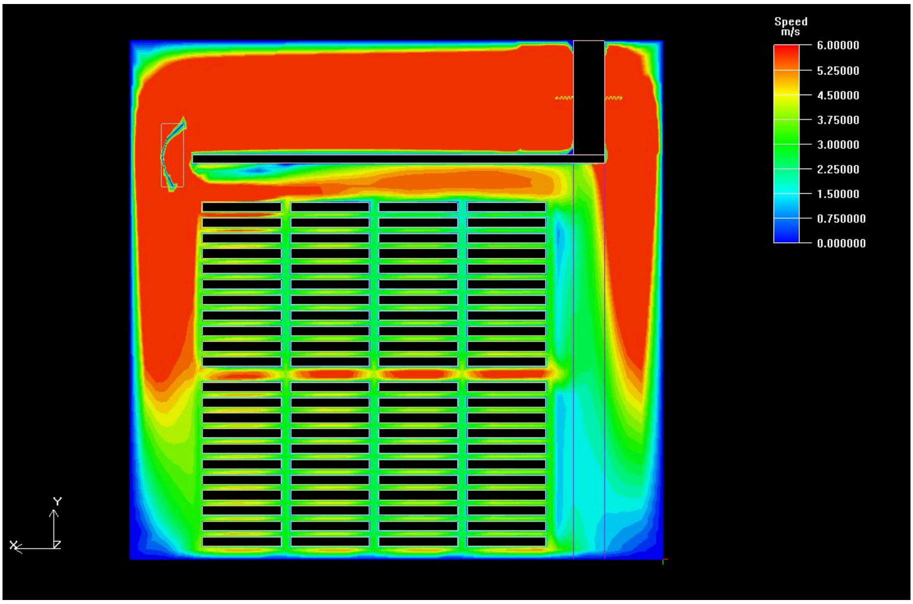

Figure 3.5: *Air velocities inside a freezing tunnel simulated using Airpak.*

and not to take short-cuts where it does not remove heat. The challenge is where to place the extra baffles so that the velocity distribution is improved without affecting other operations, such as the loading and unloading of products. Paper III shows that a guide blade after the false ceiling will increase air velocities in the worst sections of the tunnel. The simulated air velocities can be seen in Figure 3.5 (see also Figure 2 in Paper VI). The highest velocities were found in the largest voids, between the product racks and below the false ceiling. About 1/3 of the air volume passes here. Some of the cooling effect will be unused, since the cold air will travel directly back to the evaporator without cooling the products.

Alonso et al. (2011) simulated different design alternatives and found that a thick ceiling with a vertical guide plate was better than the normal (thinner) false ceiling. This design decreased backflow through the evaporator and increased velocities across the products. Kolbe et al. (2004) used plywood and plastic sheeting to prevent air by-pass in a freezing tunnel which resulted in 15% shorter freezing time and reduced fan energy use of 6%. The freezing times of the products were also more equal.

3.4.3 Fan operation

The freezing tunnel fans not only use electricity, but they also add heat to the tunnel. Towards the end of the freezing period, the amount of heat load from the fans can be as large as the heat load from the products. A program was built in MATLAB to simulate different operation alternatives (see also Paper VI). It consisted of several functions, visualized in Figure 3.6. The function `f_tunnel` was used with the input variables `reduce_time`, `reduce_vel` and `reduce_temp`. The variables `reduce_vel` and `reduce_temp` gave the reduced air velocities and reduced air temperatures at the point of time `reduce_time`. For example, a reduction in air fan velocities to 67% of full air fan speed after 14 h together with a reduction in air temperatures to -33°C was called with `f_tunnel(14, 67, -33)`. The program would then load variables from an earlier simulation (the baseline case) at the 14 h point, and continue the simulation with the new air velocities and temperatures. Using already calculated simulations saved total calculation time.

The function `f_tunnel` loaded property tables for the product and air velocity tables. The function `f_product` constructed the conductivity tables and calculated the product temperature in `f_prodTemp`. Heat transfer from each of the node of the product was also calculated at this step. When temperatures had been calculated, a function tested if the highest temperature was below -18°C. If it was, the results were written to a file. If not, a new time step was made and new calculations were done. The baseline case was calculated using a similar approach, but with no reductions. Dimensions and other product and tunnel properties were set at the beginning of this simulation. Refrigeration system power consumption and fan power load was calculated in a separate function, from assumptions of the system COP, the fans and the product temperatures and heat loads.

The simulation results showed that reducing fan air speed resulted in longer freezing times and lower total energy consumption. Many different alternatives were simulated and the results are shown in Paper VI.

Another simulation of a refrigeration system with a freezing tunnel was made by Walnum et al. (2011). They found that a 33% reduction in power consumption could be accomplished by reducing the fan speed. This increased the total freezing time by 14%. Kolbe et al. (2004) tested three different fan operation schemes, which increased the freezing time by 8-12% and decreased the total energy use by 11-8%, respectively.

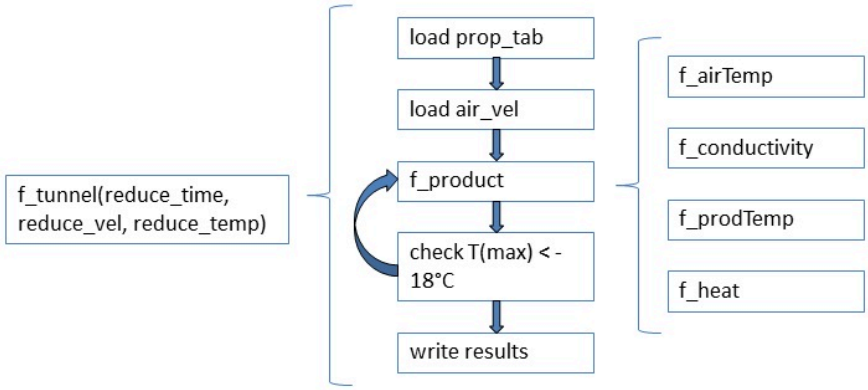

Figure 3.6: *Structure of freezing tunnel simulation program in MATLAB.*

3.5 Integration with a heating system

The vapour compression system pumps heat from a cold area to a warm area. The cold side is the main focus, but if the excessive heat is used cleverly, total system energy consumption can be decreased. The main heat sources are the compressor oil cooling and the condensers.

Compressors in industrial refrigeration systems need oil for lubrication and seals between the moving and static parts. After the compressor, the oil is separated from the refrigerant. Since the refrigerant and the oil are heated in the compression process, it is necessary to cool the oil before reinjecting it into the compressor. Typical oil temperatures are 50 - 70°C and Figure 5.25 by Stoecker (1998) indicates that 25% of the total heat input (compressor work and refrigeration load) is absorbed by the oil cooling system. Even though the temperature is not very high, it is enough for preheating cleaning water and for floor heating.

Condensing temperatures are about 20°C and the exergy of this heat is therefore low, but this heat could be used with a heat pump to heat buildings or a swimming pool, for example.

3.6 Cold thermal energy storage

Another promising technology is cold thermal energy storage (CTES). Hafner et al. (2011) describe a cascade system with carbon dioxide and ammonia, also shown in Figure 3.7. Carbon dioxide is frozen in a shell and a tube heat

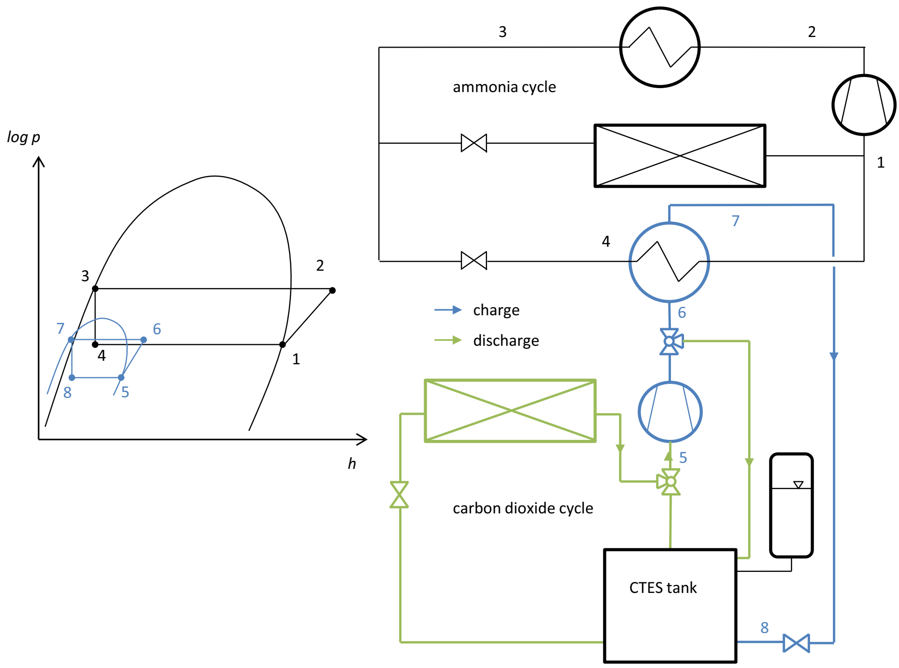

Figure 3.7: Example of cold thermal energy storage system (Hafner et al., 2011).

exchanger and used later in the process when the compressor is off. The part-load operation of the compressor can thus be avoided. They found that 30% of the energy consumption can be saved by using this system instead of a conventional system where the compressors are operated at part-load during the freezing process. Saito (2002) reviewed the field of cold thermal energy storage. Applications other than the previously described include ice slurries and ice-making types. One significant advantage of CTES is the possibility of moving energy consumption from one time to another, for example by using electricity during the night instead of the daytime, when energy costs and refrigeration loads are higher. Peak loads can be avoided and power costs decreased. A cold store that only uses the refrigeration system during the night can be used as a CTES with low cost and effort.

Chapter 4

Summary of the papers

4.1 Compressor operation: Papers I and II

Screw compressors with slide valve regulators are often used in industrial refrigeration. The slide valve makes it easy to reduce cooling capacity, but this set-up is unfortunately not energy efficient. Papers I and II analyse alternative ways of operating the compressor. Paper I introduces an optimization model which is also used in Paper II. Operation optimization using variable frequency drives is discussed in the second paper.

When a slide valve is in use, a slot opens for the refrigerant to vent back to the suction side. In this way the capacity can be easily reduced from 100% to 10%, but due to friction in the gas and volume ratio change, the energy efficiency is reduced. The efficiency will decrease rapidly as soon as the slide valve is in use. A better approach to capacity regulation is to use a variable frequency drive. This changes the speed of the compressor motor and therefore also reduces power consumption.

The refrigeration system that was analysed had 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW. The compressors were divided into two subsystems, with evaporation temperatures of -38°C and -42°C. The first system, with 5 compressors, was evaluated, including an analysis of measured and logged refrigeration system data from two periods of 9 days each. The refrigeration system was both operated at full capacity and at reduced capacity during these periods. Part-load curves for the compressors were calculated in Paper I. These were used in an optimization model that gave the minimum power consumption for a certain cooling capacity load.

Results showed that largest energy saving potential was during part-load operation, when the compressors operated far from the design point. There was nearly full production every day during the first period, and Paper I shows that COP could be improved by 1.6% and 11.8% compared with the existing system. Paper II analysed the second period, which had reduced cooling capacity. The system was optimized both with and without variable frequency drives and the difference in improvement was small between these two alternatives. A system with several different sized compressors can be operated efficiently without variable frequency drives, but these are still recommended for smaller systems.

4.2 Air velocity field: Paper III

Air blast freezing tunnels are common in large fish freezing plants because they feature simple construction and can contain large quantities of fish. However, the freezing times are long and often varies between the products. The main reason for this is an uneven air velocity field. The quality of the fish could be decreased if products are transferred to the cold store too early, before they are fully frozen. On the other hand, extensive freezing times result in higher energy costs. The packaging of the products also reduces the heat transfer and increases the freezing time.

The air velocity field inside the freezing tunnel was analysed in Paper III. The velocities between the products could not be measured because of limited space, and therefore the Computational Fluid Dynamics software Airpak was used to simulate air velocities. The dimensions from a freezing tunnel at a fish freezing plant were used and some different configurations were tried.

The velocities from the Airpak simulation program were used to calculate an average velocity for each product shelf. A polynomial function was generated from a group of velocities, which was then integrated to find an average. Steady state calculations with a modified Plank's equation were made to find freezing times for the products. Both product type (Atlantic mackerel) and air properties affect the freezing time, and packaging properties were included in the overall heat transfer coefficient.

A tunnel was simulated both with and without a guide blade positioned after the air fans. Air velocities for the tunnel without the guide blade were between 0.97 m/s and 6.29 m/s, which resulted in freezing times between 16 h and 32 h (a span of 16 h). With the guide blade, the air velocities were between 2.57 m/s and 5.36 m/s, resulting in freezing times from 17 h to 21 h, a span of only 4 h. The freezing times found from temperature measurements in the freezing tunnel were between 13 h and 20 h, which indicates that the simulation results were valid, but the velocity distribution in reality is more uneven.

4.3 Product modelling: Papers IV and V

Numerical models and transient simulation are important in analysing freezing processes and their improvement possibilites. Temperatures during the process can be found for different locations inside the product.

Product and air temperature measurements were made at a fish freezing plant and these were compared with data from two simulation programs. Thermocouples were used to measure absolute temperatures inside the freezing tunnel during operation. Product temperatures were found with temperature loggers placed inside the product boxes. These were located between the fish in the centre of the box and on top of the fish below the lid.

The object-oriented programming language, Modelica, along with the simulation environment Dymola were used for the simulations in Paper IV. The different parts of the refrigeration system were included in this program. Paper IV shows reasonable accuracy between the measured and the simulated data.

A finite difference model was used for the product simulations of Paper V, and this was written in MATLAB. The simulation models included air and product properties, but the refrigeration system was not included. The results show that freezing time will increase by 1 h when the packaging air gap is increased from 1 cm to 3 cm, but only by 0.2 h when the air gap is increased from 3 cm to 5 cm. Air velocity clearly affected the freezing time. The total freezing time for one product was 22 h at 2 m/s and 15 h at 6 m/s. A comparison with the measured data showed similarities, so that the simulation program can be used in future freezing process simulations.

Both papers indicate that the measured data were scattered and that there are always uncertainties when food products are involved.

4.4 Optimization of air fan operation: Paper VI

Freezing times are typically around 20 h in an air blast freezing tunnel. High air velocities are important in the first part of the freezing period, when the product heat load is large. When it decreases, the air velocities can be reduced, which also reduces the fan heat load and total energy consumption.

The simulation program described in Paper V was developed and used for finding freezing times and energy consumption for alternatives with different air velocities and temperatures. The air velocities in the space between the products from the Airpak simulation program (Paper III) were used.

Reduced air volume flow resulted in decreased energy consumption but also extended freezing times, compared with the baseline case. The alternative with the lowest energy consumption used 73.8% of the baseline case, which resulted in nearly 50% longer freezing times. In this alternative, the fan speed was reduced to 33% after 8 h. Another alternative, with fan speed reduced to 50% after 10 h, resulted in an energy consumption of 80.5% of the baseline case and extended freezing times by 4 h (to 24 h).

Decreased air inlet temperatures were also simulated. This resulted in shorter freezing times, but the net effect on energy consumption was not distinct, since it depends on the rest of the refrigeration system.

Fan heat load of total heat load was also given in the paper. In the baseline case, the fan heat load was almost 50% of the total heat load at the end of freezing. In the cases where the air fan speed was reduced to 67%, the fan power load at the end was about 25%, and when the speed was reduced to 33%, it was 5%.

In further work, more detailed models of the refrigeration system could be included. A natural issue for future research would be to investigate the effect of gradually reducing the fan speed throughout the freezing process.

Chapter 5

Discussion

The main motivation for increasing energy efficiency in industrial refrigeration systems is to reduce costs. Refrigeration requires electricity, which has become increasingly expensive over the last decades. This forces industry to evaluate its energy use and introduce energy saving measures. However, improving energy efficiency is not only about changing technologies, but also making people aware of what needs to be done to obtain these efficiencies, such as knowing how freezers are turned on and off, closing doors and turning off lights, and finding the most efficient ways to use wash water. The condenser water temperature is an essential aspect of the condenser pressure and thus in determining compressor power consumption. During periods with high sea water temperatures, the freezers are more costly to operate, which could be included in the evaluation of production capacities.

Another central advantage of saving energy is the possibility of decreasing greenhouse gas emissions. The greatest impact from refrigeration on global warming is not from refrigerant release[1], but from electricity production (Pearson, 2008c). Most of Norway's onshore electricity production is from hydropower, which is essentially free of greenhouse gas emissions, but increasing demand means that some electricity (produced from nuclear power, coal, oil and gas) also has to be imported. Increasing energy efficiency can make power plant expansion less necessary and make the country less dependent on imported power.

The main electricity consumers of an industrial refrigeration system are the compressors and the freezing tunnel fans, which have been the main focus of this work. An industrial refrigeration system often has several compressors in parallel and screw compressors are common for large pressure

[1] and certainly not from ammonia systems since GWP for NH_3 is zero

ratios. The capacity regulation can be done with slide valves, which is easy but not energy efficient, since power consumption is not reduced equally. Papers I and II include an evaluation of a fish freezing plant in Norway, where an alternative compressor operation was suggested. The most important aspect of this alternative is avoiding having more than one screw compressor on part-load operation. The system had different sized compressors that could be used for many levels of cooling demand. Another possibility is to install a variable speed drive (VSD) on the compressor motor, which will reduce power consumption linearly with reduced cooling capacity. VSDs do involve investment costs, but the pay-back time is short — less than 2 years (Beggs, 2009) or less than 6 months (Tolvanen, 2008), depending on the use.

The second focus area of this work was freezing tunnels. Large air blast tunnels are common in fish freezing plants in Norway. The fish freezing plant analysed for this work had 5 freezing tunnels, each of which had a freezing capacity of more than 100 tonnes of pelagic fish. The tunnel area perpendicular to the airflow should be filled with products, leaving only small spaces where the cold air can flow. This ensures high velocities across the products and good use of the cold air. Very often there are large voids between the top of the products and the false ceiling, which will short-cut the air. It was found that about 1/3 of the air volume passed in these openings. If possible, they should be reduced or blocked to allow the air to pass over the products instead.

Another issue is uneven air velocity distribution. Products take longer to freeze when the air velocity is low. A consequence of this may be that products have not reached the desired final temperature when moved to the cold store, which could affect quality. The distribution of air velocity was analysed and discussed in Paper III. This paper also examined the influence of product packaging. Packaging results in poorer heat transfer and longer freezing times, but because of production line operation and consumer demands, it can be challenging to change or remove it. If for example the lid of the product is removed, the packages can become more unstable and off-shaped during freezing, which complicates finish handling.

Data collection and measurements were made at the fish freezing plant used as the baseline plant in this work. This gave varying results, but definitely provided insights into how such a large system works. The data from the plant was used in several of the papers, both as input data into the simulations and for validation of the simulation results. Product modelling was described in Paper V, where a finite difference model was used. This was also used in Paper VI to determine energy consumption of the refrigeration system and freezing times for products. In Paper IV, the simulation program

Modelica was used. The results from these simulations were also compared with the measurement data.

In Paper VI, the energy use of air fans and refrigeration system was calculated, along with freezing times. It was found that a reduction in air fan speed resulted in decreased energy consumption and longer freezing times. A reduction at an earlier point of time gave larger reductions in energy consumption and longer freezing times than at a later point. If it was reduced to 33% after 8 h, the freezing time was extended with 47% compared to the baseline case. The energy consumption was reduced to 73.8% of the baseline case. An extended freezing time with 20%, 4 h longer than the baseline case, resulted in energy consumption of 80.5% of the baseline case.

Another possibility that was analysed in Paper VI was to reduce the temperature of the inlet air. This resulted in shorter freezing times, but did not particularly affect the total energy consumption. This approach need to be evaluated before implemented on a refrigeration system, especially if other freezers are in the same system.

Chapter 6

Conclusions

It is profitable in many different ways to increase the energy efficiency of industrial refrigeration systems, not only for the owners, but also for society at large, and for the environment.

Screw compressors with slide valve regulators are common in the fish freezing industry in Norway, but this type of regulation should be avoided for best energy efficiency. In systems with different sized compressors, a good control and operating system can be enough to avoid unnecessary part-load operation. If that is not enough, a variable speed drive installed on one compressor motor will give easy capacity control, without excessive electricity consumption.

When designing and operating a freezing tunnel, the aim should be to have a fully stacked tunnel with even air velocity distribution. The air should only pass through small spaces between the products to obtain high velocity and high heat transfer from the products. If the air takes shortcuts in the tunnel, this results in unnecessary energy use and longer freezing times. Uneven air distribution results in products that require different freezing times, which can affect the quality of the product as well as lead to unnecessarily long freezing times. The air velocity distribution can be improved with guide blades.

The air fans use electricity, but also add heat to the freezing tunnel, which has to be removed by the refrigeration system. If the air fan speed is reduced by using a variable speed drive, energy consumption can be decreased, but the freezing time will be extended. This could be a good alternative as long as the production rate is not at a maximum. Another possibility is to decrease the evaporation temperature slightly at the end of the freezing process, which will reduce the freezing time.

Other important issues for increasing energy efficiency are system design, maintenance and upgrading, and integration with other systems, for example a heating system.

The main conclusions from this work were:

Compressors

- Part-load characteristics were established for different sized compressors in an ammonia refrigeration system, which were different from the characteristics provided by the producer.
- A system with several compressors should have a maximum of one compressor on reduced capacity, especially if the capacity is regulated using a slide valve.
- A system with several different sized compressors can be operated efficiently without variable speed drives, if the regulation of compressors using slide valves is avoided.
- Variable speed drives make capacity regulation easy and energy efficient.
- The calculated COP of the system on a day-basis was improved by 1.6% to 11.8% for a period with a high production rate.
- The COP for a period with a low production rate could be improved even further, from about 1.4 to 1.8 (29%).

Freezing tunnels

- The air velocity distribution in a freezing tunnel is always uneven, which leads to different freezing times for products.
- A simple guide blade in the air flow at the most critical location in the freezing tunnel can help even out air velocity distribution.
- Total energy consumption can be decreased by reducing air fan speed during the freezing period, but with extended freezing times.
- By reducing the air fan speed to 33% after 8 h, the simulation resulted in a total energy consumption of 73.8% of the baseline case. The freezing time was extended with 9.3 h (47% longer than the baseline case)

- When the air fan speed was reduced to 50% after 10 h, the total energy consumption was 80.5% and the freezing time was extended by 4 h (to 24 h).
- In the baseline case, the fan heat load was almost 50% of the total heat load at the end of the freezing. In the cases with air fan speed reduced to 67%, the fan heat load at the end was about 25%, and the for the 33% cases it was 5%.
- Reducing air inlet temperature resulted in shorter freezing times of all of the alternatives.
- Reducing air inlet temperature did not result in significant changes in total energy consumption, and this approach should be evaluated further.

Chapter 7

Suggestions for further research

Several energy efficiency issues are discussed in this thesis and energy efficiency should continue to get attention in the years to come. The methods used for this work, measurements and computer simulations, have been useful and are recommended as tools for future research. The simulation program for the freezing tunnels could be extended by including more detailed models for the refrigeration system and control strategies.

Modelica is an existing software program that can be used in the transient modelling of refrigeration systems. This program seems to be fairly simple and flexible, and dynamic control of the air fans is also possible, for example.

Compressor operation differs between different refrigeration plants and depends on the system's age, its size and other parameters. Operations can be improved by changing the control system and installing variable speed drives. The evaporation temperature should be further analysed, to see if a sliding set point will lead to higher or lower energy consumption. The condenser and the evaporator temperature need to be seen in a larger context and not just in connection to compressor capacity, but in association with product rate, surrounding conditions and other operation parameters.

The energy consumption of freezing tunnels can be improved. Methods for optimizing fan operation should be further evaluated. Many tunnels do not have optimal air velocity distribution, which could be improved by installing guide blades and baffles. However, it is not always clear exactly how these improvements should be made, which means that that optimization programs should be developed to enable their design.

A refrigeration system produces waste heat at different temperature levels. The amount of heat produced by condensers is often large, but is of low quality. Heat from the compressor oil is typically at 50°C and is therefore of higher quality. A more comprehensive analysis of the use of waste heat both inside the processing plant and in the nearby community could be valuable. Another interesting area of research is thermal energy storage. Accumulated heat or stored refrigeration capacity could be produced during off-peak hours, to be used when the demand for heating or cooling is larger.

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Appendix I

Paper I

REDUCING POWER LOAD IN MULTI-COMPRESSOR REFRIGERATION SYSTEMS BY LIMITING PART-LOAD OPERATION

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ABSTRACT

Compressor capacity regulation in a large refrigeration system was analysed. The system used 5 screw compressors and ammonia as the refrigerant, with slide valves to regulate the compressors and match their refrigeration capacity with product freezing loads. Compressors in the existing system can operate simultaneously with reduced capacities, which results in reduced energy efficiency. An experimental analysis of compressor operation was undertaken and a model for optimal compressor operation for energy efficiency was developed. The optimized operation was based on adjusting the cooling capacity by varying the number of compressors in full operation and selecting the compressor with the best part-load characteristics for the remaining part-load requirements. Changing from the existing to a more optimized operation would increase the coefficient of performance for the system by 1.6-11.8 %, depending on if it is full production or part-load production.

1. INTRODUCTION

Throughout history, fishing and the seafood industry have made important contributions to Norwegian society. Today, seafood is one of Norway's largest exports (after oil, gas and metals), mostly in fresh, frozen or dried form, and exported primarily to Denmark, Russia and Japan (Statistics Norway, 2006). Because of the increasing interest in Norwegian seafood and its substantial development potential, the seafood industry has been designated as one of the government's priority areas.
Norway's stationary energy system is to a large extent dependent on electricity, which is mainly based on hydropower and has been fairly inexpensive until this last decade. Consequently, industry has not focused on electricity savings. This is clearly evident in the seafood industry, where the energy consumption per kilogram of processed fish varies significantly from one factory to another. (Enova, 2002) Electricity prices are expected to continue to increase, which will force the industry to re-evaluate its energy systems and process controls.
The opportunities for energy savings in the seafood industry are many. Refrigeration systems are the main electrical consumer, and are used for chilling, ice production, cold storage and freezing of fish. Possible improvements include better dimensioning of the system (currently, the design cooling load is often higher than operating cooling load), better system regulation of the components, and a more uniform use of energy over a 24-hour period. A larger analysis showed that much could be gained by integrating refrigeration systems with heating systems. (Gjøvåg 2003), (Aprea et al, 2007)
One objective of this paper was to present an analysis of the behaviour of compressors under part-load operation. Another objective was to use the data from part-load operation in a model that provided optimized operation for the compressor system. The economic benefits of this approach as compared to today's systems were then calculated.

2. REDUCING REFRIGERATION CAPACITY WITH SLIDE VALVE REGULATION

Screw compressors have often been used by the Norwegian seafood industry when large capacity is needed and only one stage is required. A one-stage screw compressor can work with a higher pressure ratio than a one-stage reciprocating compressor. Many screw compressors employ slide valve regulation, which should give continuous capacity control between 10 % and 100 %. A clear drawback of this system is its poor efficiency when refrigerating capacity is reduced below 100 %. The power requirement is almost the same at part-load operation as it is at full load. (Brendeng 1979)

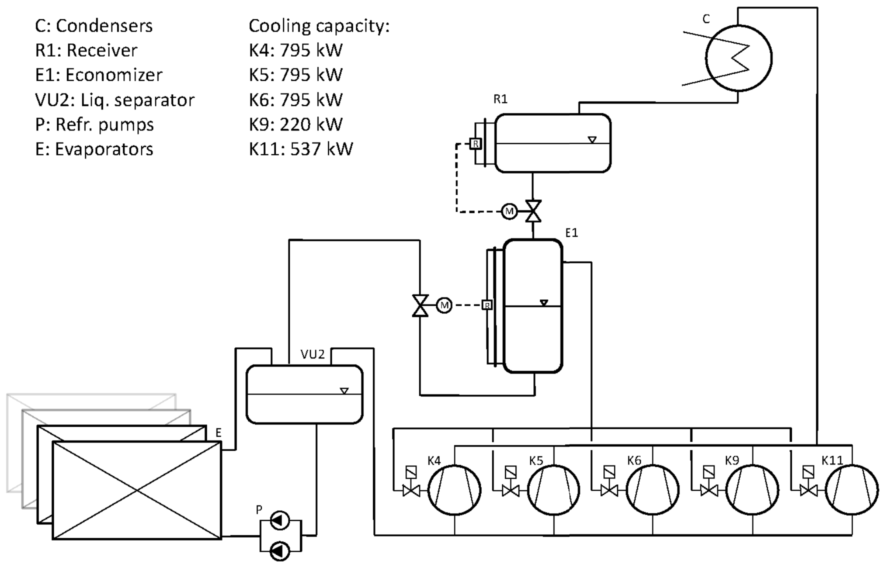

Figure 1. A simplified model of the refrigeration plant (5 of 11 compressors shown)

The poor efficiency of part load operation is mainly related to two factors, friction and volume ratio change. When the slide valve is in use, a slot opens for the refrigerant to vent back to the suction side. This leads to friction in the gas and a change in the volume ratio of the compressor. (Stoecker 1998)
Design data for part-load efficiencies of screw compressors are often higher than they are in reality. Because the edge of the slide valve slot usually is situated at 80 % of the swept volume, efficiencies fall rapidly as soon as the slide valve is opened. The slide valve position is not the same as the cooling capacity. At 90 % of slide valve position, the cooling capacity can be as low as 50 % of full capacity. (Gosney, 1982)
Another important aspect is the control system. Many fish processing industries have very simple control systems, where there is little communication between components. The compressors are regulated with a thermostat and more than one compressor can work at a partial load at the same time.

3. MEASUREMENT AND METHODS

A processing plant for pelagic fish was chosen for the analysis of the compressor regulation system. The plant is situated in the south-western part of Norway, and has an annual production volume of 50 000 tonnes of frozen fish and fish products. The plant was built in 1993.

3.1. The processing plant refrigeration system

The refrigerant system has 3 reciprocating and 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW. The reciprocating compressors work in two stages and an economizer is used with the screw compressors. The refrigeration system supplies cooling to 5 batch freezing tunnels, 10 plate freezers, 3 spiral freezers and a freezing storage. The plant also has RSW units and ice machines. The system contains 30 tonnes of ammonia as the refrigerant. The freezing capacity is 625 tonnes of fish per day with the storage capacity at 10 000 tonnes. (Solheim, 2006)
The screw compressors are divided into two subsystems that have different pressure levels. The first has an evaporator temperature of -38 °C and contains of 5 screw compressors. The other subsystem includes 3 screw compressors and has an evaporator temperature level at approximately -42 °C. The first subsystem and its compressors have been evaluated in this paper. A simplified model can be seen in Figure 1.
Since the conditions in a large industrial plant are complex, making measurements is quite complicated and it is not possible to measure all variables. The plant is equipped with a control and monitoring system that continuously logs temperatures, pressures and other data from the refrigeration system. A period of 9 days has been selected and data from this period have been analysed. The refrigeration system was operated at full capacity during most of this period. The focus was on the compressors.

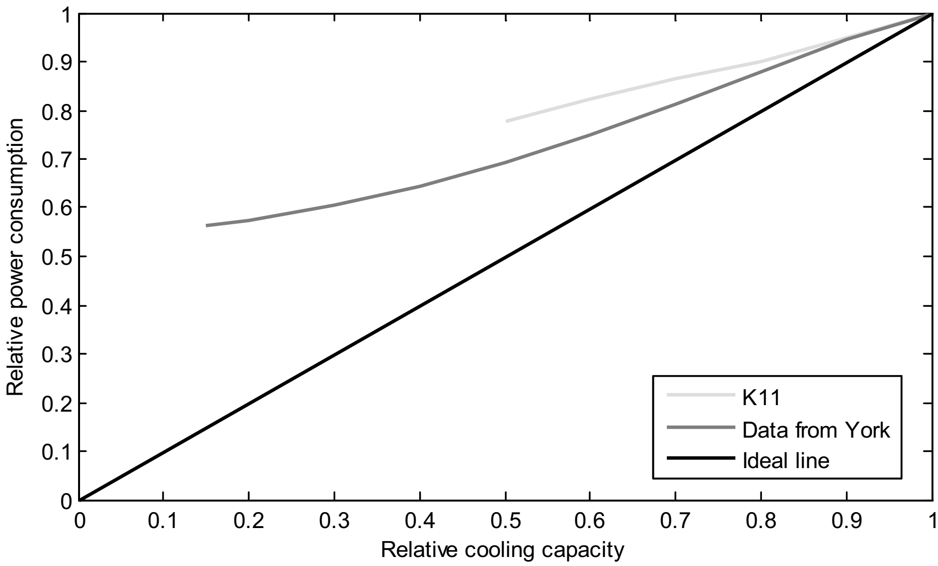

Figure 2. Part-load capacity curve for compressor No. 11

3.2. Part-load capacity curves of the screw compressors

One objective of this paper was to analyse the behaviour of the compressors when the cooling capacity was reduced by screw compressor slide valves. This is illustrated in a part-load capacity curve (an example is given in Figure 2). The curve shows the relative power consumption by relative cooling capacity. Part-load capacity curves have been constructed for each of the screw compressors.

The real power drawn from the electricity net was calculated from the net voltage, the current to the compressor motor and the power factor. The calculated real power divided by the dimensioned power of the compressor motor (given by the compressor manufacturer) is shown in the part-load capacity curve.

The isentropic efficiency and the motor efficiency are represented by the total efficiency. Since it was not possible to measure the mass flow of ammonia, it was difficult to establish an accurate isentropic efficiency. Instead, the mass flow of ammonia had to be calculated from an assumed isentropic efficiency. Vittersø (1987) showed a fairly constant isentropic efficiency for a screw compressor at different pressure ratios. The pressure ratio at dimensioning capacities at the plant was 10.8, which gives an isentropic efficiency of 56 %. The motor efficiency was assumed to be 100 %. In calculating the mass flow rate, it was assumed that the efficiencies were constant during the period. (Haugland, 1993)

Pressures and temperatures were measured on both the evaporator and the condenser side. Pressures are gauge pressures, and measured atmospheric pressures were therefore also included in the calculations. Enthalpies were calculated with an MS Excel application developed at NTNU and Sintef Energiforskning. These calculations were based on the Martin-Hou state equation described by Chan and Haselden (1981). Slide valve positions for the screw compressors were measured and an equation from Sabroe was used to convert the position percentage to capacity percentage. (Solheim, 2006)

3.3. Optimization model

A model for optimized operation of the compressors was developed to give the optimized operation for each compressor in the system, with minimized power consumption, provided that the total refrigeration load requirement is met. The object function is:

$$\min_x f(x) = \sum_i \sum_j P_{\max}(i) \cdot \left[x(i,j) \cdot a(i,j) + d(i,j) \cdot b(i,j) \right] \tag{1}$$

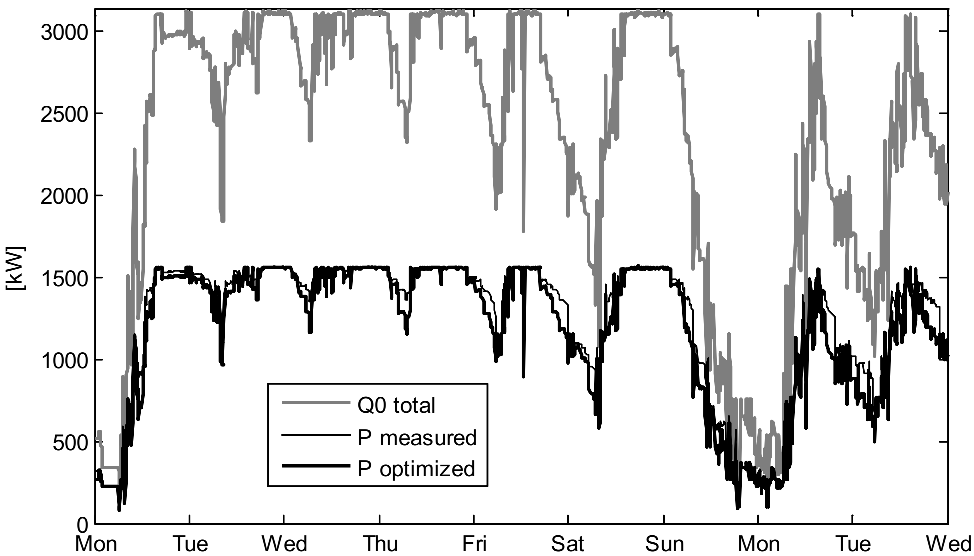

Figure 3. Cooling capacity, measured and optimized power load for the selected period

$P_{max}(i)$ is the power load of compressor i at maximum capacity, i stands for the different compressors and j is the interval in the part-load curve. x is a matrix of the relative power load, a and b are the slope and the intercept from the part-load curves for each compressor, and d is an operational variable used to make sure that each compressor is only in one interval at a time.
The optimization problem has both equality and inequality constraints. The equality constraint is:

$$Q_{0tot} = \sum_i \sum_j x(i,j) \cdot Q_{0\max}(i) \tag{2}$$

Q_{0tot} is the total cooling capacity; this is the input to the optimization problem. $Q_{0max}(i)$ is the maximum compressor capacity for each compressor. The capacity curves are divided into intervals so that each interval has its own set of linear regression constants. The first two inequality constraints set the limits for x in each interval.

4. RESULTS AND DISCUSSION

4.1. Part-load capacity curves

The part-load capacity curves demonstrated the relation between the relative cooling capacity and the relative power load. The relative values were calculated using a maximum cooling capacity (Q_{0max}) and maximum power load (P_{max}) for each compressor. Maximum values for three of the compressors were provided by the manufacturer. For compressors 9 and 11 it was necessary to adjust these values to better suit the measured values. This was done by examining the graph with all measured values for the power load to locate a mean value where most of the maximum values were located. The difference between design data and measured data was not analysed further.
It is important to note that the curve in Figure 2 diverges from the ideal line when the relative cooling capacity changes from 1 to 0.9. This leads to considerably lower energy efficiency when the compressor runs at any capacity lower than 100 %. The calculated curve and the curve from the compressor manufacturer are rather similar for cooling capacities between 0.8 and 1, but differ at lower cooling capacities.
Power loads for cooling capacities lower than 0.5 were not calculated because there were not enough data to make an accurate calculation.

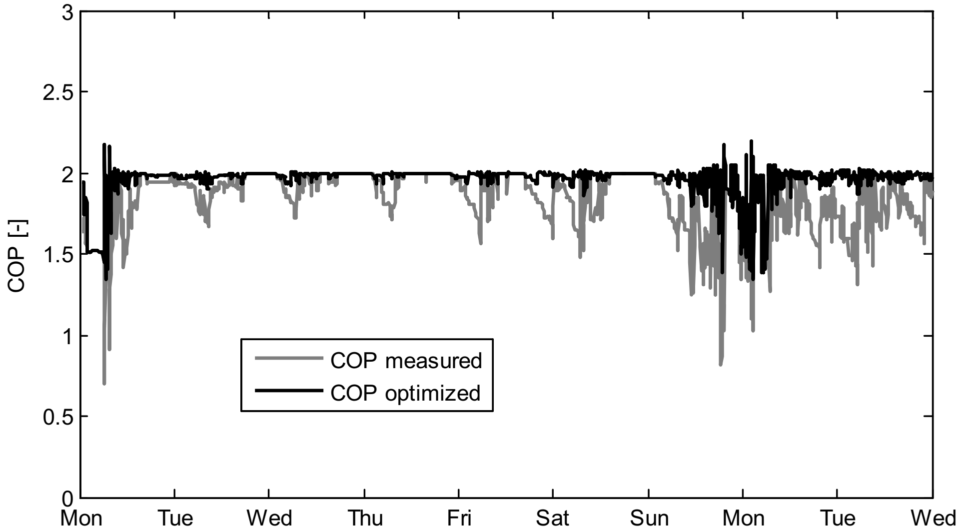

Figure 4. COP for measured and optimized system

The pressure difference in the system was around 11 for dimensioning values. Depending on many factors, the operational pressure difference varies. It is possible to make different part-load capacity curves for different pressure levels, but a simpler method was used here. The measured data gave a variation in pressure difference between 9 and 11. Mosemann et al. (2007) showed that the differences in part-load characteristic curves were not significant with these small variations in pressure differences.

4.2. Optimized system performance for a given period

The part-load capacity curves were calculated based on measurement data from a period of nine days. The same period was used in the following calculations. The cooling capacity was calculated from the slide valve position, maximum cooling capacity and a conversion graph given by Sabroe (Solheim, 2006). The power load was calculated from relative cooling capacity, part-load capacity curves and a maximum power load. The values are shown in Figure 3. This simplified approach was used to match the calculated values with the optimized results. The power load of the optimized system should be the same as in the measured system when the compressors were all in 100% operation.

The measurement period took place during fishing season and the plant was under full production for 8 of 9 days. The processing plant operates all year round, but has maximum production for only about 60 days of the year. At peak production time, the large freezing tunnels are full, with a maximum capacity of 625 tonnes per 24 hours.

The period was selected because of its large production load. Large production leads to a more stable system with more compressors operating at full load, which is beneficial for the part-load capacity curves. However, the greatest number of improvements can be made when the system is at part-load operation. Systems like this use a great deal of energy even at low production rates because the compressors are designed for their most efficient operation at a high load.

Table 1. Improvement in COP between measured and optimized system.

	Mon	*Tue*	*Wed*	*Thu*	*Fri*	*Sat*	*Sun*	*Mon*	*Tue*	*Total*
COP measured [-]	1.75	1.92	1.96	1.96	1.88	1.90	1.74	1.75	1.77	1.85
COP optimized [-]	1.88	1.98	1.99	1.99	1.98	1.99	1.94	1.91	1.98	1.96
Improvement [%]	7.4 %	3.1 %	1.6 %	1.7 %	5.3 %	4.5 %	11.6 %	9.2 %	11.8 %	6.1 %

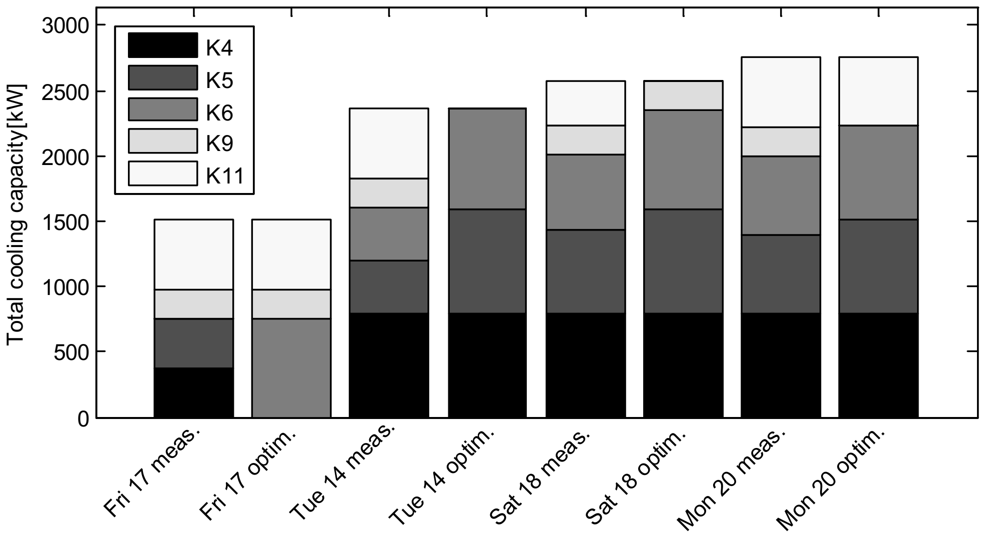

Figure 5. Examples of different compressor operations for measured and optimized systems

The coefficient of performance (COP) was calculated from the cooling capacity and the power load. The capacity and load for each compressor was added to give the total values. Graphs for COP for both the measured system and the optimized are shown in Figure 4. Table 1 gives the average values for COP for each of the days and the total period. It can be seen that the largest improvements can be made during the days of low production (end of the period). A more optimized operation can increase COP by 11.8 % on the second Tuesday, but only 1.6 % on Wednesday. The average improvement in COP for the period is 6.1 %.
The major reason for the increase in COP is a more stable system with less part-load operation. During lower production periods for the existing system, more than one compressor may operate at part-load operation. Only one compressor operates at part-load in the optimized system.
The optimization model does not account for the rate of on/off regulating of the compressors. In reality this is of importance. There are set values for the time period between each start and each stop and vice versa. Otherwise the compressors can be damaged. A more advanced optimization model would include this. The error from excluding this from the model is probably not large, especially since three of the compressors were the same size.

4.3. Examples of four different compressor operation schemes

Four different compressor part-load operation combinations were selected, see Figure 5. The cases included both measured values and optimized values, all of which clearly showed the improvement potential. It is possible to cut the amount of compressors by 1-2 in each instant. The y-axis had a maximum value of 3142 kW, which was the maximum total capacity of the system. This illustrates how the system load compared to the total.

4.4. Electrical energy savings and cost differences

The total electrical energy load and the electricity cost were calculated for the selected period, both for the measured system and the optimized system. The difference was 13 MWh (268 - 255 MWh) when the cooling capacity was 505 MWh. With an electrical energy cost of 6 cents/kWh, the cost savings would be € 770 for the period. The saving during days with high production was about € 40, whereas it could be up to € 170 during days of low production. The processing plant normally operates about 60 days per year at full production. A rough estimation based on these numbers gives an annual saving of € 50 000.
The average specific energy consumption for the pelagic industry was 218 kWh/tonnes in 2002 (Enova, 2002). It is not possible to calculate this value for this example, since the existing data apply only to a part of the total plant, but it is possible to show the improvement compared with the average values. With 3400

tonnes of fish frozen during the period, it would be possible to decrease the specific energy consumption by 2% relative to the average values for the pelagic industry.

5. CONCLUSIONS

A large amount of the electricity used in refrigeration systems goes to operating the compressors. This research has therefore focused on energy optimization in the compressor system. Since screw compressors with slide valve regulation are often used in the fish processing industry in Norway, improvements in this area can result in large energy savings for the whole industry.

Measurement data were assembled and analysed for a Norwegian fish processing plant. Part-load capacity curves were created for 5 screw compressors. These curves showed a behaviour that was worse than the design data provided by the manufacturer. The part-load capacity curves also illustrated that part-load operation was significantly less efficient than full load and that the differences were larger as the loads were lowered.

One of the challenges in the freezing plant studied was to use already installed cooling capacity in a more optimal way. The compressors were often running on part-load operation, which resulted in high energy consumption per tonne of processed fish. Optimal control of the compressors was an important factor for reducing the refrigeration energy demand.

Based on measurement values from a nine-day period where the industrial plant was in full production, cooling capacities and power load were calculated. An optimization program was used to optimize the operation of the compressors. The results showed differences between the measured and optimized system. The average COP for the system could be improved from 1.85 to 1.96 for the period. If the average COP was calculated per day, the improvement differed from 1.6 % to 11.8 % depending upon whether the production was high or low on that day. The COP increased because the optimized system had only one compressor in part-load operation when the existing system had more.

With large systems like the one described in this paper, it is possible to optimize the operation of the compressors using our approaches. There are also other ways of controlling the capacity in a multi-compressor system. Other researchers have suggested changing the cooling load by using variable speed compressors (Qureshi, Tassou, 1996), (Aprea et al, 2007), (Mosemann et al, 2007), especially for small-size refrigeration systems. However, for large refrigeration systems with several different size compressors, optimal control could probably be as efficient as installing a frequency inverter.

NOMENCLATURE

a(i,j)	slope for part-load curve	j	interval (5 alternatives)
b(i,j)	intercept for part-load curve	P_{max}	power load at maximum capacity [kW]
COP	coefficient of performance	Q_{0max}	maximum cooling capacity [kW]
d(i,j)	operational variable	Q_{0tot}	total cooling capacity [kW]
f(x)	object function in optimization problem	RSW	refrigerated seawater
i	compressor (5 alternatives)	x(i,j)	relative cooling capacity

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Appendix II

Paper II

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Reducing power consumption in multi-compressor refrigeration systems

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ABSTRACT

An experimental analysis of compressor operation in a large refrigeration system was undertaken and a model for optimal compressor operation for energy efficiency was developed. The system used 5 screw compressors and ammonia as the refrigerant, with slide valves to regulate the compressors and match their refrigeration capacity with product freezing loads. Compressors in the existing system can operate simultaneously with reduced capacities, which results in reduced energy efficiency. Optimized operation was made both with and without a variable frequency drive. The results showed that the most electrical energy can be saved during days when not all of the tunnels were loaded. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation of the refrigeration system.

Diminution de la consommation d'électricité des systèmes frigorifiques à plusieurs compresseurs

Mots clés : système frigorifique ; système à compression ; compresseur à vis : modélisation ; optimisation : régulation ; vitesse variable ; économie d'énergie

1. Introduction

Throughout history, fishing and the seafood industry have made important contributions to Norwegian society. Today, seafood is one of Norway's largest exports (after oil, gas and metals), mostly in fresh, frozen or dried form, and exported primarily to Denmark, Russia and Japan (Statistics Norway, 2006). Because of the increasing interest in Norwegian seafood and its substantial development potential, the seafood industry has been designated as one of the government's priority areas. Norway's stationary energy system is to a large extent dependent on electricity, which is mainly based on hydropower and has been fairly inexpensive until this last

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Nomenclature

AC	alternating current
$a_{i,j}$	slope for part-load curve
$b_{i,j}$	intercept for part-load curve
COP	coefficient of performance
DC	direct current
$d_{i,j}$	operational variable
$f(x)$	object function in optimization problem
i	compressor (5 alternatives)
j	interval (5 alternatives)
K4, K5, K6, K9, K11	included compressors
P_{max}	power consumption at maximum capacity [kW]
PWM	pulse-width-modulated drive
Q_{0max}	maximum cooling capacity [kW]
Q_{0tot}	total cooling capacity [kW]
RSW	refrigerated seawater
$x_{i,j}$	relative cooling capacity

decade. Consequently, industry has not focused on electricity savings. This is clearly evident in the seafood industry, where the energy consumption per kilogram of processed fish varies significantly from one factory to another (Enova, 2002). Electricity prices are expected to continue to increase, which will force the industry to re-evaluate its energy systems and process controls. There is also an interest for more eco-products on the market. A reduction in electrical energy usage will not only be environmental friendly in itself, but would also lead to lower emissions of pollutants such as CO_2. The opportunities for energy savings in the seafood industry are many. Refrigeration systems are the main electrical consumer, and are used for chilling, ice production, cold storage and freezing of fish. Possible improvements include better dimensioning of the system (currently, the design cooling load is often higher than the operating cooling load), better system regulation of the components, and a more uniform use of energy over a 24-h period. A larger analysis showed that much could be gained by integrating refrigeration systems with heating systems (Gjøvåg, 2004; Aprea et al., 2007).

The objective of this paper was to use data from part-load operation (Widell and Eikevik, 2008) in a model that provided a more optimal operation of the compressor system, both with and without a variable frequency drive. The model gives the potential of electricity savings. The economic benefits of this approach as compared to today's systems have been calculated. We have not detailed how to implement this optimizing model in an actual control system. Examples of this can be found in Leducq et al. (2006). They have described a non-linear control algorithm for predicting optimal operation. Several objectives can be defined simultaneously and they have to be weighted against each other. The laboratory plant described in this paper has a single compressor with a variable speed drive, but the algorithm can be used on multi-unit systems also.

2. Reducing refrigeration capacity with slide valve regulation

Screw compressors have often been used by the Norwegian seafood industry when large freezing capacity is needed and only one-stage compression is required. A one-stage screw compressor can work with a higher pressure ratio than a one-stage reciprocating compressor. Many screw compressors employ slide valve regulation, which should give continuous capacity control between 10% and 100%. A clear drawback of this system is its poor efficiency when refrigerating capacity is reduced below 100%. The power requirement is almost the same at part-load operation as it is at full load (Brendeng, 1979). The poor efficiency of part-load operation is mainly related to two factors, friction and volume ratio change. When the slide valve is in use, a slot opens for the refrigerant to vent back to the suction side. This leads to friction in the gas and a change in the volume ratio of the compressor (Stoecker, 1998). The design data for part-load efficiencies of screw compressors are often higher than they are in reality. Because the edge of the slide valve slot usually is situated at 80% of the swept volume, efficiencies fall rapidly as soon as the slide valve is opened. The slide valve position is not the same as the cooling capacity. At 80% of slide valve position, the cooling capacity can be as low as 50% of full capacity (Gosney, 1982). Fig. 1 illustrates this relationship. The curve is different for different types of compressors. Values from the compressor manufacturer have been used in our calculations. Another important aspect is the control system. Many fish processing industries have very simple control systems, where there is little communication between components. The compressors are controlled with a thermostat and more than one compressor can work at a partial load at the same time.

3. Reducing refrigeration capacity by varying compressor speed

A more efficient way to modify the refrigeration capacity than with the slide valve is to vary the compressor speed. This can be done with a variable frequency drive connected to the AC

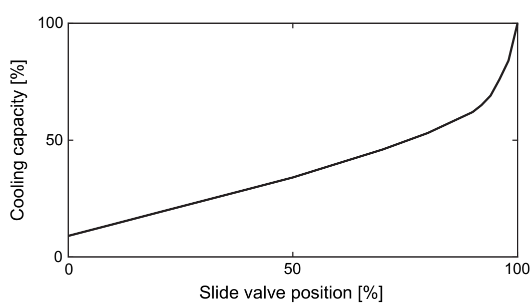

Fig. 1 – Theoretical curve showing cooling capacity as a function of compressor slide valve position.

motor. The most common one is a pulse-width-modulated drive (PWM drive). This operates firstly by converting the AC voltage into a fixed DC voltage. Secondly, the DC voltage is inverted back to AC in an inverter, where the frequency and voltage can be selected within a certain range (Avallone and Baumeister, 1996). Most of the literature found in this area analyses varying compressor speed using reciprocating or scroll compressors and not screw compressors. The frequency range for these types of compressors is 30–50 Hz for reciprocating compressors and 15–50 Hz for scroll compressors. Reciprocating compressors cannot use lower frequency than 30 Hz because of noise, vibration and lubrication troubles (Aprea and Renno, 2004; Aprea et al., 2006). We assume in this paper that screw compressors also have a lower frequency limit of 30 Hz. The suggested frequency range for the compressors at the processing plant (described in Section 4.1) is 30–67 Hz. Most motors are built for frequencies from 50 Hz to 67 Hz, so that it is possible increase the frequency of the compressors beyond 50 Hz.

4. Measurement and methods

A processing plant for pelagic fish was chosen for the analysis of the compressor regulation system. The plant is situated in the south-western part of Norway, and has an annual production volume of 50 000 ton of frozen fish and fish products. The plant was built in 1993.

4.1. *The processing plant refrigeration system*

The refrigerant system consists of two subsystems, which together have 3 reciprocating and 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW (−40/+20 C). The reciprocating compressors work in two stages and an economizer is operated with the screw compressors. The refrigeration system supplies cooling to 5 batch freezing tunnels, 10 plate freezers, 3 spiral freezers and a large freezing storage area. The plant also has RSW units and ice machines. The system contains 30 ton of ammonia as the refrigerant. The freezing capacity is 625 ton of fish per day with the storage capacity at 10 000 ton (Solheim, 2006). The screw compressors are divided into two subsystems that have different pressure levels. The first has an evaporator temperature of −38 C and contains 5 screw compressors. The other subsystem includes 3 screw compressors and has an evaporator temperature level at approximately −42 C. The first subsystem (−38 C) and its compressors have been evaluated in this paper. A simplified sketch of the system can be seen in Fig. 2. Since the conditions in a large industrial plant are complex, making measurements is quite complicated and it is not possible to measure all variables. The plant is equipped with a control and monitoring system that continuously logs temperatures, pressures and other data from the refrigeration system. Two periods of 9 days each have been selected and data from these periods have been analysed. The refrigeration system was operated at full capacity during most of the first period and at reduced capacity on the second. The focus was on the compressors.

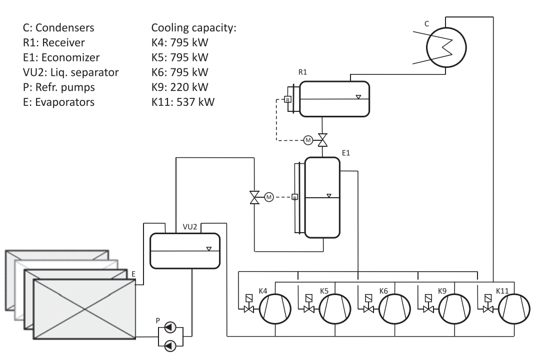

Fig. 2 – A simplified model of a part of the refrigeration plant. Only one of the two subsystems was analysed in this paper and it includes 5 screw compressors.

4.2. *Part-load capacity curves of the screw compressors*

The behaviour of the compressors was analysed and described by Widell and Eikevik (2008) and part-load capacity curves were constructed for each of the screw compressors in the system. These curves show the relative power consumption by relative cooling capacity and are used in the optimization. The closer to the ideal line, the better is the efficiency of the system. The part-load capacity curve for a compressor with variable frequency drive is given by York (Solheim, 2006) and is illustrated in Fig. 3. It can be seen from this that varying compressor speed gives a more energy efficient operation than slide valve regulation.

4.3. *Optimization model*

A linear programming model was developed for analyzing the difference between the actual (measured) system and a system with optimized operation of each compressor. The objective was to minimize power consumption, provided that the total refrigeration load requirement was met. The refrigeration loads entered into the model were from the measured period and each measured value was analysed with the model. The object function was:

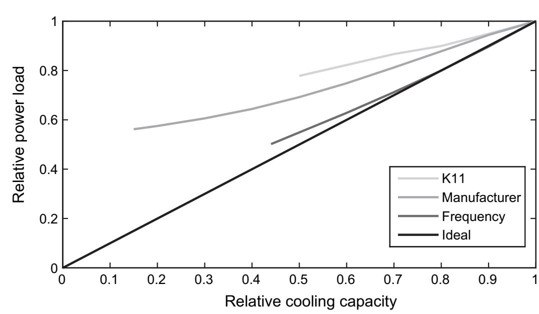

Fig. 3 – Part-load capacity curves for compressor No. 11, with calculated values, values from the manufacturer and with a variable frequency drive.

$$\min_x f(x) = \sum_i \sum_j P_{\max,i} \cdot [x_{i,j} \cdot a_{i,j} + d_{i,j} \cdot b_{i,j}] \quad (1)$$

where $P_{\max,i}$ is the power load of compressor i at maximum capacity, i stands for the different compressors and j is the interval in the part-load curve. $x_{i,j}$ is a matrix of the relative power consumption, $a_{i,j}$ and $b_{i,j}$ are the slope and the intercept from the part-load curves for each compressor. $d_{i,j}$ is a matrix of binary variables that is found together with $x_{i,j}$. It is also used in the constraints and it controls that the computer program does not allow a solution where one compressor is operated in more than one interval. The optimization problem has both equality and inequality constraints. The equality constraint is:

$$Q_{0,\mathrm{tot}} = \sum_i \sum_j x_{i,j} \cdot Q_{0\max,i} \quad (2)$$

where $Q_{0,\mathrm{tot}}$ is the total cooling capacity. This is the input for the optimization problem. $Q_{0\max,i}$ is the maximum compressor capacity for each compressor. The capacity curves are divided into intervals so that each interval has its own set of linear regression constants. The first two inequality constraints set the limits for x in each interval. The third inequality constraint sets the sum of the d-variables for one instant equal to or less than 1, which ensures that a compressor is only operating in one interval at the time. When the compressors with variable frequency drives were simulated, the data from Fig. 3 were used to find a and b. Several combinations of compressors and variable frequency drives were evaluated, but only the best ones have been presented.

4.4. *Limitations*

In reality, to avoid compressor damage, there are limitations on how often, or the times between when a compressor can be stopped and started again. This is not included in the optimization equations since these only apply to one cooling capacity at the time and not to a vector of numbers. The error from excluding this from the model is estimated to be low, due to the fact that the three of the compressors were the same size. Another limitation is that the compressors are always started and stopped in a certain order and this is also not included in the optimization. This means that a compressor with lower efficiency can be chosen instead of one with higher efficiency. It would be better for the actual system to have a more flexible way of selecting which compressor to start or stop.

5. Results and discussion

5.1. *Optimized system performance*

The part-load capacity curves were calculated based on measurement data from the first period of nine days (Widell and Eikevik, 2008). Values from both periods were used to find optimized operation of the system. The cooling capacity was calculated from the slide valve position, maximum cooling capacity and a conversion graph given by Sabroe (Solheim, 2006). Cooling capacities for both periods are shown in Fig. 4. The power consumption was calculated from relative cooling capacity, part-load capacity curves and a maximum power consumption. This simplified approach was used to make the comparison between the measured and optimized system more reliable. It was important that we compared relative values (multiplied with constant maximum values) instead of measured absolute values, since these latter varies with different factors, such as seawater temperature and mass flow in the condensers.

The first period was during the fishing season when the plant was under full production for most days. The processing plant operates all year round, but uses the freezing tunnels for only about 60 days a year. At peak production time, the large freezing tunnels are full, with a maximum capacity of 625 ton per 24 h. The first period was selected because of its large

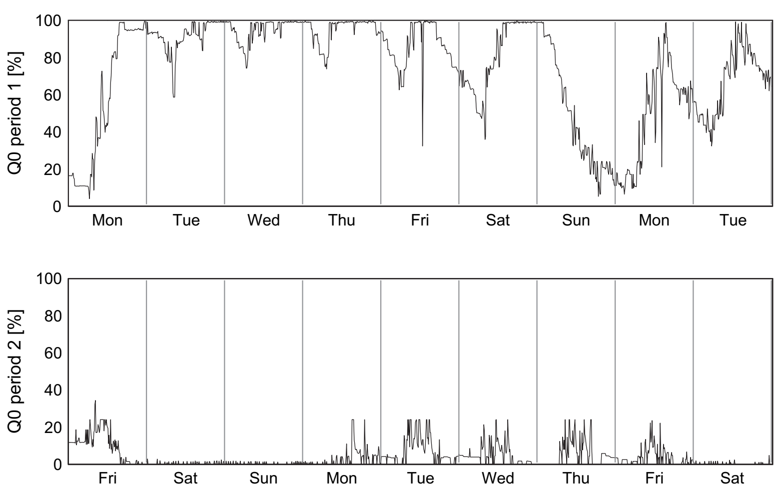

Fig. 4 – The relative total cooling capacity during periods 1 and 2.

production load. Large production leads to a more stable system with more compressors operating at full load, which was beneficial for the part-load capacity curves. We see that the greatest number of improvements can be made when the system is at part-load operation, which is during days when not all of the tunnels are loaded. Systems like this use a great deal of energy even at lower production rates because the compressors are designed for their most efficient operation at a high load.

The freezing tunnels were not in operation during the second period, which can be seen in Fig. 4. This period illustrates how the system is operated for most of the year. There is no production during the weekends; the only cooling capacity needed then is to support a small storage freezer.

The system was first optimized without variable frequency drives. All of the compressors were regulated with slide valves. In these simulations, more than one compressor could be at part-load operation, but then often above 90%. It seems to be more efficient to have two compressors at a high part-load operation than one compressor at a low part-load operation. The measured system had often more than one compressor at a low part-load operation.

The next optimizations included different alternatives for variable frequency drives. The system has 5 screw compressors, of which three are of the same size. Frequency drives were simulated in compressors K5, K9 and K11. The part-load characteristic shown in Fig. 3 was used for all of the compressors. Three different combinations of two compressors with variable frequency drives were also analysed; these were K5 + K9, K5 + K11 and K9 + K11. The results showed that it could perhaps be profitable to install one variable frequency drive, but not two.

5.2. *Coefficient of performance*

The coefficient of performance (COP) was calculated from the cooling capacity and the power consumption. Average values were calculated for each day. Graphs for the COP for both the measured and the optimized system are shown in Fig. 5 for period 1 and Fig. 6 for period 2. The greatest improvements can be made during days when not all of the tunnels were

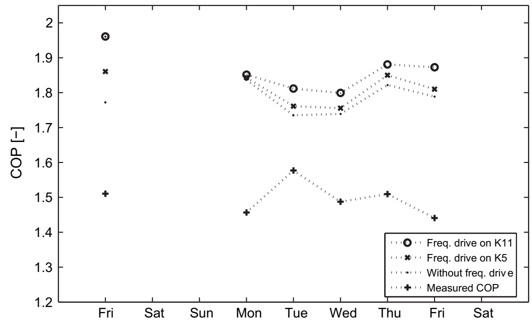

Fig. 6 – Daily average of COP for the system for period 2. The lines have been drawn to better illustrate the connections and differences between the values.

loaded, but not when the system is almost off, as it is during the weekends in period 2. Here we do not have enough data to calculate a reliable average COP. During these days the cooling load varied between zero and very low (typically with one compressor at part-load below 50%). A zero COP is not included in the average values and the optimization model could not find a correct value when the part-load was below 50%.

Cooling capacities and COPs have been sorted by cooling capacities, as can be seen in Fig. 7. From this we see that the COPs for the optimized system (gray rings and crosses) are much higher and more even than for the measured system (black dots). For high cooling capacities, COPs for both optimized and measured system are high. When cooling capacities are low, there are larger differences between the optimized system and the measured system. In the latter the COP can be different at one cooling load; this means in reality that a compressor with lower efficiency was used instead of one with higher efficiency.

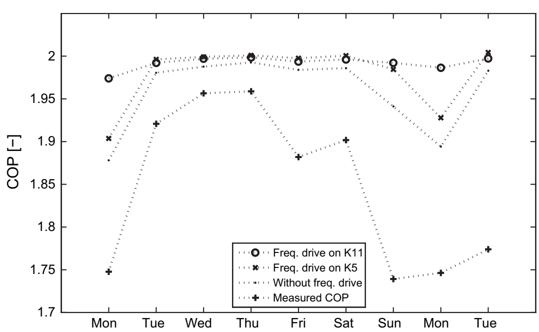

Fig. 5 – Daily average of COP for the system for period 1. The lines have been drawn to better illustrate the connections and differences between the values.

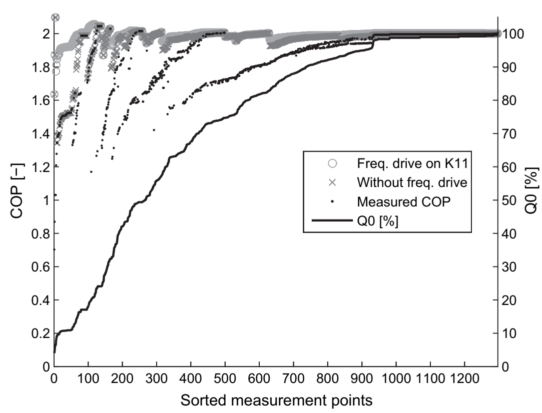

Fig. 7 – Values for COP sorted by the increasing relative cooling capacity (relative to maximum load).

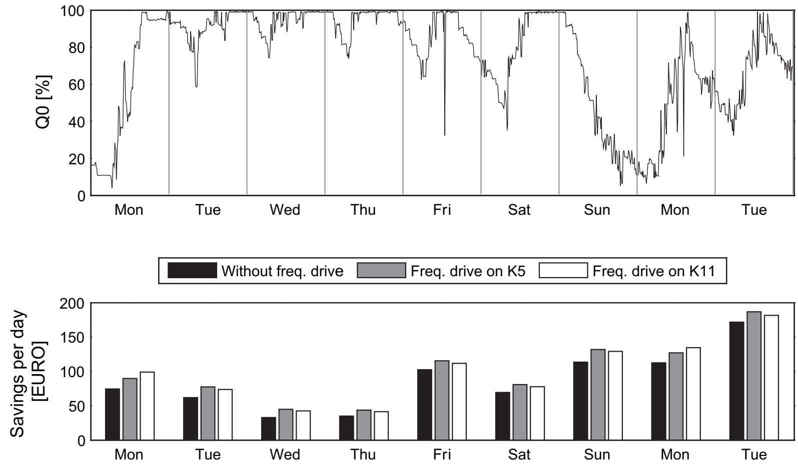

Fig. 8 – Savings per day [€] for period 1, compared with measured system, for three different optimization alternatives. Cooling capacities [kW] for the period are included.

5.3. *Electrical energy savings and cost differences*

The total electrical energy load and the electricity cost were calculated for the selected periods. The electrical energy cost was set to 0.06 €/kWh. The savings in € per day for period 1 can be seen in Fig. 8. Three different optimization alternatives were chosen, one without a variable frequency drive and two with a variable frequency drive, in compressors K5 and K11. By looking at the cooling load also displayed in the graph, we can conclude that the smallest differences between the measured and optimized system are when the production is high. It is during days when not all of the tunnels were loaded that most of the savings can be had. The difference between the optimized system with and without frequency drives is not significant. Variable frequency drive in the largest compressor, K5, seems to be favourable for most of the days during period 1. However, it cannot be concluded from this that K5 is the best compressor to connect to a variable speed drive. This kind of operation only happens for about 16 days a year. An analysis over the usage during a whole year must be undertaken.

Since the system is very complex, with many uncertainties and variation during the year, it is not easy to make a suggestion for possible savings over a whole year's operation. We have to make assumptions about the number of days with different kinds of cooling loads and assume how much

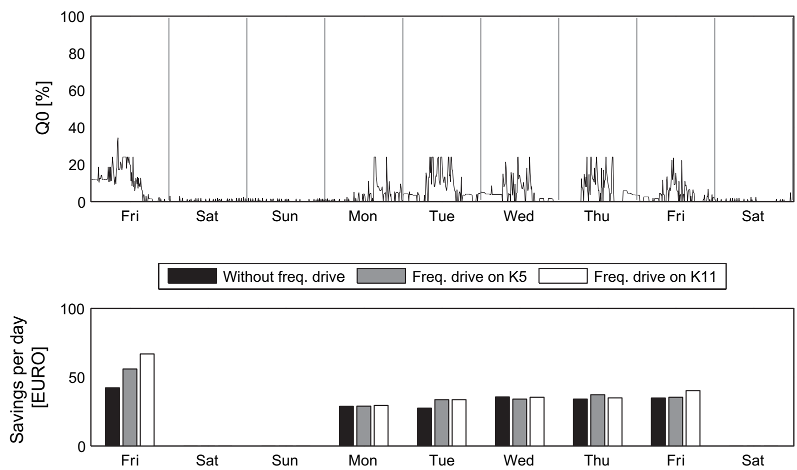

Fig. 9 – Savings per day [€] for period 2, compared with measured system, for three different optimization alternatives. Cooling capacities [kW] for the period are included. Three days did not have enough data for the graph.

can be typically saved during these days. Both periods have to be used for this analysis. The results from this analysis vary too much to provide a precise answer. The system operates much like the working days in period 2 during most of the year (about 200 days). Fig. 9 shows how much can be saved during these days. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation. There does not seem to be a large difference between optimizing with and without a variable frequency drive, but the difference is probably larger in reality. There are limitations on how often a compressor can be stopped and started, but this has not been included in the optimization model, as described in Section 4.3. The inability to include the stop/start limitations leads to more optimistic values for the system without a variable speed drive. This error is probably lower for the alternatives with a variable speed drive, since this gives better and more flexible regulation. The system tends to regulate only one compressor.

The results showed that most electrical energy can be saved during days when not all of the tunnels were loaded. This is not the case for more than about 20–40 days per year. Most typical operation is below 10% of the total cooling capacity. Our calculations gave very different results for these days, but the savings per day for compressor K11 are best in most cases. We conclude that this compressor should have a variable speed drive. This is also logical from a cooling capacity view, as it is in mid-range (see also Fig. 2).

6. Conclusions

A large amount of the electricity used in refrigeration systems goes to operate the compressors. This research has therefore focused on energy optimization in the compressor system. Since screw compressors with slide valve regulation often are used in the fish processing industry in Norway, improvements in this area can result in large energy savings for the whole industry. One of the challenges in the freezing plant studied was to use the already installed cooling capacity in a more optimal way. The compressors were often running at part-load operation, which resulted in high energy consumption per tonne of processed fish. Optimal control of the compressors was an important factor for reducing the refrigeration energy demand.

A linear programming model was developed to give the optimized operation for each compressor in the system, with minimized power consumption, provided that the total refrigeration load requirement was met. The model was used to optimize the operation of the compressors during two measurement periods. There was high production during the first period and low production during the second. The optimization was done both with and without a variable frequency drive.

The model calculated power consumptions for all of the optimization alternatives. These power consumptions were used together with the cooling loads to find the coefficient of performance (COP) for the system. Costs in € were also calculated and the difference between measured and optimized system provided possible savings amounts. Savings per day varied with different cooling loads. The results showed that most electrical energy can be saved during days when not all of the tunnels were loaded. Systems like this use a great deal of energy even at lower production rates because the compressors are designed for their most efficient operation at a high load.

The most typical operation is below 10% of total cooling capacity. Our calculations gave very different results for these days, so very precise calculations for savings per year were not possible. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation of the refrigeration system. There does not seem to be a large difference between optimizing with and without a variable frequency drive, but the difference is probably larger in reality. Our calculations show that the mid-range compressor should have a variable speed drive.

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Appendix III

Paper III

AIR VELOCITY FIELD IN AN AIR BLAST FREEZING TUNNEL

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ABSTRACT

The air velocity field inside an air blast freezing tunnel is important in heat transfer from a product and highly influences freezing time. The freezing time is a function of air properties, air velocity and product characteristics. An uneven air velocity field results in different product freezing times and inefficient energy use by the fan. Compressors and fans are the biggest consumers of electricity in a freezing tunnel, which means it is important to optimize their use. This paper presents simulations of the air velocity field in an existing freezing tunnel using Computational Fluid Dynamics (CFD) software. The air velocity field in the blast freezer is difficult to measure, but simulations can provide indications of the real conditions in the tunnel. The air velocity distribution in the tunnel is discussed as well as how this output can be used to calculate heat transfer coefficients to predict product freezing times. Alternative tunnel construction options are also analysed. A guide blade in the tunnel was installed to provide a more even air velocity field, which resulted in more uniform product freezing times. Freezing times were calculated with a modified Plank's equation and then used to visualize the influence of air velocity. The freezing times in the tunnel without a guide blade varied between 16 and 32 h. For a tunnel with a guide blade the freezing times were between 17 h and 21 h. Freezing times in the existing tunnel were between 13 and 20 h. Future work should include the air velocity field in a numerical model for product freezing.

Key words: simulation, air distribution, velocity, freezing time, food, heat transfer coefficient

1. INTRODUCTION

The freezing time is defined as the total time needed to freeze a food product from an initial temperature to a given final temperature in its thermal centre (Delgado and Sun, 2001). The freezing time is a function of air properties, air velocity and product characteristics (Kiranoudis and Markatos, 1999). The product is often determined by the customer, and air properties and velocities therefore have to be changed in order to improve the process. There is always an uneven velocity field inside a large air blast freezing tunnel because of its construction. Different velocities inside a freezing tunnel lead to different heat transfer coefficients and different freezing times (Becker and Fricke, 2004).

Cost is the main issue in food processing, along with food quality. In air blast freezing tunnels it is important for food quality that all of the products reach the designated temperature before they are stored. After-freezing in a storeroom will be slow because of lower temperature differences and often no forced airflow. This could affect product quality. It is difficult to determine when products have reached a certain temperature, and temperature measurements are often made using a sampling of products. If the freezing times are similar, this can be used to determine the correct time to unload the tunnel. If not, the products will either be unloaded too soon or the operating costs will be unnecessary high (Cleland, 1990).

In an analysis of electricity consumption in automatic meat freezing tunnels, Edwards and Fleming (1979) found that heat from the fans contributes significantly to the refrigeration load and was sometimes larger than the product refrigeration load. The fans not only use electricity to operate, they also add heat to the freezing tunnel, which then has to be removed by the refrigeration system. Uneven freezing times because of non-uniform air flows have also been found. Relatively simple design modifications can improve the freezing process and decrease electricity consumption. Examples are modules with individually regulated fans and refrigeration loads, baffles that decrease the air bypass and two-speed fans. The air distribution ceiling (false ceiling) is also important for air flow.

Huan et al. (2003) analysed the airflow inside a spiral quick freezer in which different blockage board dimensions were simulated. They found that the average velocity in the freezing zone increased with the increasing width of the blockage board, because it prevented airflow short-circuiting.

Brown and James (2006) analysed the effect of air temperature and velocity on tempering times. Although tempering is the opposite of freezing, the processes are similar, apart from the heat transfer inside the products, which are much slower in thawed than in frozen product. Packaging and air gaps between the product and the cardboard box made the heat transfer coefficient significantly lower than if no packaging had been used. Temperature differences between the product and the air also affected the freezing time.

This paper presents steady state simulations of velocity fields in air blast freezing tunnels. Different constructions were simulated to visualize the non-uniformity of the velocity field. The simulation model was based on an existing freezing tunnel. The construction of this tunnel was quite good; the different results can be used to suggest general improvements to a freezing tunnel. It was not possible to measure the velocity field inside the existing freezing tunnel because the space between the products was too small.

Models for calculating the heat transfer coefficient and freezing time are also presented. The heat transfer coefficients for different velocities can be used in other papers where a transient analysis is made. There should be more weight on the connection between air velocity, freezing times and energy usage in the industry, especially in existing processing plants.

2. METHODS

2.1. Simulation with Airpak

We made steady state simulations with Airpak, a Computational Fluid Dynamics (CFD) design tool that uses FLUENT's solver engine. The air velocity field for different constructions was visualized. Other properties could also have been determined with the software, but the focus was on air velocities (absolute). Freezing times were calculated with a simplified model to compare the different constructions. CFD was used for modelling fluid flow and for solving fluid motion and energy conservation equations. The conservation of other factors can also be included. Examples of equations for fluid motion are continuity equations and Navier-Stokes equations. (Smale et al., 2006)
Properties from an existing fish freezing plant were used to set the geometry for a model of an air blast freezing tunnel in Airpak. Figure 1 shows one of the different designs that was visualized. Air is blown by the fans across the false ceiling, through the product carriages and finally through the evaporator. The freezing tunnel has a guide blade that spreads the air flow after the channel below the ceiling. Simulations have also been done without this guide blade.
After the model was built, a mesh was generated in Airpak. The mesh should be finer closer to objects and no mesh element should have angles that are overly acute. "A good mesh is essential for a good solution" (Fluent Inc., 2007). The mesh used here was a hexahedral unstructured mesh. A solution was calculated when the mesh was determined to be acceptable. An indoor zero-equation turbulence model was used. The results were examined with plane cuts, plots and a number of other tools.

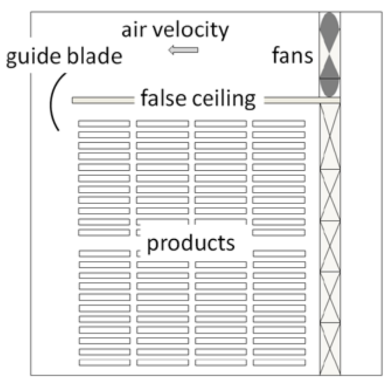


Figure 1. The air blast freezing tunnel model, drawn in Airpak, a design tool that uses FLUENT's solver engine. The figure includes fans, evaporator units, a false ceiling, a guide blade and product blocks.

2.2. Air velocity

The air velocity distribution between the products in the tunnel has a profile similar to what is shown in Figure 2. The velocity is zero at the surface of the product. The highest velocity is in the middle of the air space for most of the tunnel. The profile is different close to the guide blade. Airpak calculates the velocities for all of the nodes in the mesh. With this data, we have generated polynomial functions of different degrees (2nd, 3rd and 4th). The functions have been integrated across the height and an average velocity is found. Each product air space has 3-4 groups of velocity data that give an average velocity. The standard deviation for each air space was calculated. The average velocity should be the same for the same air space (Incropera and DeWitt, 2002), which means the polynomial function with lowest standard deviation values has been selected. Figure 4 shows simulated data and polynomial functions.

2.3. Calculating freezing time

The freezing time for a product is the total time from the initial temperature to when the thermal centre has reached final storage temperature (-18°C for this example). There are several different ways of calculating the freezing time. Analytical methods such as Plank's equation and modifications to this are easy to implement and provide a quick solution. However, these methods do not present any information about the process between the start and the end state. They are also best suited for simple geometries. Plank's equation does not include sensible heat, but that was corrected in the modifications. Numerical methods are often more accurate and detailed, but also require more information. Temperature profiles can be found and more complex geometries can be modelled. Thermophysical properties can change with time. However, the drawback is that numerical methods are time consuming to implement. (Delgado and Sun, 2001), (Salvadori et al., 1997), (Pham, 1984)

A modified Plank's equation with a geometric factor is used in this paper to visualize different constructions (Valentas et al., 1997). This cannot simulate transient progress, but can indicate the quality of the different alternatives. Analytical methods like this can only include constant transport properties and are not good enough for analysing energy efficiency, where a transient view is necessary.

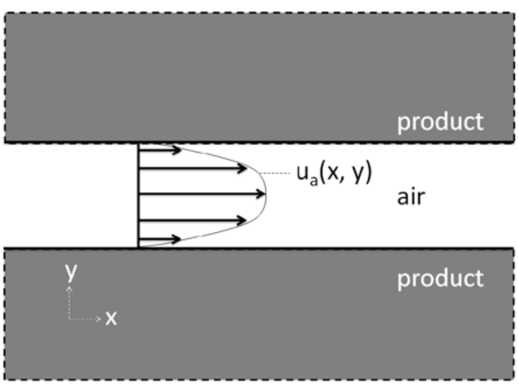


Figure 2. Air velocity profile between two products. The arrows indicate the size of the velocity for a given y value. The coordinate system shows only the direction of the coordinates, not the origin.

The freezing time is affected by both product and air properties. Thermodynamic properties for Atlantic mackerel are predicted based on tables and equations from ASHRAE (2006). The convective heat transfer coefficient has been calculated with this equation, which is valid for objects with planar surfaces (Valentas et al., 1997):

$$h_c = 7.3 \cdot u_a^{0.8} \tag{1}$$

The velocity used in the equation is an average of the mean velocities between the products (described in section 2.2). Each air space between the products has one average air velocity, which is used for calculating the convective heat transfer coefficient. The radiation heat transfer coefficient was also calculated using the following equation and included to see if radiation affects the overall coefficient (Incropera and DeWitt, 2002):

$$h_r = \varepsilon \cdot \sigma \cdot \left(T_s + T_{room}\right) \cdot \left(T_s^2 + T_{room}^2\right) \tag{2}$$

Overall heat transfer coefficients will be significantly lower if the products are packed inside a cardboard box. The packaging and the air gaps between the product and the cardboard both affect heat transfer (Brown and James, 2006). The overall heat transfer coefficients were calculated with this equation (Incropera and DeWitt, 2002):

$$U = \frac{1}{\dfrac{1}{h_c + h_r} + \dfrac{d_a}{k_a} + \dfrac{d_p}{k_p}} \tag{3}$$

Thermal conductivities were found in Incropera and DeWitt (2002) and both air and carton thicknesses were set to 1 mm, see Figure 3. The overall heat transfer coefficient for different heat transfer coefficient and packaging is plotted in Figure 6.

3. RESULTS AND DISCUSSION

Airpak was used to calculate simulated air velocities inside a freezing tunnel because it was not possible to actually measure velocity inside the tunnel. The air velocities found were used later in a larger numerical model that calculates the transient development of the temperature inside the tunnel and the products. When the model was built in Airpak, different constructions were also analysed to see how changes affected air distribution.

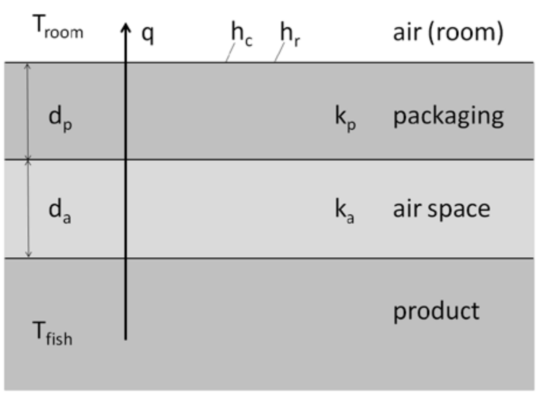


Figure 3. Illustration of the components in equation 3. The product is packed inside a cardboard carton with an air space between the packaging and the product. Heat (q) is transported from the product to the room air because of the temperature difference (T_{room}<T_{fish}).

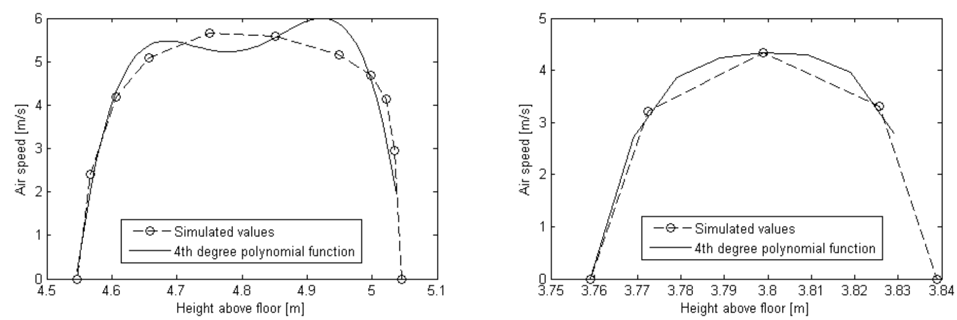
The results from Airpak had to be analysed and average velocities found. Average air velocity is used when calculating the heat transfer coefficient, which is important for finding the freezing time. We have selected a modified Plank's equation for calculating the freezing time in this paper. The starting assumption is that a low velocity results in long freezing times and vice versa. In reality it is much more complex than this, and a numerical solution provides better results.

3.1. Velocity

Using the calculations described in Section 2.2, we found that 4th grade polynomial functions suited the Airpak velocity data best. Examples of simulated values from Airpak together with the polynomial function can be seen in Figure 4. A fully developed velocity profile is parabolic (2nd degree polynomial function) for laminar flow in a circular tube. However, the air flow is not laminar between the products; the Reynolds number is between 43 000 and 90 000, which indicates turbulent flow and a flatter profile, which is best provided by 4th grade polynomial function. Fully developed turbulent flow can be found when $(L/D_h)>10$, which would be for an L larger than 2 m. The length of the product boxes was 1 m, so we can assume that turbulent flow never fully develops (Incropera and DeWitt, 2002).
We simulated a tunnel both with and without a guide blade to examine the consequence of uneven air velocity distribution. Sorted average air velocities for the air spaces around all of the products can be seen in Figure 5. The simulation without a guide blade results in velocities between 0.97 m/s and 6.29 m/s, where the simulation with guide blade results in a narrower span of velocities of 2.57-5.36 m/s.

3.2. Heat transfer coefficient

The heat transfer coefficients are calculated as described in Section 2.3. A graph with and without packaging is shown in Figure 6. The contribution from the packaging and the air can also be seen. Both are 1 mm thick, but the thermal conductivity of air is lower. Overall heat transfer coefficients with packaging were significantly lower than the original heat transfer coefficient. It seems that the importance of the air velocity decreases as the amount of packaging increases.



a) b)

Figure 4. Simulated data are fitted to a 4th degree polynomial function. The function is integrated to find a medium velocity between each product box. a) Velocity profile for the space between top product and false ceiling. b) Velocity profile for space between products in the upper carrier.

Table 1. *Heat transfer coefficient for different packaging.*

	without radiation	with radiation
Convective heat transfer coefficient (h_c)	20.0	20.0
Overall heat transfer coefficient (U) without packaging	20.0	23.5
U with 1 mm cardboard packaging	18.0	20.8
U with 1 mm air layer	10.5	11.4
U with 1 mm cardboard packaging and 1 mm air layer	10.0	10.8

A radiation heat transfer coefficient was also calculated for different temperatures. From Table 1 it can be seen that radiation increases the overall heat transfer coefficient by 8% for a product with packaging.

3.3. Freezing time

The main objective of this paper was to simulate air velocities inside an existing freezing tunnel. However, the differences between air velocities in different locations are more visible if different freezing times are calculated. Figure 5 shows that air velocities have a larger range for the simulated tunnel with no guide blade. Freezing times calculated with modified Plank's equation (no radiation) would be between 16 h and 32 h (where the second worst case is 28 h), a span of 16 h. With guide blade, the freezing times are between 17 h and 21 h, with only 4 h difference between the most rapid and the slowest frozen products. The addition of radiation effects mostly affected the longest freezing times. With radiation and without a guide blade, the span between worst and best cases was 12 h, while with a guide blade, the span was 4 h.

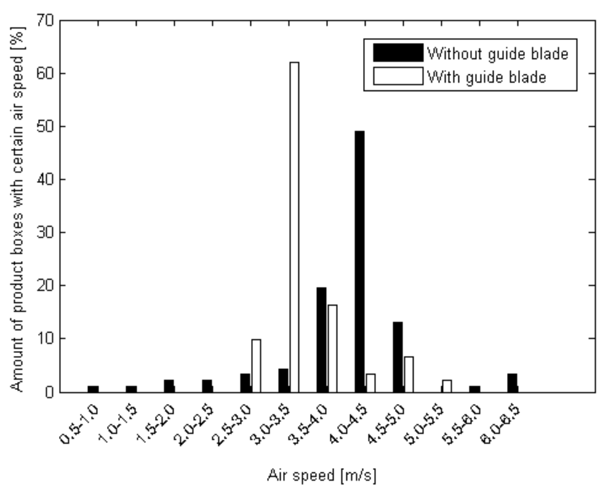


Figure 5. Air velocity distribution for a tunnel with and without a guide blade. Only velocities above and between the products are included.

It was not possible to measure the air velocities inside the freezing tunnel. Instead we have measured the temperatures inside the product boxes. From this we see that the actual freezing time is between 13 h and 20 h, which matches the calculated freezing time quite well. The measured freezing times indicate that the air velocity distribution is more uneven in reality than in the simulated tunnel. A reason for this is that the carriers are put into the tunnel by truck (resulting in random gaps between carriers) and not automatically.

3.4. Uncertainties

The Airpak model is subject to some uncertainties, most of which are caused by the simplification of geometries used in the program, which are not as complex as reality. The carriages have not been included, only the product boxes. This speeds the generation of the solution. The evaporator is also simplified to a resistance object, which has a certain air pressure drop. We were not able to make air velocity measurements inside the tunnel, so we have no data to confirm these results. The calculated average velocity for each product is based on a limited number of node velocities.

The heat transfer coefficient was calculated with a simple equation. More complicated equations could have been used, but this would not necessarily result in greater accuracy. The packaging is also simplified and assumed to be equal around the whole product. In reality there would be more air gaps on the top of the products than on the bottom. The freezing time prediction was also calculated using a simple method, but this was good enough for this study since our main focus was not freezing times.

Despite these limitations and uncertainties, the velocity fields that were generated seem valid and can be used in further analyses of the freezing process.

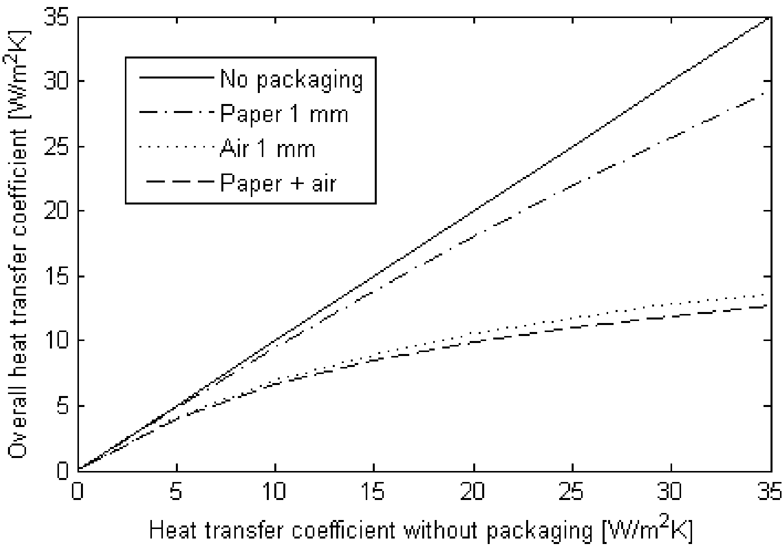


Figure 6. Heat transfer coefficients for different packaging thicknesses (no radiation included).

4. CONCLUSIONS AND FURTHER WORK

The main objective of this study was to simulate air velocities inside an air blast freezing tunnel. It was also important to analyse the simulated data and find valid average velocities that can be used later in numerical models for a freezing process. A discussion of heat transfer coefficients with and without packaging was also included.

Several earlier papers have shown that air velocity is important in the freezing of a product. The air velocity field generated by the fans is directly connected with electricity usage. The fans not only use electricity for operation, they also add heat to the freezing tunnel which then has to be removed by the refrigeration system. An optimal air velocity field will result in even freezing times without excess use of electricity.

We used Airpak to simulate the air velocity field inside an existing air blast freezing tunnel. It was simple to use and gave acceptable results. Average velocities for each product air space were found using polynomial fitting and integration. A tunnel both with and without guide blade was simulated and the results were considerably different. The air velocity field was much more uneven without the guide blade and the average velocities were between 0.97 m/s and 6.29 m/s. With a guide blade the velocities were 2.57-5.36 m/s.

Overall heat transfer coefficients with packaging were significantly lower than the original heat transfer coefficient. It seems that the importance of the air velocity decreases as the amount of packaging increases. The convective heat transfer coefficient was used when calculating the freezing time. Radiation heat transfer coefficient was also calculated and included. A modified Plank's equation was used. These results show the consequences of an uneven velocity field. Without a guide blade (and no radiation effects) the freezing times varied between 16 h and 32 h (where the second worst case is 29 h), a span of 16 h. With guide blade the freezing times were between 17 h and 21 h, only 4 h different between the quickest and the slowest frozen product times. A tunnel where the quickest products are frozen at 17 h and the slowest at 33 h should be reconstructed. Such a tunnel either uses too much electricity or several of the products are not completely frozen, or both. With radiation effects and without a guide blade, the span between the worst and best case was 12 h, while with guide blade the span was 4 h.

This paper has shown that air velocity fields can be easily simulated and analysed to obtain valid results. Suggestions for further work include inserting these velocities into numerical models and conducting transient analyses of the freezing process. These models should include the fan electricity usage and the refrigeration load. With this it should be possible to make suggestions about how to regulate the fan speed during the freezing process in order to decrease electricity usage and hopefully to also increase the product quality.

NOMENCLATURE

CFD	Computational Fluid Dynamics	T_{fish}	product temperature
d_a	thickness air gap	T_{room}	tunnel air temperature
D_h	hydraulic diameter	T_s	surface temperature
d_p	thickness packaging	U	overall heat transfer coefficient
h_c	convective heat transfer coefficient	u_a	air velocity
h_r	radiation heat transfer coefficient	x	horizontal coordinate parallel with air flow
k_a	thermal conductivity air	y	vertical coordinate in tunnel
k_p	thermal conductivity packaging	z	horizontal coordinate normal to air flow
L	length of product box in x-direction	ε	emissivity (0.95)
q	heat	σ	Stefan-Boltzmann constant

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Appendix IV

Paper IV

VERIFICATION OF A MODELICA-BASED DYNAMIC SIMULATION MODEL FOR BATCH FREEZING TUNNELS

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ABSTRACT

There is a focus on increasing energy efficiency in the fish processing industry in Norway. Refrigeration, and particularly freezing, is energy demanding and often not optimized with regard to energy usage. To evaluate different energy saving measures, a dynamic simulation model of a batch freezing tunnel for pelagic fish has been developed. To validate the simulation model, temperatures of air and product during a freezing process at an industrial plant have been measured. The results are compared and the model seams able to predict the temperature development with reasonable accuracy. The various uncontrollable variables of the product packages (e.g. air pockets) have significant effect on the results and are difficult to account for in a simulation model. However, the calibrated model is well suited for investigation of several energy saving measures.

1. INTRODUCTION

There is a focus on increasing energy efficiency in the fish processing industry in Norway. Refrigeration, and particularly freezing, is energy demanding and often not optimized with regard to energy usage. An estimated 15% of the world's electricity is used for refrigeration (Coulomb, 2008) and we have a great potential for development in the area of energy efficiency in food processing. Freezing processes need to be improved without deteriorating the food quality.

The freezing process for packed pelagic fish is usually 12-20 hours. It is difficult to make changes to the process without affecting the quality or the work schedule on a freezing plant. Performance improvements (e.g. reduced energy usage) of an existing plant can be predicted by simulating a refrigeration system in a computer model. There are many ways to model these systems and the selected method should be stable, rapid and accurate(Ding, 2007).

During freezing, the operating conditions for the refrigeration cycle, as well as the driving temperature difference over the product changes significantly from start to finish. Due to product stacking and placement inside the boxes, the temperature and freezing rate in the fish will progress differently. Transient modelling can take these effects into account.

In most industrial tunnel freezer plants, the air fans run at full power during the whole freezing process. It is well known that this is not energy effective (Edwards and Fleming, 1979), as the refrigeration system needs to remove the heat added by the fan, in addition to remove heat form the product. Normally, fan power represents 25 – 30 % of the total average refrigeration requirement, but at the end of the freezing process, heat from the fans can represent up to 95-99 % of the refrigeration load (Magnussen and Nordtvedt, 2006). Therefore, improving fan control can have large impact on increased energy efficiency.

A transient model including a refrigeration plant, an air blast freezing tunnel and food products has been built. The focus of this work is validating this model against measurements from an industrial freezing plant.

2. MATERIALS AND METHODS

2.1. Measurements

A pelagic fish processing plant was chosen for the analysis of the air blast freezing tunnels. The plant is situated in the south-western part of Norway, and has an annual production volume of 50 000 tonnes of frozen fish and fish products. The plant was built in 1993. The refrigeration systems at these plants are often complex and designed to supply refrigeration to both several freezing tunnels and storage rooms at different temperature levels. The focus point for the measurements has therefore been the temperature changes of the air and product in one single tunnel.

2.1.1. Processing plant description

The refrigerant system has 3 reciprocating and 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW. The reciprocating compressors work in two stages and the screw compressors operate in one stage with an economizer solution. The refrigeration system supplies cooling to 5 air blast freezing tunnels, 10 plate freezers, 3 spiral freezers and freezing store. The system contains 30 tonnes of ammonia as the refrigerant. The freezing capacity of the tunnels is 625 tonnes of fish per day with a storage capacity of 10 000 tonnes (Solheim, 2006).

Each tunnel is divided into three sections. Each section has three fans, dimensioned to blow 22.23 m^3/s of air through the tunnel. The side and top view of a tunnel section are illustrated in Figure 1. The letters shown in the top view indicates the position of the temperature logger series.

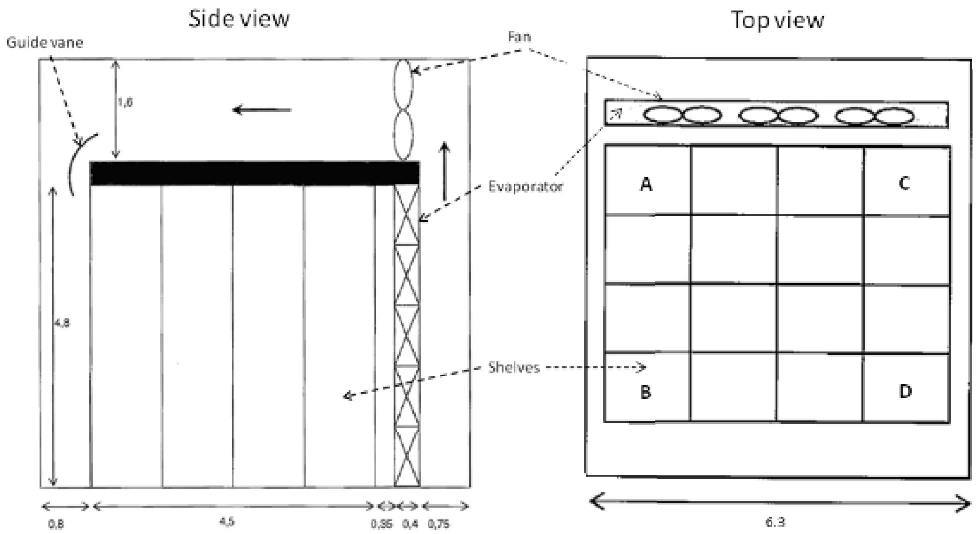

Figure 1: Tunnel section layout

Each square on the top view in Figure 1 illustrates a shelf rack, and on each shelf rack there are 22 shelves. On each shelf, 5 packages of 20 kg each are stacked together. Figure 2 shows the measurements of the stacked packages, and also illustrates the simulation model discretization and location of the temperature elements. On each shelf with temperature elements, only one of the five boxes had loggers, but which of them was not registered. This may contribute to uncertainties in the comparison with the simulation results.

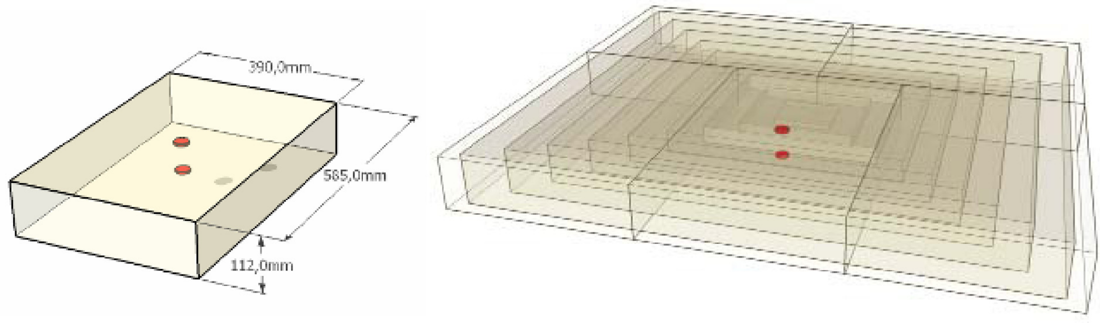

Figure 2: Illustration of packages, discretization and location of temperature elements.

2.1.2. Measurement setup

Thermocouples were used to measure absolute temperatures in the freezing tunnel. These were positioned before and after the evaporators, and in the channel on the pressure side of the fans, above the products. Temperature loggers were also placed inside the product boxes. These were located in different heights and rows.

Four series (A, B, C and D) of temperature elements where placed according to the letters in Figure 1, and for each series, one box on shelf 11 and one box on shelf 22 had loggers placed in the middle and just below the lid.

2.2. The simulation model

Modelica has been used together with the simulation environment Dymola(Dassault Systems) as the platform for the model. Modelica is an open object oriented equation based programming language for modelling of complex physical systems including mechanical, electrical, thermo physical subcomponents etc. (Modelica Association). The object oriented structure allows reuse of developed components, such as compressors, heat exchangers, valves, etc. The objects (or classes) are translated and exercised with a simulation engine (compiler). Modelica allows easy coupling of complex systems and cycles. All components in the system, except the product model, has been modelled with components from the TIL library(Richter, 2008).

2.2.1. The product model

A finite volume model has been built for the product. It is assumed that the fish is vacuum packed in boxes. The boxes are discretized with equal thickness of each layer (t_{hight}, t_{length} and t_{with}). This means that each layer is represented as a box with equal shape, but different size (see Figure 2). However, the outermost layer has half the thickness of the other layers, to better represent effects near the surface.

Each layer is modelled with a temperature dependent thermal heat capacity and thermal conductivity found in ASHRAE (2006). In all results presented in this paper, composition data for mackerel are used. The specific heat capacity and thermal conductivity is shown in Figure 3.

When the product is packed in boxes, the real weight of the box is smaller than the calculated weight (volume multiplied with density). The density is found in the same way as the heat capacity and thermal conductivity. This means that there are air gaps between the fish inside the box. Due to this, the thermal conductivity is corrected by a porosity factor found in Valentas et al. (1997).

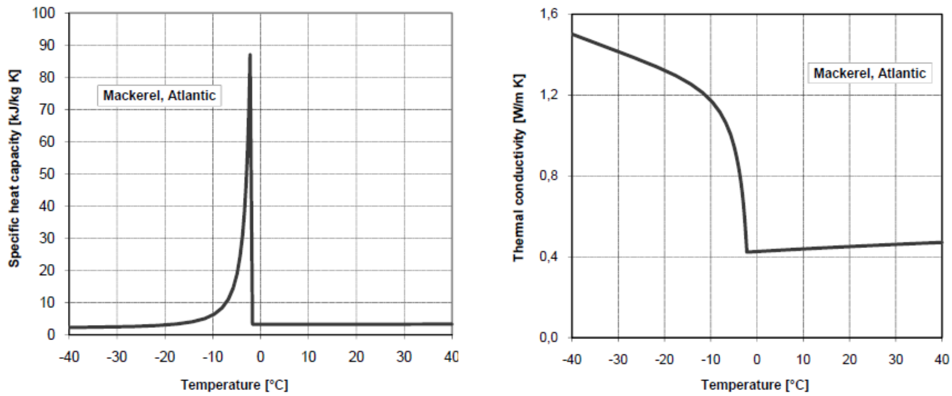

Figure 3: Thermal properties of Mackerel.

In addition to the thermal resistance in the product, a resistance element representing an air layer between the box and the product is modelled. The thickness of this layer is used as a tuning parameter for the model and for the simulations. A 1,6 mm layer was found to yield the most accurate results for this case. However this could be different for other cases, and dependent on fish sizes and package configurations. Widell and Frydenlund (2009) showed that the thickness of the air layer has a great influence on the total freezing time.

2.2.2. The tunnel model

The tunnel model is separated into three parts: The main tunnel (where the fish is frozen), the fan and the evaporator. The tunnel is modelled as a channel with the face area equal to the total face area of the tunnel subtracting the total face area of the boxes. The air distribution in the channel is assumed to be ideal (equal velocity over the whole cross-section). Lengthwise, the tunnel is divided into the same number of segments as the defined number rows, to get the effect of air temperature rise through the tunnel.

The convective heat transfer coefficient (h) between the air and the product is given by the following formula (Valentas et al., 1997):

$$h = 7{,}3 \times u_a^{0{,}8} \tag{1}$$

Where u_a=air velocity [m/s]. Typical velocities inside the tunnel range from 1–6 m/s. The pressure drop is calculated with the following formula:

$$\Delta p = k \times \frac{1}{2\rho} u_a^{2} \tag{2}$$

where k=calibration factor [-]. ρ=density of air [kg/m^3].

The evaporator is modelled as a tube and fin heat exchanger. The evaporator has to be dimensioned according to size of the tunnel and together with the rest of the refrigeration plant. The Haaf (Haaf, 1988) correlations are used for air side heat transfer and pressure drop.

The Fans are modelled as a single fan, with the efficiency linearly dependent on the rotational speed. Since the fan motor is placed inside the tunnel all the power supplied to the fan is transferred to the air, either as hydraulic energy or as heat.

2.2.3. The refrigeration plant

The refrigeration plant is connected with the freezing tunnel through the evaporator. The compressors are modelled based on pressure ratio dependent efficiencies. The basic refrigeration system is designed as a classical two-stage system with an open intercooler and ammonia as working fluid. Figure 4 shows the flow sheet of the assembled model.

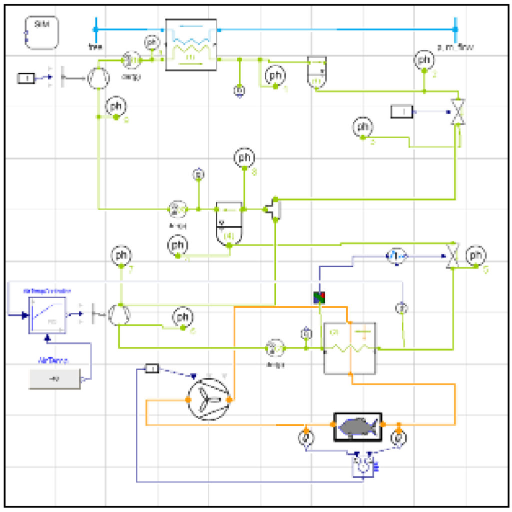

Figure 4: Refrigeration system layout

The two-stage systems require several control strategies. The high pressure receiver will make sure that the condenser outlet is saturated, and will therefore automatically control the high pressure together with the high pressure compressor. The high pressure valve is used as a floating valve for the intercooler. The low pressure valve is designed as a thermostatic valve. This is not normal in ammonia plants (due to the high heat of evaporation), but will work fine in for the simulations, and the results should be similar to a system with a receiver and a self circulating system.

The low pressure compressor is controlled to reach a defined minimum evaporation temperature. This means that it will run at maximum speed until the temperature is reached, and then reduce the speed to keep the evaporation temperature constant. The high pressure compressor is used to set the middle pressure to the geometric mean of the high and low pressure.

As discussed in 2.1.1, the refrigeration plant is designed for several freezing and refrigeration purposes, and it is not adequate to simulate the whole process. However, this means that the model is not suited for evaluation of compressor control strategies, and they are therefore simulated with ideal control of the rotational speed.

The evaporator is designed for a cooling duty of 205 kW at a temperature difference of 8 K, which is one third of the dimensioning duty at the plant, i.e. equal to one section. The rest of the refrigeration plant is dimensioned to match this duty, with a condensing temperature of 20 °C.

3. RESULTS AND DISCUSSION

3.1. Comparison of measurements and simulation

In the following, the results from the temperature measurements will be compared with the simulation results.

3.1.1. Air temperatures

Figure 5 compares the simulated and measured air temperatures at three different places. In general, the simulation seems to yield a slightly higher temperature than the measurements. The air temperature is dependent on several parameters, such as air velocity, design of the evaporator, refrigerant evaporation temperature etc. Using a relatively simple model one must therefore expect some deviation.

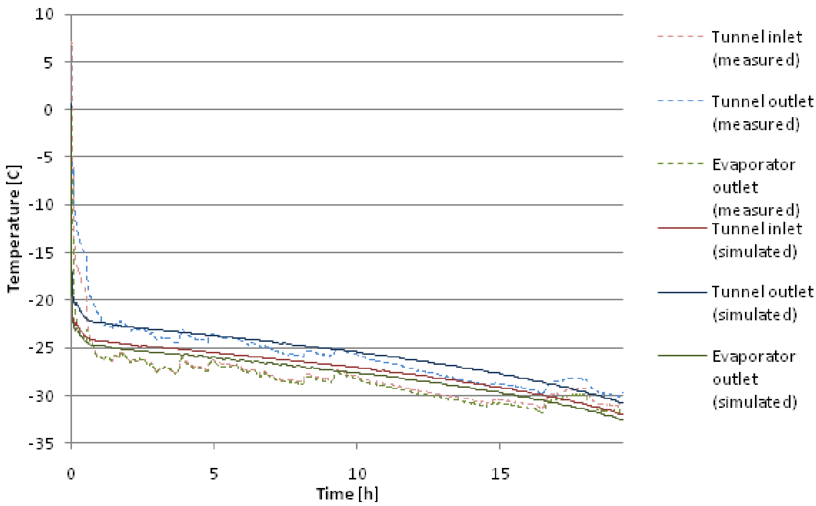

Figure 5: Air temperatures

3.1.2. Box surface temperatures

Figure 6 shows the temperatures just under the box lid. The dotted lines show the measurements (the legend letter indicate the position according to Figure 1 and the number indicates the shelf number, from the floor), while the solid lines show the simulation results. The measured temperatures show relatively large variance. Many factors, such as the temperature logger positions and thickness of the air layer, contribute to these deviations. From the simulation results, the box surface temperature and the product surface temperature are included. One can see that these two simulated temperatures represents the upper and lower boundaries for the measurements. This is natural, since the temperature loggers are placed between these two simulation nodes.

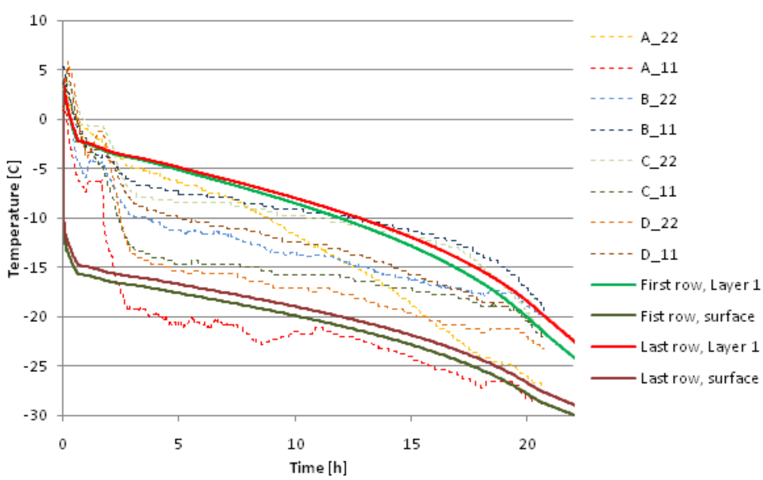

Figure 6: Product box outer temperatures

3.1.3. Box centre temperatures

Figure 7 shows the temperatures in the middle of the boxes during freezing. Also for these measurements, there are large differences. From the simulations, the innermost layer (layer 10) and layer 7 is included.

Since the elements are positioned in the centre of a single box (as shown in Figure 2, and not in the centre of the 5 boxes it might be reasonable to compare these measurements with a simulation result of layers closer to the box surface.

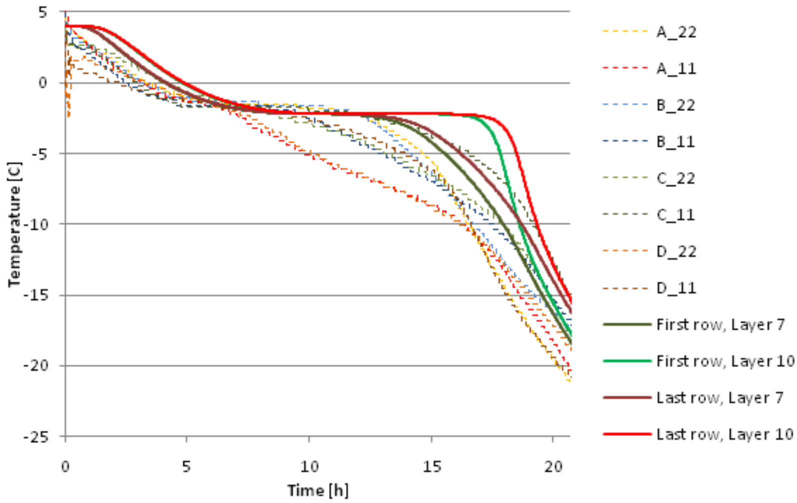

Figure 7: Product box inner temperature

3.2. Energy consumption

As mentioned earlier, an important measure for reducing the energy consumption in a batch freezing process is to improve the fan operation. Normally, fans run at full speed throughout the process. This is highly inefficient, since heat must be removed both from the products and the fans.

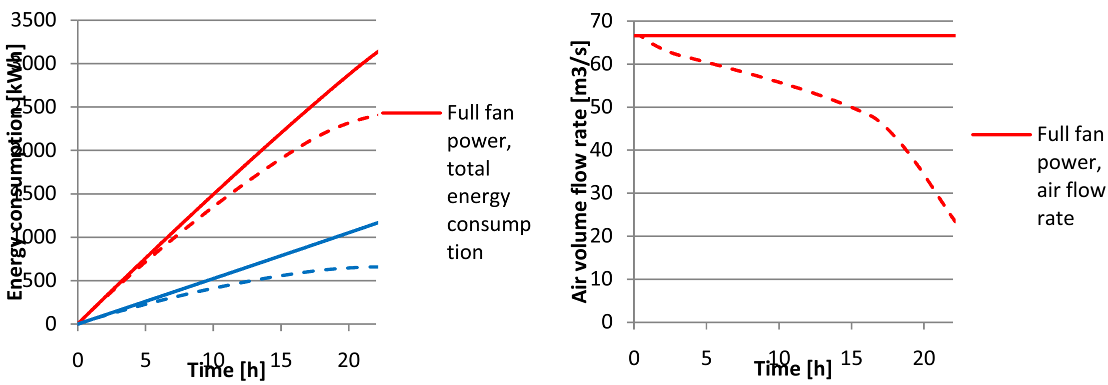

Figure 8: Energy consumption and air volume flow rate with and without fan speed control

Figure 8 compares the energy consumption and volume flow rate of air, with and without fan speed control. The fan controller is set to keep the air temperature difference through the tunnel at 2 K. The total efficiency of the fan and motor is reduced from 0.45 at full speed to 0.35 at 25 % speed. The pressure drop through the tunnel is set to yield a fan power of approximately 53 kW at full speed, according to the tunnel specification.

The results show that the total energy consumption can be reduced by about 25 %, while the freezing time is hardly affected. The main reason for this is that the thermal resistance inside the box (especially the thin air layer) is the governing factor in the total resistance, and not the heat transfer between the tunnel air and the box. This effect is also shown by Widell and Frydenlund (2009).

4. CONCLUSION

In general, the simulation results fit the measurement data with reasonable accuracy. The difference between the simulations and measurements are generally not larger than the variance of the measurements themselves. The measurements show how difficult it is to make accurate measurements, but also show the uncertainties when working with inhomogeneous material. Effects, such as package density and insulating air layers inside the box, have great influence on the results, but are very difficult to represent in simulations.

However, this simulation model is a great tool for illustrating processes at a fish freezing plant, and can be used for investigation measures to increase energy efficiency and also for predicting freezing time (when to transfer the product to the storage facility).

Further work would be to develop optimized fan operation strategies for reducing energy efficiency without affecting the freezing time and product quality too much. A second task will be to do a sensitivity analysis of other factors, such as package density, insulating air layers and tunnel filling ratio, on the energy consumption.

5. ACKNOWLEDGEMENT

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Appendix V

Paper V

NUMERICAL AND EXPERIMENTAL ANALYSIS OF FOOD PRODUCTS IN A BATCH FREEZING TUNNEL

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ABSTRACT

This paper presents models and assumptions for simulating freezing of fish in a freezing tunnel. The results are compared with measured data from a processing plant. A finite difference method has been used to calculate the transient temperature development inside the product during freezing. The product model is two-dimensional and includes latent heat release around the freezing point. The thermophysical properties for both the air and the product are dependent on temperature. The studied refrigeration system consists of 5 air blast freezing tunnels with total capacity of 625 tonnes of fish per day. The system contains 30 tonnes of ammonia as the working fluid. Modelling is an important part of developing the freezing processes and a tool to improve the energy efficiency of the refrigeration system. Reliable models and simulation programs can be further extended by adding models for improving freezing systems, thus decreasing energy consumption.

1. INTRODUCTION

Temperature is one of the most important factors in the food handling process. Lower temperature means lower activity of enzymes, bacteria and chemical processes and therefore longer storage life. The temperature also affects the freezing time, which is the time it takes to freeze the product from beginning of chilling to the desired final temperature in the centre of the product (for example -18 °C). Factors that influence the temperature and the freezing time are temperature differences between the environment and the product, size and shape of the product and the manner in which the heat is transferred. Properties of the product and the surrounding media are also important. (Müftügil 1986) It is possible to measure the freezing time experimentally, but since the conditions in a large freezing tunnel change with time and location inside the tunnel, such measurements are not always reliable. Calculation of the freezing time is also possible, but difficult due to the complexity of the system. Thermophysical properties change with temperature and all food products have changes in their properties due to the phase change of water starting at the initial freezing point. The Plank formula has been used frequently for calculating the freezing times. However, it assumes that freezing occurs at a constant temperature and that thermophysical properties do not vary with temperature. It also excludes the sensible heat removal, which means the time for precooling and subcooling. López-Leiva and Hallström (2003) have reviewed how Plank's equation has been modified and compared calculated freezing times with experimental; there is no apparent conclusion regarding which modification is preferred. Complicated analytical models do not necessarily give a more reliable result than simple ones.

Instead of using analytical models, such as Plank's equation, numerical methods can be used. These are more accurate and more flexible to use. Today's computer capacity gives solutions with shorter CPU time. A two-dimensional finite difference model for regular shapes has been described by Cleland (1990) and is used in this paper.

Normal production periods at a processing plant are very hectic and there is no room for experimenting on the operation of the refrigeration system. Modelling is therefore an important part of developing freezing processes and improving energy efficiency. The objective of this study was to find mathematical models for products in a freezing tunnel and to verify these models by comparing simulated temperature profiles and freezing times with experimental data. When reliable models and simulation programs exist, it is possible to further extend these by adding models for a refrigeration system, including the air fans, for optimization of the capacity and energy use of the freezing tunnel.

2. MEASUREMENTS

A pelagic fish processing plant was selected for the analysis of the air blast freezing tunnels. The plant is situated in the south-western part of Norway, and has an annual production volume of 50 000 tonnes of frozen fish and fish products. The plant was built in 1993.

2.1. Processing plant description

The refrigerant system has 3 reciprocating and 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW. The reciprocating compressors work in two stages and the screw compressors operate in one stage with an economizer solution. The refrigeration system supplies cooling to 5 air blast freezing tunnels, 10 plate freezers, 3 spiral freezers and freezing storage. The system contains 30 tonnes of ammonia as the refrigerant and the evaporation temperature is -39°C. The freezing capacity of the tunnels is 625 tonnes of fish per day with a storage capacity of 10 000 tonnes. (Solheim, 2006)

2.2. Measurement setup

Thermocouples were used to measure absolute temperatures in the freezing tunnel. They were positioned before and after the evaporators at different heights in the freezing tunnel. Thermocouples were also placed in the channel on the pressure side of the fans, above and before the products. Figure 1 illustrates the freezing tunnel and the measurement setup.

Temperature loggers were also placed inside the product boxes. These were located at different heights and rows. Temperatures from two of these boxes were analyzed in section 4.1. The boxes were both placed on shelf 11 (of 22) and in the first row, where air from the fans meets the products. Loggers were placed inside the packing on top of the fish and in the geometric centre of the box, between the fish. Temperatures from a box in the first row and close to the floor were compared with simulation data and are described in section 4.3.

It was not possible to measure the air velocity around the products because there was not enough space between them.

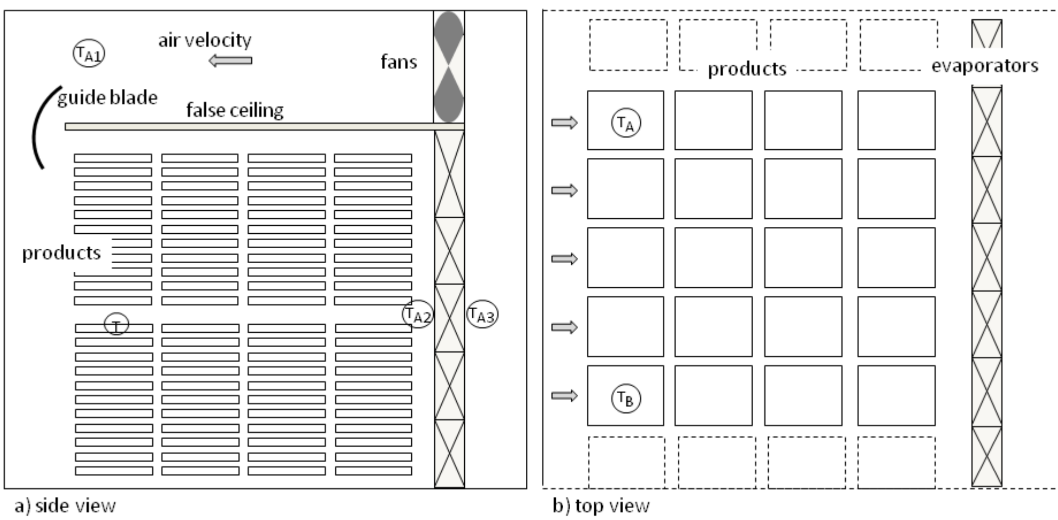


Figure 1. Freezing tunnel layout. Termocouples and temperature loggers are illustrated with a T. The top view only shows the middle section (of 3) of the freezing tunnel.

3. MODELS USED IN THIS STUDY

To calculate the temperature development inside products during freezing, a finite difference method was used. This approach was also described by Cleland (1990). The product model is two-dimensional and it includes latent heat release around the freezing point. The thermophysical properties for both the air and the product are dependent on temperature.

3.1. Product and packing configuration

The products were whole mackerel packed inside cardboard boxes, each containing 20 kg of fish. Each shelf in the freezing tunnel had 5 boxes (in full contact with each other) and for simplification, these were simulated as one box with the dimensions 1.09x0.89x0.1 m. Thermophysical properties for the product were calculated with functions and data from ASHRAE Handbook - Refrigeration (2006). These were specific for Atlantic mackerel and dependent on product temperature. Functions for specific heat, thermal conductivity, and enthalpy were used. Density was also used, but set to a constant value of 1030 kgm^{-3}. The cardboard boxes consisted of two equal parts, one base part and one lid part. The carton thickness was set to 10^{-3} m and the thermal conductivity to 0.180 $Wm^{-1}K^{-1}$ (Incropera and DeWitt, 2002).

There will always be small air spaces between the fish inside the product boxes. We assumed that this did not affect the conductivity because the fish will still be mostly in contact with each other. It was not necessary to adjust enthalpy and specific heat, since these were on mass basis (the air has practically no mass compared to the fish).

The air gaps between the fish and the packing in the real product boxes varied, but the variation was not possible to include in the simulation program. Instead, a constant air gap was used. Several simulations with different air gaps were made and the effects on the freezing time were found. Radiation inside the air gap was included and air gap size was varied between 0.5 and 5 cm.

It takes about 3 hours to load the tunnel and the section freezing system starts when each of the three sections of the tunnel is filled. This has been simplified in the simulations by setting an equal start temperature field in the product. This temperature was set to 0°C when comparing different simulation alternatives and to the measured initial temperatures when comparing simulation and measurement results. The simulation stopped when all of the nodes had reached -18°C. This number was also found from the measured data.

3.2. Nodes and time steps

The product was divided into control volumes in two dimensions, where each control volume had one centre node. In addition, each control volume at the surface of the product had an extra surface node (three extra for corner volumes). The network was uniformly shaped. The temperatures of the surface nodes were equal to the air temperature.

Simulations with different numbers of nodes and time step lengths were made. The product height was 0.1 m, and 5, 10, 15 and 20 nodes were tried. The suitable number of nodes was found by comparing integrated heat from the products calculated by the simulation program with a calculation of the total heat from the product based on the start temperature and a mean end temperature of the product. CPU time was also considered when deciding the number of nodes and time steps.

3.3. Air properties

It was not possible to measure the air velocity between the products. Air velocities were simulated with the Computational Fluid Dynamic (CFD) software Airpak (Widell and Frydenlund, 2009) to find the velocity range. Several simulations of one product with constant air velocities were made and the effect on the freezing time was found. Air velocities were set between 2 m/s and 6 m/s.

The simulated tunnel air temperature was found by equation regression of the measured temperatures at the inlet of the products; it had a falling rate during the freezing period. A graph with measured air temperatures and the temperature function can be seen in Figure 2. The air temperature in the simulation model was constant after it had reached -31°C.

3.4. Conduction and convection coefficients

Thermodynamic properties for Atlantic mackerel were based on tables and equations from ASHRAE (2006). The arithmetic mean of the conductivity was calculated on the boundary between two control volumes (Bonacina and Comini, 1973). The convective heat transfer coefficient (h_c) was calculated with this equation (Valentas et al., 1997):

$$h_c = 7.3 \cdot u_a^{0.8} \tag{1}$$

u_a is the average air velocity. The equation is valid for objects with planar surfaces.

Overall heat transfer coefficients will be significantly lower when the products are packed inside a cardboard box. The packaging and air gaps between the product and the cardboard contribute to this (Brown and James, 2006). The overall heat transfer coefficient was calculated with equation 2 (Incropera and DeWitt, 2002), where R_{side} is different for the top, sides and bottom of the product. Air gaps were included between the fish and the packaging on the sides and the top of the product, but not in the bottom. Paper carton was included everywhere, but with double thickness on the sides because of the lid. Radiation inside the air gap was included and air gap size was varied between 0.5 and 5 cm.

$$U_{side} = \left((h_c)^{-1} + R''_{side} \right)^{-1} \tag{2}$$

3.5. Product modelling

The two-dimensional finite difference approximations to heat conduction equations given by Bonacina and Comini (1973) were selected for calculating the temperature profiles, see equations 2 and 3. This strategy was also suggested by Cleland (1990).

$$\begin{aligned}
-\frac{2}{3}\frac{\Delta\tau}{C\Delta x^2}\cdot k_x^- T_{i,j-1}^{(m+1)^*} + \left[1+\frac{2}{3}\frac{\Delta\tau}{C\Delta x^2}\cdot\left(k_x^+ + k_x^-\right)\right]T_{i,j}^{(m+1)^*} - \frac{2}{3}\frac{\Delta\tau}{C\Delta x^2}\cdot k_x^+ T_{i,j+1}^{(m+1)^*} = \Bigg\{\frac{2}{3}\frac{\Delta\tau}{C\Delta x^2}\left[k_x^+\left(T_{i,j+1}^m - T_{i,j}^m\right) - k_x^-\left(T_{i,j}^m - T_{i,j-1}^m\right)\right] \\
+\frac{2}{3}\frac{\Delta\tau}{C\Delta y^2}\left[k_y^+\left(T_{i+1,j}^m - T_{i,j}^m\right) - k_y^-\left(T_{i,j}^m - T_{i-1,j}^m\right)\right]\Bigg\} + \Bigg\{T_{i,j}^{m-1} + \frac{2}{3}\frac{\Delta\tau}{C\Delta x^2}\left[k_x^+\left(T_{i,j+1}^{m-1} - T_{i,j}^{m-1}\right) - k_x^-\left(T_{i,j}^{m-1} - T_{i,j-1}^{m-1}\right)\right] \\
+\frac{4}{3}\frac{\Delta\tau}{C\Delta y^2}\left[k_y^+\left(T_{i+1,j}^{m-1} - T_{i,j}^{m-1}\right) - k_y^-\left(T_{i,j}^{m-1} - T_{i-1,j}^{m-1}\right)\right]\Bigg\}
\end{aligned} \tag{2}$$

$$-\frac{2}{3}\frac{\Delta\tau}{C\Delta y^2}\cdot k_y^- T_{i-1,j}^{m+1} + \left[1+\frac{2}{3}\frac{\Delta\tau}{C\Delta y^2}\left(k_y^+ + k_y^-\right)\right]T_{i,j}^{m+1} - \frac{2}{3}\frac{\Delta\tau}{C\Delta y^2}\cdot k_y^+ T_{i+1,j}^{m+1} = T_{i,j}^{(m+1)^*} - \frac{2}{3}\frac{\Delta\tau}{C\Delta y^2}\left[k_y^+\left(T_{i+1,j}^{m-1} - T_{i,j}^{m-1}\right) - k_y^-\left(T_{i,j}^{m-1} - T_{i-1,j}^{m-1}\right)\right] \tag{3}$$

Firstly, Equation 2 was used to find temperatures (*T*) in the nodes row-wise by using tridiagonal matrix algorithm (TDMA). Then the temperatures in each column of nodes were found by Equation 3, also solved with TDMA. The temperatures (marked with asterix) from Equation 2 were used in Equation 3. *C* was volumetric heat capacity [$Jm^{-3}K^{-1}$], and it was always based on temperatures in the previous time step. When the external nodes of the product were calculated, the boundary conditions describing the heat transfer on the surfaces were included in the thermal conductivity, *k* [$Wm^{-1}K^{-1}$]. Pham (1985) describes a temperature correction equation to ensure that the latent heat from the phase change is included (avoiding "jumping off the latent heat peak"). The following equation was used on all of the temperatures found by equations 2 and 3:

$$T_{i,j}^{m+1}(corrected) = f_T\left[f_H\left(T_{i,j}^{m-1}\right) + C_{i,j}^m\left(T_{i,j}^{m+1} - T_{i,j}^{m-1}\right)\right] \tag{4}$$

The functions f_H and f_T calculated, respectively, the enthalpy and the temperature from property tables created from equations and data for Atlantic mackerel (ASHRAE, 2006).

The heat flow out of the product was calculated from the sum of the heat flows from every control volume along the surface of the product. The total heat flow from the product during the freezing period was calculated by integrating the heat flows from the product for all of the time steps.

A two-dimensional instead of a one-dimensional model was selected because it was necessary to have different properties on the sides, top and bottom of the box. A three-dimensional model would be an alternative, but was not necessary since the variables (for example air velocity) were not changed in the third dimension along the product. Packing air gap will in reality be different in all directions, but a constant air gap was set for each side of the product.

4. RESULTS AND DISCUSSION

4.1. Measurement results

Air temperatures at different positions in the freezing tunnel can be seen in Figure 2. The air temperature marked "Input model" was the temperature function used in the simulation program and it is an approach to the measured temperatures at the inlet of the freezing tunnel. This function gives a slightly lower temperature than the measurements. Large amounts of heat rejected from the products give larger differences between temperatures at A2 and A1 in the first part of the freezing than towards the end.

Temperatures logged in the product boxes can be seen in Figure 3, where inner and outer temperatures from two boxes are displayed. These boxes were both placed in the first row, where the air from the fans meets the products (see also Figure 1). The outer temperature loggers ("outer") were placed inside the packing on top of the fish. The inner temperature loggers ("inner") were placed between the fish as close to the geometric centre of the box as possible.
Some trends can be seen in the graph. Most of the curves follow a typical freezing progress. At the beginning and the end are two steeper parts where mostly sensible heat is released. In between is a more level part where most of the latent heat is released. This part is around 70 % of the total freezing time. As expected, the temperature decrease is slower in the inner nodes than in the outer nodes.

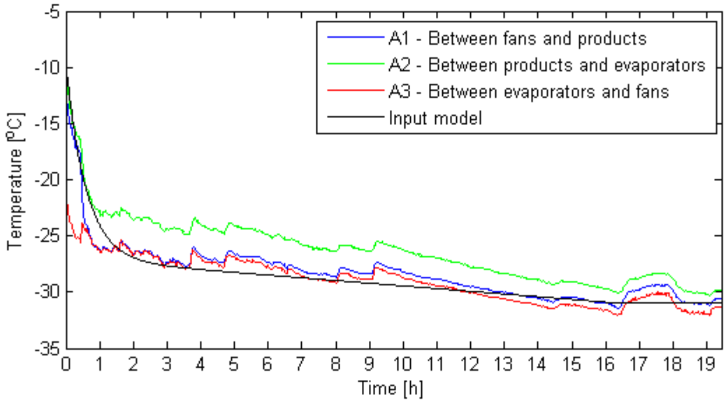


Figure 2. Measured air temperatures in the freezing tunnel and a temperature function used in the computer simulation ("Input model").

The inner nodes have a more equal behaviour than the outer nodes. This is probably because the inner nodes were not as affected as the outer nodes by the velocity field. It would have been natural that outer temperatures of A and B followed the same trend since they were in the same row and on the same height. The reason for the differences is possibly a combination of how the temperature loggers were placed inside the boxes and how the carriages were placed. There will always be a varying velocity field inside a freezing tunnel, in all directions (Widell and Frydenlund, 2009).
The freezing time was defined as the time from initial freezing until the node reached -18 °C. The graph shows that the freezing time is between 17.5 h and 19 h for most of the nodes.

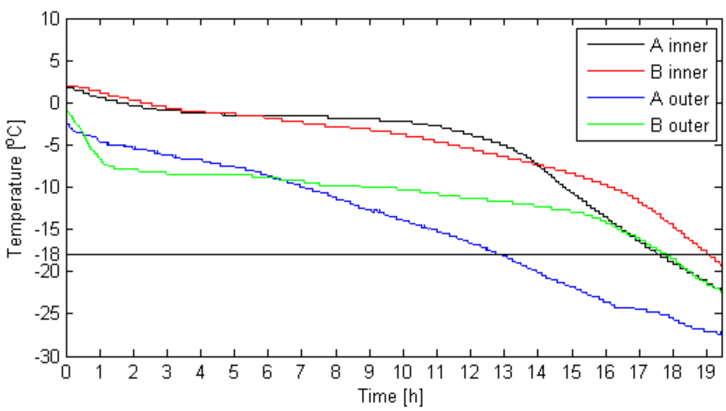


Figure 3. Measured temperatures inside the product boxes at different locations in the tunnel.

4.2. Simulation results for one product

Results from the simulation can be seen in Figures 4 and 5. Figure 4 illustrates the temperatures at different heights inside the product boxes. Figure 4 shows that air gaps between the fish and the packing made the freezing process slower and the bottom node froze much faster. "Warmest" is the slowest cooling node and it is situated 1/3 from the top of the product. This had the highest temperatures, but the curve was very close to the top node curve during the last hours. "Centre" is the node at the geometric centre of the product. Initially, the centre node froze more slowly than the top node, but seemed to be affected by the heat transport through the bottom of the product; after about 10 h, it was colder than the top node.

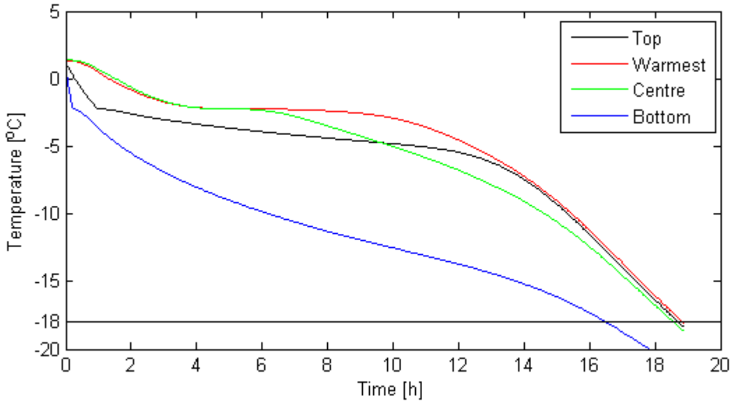
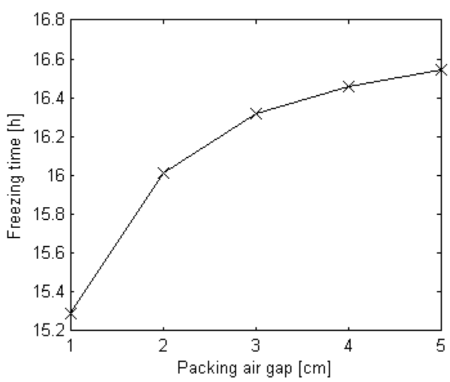


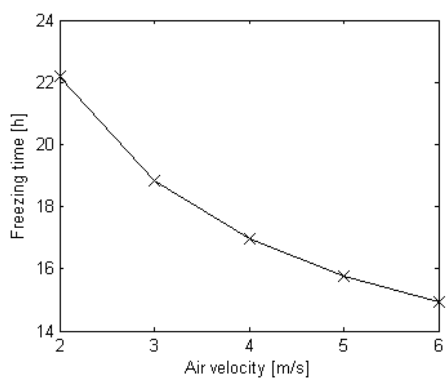
Figure 4. Simulated temperature development in different nodes in one product (at different heights)

Different air gaps caused different total freezing times, as can be seen in Figure 5a. These air gaps will have an insulating effect on the freezing process. The freezing time will increase with an increasing air gap, but only up to a certain gap size; after that, the radiation within the air gap will have a greater affect. The freezing time will increase by approximately 1 hour when the air gap increases from 1 cm to 3 cm, but only with 0.2 h when the air gap increases from 3 cm to 5 cm.

Total freezing times with different air velocities were also calculated. Figure 5b shows that the total freezing time with an air velocity of 2 m/s were about 22 h and about 15 h with 6 m/s.



a)



b)

Figure 5. Simulated total freezing time with a) different packing air gaps b) different air velocities.

Simulations with different numbers of nodes and time step lengths were also made. These showed that the length of the time step had to be decreased when the number of nodes was increased. The product height was 0.1 m, and 5, 10, 15 and 20 nodes were tried. The resulting curves show a rather large divergence for 5 nodes, whereas the differences between 10, 15 and 20 were smaller. The calculations were unstable if the number of nodes were increased above a certain number, without decreasing the time step length.

4.3. Comparison between simulation and experimental temperature profiles

Both measured and simulated temperature profiles are included in Figures 6 and 7. Measured temperatures were compared with simulated temperatures with different air velocities in Figure 6. The air velocity affects the profiles mostly at the second part of the freezing period. The measured freezing time is most equal to the lowest velocity (2.5 m/s) profile. This seems correct, since the product was situated close to the floor, where air velocities were low. Simulated temperatures in the first part of the freezing are lower than the measured temperatures, and are explained by a too low input air temperature function (showed in Figure 2).

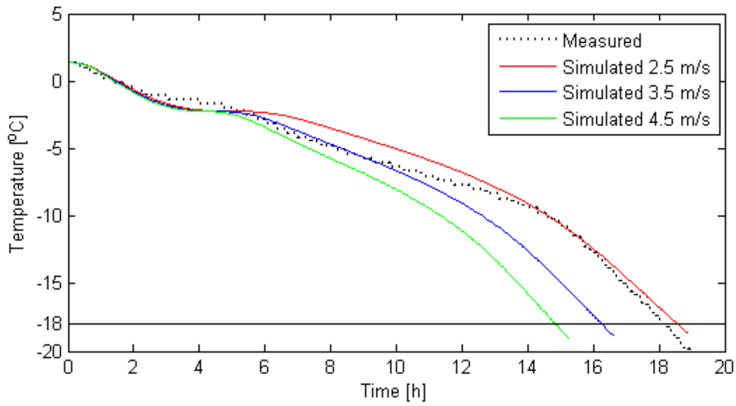


Figure 6. Measured and simulated temperature development in product with different air velocities.

The best temperature graph from Figure 6 was compared with two new graphs shown in Figure 7. Different air gaps between the fish and the product box were simulated. It is during the last hours of the freezing period that differences in temperatures can be seen. An air gap of 10 mm gives simulated temperatures most similar to the measured temperatures.

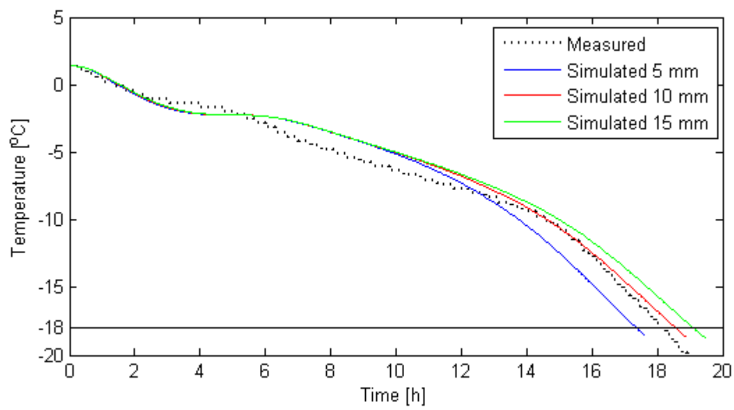


Figure 7. Measured and simulated temperature development in product with different packaging air gaps.

4.4. Uncertainties

There are always uncertainties when calculations related to food are made. Food properties have to be predicted based on experimental data. There is a shift in thermophysical properties around the start of the freezing. When simulating, the results depend on how the model is programmed. Starting values and numbers of nodes are examples of properties that can affect the results. Simplifications of reality must be made. The simulation model in this study was based on findings in the literature and our own measurements. Where there have been different possibilities, several alternatives have been tried out to find the alternative closest to the measurement data. Measured starting temperatures in the products are different at different locations in the product. It was necessary to set one starting temperature in the simulation. The air gaps between the product and the packing varied in the real product boxes, but this was not possible to include in the simulation program. Instead, a constant air gap was used. Placement of the temperature loggers will be arbitrary because of movement of the boxes.

5. CONCLUSIONS

In this paper, the focus was on freezing of food products in batch freezing tunnels. Results both from measurements at a fish processing plant and from simulations with computer models were presented. The numerical models used have been found to be flexible, but it was also challenging, since many input values were either not known or varied in the real situation.

It is shown that the influence of the air velocity field in the freezing tunnel reduces the freezing time from 22 hours to 15 hours by increasing the air velocity from 2 m/s to 6 m/s. It was not possible to fill the boxes with fish without having a small air gap between the fish and the lid. This air gap will have an insulation effect on the freezing process. The freezing time will increase by approximately 1 hour when the air gap increases from 1 cm to 3 cm.

Evaluation of the measured and the simulated temperature profiles show that air velocity and the air gap between the fish and the packing affect the results. Values that make the results comparable were found.

Modelling is an important part of developing freezing processes and improving energy efficiency. When reliable models and simulation programs exist, these can be further extended, for example by adding models for optimized freezing systems with minimized energy consumption of the compressors and the air circulation fans.

NOMENCLATURE

C	volumetric heat capacity [$Jm^{-3}K^{-1}$]	x, y	position coordinates [m]
f_T, f_H	functions to calculate temperature from enthalpy and vice versa	$\Delta x, \Delta y$	grid spacing in the x, y direction [m]
h_c	heat transfer coefficient [$Wm^{-2}K^{-1}$]	*Subscripts and superscripts*	
k	thermal conductivity [$Wm^{-1}K^{-1}$]	i, j	lattice parameters in the two-dim. grid
R	thermal resistance [KW^{-1}]	x, y	in direction of x, y
T	temperature [°C] [K]	+	between nodes i and i+1
$\Delta\tau$	time step [s]	-	between nodes i and i-1
u_a	average air velocity [m/s]	m	time level
U	overall heat transfer coefficient [$Wm^{-2}K^{-1}$]	*side*	product side

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Appendix VI

Paper VI

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