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Abstract

The content of the master thesis work performed by Espen Halvorsen Verpe at NTNU contains the following: (the entire master thesis can be found at: https://daim.idi.ntnu.no/)

The introduction of R744 based freezing systems enables lower evaporating temperatures and faster freezing times, however, also a lower system efficiency (COP). This thesis will investigate how low temperature CO2 compares to traditional refrigerants in plate freezers, regarding energy efficiency and product capacity, where the latter is of special importance for fishing vessels.
A numerical freezing model was made to simulate freezing of fish blocks in a plate freezer, and was validated by freezing of a phase changing test material in an industrial plate freezer. COP was estimated for the freezing system, and selected natural refrigerants was modelled. The freezing model relied on a two-dimensional, heat capacity-temperature, implicit finite difference method with time dependent temperature boundary conditions. The evaporating temperature is not static, as one might expect, but dynamic due to varying heat load from the product side.
The reason being the compressor cannot deliver the required freezing capacity for peak loads.

The elevation of the evaporating temperature results in prolonged freezing times and reduced product capacity, compared to ideal freezing with constant plate temperature. The conditions in the low pressure receiver was modelled to estimate the temperature increase, which was also validated. A thermal storage with CO_2 as phase change material was investigated, with objective to eliminate the elevated temperature in the beginning of the freezing process by storing energy when compressor capacity is larger than the heat load, and release that energy when the compressor struggles to maintain the low evaporating pressure. Ice formation/melting in the storage tank was modelled to determine the dimensioning properties.

The numerical freezing model demonstrated good agreement with experiments, with a deviation in freezing time of only 3 %. Numerical calculations revealed that low temperature freezing, down to -50 °C, require 71%and 57%more energy per kilo fish than for -30 °C and -40 °C, respectively, assuming R744 COP and 100mmthick blocks. The higher energy consumption is mainly due to decreased COP for lower evaporation temperatures, which was estimated to be 1.75, 2.25 and 2.98 for the abovementioned temperatures, using R744. In addition, product capacity (kg frozen fish per hour) is increased by correspondingly 66 % and 34 % by lowering the evaporating temperature to -50 °C. When implementing a energy storage system, shorter freezing times were obtained, because the pressure in the low pressure receiver is not elevated as much in the beginning of the freezing process. Results suggest a product capacity increase of 2.92 % for a low temperature R744 freezer with thermal energy storage solution. The tank volume was determined to be between 614 and 422 liter, and required between 1377 and 702 tubes, of 1 meter length and 10 mm radius.

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1 Introduction

The following parts of the thesis are relevant for this deliverable.

2 Theory

See: https://daim.idi.ntnu.no/

3 Method

First four parts: See: https://daim.idi.ntnu.no/

3.1

3.2

3.3

3.4

3.5 3CTES solution

As already mentioned, one of this thesis main objectives is to optimize a fish freezing system, regarding freezing time and product capacity. Installed compressor capacity is not dimensioned to cover peak heat loads, leading to increased pressure and temperature in the evaporator, which in turn prolongs freezing time. A CTES solution will aim to reduce this elevation in pressure by storing energy when available, and releasing it when needed, during peak loads.

3.5.1 Strategy and energy calculations

To introduce energy storage solution, it was necessary to combine it with a flooded evaporator. Simplified system solution from Kuldeteknisk AS, was used as reference, see figure 3.11.

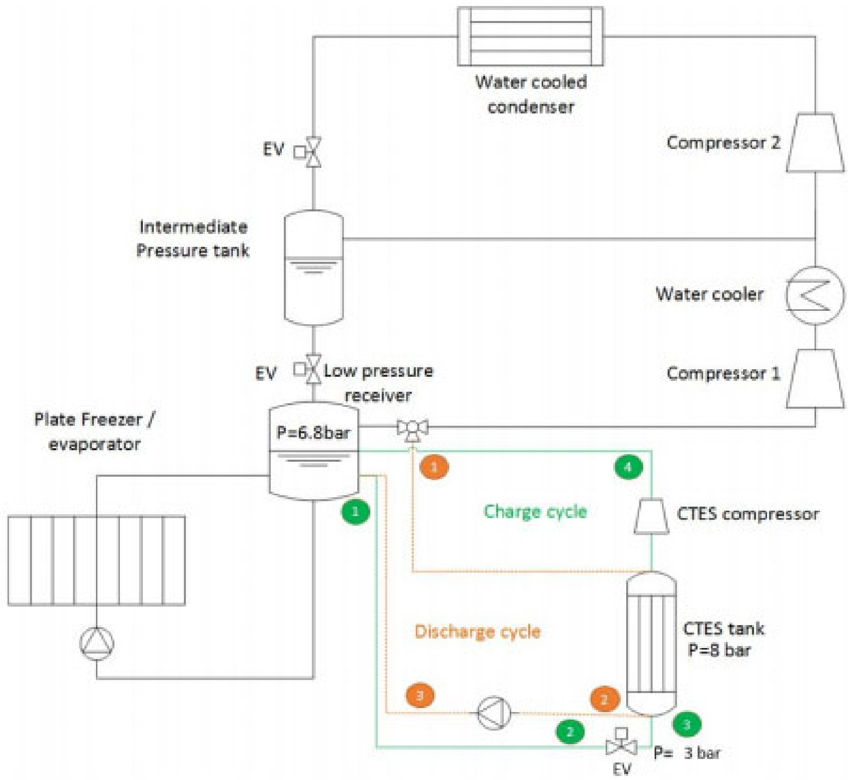

Figure 3.11: Suggested process diagram for a CTES add-on system

The intention of implementing a CTES system was to reduce the negative effect which follows by having a mismatch between supply and demand in freezing energy. The proposed solution was to run the main compressor(s) at design point, equal to the average heat load, throughout the freezing process. When the heat load is lower than the compressor capacity, the CTES system can start to run and store the available energy by expanding liquid from the low receiver to a low pressure and freeze the PCM in the CTES tank, called charging the tank. To be able to store energy at -50°, including a temperature difference in the CTES tank, pressures below the triple point of the CO2 refrigerant had to be assumed, meaning dry ice is generated after the expansion device before the CTES tank. Energy is available for storage when the pressure in the receiver is lower than designed, and would signal the expansion valve to open. When freezing the PCM, dry ice is sublimated and refrigerant gas exits the CTES tank. The refrigerant gas is finally compressed back to the receiver. During discharge, energy is released from the CTES tank by melting the PCM. Now, the compressor is struggling to maintain the low pressure in the receiver, due to excessive refrigerant vapour generation from the freezer. The receiver pressure starts to rise and signals the tree way valve to open, guiding some of the refrigerant gas through the CTES tank. The colder PCM is condensing the refrigerant, essentially helping the compressor to remove refrigerant gas in the low receiver. The liquid refrigerant from the CTES tank is pumped back to receiver.

Figure 3.12 illustrates the charge and discharge cycle in the P-H diagram. Note that logarithmic vertical axis are used above the triple point and linear axis below. Pressure and temperature levels are tabulated in table 3.1

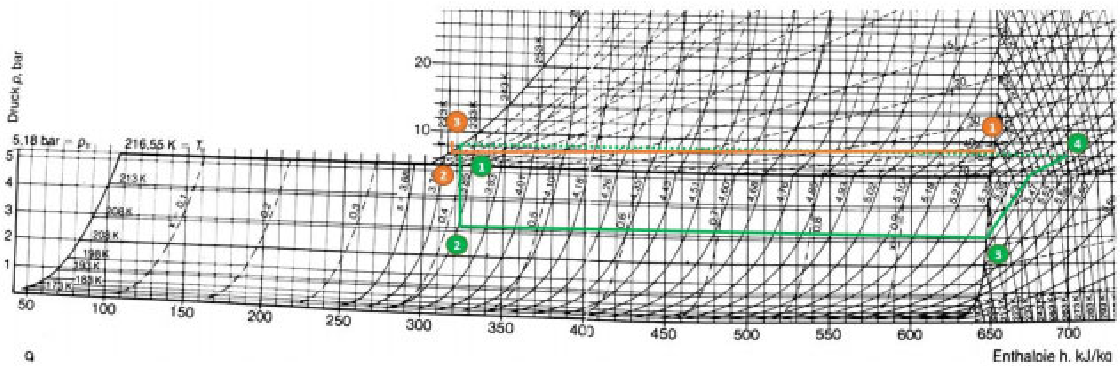

Figure 3.12: CTES cycle in P-H diagram

Illustration below takes a closer look on the CTES tank in figure 3.11. Notice the liquid supply tank, necessary to supply the tank with liquid during charging, as dry ice has approximately 20 % lower specific volume.

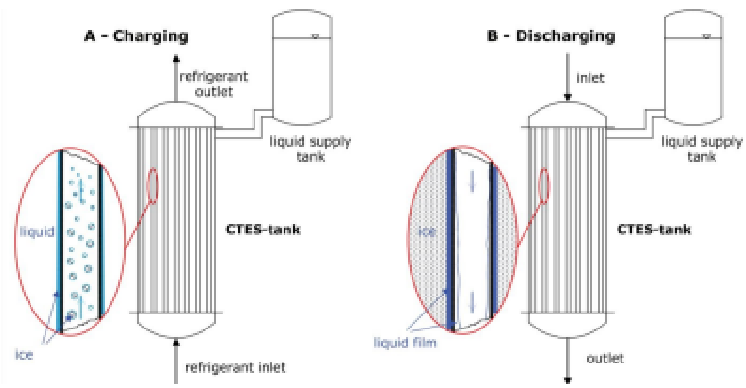

Figure 3.13: Close-up of CTES tank in figure 3.11 during charge and discharge [6]

As can bee seen from figure 3.11, an extra compressor is needed, possibly influencing the total energy input. In addition, it was important to calculate how much energy could be stored. The calculations was done as follows:

Heat load from fish was already known from simulations made to calculate the freezing time. Dimensioned compressor size was assumed to be average of the heat load. A script in MATLAB was made to calculate when heat load was smaller than compressor capacity and amount of storable energy after that point. That exact amount of energy was then added to compressor capacity in the first 15 minutes in the freezing process, simulating the discharging of the CTES tank.

Power consumption from CTES compressor could also be calculated by knowing the running time, storage time, and storable energy. Data from below the triple point of R744 is scarce, but PH- and TS-diagrams are presented in Appendix B.

3.5.2 Ice melting and dimensioning of CTES tank

To determine the dimensioning factors of the CTES tank, melting and solidification of PCM had to be studied. It was assumed a shell and tube heat exchanger, with PCM at the shell side. An important assumption is whether to have a two- or three dimensional model. Agyenim et al. (2010) [5] found that axial heat transfer was 2.5-3.5 % of the radial heat transfer, making the problem essentially a 2-D problem. Heat transfer was assumed from the inside of the condensing tubes, to the interface of solid/liquid PCM. Proper values for heat transfer from condensing CO2 is not readily available. Therefore, a preliminary analysis was done by varying the heat transfer in the tubes from relative low values (1700 W /m2K), to high values (4000 W /m2K), suggested by [53]. It turns out condensing transfer has relative impact on the total melting thickness of the PCM, differencing only 5 %. This validated the assumption that heat transfer resistance is dominated by the PCM, except for the beginning.

Aluminium pipe diameter was set to be 10 mm, and pipe thickness was assumed to be 3 mm. Temperature difference between refrigerant and PCM was set to constant 8K. Assumed temperatures and pressures in the tank are tabulated in table 3.1.

Table 3.1: Designed temperatures and pressures in CTES tank

	Tube side		Shell side
	Refrigerant, charge	Refrigerant, discharge	PCM
Temperature	-65 °C	-49 °C	-57 °C
Pressure	3 bar	7 bar	8 bar

Figure 3.14 illustrates melting of PCM around tubes, inside which refrigerant is condensing. The tube is only one of many tubes in the shell and tube heat exchanger, illustrated in figure 2.12.

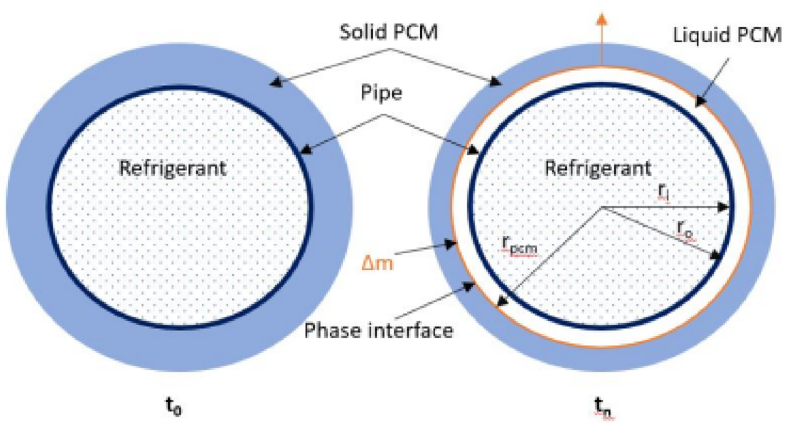

Figure 3.14: Illustration how solid PCM melts on tubes when refrigerant condenses inside

Time instance t0 corresponds to the onset of discharge, while time instance tn corresponds to an arbitrary time after t0.

Overall heat transfer coefficient product, UA, could be calculated by applying conductive and convective heat transfer in polar coordinates [29].

$$UA = \frac{2\pi L}{\frac{1}{r_i h_{conv}} + \frac{ln(r_o/r_i)}{k_{pipe}} + \frac{ln(r_{pcm}/r_o)}{k_{pcm} Nu_d}} \tag{3.17}$$

Equation 3.15 is valid for liquid PCM at rest. However, density difference in the liquid PCM gives rise to circulation in the fluid, called natural convection, resulting in higher heat transfer in liquid layer. Nusselt number, which relates heat transfer in convective fluids to pure conduction in fluids at rest, is defined by:

$$Nu_d = \frac{hd}{k} \tag{3.16}$$

Inserting Nusselt number into equation 3.15 yields:

$$UA = \frac{2\pi L}{\frac{1}{r_i h_{conv}} + \frac{ln(r_o/r_i)}{k_{pipe}} + \frac{ln(r_{pcm}/r_o)}{k_{pcm} Nu_d}} \tag{3.17}$$

Inserting equation 3.17 into equation 2.28, and expressing Δm by an increase in radius yields,

$$\Delta m = 2\pi r_{pcm} \Delta r \rho_{pcm} L \tag{3.18}$$

$$\Delta r = \frac{1}{\frac{1}{r_i h_{sub}} + \frac{ln(r_o/r_i)}{k_{pipe}} + \frac{ln(r_{pcm}/r_o)}{k_{pcm} Nu_d}} \cdot \frac{\Delta T \Delta t}{\Delta H \rho_{pcm^n} r_{pcm}^n} \tag{3.19}$$

Finally,

$$r_{pcm}^{n+1} = r_{pcm}^n + \Delta r \tag{3.20}$$

where radii can be seen from figure 3.14 n is time iteration Δr is the increase in liquid PCM radius from time iteration n to n +1 k is thermal conductivity, 0.17 W/mK and 237 W/mK for CO_2 and aluminium pipes, respectively

Now, the final thickness of the dry ice could be calculated, by knowing the storing time. Furthermore, the total mass of solid PCM could be calculated, by knowing the amount of required stored energy.

4 Results

4.1

4.2

4.3

4.4 CTES

4.4.1 Storable energy

Method from chapter 3.5.1 was done to simulate compressor capacity and heat load from fish, illustrated in figure below:

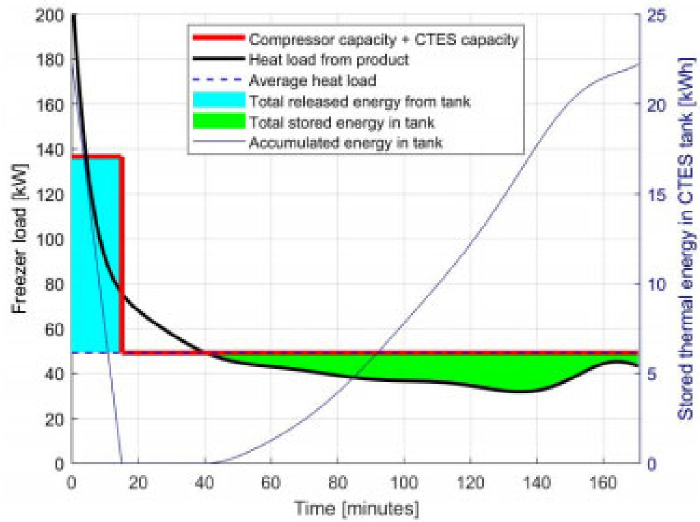

Figure 4.10: Heat load and total freezing capacity for 1250 kg fish in one plate freezer

Heat load from the fish is high in the beginning due to large temperature difference and high ice formation rate. After a while, 41 minutes, the heat load is smaller than the compressor capacity, meaning that it is possible to store the additional energy. Storing last throughout the freezing process and accumulates to around 22.2 at the end. This energy is released in the next freezing process, by condensing gas from the low pressure receiver, essentially helping the compressor to keep the low pressure. All the stored energy is now released in 15 minutes. This results in an added freezing capacity of around 89kW, 137kW in total. The added freezing capacity is enough the keep the pressure and temperature around the designed point, with very little variation.

4.4.2 Ice melting and dimensioning of CTES tank

One can imagine that the discharge, or melting, of the PCM tank would be the limiting factor, since it require larger heat transfer rates. Therefore, melting of PCM was modeled, described in chapter 3.5.2 and 2.4. Results from PCM melting thickness development is illustrated in figure 4.11.

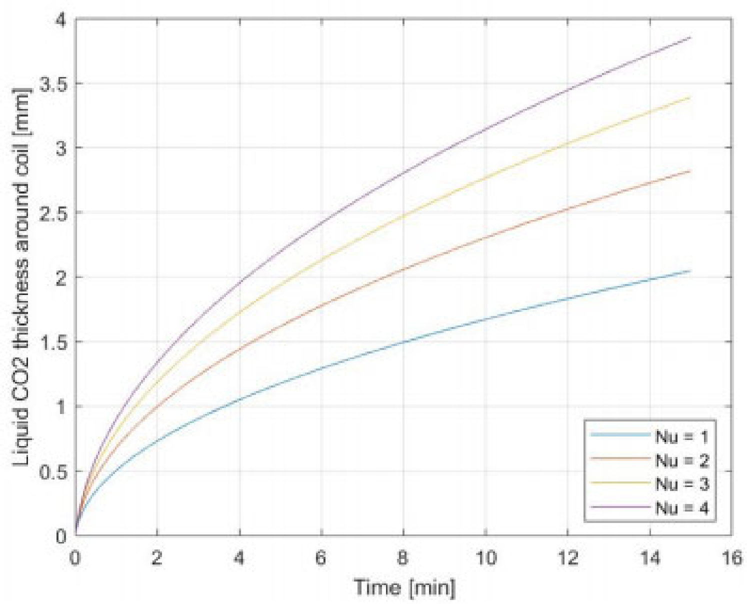

Figure 4.11: Liquid melted PCM layer development, for different Nusselt numbers

Results are dependent on the Nusselt number, as expected. Precise Nusselt number prediction is very complex. Small d/L, thickness of liquid layer divided by length of tube, generate large shear force between liquid moving up versus liquid going down. This will break up the typical circulation movement of one large circulation pattern along the whole pipe to smaller circulations. In light of this, it was chosen to implement the Nusselt number which relates convective heat transfer of moving fluid to a fluid at rest (essentially only conduction). Nu=3 means heat transfer three times as large compared to if the liquid would be at rest. Nusselt number tends to be around 2-4 for similar cases where natural convection is dominant.

Small Nusselt numbers results in a thin melted PCM liquid layer, meaning that more tubes, or larger heat transfer area (fins), are required. This is clear from the fact that 22.7 kWh of heat needs to be released and since the heat of fusion for CO2 is fixed, the mass of solid PCM that needs to be melted is constant. Dimensioning tank properties are tabulated below.

Table 4.8: Dimensioning properties for the CTES tank

	Nusselt number			
	1	2	3	4
Melted PCM thickness [mm]	2.02	2.74	3.24	3.64
No. of 1m tubes required	1377	979	805	702
Total volume of tank [L]	614	501	451	422
Mean heat transfer, $U[W/m^2K]$	264	372	452	518

The overall heat transfer coefficient, from the solid/liquid interface in PCM, to the inside of the tubes, is illustrated in figure below. The average value is tabulated in table 4.8

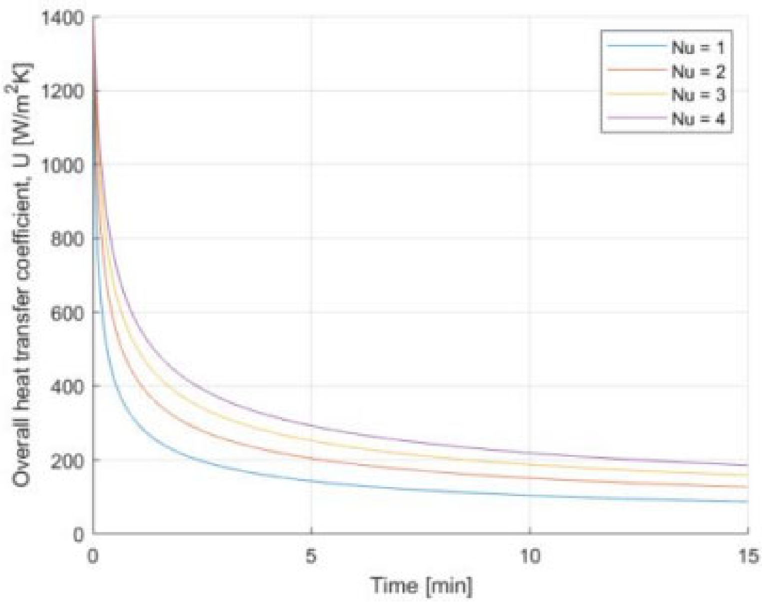

Figure 4.12: Overall heat transfer coefficient in CTES tank for different Nusselt numbers

In the beginning, the heat transfer is large due to low heat transfer resistance in the pipes. As the liquid layer starts to grow, the overall heat transfer is quickly dominated by the resistance of the liquid PCM. The high heat transfer might not be possible in real applications, thus, further research or experiments is required. Implementing fins to increase contact area is possible, and could be necessary to accomplish such high heat transfer rates. However, this will increase capital investment. Experiments or more precise numerical calculations is needed to validate the results.

4.4.3 Energy use and product capacity with CTES

One must not forget that the CTES system requires an additional compressor. A simple calculation was done to show that the CTES compressor is relatively small compared to the main heat pump compressors. Average mass flow was calculated by:

$$\Delta \dot{m}_{CTES} = \frac{\dot{Q}_{tot}}{\Delta S} = \frac{10.5kW}{219kJ/kg} = 0.048kg/s \tag{4.3}$$

Where Qtot is the averaged freezing capacity of the CTES tank in charge mode [kW/kg], and ΔS is the sublimation energy for the solid CO2 [kJ/kg].

Compressor energy can be calculated by having data on CO2 below, triple point. These are not easily gathered as most software focus on pressures above triple point. This thesis used data from Appendix B.

$$W_{c,CTES} = \dot{m}_{CTES}\Delta h = 0.048(701 - 649) = 2.5kW \quad (4.4)$$

Counting in that the CTES compressor's run time equals 75 % that of the main compressors, reveals that the CTES compressor power input is around 7% of the main compressors. This is a relatively small addition, but large enough to be considered in the main energy calculations.

Figure 4.10 reveals the mismatch in heat load and compressor capacity can be heavily downsized by use of CTES. In fact, pressure in the low pressure receiver is hardly influenced by the initially large heat load, with CTES system installed. Ideally, freezing times can be assumed to be close that of freezing times considering constant plate temperature. Table 4.9 evaluates the freezers KPIs, with and without a CTES system.

Table 4.9: Comparing dimensioning properties with and without CTES

	With CTES	Without CTES	Difference [%]
Freezing time [min]	168	174	-3.17
Energy use [kJ/kg]	289	268	7.90
Product capacity [kg/h]	399	388	2.92

Table 4.9 reveals a decrease in freezing time by 3.17%, which results in increase of product capacity by 2.92%. The slight difference in product capacity increase and freezing time is because there was assumed a reset time (defrost and unloading/loading of product) of 20 minutes. Results conclude that both energy use and product capacity slightly increases with a CTES system installed. A relative small boat, like MS Arnøytind which have 3 freezers (1250 kg fish each), could potentially freeze 800kg more fish each day by implementing a CTES system. Further cost analysis must be done before concluding with potential profit. Thermal storage is in this thesis only evaluated for low temperature R744, but could in theory also be used for higher temperatures, displayed by R717 and R290.

5 Summary and conclusions

In this thesis, a numerical freezing model was developed to predict the freezing time for fish in plate freezer. The freezing model was validated by conducting experiments on a phase changing test material in a CO2 based, industrial plate freezer. Comparison from experiment and numerical model concluded a deviation of only 3 %. Furthermore, the pressure in the low receiver was modeled to cover the dynamic plate temperatures arising from insufficient compressor capacity. The pressure model was validated by comparison with data from similar facility.

Another model was made to estimate system efficiency, COP, for two different freezing systems using various natural refrigerants. Evaporating temperatures from -30 °C down to -50 °C was studied. CO2 yields the lowest efficiencies, with COP ranging from 1.8 to 3.0 while COP for ammonia and propane was approximately 11 % higher for the same system. A CO2/NH3 cascade system was found to be on average 6 % more efficient than the pure CO2 system.

For low temperature CO2 systems, energy required to freeze fish was increased by 70 %, compared to -30°evaporating temperature for the same system. However, the freezing cost was calculated not to be significant to the total sales income. Freezing cost was estimated to be 90-190 NOK/tonne (10-20 e), while sales cost of frozen cod is around 32000 NOK/tonne (3263 e).

Focus was therefore redirected to freezing rate and product capacity increase, which is essential for fishermen. Before freezing, the fish is intermediately stored in RSW tanks, where temperature is around 0 °C. The time spent in this tank influences the fish quality. Furthermore, faster freezing enables the fishing vessel to catch more fish while there is fish present, by faster removal of fish in the RSW tanks. This ultimately reduces time at sea, and therefore cost and fuel usage for the vessel. Results revealed a product capacity increase of up to 66%, when freezing with -50 °C compared to -30°C. Even higher product capacity increase is possible when reducing fish block thickness. This is especially of interest when freezing pelagic fish, like mackerel and herring, which is smaller in size and can be closely packed, even in thin blocks.

A cold thermal storage system (CTES) was investigated, aiming to reduce the pressure elevation in the low receiver, which in turn causes prolonged freezing times. The CTES system utilized a shell and tube heat exchanger, containing PCM. Surplus "cold" energy from the freezing system was stored in the CTES tank by freezing the PCM. Likewise, when the compressor struggled to keep the low pressure, the stored energy was released, boosting the total refrigeration capacity. The CTES tank size was determined to be between 614 and 422 liters, and required 1377-702 tubes. When implementing the proposed CTES system, shorter freezing times was obtained. Results suggest a product capacity increase of 2.92% for a freezer with thermal storage solution.

6 Discussion

The numerical model fits experimental data very well, when comparing with test material with excellent contact with plates in the freezer. However, fish will have less contact with freezer plates due to the irregular shape of the product. Therefore, freezing times for fish is expected to be longer. This was modeled by implementing altered formulas for conductivity, assuming porous medias with a specified air void fraction. If possible, the numerical results presented in this thesis should be validated using industrial plate freezers and frozen fish. This was originally planned for the thesis, however, no plate freezer were available for testing.

When calculating COP for the different systems and refrigerants, simplified assumptions are made. COP will be reduced if all losses are included, but especially CO2 can have less heat loss in compressors due to relative low compressor outlet temperature, and less friction loss in pipes due to low viscosity.

Results from the product capacity calculations demonstrates an increase, both when reducing evaporating temperature and fish block thickness. However, too thin fish blocks will result in poor contact between product and plate, in addition to higher air void fractions, which increases freezing time. Following this logic, there exists an optimum thickness, regarding product capacity. Better models, with correlation between fish size, block thickness and air void fraction, along with more experiments must be done to find this optimum thickness.

One might argue on the two-dimensional assumption when modeling the CTES tank. When charging and discharging the tank, heat transfer rates are likely to vary along the pipes. In discharge mode, gas starts to condense on the inside surface of the pipes, causing a buildup of a liquid boundary layer. The boundary layer acts as a thermal resistance, reducing the overall heat transfer coefficient and reducing the melting rate on the PCM side. The same problem arises when charging the tank. The dry ice particles generated in the bottom of the pipes must be sublimated before entering the compressor, or some kind of cyclone separator must be installed ahead of the compressor. If not, somewhere in the pipe, all dry ice is sublimated and very low heat transfer rates is achieved. Stratification of the liquid/solid PCM layer is the result of this, which is not a possible result for a two-dimensional analysis. Also, it is assumed noe sub cooling of the PCM, which often is the case for pure liquids. The effect of excluding the assumptions above might result in underestimation of size and number of tubes in the tank.

The chosen PCM for thesis thesis was CO2, because of cost effectiveness, good temperature match and knowledge on the subject. CO2 has very high energy storage density, roughly 400 MJ/m3 , but has generally low thermal conductivity due to its non polar structure. Maybe other commercially available PCMs are better suited for the task, when considering volume of tank and achievable heat transfer rates.

More practical problems needs to be addressed, like control system, defrost, more than one freezer in parallel, plugging of ice in the CTES tank and perhaps the need for a "cold finger" when the freezing system is not running. A cold finger is a small heat pump unit maintaining the low temperature in liquid receiver and CTES tank, when the system is not running. Obviously, heat is leaking no matter how good insulated the components are.

6.1 6.1 Control system description

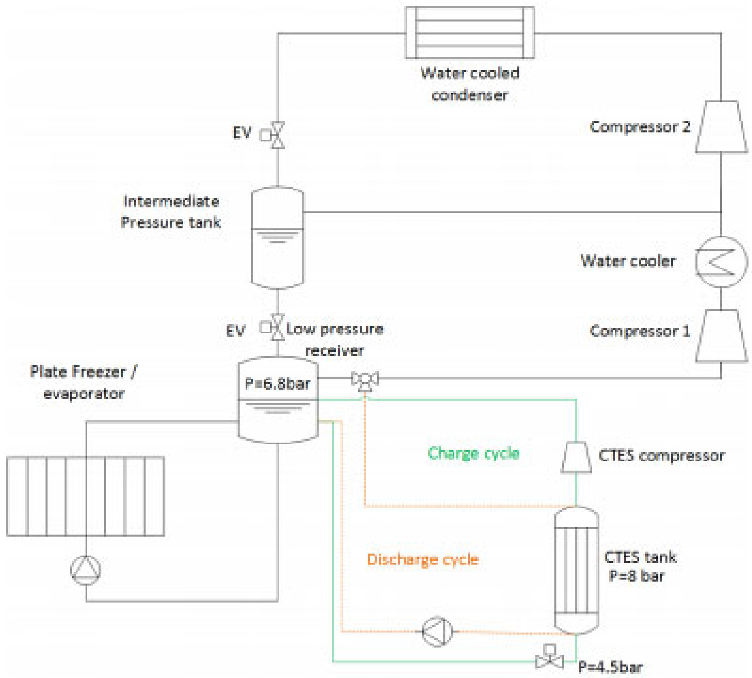

Figure 6.1: Suggested process diagram for a CTES add-on system

The most important control parameter in the freezing system, displayed in figure 6.1 is the pressure in low pressure receiver. As proven in this thesis, evaporating temperature is crucial for maximizing the product capacity of the system. The pressure in the receiver could be controlled by turning on/off the CTES system. This can replace frequency controlled compressors, which are expensive and more energy demanding.

Higher pressure than the designed 6.8 bar is a sign that the compressor cannot remove the gas generated by the freezer. This should signal a valve to open up for gas to be condensed in to the CTES tank. Too low pressure is a sign that compressor capacity is higher than the heat load, and might generate solid CO_2 in the receiver and freezer, which is not desirable. This should signal the CTES compressor to start and the expansion valve in to the CTES tank to open, allowing liquid to be removed and sublimated refrigerant to re-enter the liquid receiver as gas. This will stabilize the pressure to the designed level. Frequency converter should, however, be installed on the small CTES compressor, since available energy to be stored is varying during the freezing process, see figure 4.10.

The liquid pump to the freezer should ensure enough liquid to be evaporated. Insufficient liquid delivery results in too excessive gas formation in the tubes, leading to low heat transfer. Too much liquid results in high pressure drop and oddly enough, also low heat transfer because of a different flow regime. Studies have shown that circulation ratios around 3-6 results yields optimum heat transfer. However, heat transfer quickly drop past 6, so a circulation rate of 3-4 is recommended. Circulation ratio cannot be measured directly, so to ensure sufficient liquid supply, the pump is controlled by the pressure drop through the freezer. Freezing system suppliers often have a designed pressure drop that has proven to be effective when it comes to heat transfer rates, so the pump can be frequency controlled to match that specific pressure drop.

The intermediate pressure is, by some suppliers, set to a constant value. For R744 systems with sea water cooling, a temperature of -10°C is sometimes set as design point. This may not be the most effective pressure level, but it ensures a more stable liquid level in both receivers, and stable low level compression.

Condensing temperature is mostly dependent on sea water temperature, which is relatively constant in the north sea (9±5°C). The sea water is supplied by a sea water pump, and condensing temperature will be somewhat dependent on the mass flow, delivered by this pump. A high mass flow results is higher power consumption, but also lower condensing pressure.

6.2 6.2 Multiple freezers in parallel

One important thing to notice is that this thesis assumes only one freezer supported by one simple compressor. In reality, it is common to have multiple freezers connected in parallel, supported by multiple compressors. Some fish trawlers are basically frozen fish factories at sea, capable of having over 10 ten plate freezers in addition to blast freezers and a large frozen storage. For these vessels, energy storage may not be needed, since the base heat load is huge and the fluctuations due to varying heat load is small in comparison.

However, it is of interest to look at fishing vessels, or land based facilities, having a few plate freezers in parallel. A smart strategy would be to start the freezers with an interval, of maybe 20 to 45 minutes. Then, peak heat load would be spread out instead of starting all the freezers at once, resulting in a huge peak heat load. If multiple compressors are used, some of the compressor capacity regulation may be done by turning on/off the compressors.

Below, a simulation of the total heat load in a system with 3 freezers.

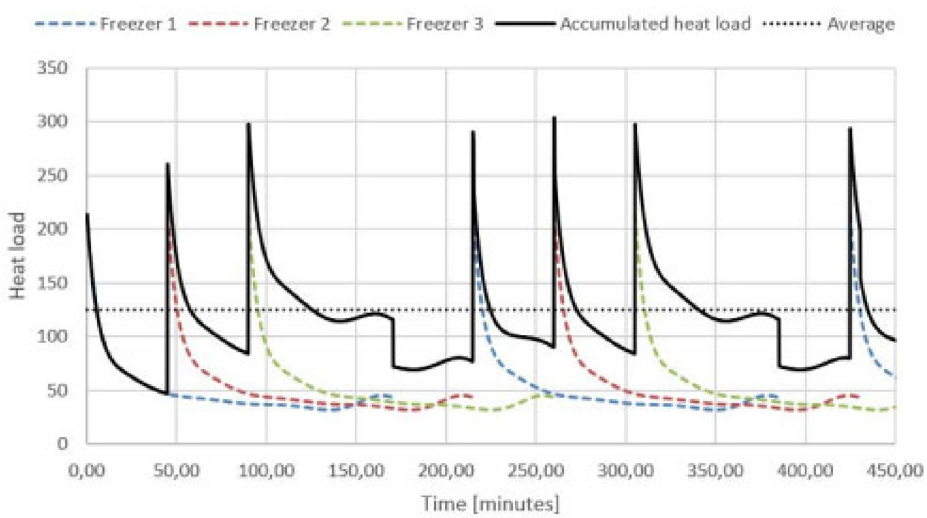

Figure 6.2: Three freezers in parallel, starting with intervals of 45 minutes

Figure 6.2 illustrates the respective and combined heat load of three plate freezers in parallel, with 45 minutes intervals. Maximum heat load is approximately 300 kW, and the dimensioning compressor capacity is approximately 115 kW, provided by two or three compressors. If thermal storage is used, the mismatch between compressor capacity and heat load can be reduced by the same principle shown in figure 4.10. Storage tank dimensions are likely to be equal as in table 4.8, since time between peaks are longer than time required to charge the tank. The tank was able to discharge almost at almost 100kW capacity for 15 minutes. Recharge time is shorter due to higher conductivity of dry ice, compared to liquid CO2. As mentioned before, these results require more research and experimental validation.

MS Arnøytind used four 45 kW low stage compressors, adding up to a maximum capacity of 180 kW, whereas one of them is frequency controlled. As can be seen from figure 2.5, heat load rarely exceeds 180 kW, which is essentially an overdimensioned system, but is needed to keep the low pressure during peak loads. Also, a freezer storage with unknown load must be covered. If the system included a CTES system, a lower installed compressor capacity could be installed. The cost effectiveness of such a system is, for now, unknown. One would have to balance the reduction in price due to lower compressor capacity, and the added cost of a CTES system. Also, benefits of a possible increase in capacity must be included.