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Evaluating Plate Freezing of Fish Using Natural Refrigerants and Comparison with Numerical Freezing Model

Espen Verpe[a], Michael Bantle[b], Ignat Tolstorebrov[a]
[a] Norwegian University of Science and Technology, Department of Energy and Process Technology, Ignat.Tolstorebrov@ntnu.no
[b] SINTEF Energi AS, Department of Thermal Energy, Michael.Bantle@sintef.no

ABSTRACT

This paper compares experimentally measured freezing time and predicted numerical freezing time in industrial CO_2 plate freezer. The model uses a two-dimensional, heat capacity-temperature, finite difference method with constant temperature boundary conditions. Also, COP was estimated for two different systems using different refrigerants and evaporating temperatures. Freezing time and COP was used to calculate relative energy requirements for fish freezing and freezing capacity for the different system solutions.

The numerical model demonstrated good agreement with experiments, with a difference in freezing time of only 3%. Results reveals that low temperature freezing, down to -50°C, require 51% and 20% more energy per kilo fish than for -30°C and -40°C, respectively, using R744. The higher energy consumption is due to lower COP, which are 1.8, 2.3 and 3.0 for the abovementioned temperatures for a two stage R744 system. However, capacity (kg frozen fish per hour) is increased by correspondingly 65% and 33%. Different freezing times results in varying refrigeration capacity, ranging from 47.9 to 25.9 kW for -50°C and -30°C, regardless of refrigerant.

1. INTRODUCTION

For many years, synthetical refrigerants like R22 has been the dominant refrigerant on-board fishing vessel due to high efficiency, manageable operation pressure, non-toxicity and non-flammability. However, recent change in industries all over the world have shifted focus to more environmentally friendly alternatives, with help from political induced taxes and phase-out of synthetically refrigerants. Global Warming Potential and Ozone Depletion Potential, referred to as GWP and ODP, are central values to estimate the harm when the refrigerant is leaked into the atmosphere. For example, R22 has a GWP of 1810, which means it contributes to global warming nearly 2000 times as much as for the same amount of CO_2.

This has led to recent development in systems using naturally occurring refrigerants, like CO_2, ammonia and other hydro carbons which have no or low GWP and ODP values. Nowadays, CO_2 and ammonia are the predominant refrigerants in the industry, but ammonia have a crucial weakness when it comes to evaporating temperature. Below -33.3°C, the evaporating pressure is sub-atmospheric, risking leakage of water and air *into* the system. This is not the case for CO_2 which can, in theory, reach temperatures to -56.5°C. In practice it is limited to around -50°C, reducing the risk of dry ice formation.

One of the main reasons of this project is to predict freezing times of fish blocks in a plate freezer using CO_2, which enables faster freezing than can be expected for ammonia based systems. In fact, plate freezer manufacturers, like Dybvad Stål Industri in Denmark, claim that freezing time can be reduced by 25-50%. At the same time, lowering the evaporation temperature increases the pressure ratio of the compressors, resulting in decreased system COP. Moreover, different thermodynamic properties of the different refrigerants will influence the efficiency.

Fast freezing time, leading to increased capacity, is crucial to fishing vessels. Higher capacity means that fishing boats can empty the RSW tank faster, where unfrozen fish is stored. Reduced time in RSW tank improve the quality of fish. Furthermore, boats can harvest more fish *while* the fish is present. This reduces time at sea, and therefore fuel consumption and cost. At last, higher capacity means fewer boats can harvest the same amount of fish, further reducing fuel consumption and economy of the owner.

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Prediction of freezing time is an important parameter when designing new freezing systems or freezing facilities. As of today, there was not found available software specifically designed to predict freezing time in plate freezer.
This project will develop numerical simulations for plate freezing, and compare with experimental data in industrial plate freezer using R744. Furthermore, investigation of block thickness and plate temperature's influence on freezing time, energy requirements and freezing capacity will be conducted.

2. METHODS AND MATERIALS

2.1 Installation

The heat pump system where the experiment was conducted was delivered by Kuldeteknisk AS. It is a two-stage seawater condensed system with a pump circulated evaporator (plate freezer), illustrated below. Evaporating temperature was held close to -50^oC. Lower temperatures were regarded as unsafe, risking local low pressure and dry ice formation.

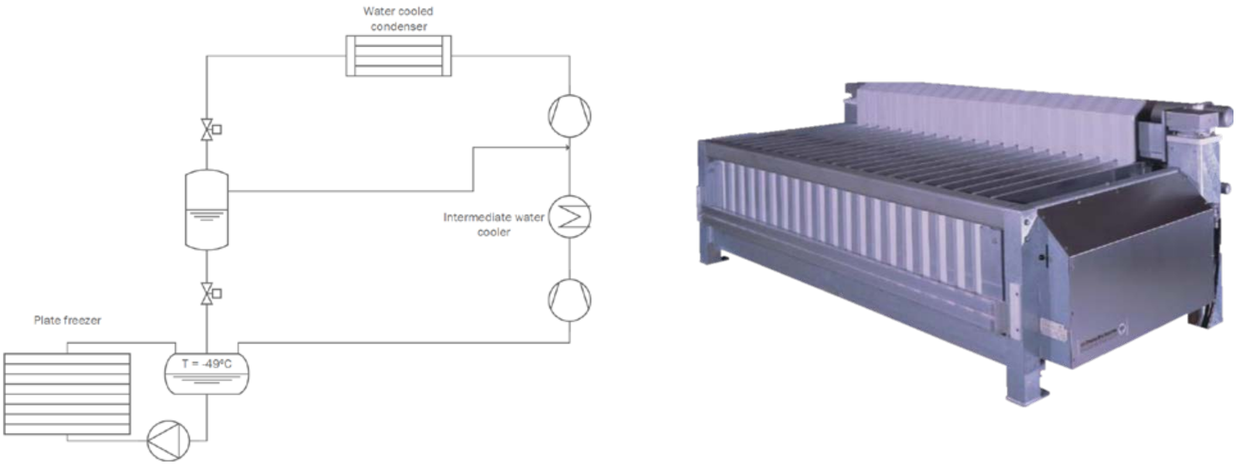

Figure 1: Simplified system solution and corresponding plate freezer used in the experiment

The vertical plate freezers are of the type DSI V3. Each freezer has capacity of 48 fish blocks with a block size of 530x520x100mm, corresponding to approximately 1200 kg fish.

2.2 Description of experiment

In the industrial plate freezer, illustrated in Figure 1, a 103 mm thick block of test material was placed between the evaporating plates. Temperature was measured using Accsense VersaLog TC-monitor in the geometrical center, 51.5 mm away from the wall. Evaporating temperature was measured to be -49±0.5^oC during the experiment.

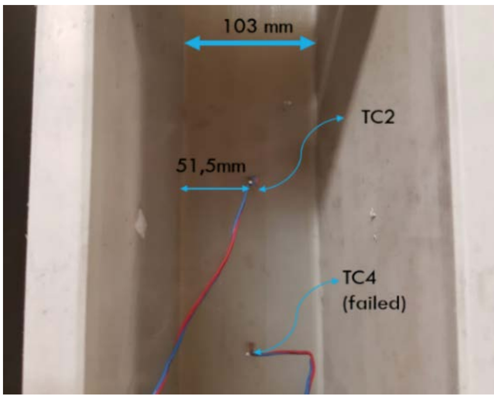

Figure 2: Experimental setup

2.3 Test Material

A phase change material (PCM), was chosen as test material due to availability and simple measurement probing. A readily available PCM is a water/agar-agar mix. Agar-gel will have similar freezing point and

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thermodynamical properties as water and is non-toxic. Since block thickness was measured to be 103mm at the industrial plate freezer, the test material had to be made to the same thickness. Agar-agar was mixed with water (10g/L water) and brought to boiling until the liquid was clear again. Salt was added to correct for the lower initial freezing point of fish (20g/L water). The boiling solution was quickly poured into a mold with a flat bottom.

The test material consists of almost 99% water, so the thermal conductivity could be assumed to be similar to values of water. However, the gel has a much more structured arrangement, possibly leading to higher conductivity. Therefore, a series of tests, using Transient Hot Disk Method, was performed. Results revealed around 10% higher conductivity for agar-gel, compared to water, for temperatures above freezing. Other thermodynamical properties like c_p and ρ was measured to be equal to water.

2.4 CFD Model

Numerical methods are far superior to analytical models when it comes to accuracy, because of the possibility to have temperature dependent thermal properties, complex shapes and changing surface heat transfer coefficient. They are, however, far more time consuming, both in computation and implementation (Valentas et al. 1997)

The chosen method in this project is to solve the two-dimensional heat diffusion equation using finite difference and apparent heat capacity, c_a, which includes both latent and sensible energy (Schwartzberg, 1976)

$$\frac{\partial}{\partial t}(\rho(T)c_a(T)\cdot T) = \vec{\nabla}\cdot\left(k(T)\cdot\vec{\nabla}T\right) \quad (1)$$

Equation (1) is expanded in two dimensions and non-dimensionalized. The implicit forward time central space is chosen as numerical scheme. Implicit FTCS is unconditionally stable, enabling larger step size for faster simulations (Abbasi, 2015). Thermal properties are updated on every time step with values calculated from methods described in ASHRAE (2010). Fixed temperature at evaporating plates, and insulated short sides is used as boundary conditions. More detailed model description is found in Verpe (2017).

Other methods to predict freezing time include rewriting the left hand side of Equation (1) to solve for enthalpy (*Mannapperuma & Singh, 1989*). Since enthalpy curve is known for many foods, one can calculate enthalpy decrease for a given time.

2.5 System solution model

To calculate COP for different system solutions and refrigerants, model using RnLib, was developed. The model calculated necessary thermodynamic properties at several locations in the system, enabling a COP estimation.

In all systems it was assumed water cooled condenser at 8ºC and 5K temperature difference. Internal heat exchangers before compressors ensured 10K overheat. The pressure ratios were set equal for both compressor stages, while isentropic efficiency was set to 0.71 for all compressors. No heat loss was calculated for compressors or condensers, but heat loss in evaporators was calculated for the different evaporating temperatures.

Also, pressure loss in evaporators was calculated using methods from Sardeshpande et al. (2014). In this method, pressure loss arises from acceleration loss, pipe friction and height difference.

For the rest of the calculations a two stage system with intermediate pressure receiver and a cascade system is used, illustrated in Figure 3a. Here, a direct expansion evaporator is assumed as it has a faster set-temperature in evaporator, because the refrigerant starts to boil immediately. This might be important for heat pump systems which are often turned on/off.

Similar configuration was used for cascade system. There, the intermediate pressure receiver is substituted by a cascade heat exchanger which operates as a condenser for the low temperature side and an evaporator for the high temperature side.

Evaporating temperature could be controlled by a thermal expansion valve, using the temperature at the outlet of the internal heat exchanger. Frequency controlled compressors sustains the low pressure in evaporator and regulates the mass flow. Varying evaporating temperatures results in different freezing times and therefore different compressor capacity.

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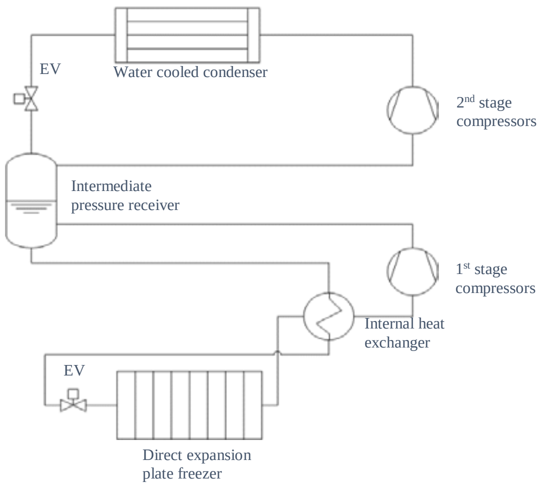


Figure 3a: Two stage with internal heat exchanger and pressure receiver

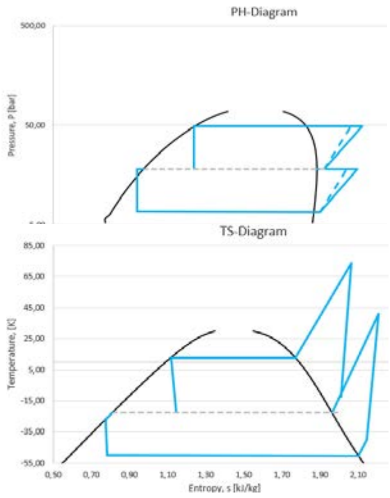
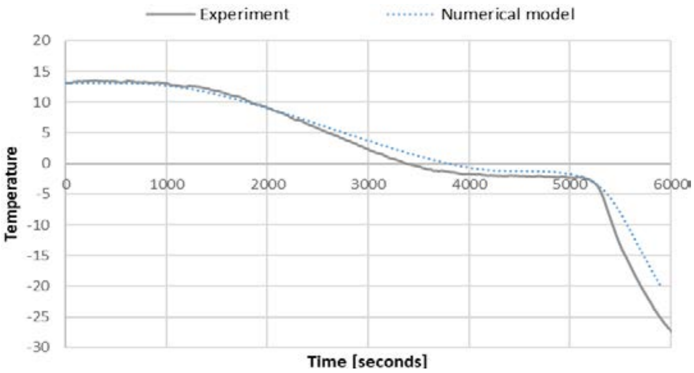


Figure 3b: Corresponding Ph and Ts-diagrams for R744 at T_0=-50°C

3. RESULTS AND DISCUSSION

3.1 Experimental freezing time in vertical plate freezer

Results from experiment and numerical model prediction, using similar conditions as the experiment (thermal properties for agar-gel, evaporating temperature: -49 °C, thickness: 103mm), is compared in figure below:



	Freezing time
Numerical model, gel	98.5 min
Experimental, gel	95.7 min

Figure 4: Results from plate freezer experiment and numerical model

Time for agar-gel to be frozen (from +13°C to -20°C in center) was measured to be 95.7 minutes. The numerical model predicts freezing time to be 98.5 minutes, which is 2.6% over-estimation. The numerical model follows the measurements closely at the beginning, but freezing rate decreases after about 40 minutes. Initial freezing, in center, occurred later than measured in experiment. Furthermore, the temperature drop after “the critical zone” is steeper than for the numerical model. Similar results are found in Seara et al. (2012), where freezing time was measured to be between 5500 and 6400 seconds, using 9.5 cm water blocks and slightly higher evaporating temperatures.

The temperature measurements could be affected by the measuring probe itself. The TC was inserted in the gel using a thin (~1mm) steel rod, see Figure 2. The rod’s contact with warmer surrounding air might have influenced the measured temperature, even though the TC tip was placed 3cm away from the steel rod end.

3.2 Pressure drop calculations

Pressure loss can be calculated for the relevant evaporating temperatures for the selected refrigerants, assuming 1200kg fish in freezer. Results are tabulated below:

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Table 1: Important values in plate freezer for selected refrigerants and applicable evaporating temperatures

		$T_{e,avg}$ [°C]				
		-50	-45	-40	-35	-30
R744	G [kg m^{-2} s^{-1}]	236	205	179	158	137
	W_c [kW]	27.3	21.5	15.7	12.2	8.7
	ΔP [bar]	0.55	0.39	0.29	0.22	0.16
	ΔT [K]	1.96	1.2	0.77	0.52	0.34
R290	G [kg m^{-2} s^{-1}]			130	113	97.8
	Wc [kW]			14.1	10.6	7.7
	ΔP [bar]			0.19	0.14	0.10
	ΔT [K]			3.97	2.54	1.52
R717	G [kg m^{-2} s^{-1}]				33.9	28.9
	Wc [kW]				10.8	7.8
	ΔP [bar]				0.13	0.08
	ΔT [K]				2.7	1.3

Relevant temperature is assumed to be corresponding temperature where pressure at the evaporator outlet is above 0.9 bar. This means for example limiting the T_e to -45°C and -36°C for R290 and R717, respectively. Pressure loss in dependent on thermodynamical properties, and mass flux squared. R717 has low mass flux, and therefore low pressure loss, in evaporator due to high latent heat of evaporation. However, R717 (and R290) has high $\Delta T/\Delta P$, resulting in higher drop in evaporating temperature. R744 on the other hand, has one of the lowest $\Delta T/\Delta P$ of all refrigerants.
Evaporating temperature is regarded as average temperature across evaporator. R290 has relative high temperature and pressure loss. Therefore, -45°C might not be an applicable evaporating temperature as the pressure at the outlet of evaporator and internal heat exchanger is too far below atmospheric pressure (0.7 bar). Also, -35°C for R717 is a questionable evaporating temperature, as the lowest pressure is just below 0.9 bar.

3.3 Evaluation of system efficiency using natural refrigerants

Different natural refrigerants are used to calculate COP for system illustrated in Figure 3a with assumptions explained in chapter 2.5. Also, a cascade system with same assumptions, and a 5K temperature difference in cascade heat exchanger are used in the comparison. Results are shown in figure below:

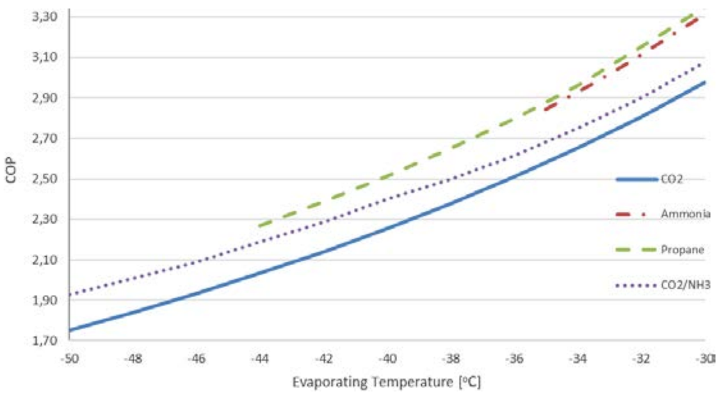

Figure 5: COP for cascade and two stage system and selected natural refrigerants. Varying evaporating temperature and T_c=13°C.

Figure 5 shows that R744 has the lowest COP while R290 and R717 has the highest. Cascade system is somewhere in between as it uses two refrigerants, but has an additional temperature difference in the cascade heat exchanger. COP for all refrigerants would be lower if compressor heat loss and friction in pipes is included. However, the losses would have higher impact for R290 and R717 because of high compressor outlet temperature and high viscosity compared to R744.

3.4 Evaporating temperature and fish block thickness influence on freezing time

As one can expect from the widely used Plank equation (Plank, 1941), increased thickness and higher evaporating temperature results in longer freezing times. Relative freezing times is calculated in the numerical model described in Chapter 2.4 and results are tabulated below.

Table 2: Relative freezing times for cod with varying thickness and plate temperature

		Evaporating Temperature [°C]		
Relative Freezing time [-]		-50	-40	-30
Thickness [mm]	75	**33 %**	**41 %**	**56 %**
	100	**58 %**	**73 %**	**100 %**

As can be calculated from Table 2, freezing times are dependent on the thickness squared for the same evaporating temperatures.
DSI, a Danish plate freezer manufacturer, claims that using low temperature R744 reduces freezing time by 25-50% compared to standard refrigerants. Assuming evaporating temperature of -33°C for standard HFC and -50°C for CO_2 reveals a decrease by 33%, which is consistent with DSI results.
Freezing time for fish is calculated assuming homogenous material and regular shape. As fish is irregularly shaped, air voids and reduced contact with evaporating plates must be included in model for accurate freezing times. This will be included in further work. However, relative freezing times still might be valid

3.5 Relative energy use and capacity calculations

With known COP and freezing time has been calculated, it is possible to estimate the energy required to freeze the fish. Capacity, amount of fish frozen per hour, is also estimated.

Table 3: Specific freezing energy in cod

	T=-50°C	T=-40°C	T=-30°C	
Latent Heat	240	238	235	kJ/kg
Pre-cooling	57.2	57.2	57.2	kJ/kg
Sensible Freezing	68.9	59.3	49.9	kJ/kg
Total freezing energy	**366**	**354**	**342**	kJ/kg

Table 4: Relative compressor energy required to freeze fish and freezing capacity (COPs for R744 system are used)

		Evaporating temperature [°C]		
Relative energy use [kJ/kg]		-50	-40	-30
Thickness [mm]	75	**136 %**	**107 %**	**84 %**
	100	**151 %**	**120 %**	**100 %**

		Evaporating temperature [°C]		
Capacity [kg/h]		-50	-40	-30
Thickness [mm]	75	**201 %**	**168 %**	**127 %**
	100	**165 %**	**133 %**	**100 %**

Compressor energy use increases with reducing temperature, mainly because of lower COP. Also, energy required to freeze 1 kg of fish is calculated in Table 3, which demonstrates higher required energy to be frozen for lower temperatures, due to higher total ice fraction and steeper temperature gradient.
In the capacity calculations, a reset time (defrost and unloading) is set to 20 min and a freezer with a fixed number of plates is assumed. Therefore, the amount of fish per batch is smaller for thinner blocks. Nevertheless, the capacity still increases with reduced thickness. This can be explained by the fact that freezing time is dependent on thickness squared, whilst the amount of kilos per batch is linearly dependent on thickness. However, too thin blocks will result in bad contact between fish and plates, and larger air voids in the frozen fish block. Following that logic, there might exist an optimum block thickness, regarding capacity. Further experiments are required to find this optimum block thickness.
In Verpe (2017*)* estimations was done to calculate cost of the freezing process. It was made clear that direct cost of energy in the freezing system is under 0.5% of total sales price, and therefore not a significant part.

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3.6 Refrigeration capacity

Calculating enthalpy in the fish as it freezes, it is possible to determine refrigeration capacity during freezing.

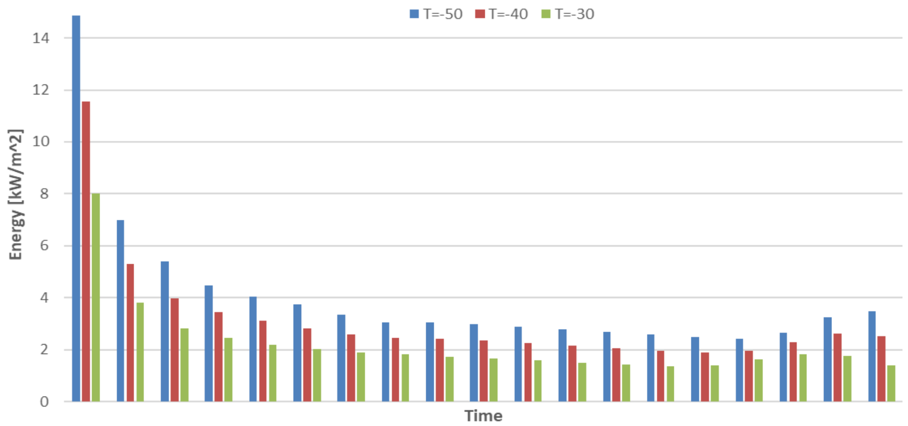

Figure 6: Refrigeration capacity per contact surface area during freezing of 100mm cod

Results reveal that refrigeration capacity is large at the beginning, when the temperature difference between plate and products is at it's highest. After some time, refrigeration capacity is almost constant. This can be explained by the "freezing plateau" where temperature in product is almost constant.

For all three cases, refrigeration capacity is increased at the end of the freezing period. This happened around the time the core temperature starts to decrease again, see *Figure 4*. The increase might arise from increased conductivity as ice fraction is near the maximum value, which increases the overall heat transfer coefficient and in turn allowing more heat to be "extracted" from the fish. Results in *Figure 6* are measured in kW/m^2, meaning refrigeration capacity in kW per surface area of one side of the fish block.

Table 5: Average capacity and freezing times for 100mm thick cod and 1200 kg fish

1200 kg	T=-50ºC	T=-40ºC	T=-30ºC	
Average refrigeration capacity	47.9	35.3	25.9	kW
Freezing time	152.8	193.8	264.8	min

As expected, refrigeration capacity is higher for lower evaporating temperatures, due to faster freezing times and similar energy to be frozen, see table 2 and 3.

4. CONCLUSION

In this investigation, a numerical freezing model was developed to predict freezing time in fish, using a plate freezer. The model was validated using experimental data from freezing a PCM test material in a R744 based plate freezer. Comparison from experiment concluded with an overestimation of 3% of the model.

A process model was made to estimate total efficiency, COP, for two different freezing system using different kinds of natural refrigerants. Evaporating temperatures from -30ºC down to -50ºC was studied. CO2 had the lowest efficiency of the selected refrigerants with COP varying from 1.8 to 3.0 while COP for ammonia and propane was approximately 11% higher, for the same system. A Cascade system using R744 at the low stage and ammonia at the high stage was on average 6% more efficient than the pure R744 system. However, the evaporating temperature when using propane and ammonia is limited to approximately -43ºC and -33ºC, respectively. Therefore, when using R744 at the low stage, faster freezing and increased capacity is enabled. Furthermore, not all losses are included in the model. By doing so, R744 efficiency is likely to decrease less, compared to the other refrigerants, due to special thermodynamic properties.

Using low temperature R744 systems, energy required to freeze 1 kg of fish is increased by almost 63%. This may sound much, but this number is likely to be smaller if more exact assumptions are made. Also, cost of freezing was calculated to not be a significant part of the total sales price.

Focus is therefore redirected to capacity increase, how much fish can be frozen per hour. This is essential for fishermen as it determines how much fish that can be caught. When the intermediate storage, RSW tanks, is filled up, no more fish can be caught. The temperature in the RSW tank is well above zero, so quality of the fish is dependent on time spent in the tank. Furthermore, faster freezing enables the fishing vessel to catch more fish when there is fish present, reducing time at sea searching for fish. This ultimately reduces cost and fuel usage. Results reveal a capacity increase of 65%, by freezing with -50°C compared to -30°C. Even higher capacity increase, up to 100% may be possible when reducing fish block thickness from 100mm to 75mm. More accurate models and experiments must be made to confirm this.
At last, results revealed different average refrigeration capacity for the different evaporation temperatures. This can be used to optimize the mass flow in the system by frequency controlled compressors and optimize compressor sizes.

NOMENCLATURE

A	Area [m^2]	α	Thermal diffusivity [$m^2 \cdot s^{-1}$]
$c_{p,u}$	Unfrozen specific capacity [$J \cdot kg^{-1} K^{-1}$]	$c_{p,f}$	Average frozen specific capacity [$J \cdot kg^{-1} K^{-1}$]
c_a	Apparent heat capacity [$J \cdot kg^{-1} \cdot K^{-1}$]	G	Mass flux [$kg \cdot m^{-2} \cdot s^{-1}$]
k	Thermal conductivity [$W \cdot m^{-1} \cdot K^{-1}$]	ρ	Density [$kg \cdot m^{-3}$]
T_e	Evaporating temperature [K]	W_c	Compressor Work [kW]

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