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Cold thermal energy storage with low-temperature plate freezing of fish on offshore vessels

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ABSTRACT

This paper presents the results of a numerical and experimental study of freezing time for fish in plate freezers, and the potential for improvement by altering fish block thickness and lowering evaporation temperature. Furthermore, the benefits of implementing cold thermal energy storage (CTES) in the system are presented.
The numerical freezing model relied on the apparent heat capacity method and was validated experimentally with an industrial plate freezer.
The freezing model demonstrated good agreement with experiments. Results suggested an increase in fish production capacity by 66 % when using -50 °C systems compared to -30 °C. Another 13 % increase in production capacity is possible if 50-mm thick fish blocks are used instead of 100-mm. Implementing CTES suggested a reduction in freezing time of 3.2 %, by maintaining constant plate temperatures during freezing.
Reduced evaporating temperature, thinner fish blocks and implementation of CTES proved to be beneficial for freezing processes on offshore vessels.
Keywords: CTES, Plate Freezing, R744, Freezing Model, Energy Efficiency.

1. INTRODUCTION

In the last decade, there has been an increase in natural refrigerants used in refrigeration systems (Oña, 2017). On offshore vessels, most systems use either R717 (ammonia) or R744 (CO_2), which can evaporate at temperatures as low as -50 ºC. Though operating at lower temperatures reduces freezing times, it also lowers the system efficiency and requires a higher installed compressor capacity.
This study focuses on plate freezing of fish onboard small fishing vessels, where compact equipment and high production capacity (kg frozen fish per day) is essential. Plate freezers are compact and uses plates, inside which the refrigerant evaporates, in direct contact with the fish to ensure fast freezing. On smaller vessels, whole, ungutted, fish are distributed between the plates and frozen to fish blocks with core temperature of at least -20 °C (Karlsen, 2018). Prediction of freezing times is an important parameter when designing freezing systems.
A new CO_2 system installed on a fishing vessel displayed significantly longer freezing time than expected (Karlsen, 2018). This is likely due to the heat load from the product side is highly varying during the freezing process, while the compressors are dimensioned for the average heat load to avoid over dimensioning the system. This results in insufficient compressor capacity during peak heat load, leading to elevated temperature in the low-pressure receiver and evaporator, causing prolonged freezing times.

1.1 Aim of the study

This study investigates the possibility to optimize the freezing process and equipment on board fishing vessels, by installing a cold thermal energy storage (CTES) system. The CTES system stores energy when the compressor capacity is larger than the heat load, and releases energy when needed, typically in the beginning of the freezing process. The effect on the freezing time and necessary equipment will be investigated.

To reach this aim, the following activities were done:

1) Development of a numerical model for the fish freezing process
2) Development of a numerical model for the temperature development in the low-pressure receiver
3) Model validation by freezing experiments and data from low-pressure receiver from freezing facility
4) Suggestion of CTES system and effects on key performance indicators of freezing system

This paper is based on the first author's master thesis (Verpe, 2018), where more details and results can be found.

2. METHODS AND MATERIALS

2.1 Description of Offshore CO_2 Freezing System

This study is based on an offshore, two-stage, sea-water cooled, flooded evaporator, R744 freezing system. A simplified model description is illustrated in Figure 1. The plate freezer has 24 stations and approximately 1250 kg fish capacity.

A model to estimate the system COP (Coefficient of Performance) was established with refrigerant data fromfree software RnLib. Reasonable assumptions included: isentropic compressor efficiency of 0.7, sea water temperature of 8 ^{o}C, temperature difference of 5 K in condenser and pressure drop in evaporator and condenser estimated by methods from Sardeshpandea et al. (2015), which encompass acceleration loss, friction loss and height difference.

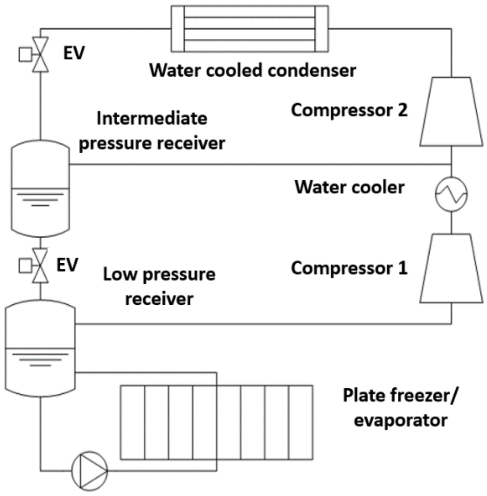


Figure 1: Simplified model description of a two-stage freezing system

2.2 Numerical Description

2.2.1 Freezing model

To be able to predict freezing times for various evaporating temperatures and fish block thicknesses, a numerical freezing model was written in MATLAB. The numerical domain contained the cross section for the fish block while the selected numerical method was to solve the two-dimensional heat diffusion equation, see Eq. 1, by an implicit finite difference scheme, described in Tannehill et al. (1997)

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

Eq. 1 uses apparent heat capacity, c_a, which includes both latent and sensible energy, described by Schwartzberg (1976). Other temperature-dependent thermodynamic properties were calculated using methods described in ASHRAE (2010). Note that the conductivity for porous media was used, to compensate for larger air voids between the packed fish, resulting in more realistic thermal conductivity values than for a homogenous material.

At first, the boundaries of the numerical domain, simulating contact with the freezer plates, were set to constant values (Dirichlet boundary condition). The required heat load, Q, to maintain boundary at constant temperature was calculated, and it was recognized such heat loads are not practically possible due to limited installed compressor capacity. Transient freezer plate temperature was therefore implemented, as described in Section 2.2.2.

2.2.2 Pressure receiver model

Limited compressor capacity results in insufficient refrigerant gas removal from the low-pressure receiver, see Figure 1**.** Increased pressure in the receiver elevates evaporating temperature in the freezer, which in turn, prolongs freezing time. Therefore, a model was made attempting to simulate the evaporating temperature during freezing of fish by assuming constant compressor capacity and calculating heat load from freezing fish. A gas mass balance was set around the receiver, enabling the possibility to calculate specific volume, v, for the tank, see more details in Verpe (2018). The simulated transient liquid temperature in the low-pressure receiver was used as transient boundary conditions in the freezing model, described in Section 2.2.1.

2.2.3 Energy Storage Tank Model and Description

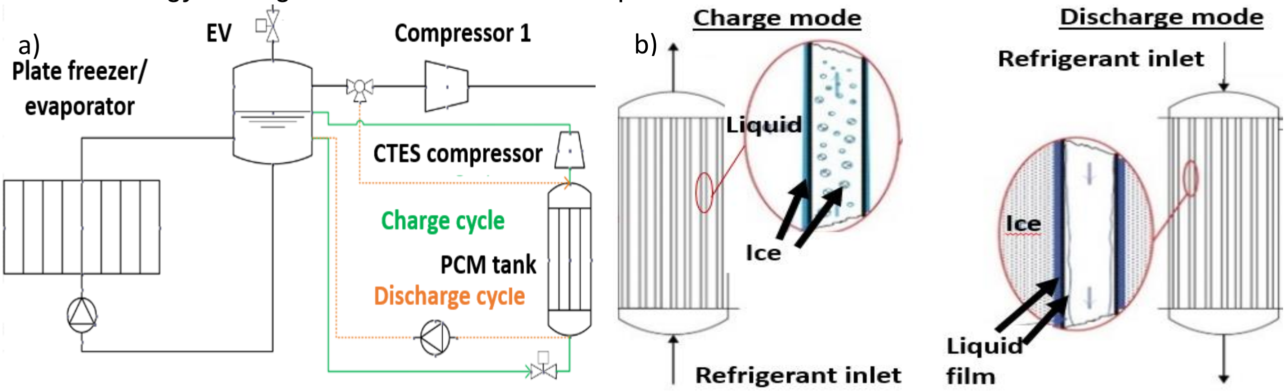


Figure 2: Charge/discharge of the CTES system (a) and a close-up on the PCM tank (b), adapted from Hafner et al. (2011)

An increase in the evaporating temperature, and therefore freezing time, due to insufficient compressor capacity could in theory be eliminated by implementing a CTES system. In this study, a theoretical shell-and-tube heat exchanger containing PCM (Phase Change Material) on the shell side was used to store energy, as suggested by Agyenim et al (2010). Charging, or freezing, of PCM is done when surplus energy is available, in other words when the compressor capacity is larger than the heat load from the product. Energy is released from the PCM tank (discharging) during the first minutes of freezing, when the heat load is largest. This essentially helps the compressor to remove gas, by condensing refrigerant in the storage tank and melting the PCM, see Figure 2.

The PCM was also chosen to be CO_2 due to matching phase changing temperatures and relatively high thermal conductivity, which is a key factor to improve the heat transfer rates. To store energy around -50 ^oC, the charging temperature of refrigerant in the tubes must be below the triple point, indicating solid dry ice particles to be formed. In charge mode, the liquid refrigerant from the receiver is expanded, generating solid ice to be sublimated in the tubes, while PCM is freezing on the shell side. During discharge, refrigerant gas is removed from the receiver, helping the compressor to maintain low pressure and temperature. The refrigerant gas is condensed when in contact with the colder PCM, which is melted and releases the stored energy. Dimensioning of equipment is fundamental, therefore PCM melting and freezing was modelled, based on the two-dimensional illustration in Figure 3.

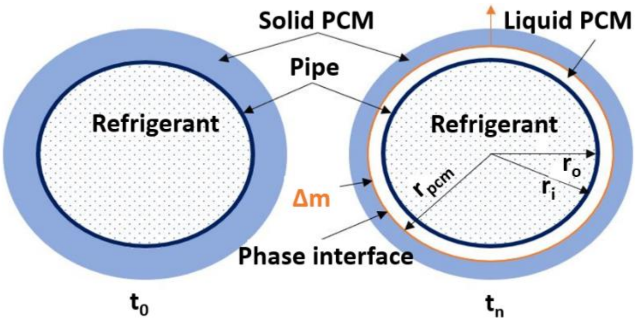


Figure 3: Close-up on pipe in PCM tank during discharge

Table 1: Suggested pressure and temperature levels in CTES tank

	Tube side		Shell side
	Charge	Discharge	PCM
Temperature	-65 °C	-49 °C	-57 °C
Pressure	3 bar	7 bar	8 bar

Assuming heat is transferred from phase changing refrigerant, through metal pipes, liquid PCM and to the interface of phase changing PCM, an expression for the overall heat transfer coefficient is described by Eq. 4 **(**Bergman et al. 2013).

$$UA = \frac{2\pi L}{\frac{1}{r_i h_{conv}} + \frac{ln(r_o/r_i)}{k_{pipe}} + \frac{ln(r_{pcm}/r_o)}{k_{pcm}}} \quad (4)$$

Eq. 4 is valid for liquid PCM at rest. However, buoyancy effects lead to heat transport by natural convection, resulting in higher heat transfer in the liquid layer. Nusselt number, *Nu*, which relates heat transfer in convective fluids to pure conduction in fluids at rest, is defined in Eq. 5.

$$Nu = \frac{h}{k} d \quad (5)$$

Heat flow to the melting PCM can be expressed by:

$$\dot{Q} = UA \cdot \Delta T = \frac{\Delta m \Delta H}{\Delta t} \quad (6)$$

Inserting Equation 4 into equation 6, by use of Equation 5 and expressing Δm by known quantities, yields:

$$\Delta r_{pcm}^{n+1} = \frac{1}{\frac{1}{r_i h_{condens}} + \frac{ln(r_o/r_i)}{k_{pipe}} + \frac{ln(r_{pcm}^n/r_o)}{k_{pcm} Nu_d}} \cdot \frac{\Delta T \Delta t}{\Delta H \rho_{pcm} r_{pcm}^n} \quad (7)$$

Equation (7) describes and incremental change in the PCM radius, Δr, for time steps n and n+1, and will be used to evaluate the required amount of PCM, and size of the tank. Pressures and temperatures assumed for the PCM tank are described in Table 1. Condensing heat transfer, $h_{condens}$, inside the tubes is hard to predict correctly. Therefore, values between 1700 and 4000 W/(m^2·K), considered to be high and low boundaries by *Zang et al. (2012)*, were implemented for sensitivity analysis. Results concluded that they had a limited impact on PCM development, because the overall heat transfer is dominated by the low conductivity of the PCM. Aluminium pipe diameter was assumed to be 10 mm with 3-mm thick wall. The Nusselt number could not be determined precisely due to the complexity of the dynamics at play. Therefore, “most probable” Nusselt numbers between 1 and 4 were assumed.

2.3 Validation of models

To validate the numerical freezing model, a test material consisting of agar-gel, was selected due to good contact with plates, no internal circulation during freezing, high density of nucleation sites and convenience to monitor temperature. Thermal conductivity of the gel was measured using HotDisk TPS (Transient Plane Source) method. The test material was frozen in an industrial plate freezer, while measuring core temperature with a TC (Temperature Couple) monitor. Evaporating temperature was measured to be -49 ºC during the whole experiment, due to almost empty freezer, giving optimal conditions for a constant plate temperature assumption.

Validation of the pressure receiver model was done by comparing model results with data, gathered from another land-based freezing facility with similar freezing system.

3. RESULT AND DISCUSSION

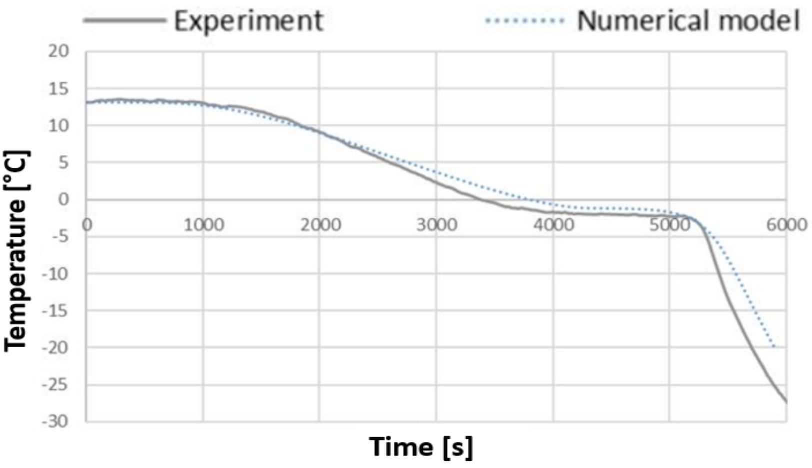
3.1 Validating Models

To validate the models, different strategies were chosen. The numerical freezing model was validated by comparing results with experiments in an industrial plate freezer, carried out by the authors.

3.1.1 Freezing time

In Figure 4, the experimentally measured core temperature is compared to the numerical freezing model. To achieve a core temperature of -20 ºC, there was only a difference of 2.9 % in freezing time.

Results from the numerical model were in good agreement with the experimental data. One can therefore assume reasonable agreement also to frozen fish, provided correct model inputs.



3.1.2 Temperature in receiver

In Figure 5, the theoretical temperature is compared to measured temperatures in the low-pressure receiver. The theoretical model overestimates the elevation of peak temperature, by 4 to 7 K. In addition, the temperature peak seems sharper than in the experiments, which may be explained by different compressor control strategies.

Modelling pressure in a receiver is a complex task since it is influenced by a number of parameters, e.g., control of compressor, heat loss, liquid level, gas and liquid in non-equilibrium and other factors difficult to implement in numerical models. However, numerical results seem to be in acceptable agreement with experimental data.

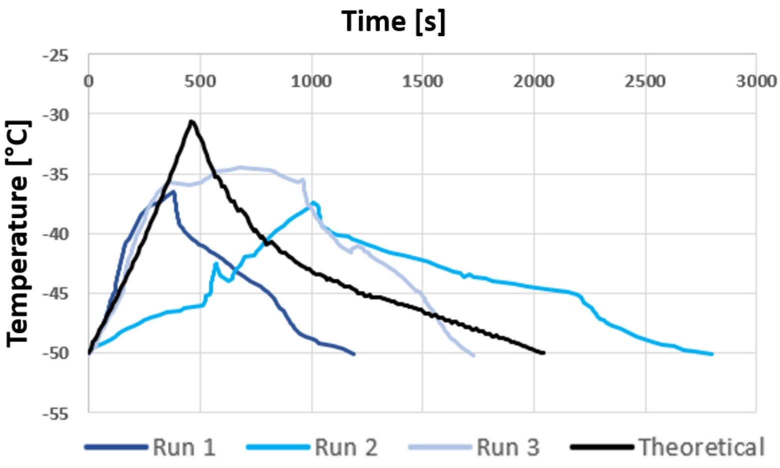


Figure 5: Model prediction and measured data on temperature in the low-pressure receiver during freezing

3.2 Results from numerical models

3.2.1 COP of system

The efficiency of the freezing system illustrated in Figure 1 was estimated by simple steady state models. The COP was calculated to be 1.75 and 3.0 for evaporation temperatures between -50 °C and -30 °C, which is just below 50 % of the theoretical Carnot efficiency for the respective temperature levels.

3.2.2 Freezing time, energy use and production capacity

Based on the calculations of the freezing times and the system COP, the key performance indicators could be determined, as shown in Table 2. A freezer with 25 plates was assumed.

Table 2: KPIs for a R744 plate freezer. Reference values: freezing time: 301 min, energy use: 43.2 kWh/ton, production capacity: 233 kg/h

		Freezing time [min]			Energy use [kWh/ton]			Production capacity [kg/h]		
Evaporating temperature		-50°C	-40°C	-30°C	-50°C	-40°C	-30°C	-50°C	-40°C	-30°C
Block thickness [mm]	50	19 %	23 %	31 %	149 %	108 %	81 %	188 %	161 %	126 %
	75	36 %	45 %	61 %	156 %	122 %	90 %	177 %	146 %	111 %
	100	58 %	73 %	100 %	172 %	133 %	100 %	166 %	134 %	100 %

Energy use is calculated using COPs for the CO_2 system, and a heat loss from the freezer is included. Production capacity calculations are made assuming a reset time (defrost, unloading/ loading of product) of 20 minutes between the freezing cycles.

The increase in the energy use by lower evaporation temperature can be explained by a decreasing COP and increasing heat loss, even though the freezing time is reduced. In addition, the energy use is lower for thinner blocks. This can be explained by the freezing times, which seem to be dependent on thickness squared, while the mass is linearly dependent on thickness. The same argument is also valid when examining production capacity. Therefore, production capacity can be greatly increased, both with reduction in evaporating temperature and plate thickness. In practice, too thin fish blocks will reduce surface contact and increase air void fraction, which in turn prolongs the freezing time.

3.3 Cold thermal energy storage

Figure 6 illustrates when and how much energy can be stored in the PCM tank, for a 1250-kg capacity freezer. Charging of the tank is done when the compressor capacity is larger than the heat load. The same energy can be released in the beginning of the next freezing cycle.

In practice, the pressure in the receiver determines whether the compressor capacity is higher than the heat load since this cannot be directly measured. Lower pressure than designed indicates surplus energy available and should signal the expansion valve to open and start the CTES compressor, see Figure 2.

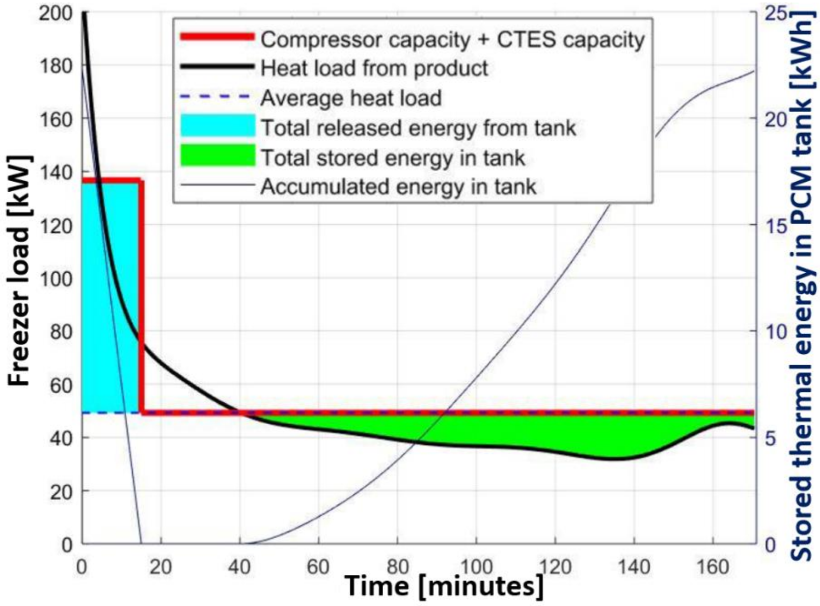


Figure 6: Heat load, compressor capacity and discharged/accumulated thermal energy during freezing

The *effective* compressor capacity, compressor + CTES freezing capacity, matches the heat load far better. This nearly eliminates the temperature elevation in the low-pressure receiver, which is clearly present if the installed compressor capacity alone does not cover the maximum heat load in the beginning of the freezing process.

3.3.1 Ice melting and sizing of the PCM system

To determine the number of tubes and the size of the storage tank, the melting (discharge) of PCM had to be modelled. It is likely that discharging of the tank would be the dimensioning criteria due to the short time and the resulting high heat transfer rates. The development of the melting layer was modelled for different Nusselt numbers, as shown in Figure 7.

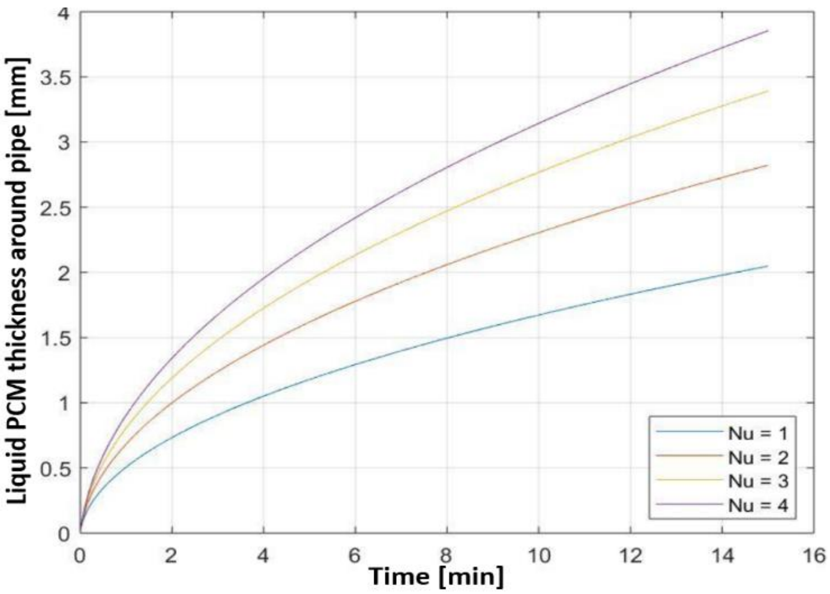


Figure 7: Evolution of the liquid PCM thickness for various Nusselt numbers

The total amount of PCM required for each 1-m long tube can then be calculated by knowing the amount of storable surplus energy, found from Figure 6. One can also determine the number of tubes, average heat transfer rates and total PCM tank volume, see Table 3.

Table 3: Dimensioning properties of CTES

	Nusselt number			
	1	2	3	4
Melted PCM thickness [mm]	2.02	2.74	3.34	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^{2}K]	264	372	452	518

The number of tubes and the total volume of the tank are especially interesting since they influence the capital investment cost and the space requirement, respectively. A 500 L tank occupies valuable space onboard fishing vessels.
Methods of heat transfer enhancements *(Agyenim et al. 2012),* may be needed to achieve the heat transfer rates, calculated in Table 3. The enhancement might also reduce the number of tubes and size, however, investment cost is likely to increase.

3.3.2 Effect on freezing system

Now, it is possible to calculate new freezing times when a CTES systems is enforcing constant plate temperature, since higher effective freezing capacity is installed. The potential benefits of installing a CTES system are also investigated.

Table 4: Potential benefits of installing a CTES system

	With CTES	Without CTES	Difference [%]
Freezing time [min]	168	174	-3.17
Specific energy use [kWh/ton]	80.2	74.3	7.94
Production capacity [kg/h]	399	388	2.92

Note the increase in energy use, due to the added CTES compressor. Freezing time is decreased by over 3 %, due to lower average plate temperature during the whole freezing process, which in turn results in 2.9 % production capacity increase. Capacity and freezing time are not directly correlated due to the assumed 20-min reset time, for defrost, unloading and loading of product between each freezer batch.

4. CONCLUSION

The numerical freezing model was validated by freezing a test material with good contact and constant plate temperature. The prediction of the temperature in the low-receiver, and therefore the freezer plate temperature, demonstrated good agreement with the experimental data, though the receiver temperature is erratic and varies from cycle to cycle.
Both decrease in block thickness and lower evaporating temperature, enabled by R744 systems, result in increased production capacity for the freezer, which is more influential than the increased energy consumption. The freezing station is the bottleneck of the production line, making the short-term storage on the boat (often refrigerated sea water, RSW-tanks) to be filled up while there is still fish to be harvested. Vessels with installed R744 systems, having fast freezing rates, can therefore spend less time at sea, which in turn reduces operational costs and fuel consumption.
Implementing a CTES system with a low-temperature R744 system, using CO_2 as PCM and storing energy when the compressor capacity is larger than the product heat load, was estimated to increase the production capacity by 2.9 %. The storage tank practically eliminates the temperature increase in the low-pressure receiver, by condensing the refrigerant gas that the compressor is unable remove. The PCM tank size was determined to be between 600 and 400 litres, depending on the achievable internal heat transfer rates. The tank required between 1400 and 700 tubes to be able to release the stored energy within 15 minutes.

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NOMENCLATURE

ρ	Density [kg/m^3]	$\dot{m}$	Mass flow [kg/s]	r	Radius [m]	Q	Heat [W]
c_a	Apparent heat capacity [kJ/(kg·K)]	υ	Specific volume [m^3/kg]	L	Length [m]	U	Overall heat transfer coefficient [W/(m^2·K)]
T	Temperature [K]	$\dot{V}$	Volume flow [m^3/s]	h	Convective heat transfer [W/(m^2·K)]]	A	Area [m^2]
k	Thermal conductivity [W/(m·K)]	x	Vapour fraction	t	Time [s]	H	Heat of fusion [kJ/kg]

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