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Simulation of a carbon dioxide (R-744) refrigeration system for fishing vessel

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ABSTRACT

The paper describes architecture of a prototype industrial CO_2 trans-critical system for production of refrigerated sea water (RSW) for either fishing vessels or land-based process plants. The refrigeration system was designed to cover cooling demands of RSW (up to 450 kW), air-conditioning (up to 170 kW) and freezing equipment (up to 82 kW at -25 °C). Four system design cases were evaluated: one stage compression, two-stage expansion with auxiliary compressor, three stage expansion with parallel compressors and two stage expansion supported by ejector. The optimum high side pressure level and the effectiveness of internal heat exchanger were investigated to optimize the four system designs with respect to capacity, seawater temperature and COP. The system performance was evaluated for fish chilling on-board , including both chilling and temperature maintenance periods. Analysis shows a 28% difference in energy demand after 42 hours of operation, when applying the ejector supported solution. Considering that electricity onboard fishing vessels is provided by fossil fuel based generators, the decrease in energy demand results in lower total greenhouse gas emissions.

Keywords: Refrigerated sea water, trans-critical CO_2, Ejector

1. INTRODUCTION

Implementation of transcritical CO_2 refrigeration systems is gaining momentum and find various application areas from commercial to industrial systems (Accelerate , 2020). This is because of the detrimental effects of non-natural (synthetic) refrigerants (also called F-gases) on the environment (J.L. Dupont, 2019) (Sovacool, 2021); legislative pressure to phase out F-gases (K. Zolcer Skačanová, 2019); the development of various solutions for increasing performance of CO_2 refrigeration systems (Paride Gullo, 2018); and heat recovery integration which enables higher system coefficient of performance (COP). In addition to these aspects, the thermodynamic and transport properties of CO_2 make it an ideal candidate in most applications (Pearson, 2005) .

One of the promising sectors of application of the transcritical CO_2 system is the fisheries. CO_2 is a non-toxic and non-flammable refrigerant, which are critical required properties particularly for the systems installed in fishing vessels. The legislative measures to lessen GHG emissions by phasing out F-gases are also creating pressure on the fishery sector due to the fact that R-22 (HCFC-22), which depletes the ozone layer and causes the greenhouse effect, is still used in 70% of the refrigeration systems installed in fishing vessels (UNEP, 2016).

In this study, a simulation model was developed for a prototype transcritical CO_2 refrigeration system, which was designed to meet air condition (AC) and refrigerated sea water (RSW) cooling demands on fishing vessels and at land-based process plants. The four different system design were investigated and discussed.

2. METHODS

2.1. System design and cases

The transcritical CO_2 system was equipped with three parallel compressors and it is schematically shown in Figure 1 . Compressor C1 was equipped with a frequency converter, while the other two were controlled by ON/OFF regulation. The compressors were activated based on the requested capacity, which increases systems flexibility regarding energy demand and operational conditions.

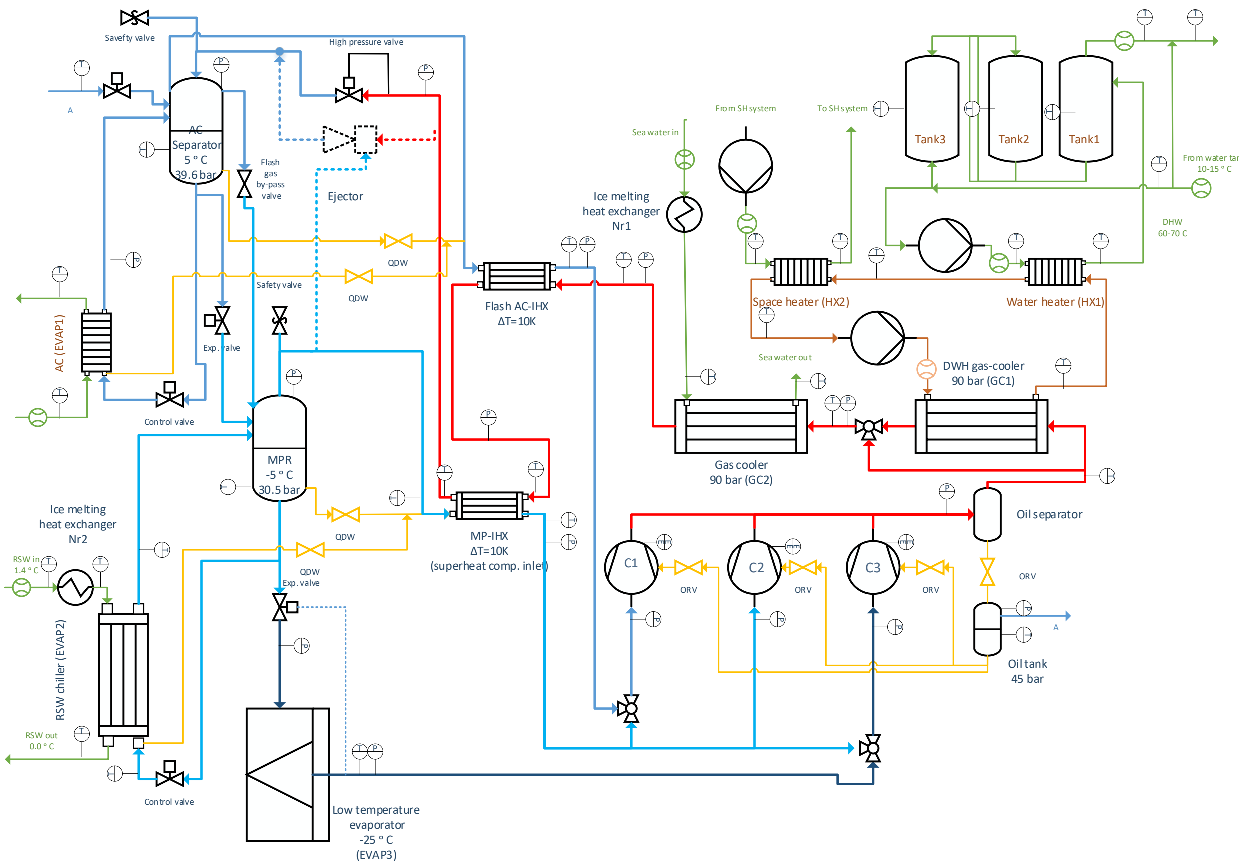


Figure 1: Principal sketch of the transcritical CO_2 refrigeration system

The system utilized the benefits of a transcritical CO_2 loop by energy recovery from the gas coolers (GC1 and GC2). It was designed to supply domestic hot water (DHW) and space heating (SH). The hydronic subsystem provided heat through two heat exchangers in series, HX1 and HX2 in Figure 1, at high and medium temperatures. During operation conditions with negligible DHW and SH demands, the heat was rejected through GC2 while avoiding GC1 by a three-way bypass valve. The CO_2 entered the gas cooler as vapour and was cooled down to by seawater. The system is equipped with two internal heat exchangers, AC-IHX and MP-IHX.

The main function of the CO_2 system was to provide RSW, where the cooling load was the controlling parameter. The set temperature in the RSW tanks was approximately -0.5 °C. To ensure sufficient cooling, CO_2 was set to enter the RSW evaporator (EVAP 2 in Figure 1) at approximately 30.5 bar and -5 °C. EVAP 2 functioned as a gravity-fed flooded evaporator and operated in conjunction with a medium-pressure receiver

(MPR in Figure 1). Integration to meet AC cooling needs was done by including a flooded evaporator (EVAP 1) and a separator (AC separator) between the high-pressure regulating valve and medium pressure receiver. The AC and the low temperature (LT) evaporating temperatures were 5°C and -25°C, respectively. The LT evaporator (EVAP 3) ran on direct expansion conditions, meaning a section of the evaporator was used to superheat the refrigerant before entering the LT compressor (C3). The system was equipped with multi-ejector rack parallel to the high-pressure valve, marked with dashed lines in Figure 1.

The described system design had multiple system configuration that could provide the requested cooling onboard. The configuration included the following options:

- CASE 1: All the three compressors (C1, C2, C3) were available to provide the requested RSW capacity.
- CASE 2: AC and RSW chilling, where compressor C1 was responsible for AC, while rest for RSW chilling.
- CASE 3: The system provided AC, RSW and LT cooling, where C1 was responsible for AC, C2 responsible for RSW and C3 responsible for LT storage. CASE 3 was a local solution of CASE 2.
- CASE 4: The parallel compression ran in conjunction with the ejector rack, providing AC and RSW cooling (CASE 2 + ejector). The ejector removed part of refrigerant vapour out of the RSW separator, hence unloading the RSW compressors.

2.2. Equipment

The CO_2 system had three semi-hermetical compressors, type: HGX46/ 400-4 ML CO2T. Each compressor had six reciprocating cylinders with a suction gas-cooled motor and a swept volume of 400 m^3. One compressor was equipped with a frequency converter with a range of 20-70 Hz.

The outlined gas coolers (GC1 and GC2) are type Alfa Laval AXP112, manufactured by Alfa Laval. The heat exchanger was a brazed plate heat exchanger with external frames made of carbon steel. The capacity of the gas cooler was determined by the number of plates. The internal heat exchangers in the presented system layout were also manufactured by Alfa Laval, type AXP52. The benefits of the AXP heat exchangers are compactness, ease of installation, self-cleaning, low level of service and are gasket free.

Isotherm manufactures RSW and AC evaporator (EVAP1 and EVAP2). Configuration of these heat exchanger was a shell and tube heat designed for usage with CO_2 and seawater. The designed cooling capacity of the RSW evaporator was 450 kW. Danfoss provides the ejector, type “Multi ejector”. Each ejector block had a range of ejectors mounted vertically and in different sizes. Multi ejector was available with 4 to 6 ejectors and matches the capacity demand using different numbers and combinations.

2.3. Simulations

To evaluate the described refrigeration unit, models were built in the simulation software Engineering Equation Solver (EES) and Dymola. EES is a software package used to solve system of nonlinear equations. It is especially well suited to build a vapour compression refrigeration system as it does not require special coding. EES has a complete database for the properties of different refrigerant applied in the refrigeration system, such as R744 (carbon dioxide). Dymola is a modelling and simulation software based on the open Modelica modelling language. The ad-on libraries, *TIL-Media* and *TIL 3.5.0* were used as they provide many common pre-modelled components and refrigerant used in the refrigeration system.

The EES tool was applied to develop steady-state performance analysis, whereas Dymola was used for dynamic load simulations. Creating simulation models accounting for every detail of the RSW system and its configurations is time-consuming and influence reliability and simulation time. Therefore, several simplifications were made. The most important were constant heat transfer coefficient and not accounting for pressure drop in components.

3. RESULTS AND DISCUSSION

3.1. Influence of internal heat exchangers on system performance

The internal heat exchangers (AC-IHX and MP-IHX) had two main functions. The IHX superheats the gas leaving the separator, and at the same time, subcools the liquid leaving the condenser. This reduces the expansion losses and increases the total refrigeration capacity of the system. The influence of IHX in a transcritical CO_2 system has been investigated in several studies, concluding with an increase in system COP by up to 12% (E. Torrella, 2011).

The refrigeration capacity of RSW production was calculated for different seawater temperatures and they are presented in Table 1. The influence of the efficiency of IHXs on refrigeration capacity depends on the system configuration. For CASE 1, the change was minor: at a seawater temperature of 15 °C with 90% effectiveness, the value of refrigeration capacity was 3% less when compared with the value at 10% effectiveness. For CASE 4, the change was 15%. However, one trend was noticed throughout all the four configurations. The RSW refrigeration capacity was larger, utilizing a lower efficient IHX at seawater temperatures below the critical temperature of CO_2 (<30 ℃ in Table 1). With an efficient IHX, the volumetric cooling effect ($\frac{m^3}{kW}$) of IHX will increase, and the start of the compression line will move to an area of greater superheat where the isentropic compression lines become flatter. This results in a higher enthalpy difference during compression, increasing power demand and lower refrigeration capacity at RSW. Based on the presented calculations, it is advised to cut off the internal heat exchangers using two bypass valves at seawater temperatures lower than 30 °C, regardless of the chosen system configuration.

Table 1: Influence of internal heat exchanger efficiency on system configurations at multiple seawater temperatures (the same efficiency for both IHX) on refrigeration capacity at RSW

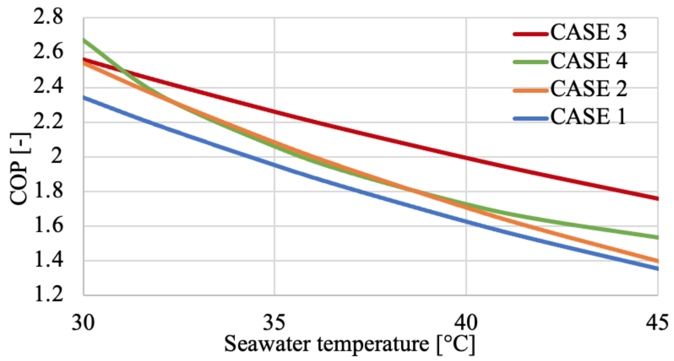
Operation mode	η_{IHX}	$\dot{Q}_{RSW}$ ($T_{SW} = 15\,°C$)	$\dot{Q}_{RSW}$ ($T_{SW} = 20\,°C$)	$\dot{Q}_{RSW}$ ($T_{SW} = 25\,°C$)	$\dot{Q}_{RSW}$ ($T_{SW} = 30\,°C$)	$\dot{Q}_{RSW}$ ($T_{SW} = 35\,°C$)
CASE 1	10 %	425.5 kW	398.7 kW	357.1 kW	299.4 kW	253.1 kW
	54 %	416.6 kW	393.0 kW	353.9 kW	300.7 kW	240.8 kW
	90 %	411.4 kW	387.0kW	352.5 kW	307.9 kW	229.4 kW
CASE 2	10 %	336.5 kW	344.9 kW	333 kW	300.73 kW	248.1 kW
	54 %	308.1 kW	304.3 kW	295.6 kW	281.96 kW	263.4 kW
	90 %	292.2 kW	284.4 kW	276.2 kW	267.5 kW	258.3 kW
CASE 3	10 %	160.9 kW	161.4 kW	159.3 kW	152.9 kW	143.8 kW
	54 %	147.6 kW	144.0 kW	141.8 kW	138.0 kW	135.5 kW
	90 %	140.5 kW	136.13 kW	132.8 kW	127.6 kW	123.8 kW
CASE 4	10 %	426.5 kW	438.8 kW	417.8 kW	363.7 kW	276.3 kW
	54 %	384.0 kW	395.1 kW	382.8 kW	347.0 kW	287.7 kW
	90 %	361.8 kW	360.5 kW	348.9 kW	326.9 kW	294.5 kW

CASE 2 provided AC refrigeration capacity in the range of [0 kW – 170 kW] for seawater temperatures [15°C – 35°C]. CASE 3 provided a stable LT refrigeration capacity of 82 kW.

3.2. Optimization of gas cooler pressure with respect to the seawater temperature

At temperatures above the critical temperature of CO_2, it was important to keep the cycles high pressure at the optimal value to ensure good systems COP. High seawater temperatures result often in low COP when applying CO_2 without special system modifications. An increase in seawater leads to a decrease in enthalpy out of gas cooler, and corresponding decrease in systems refrigeration capacity. At the same time, power input increases, thus resulting often in a substantial decrease of COP.

Based on performed simulations, it was evident that there exists an optimal discharge pressure for each gascooler exit temperature, which gives a maximum COP. The systems COP increased quickly with increasing gas cooler pressure to the optimum value, when the high side pressure was further increased COP slowly decreased. The slight decrease can be advantageous, as the lower gradient makes the system’s performance less sensitive to high-pressure control. Furthermore, based on the optimum COP values, optimum pressure levels were calculated for each of the cases and plotted in Figure 2. The optimum COP values were curve fitted at multiple seawater temperatures in the range of 30°C to 45°C. Figure 2 indicates the difference between CASE 1 and 4, when the unit produces RSW. As a result of three stage expansion for CASE 3, a comparable high efficiency was achieved.



Operation mode	Refrigeration at
CASE 1	RSW
CASE 2	RSW and AC
CASE 3	RSW, AC and LT
CASE 4	RSW

Figure 2: Optimum gas cooler pressure lines for the reviewed system configurations at seawater temperatures 30 °C or higher, IHX efficiency 30%.

According to S.M.Liao, the optimal gas cooler pressure for a transcritical CO_2 system depends on three parameters: the temperature out of the gas cooler, the evaporating temperature and the compressors isentropic efficiencies (S.M Liao, 2000). In all presented cases, the evaporating temperature was constant. Accordingly, the optimal pressure was only affected by the temperature of refrigerant out of the gas cooler, which was assumed to be 5 K higher than the seawater temperature. Curve fitting the optimum gas cooler pressure developed in Figure 2 yielded the correlations presented in Table 2. The correlations can be used in further simulations to operate at the optimum discharge pressure. One should note, at seawater temperatures above 40°C, the gas cooler pressure should be at 110 bar, due to the compressor’s application range. The correlations obtained by S.M.Liao et al. are different when compared to equations presented in Table 2, as they consider the evaporating temperature as well (S.M Liao, 2000).

Table 2: Optimum gas cooler pressure formulas assuming seawater temperature of 30 °C or higher.

Operation mode	Optimum gas cooler pressure formula [bar]
CASE 1	$2.4336 * (T_{SW} + 4) + 1.4614$
CASE 2	$2.209 * (T_{SW} + 4) + 9.1057$
CASE 3	$1.473 * (T_{SW} + 4) + 30.446$
CASE 4	$-0.0738 * T_{SW}^2 + 7.745 * T_{SW} - 81.72$

3.3. Predictions of system performance for fish chilling on board

RSW chilling onboard fishing vessels can be divided into three periods: 1) Prechilling, 2) Chilling and 3) Maintenance. Prechilling is cooling down the seawater in the RSW tank. The chilling period begins when the fish has been loaded in the RSW tank and lasts until a set temperature of seawater is reached in the chilling tank. Factors that influence the length of the chilling period are quantity of fish, amount of seawater and capacity of the refrigeration system. Maintenance chilling lasts until the unloading. During maintenance, the heat loads are primarily due to transmission losses, the length of this period can be up to 39 hours for vessels going far to the sea (E. S. Svendsen, 2021). So, high system COP, especially during the maintenance period, is crucial to ensure the overall efficiency of RSW systems.

Based on the reported chilling load data from a research cruise, which assessed the energy efficiency of onboard RSW system, two simulation scenarios were developed using Dymola to examine predictions of system performance for CASE 1, CASE 2 and CASE 4. The catch size was assumed to be 59 m^3, consisting of mackerel. Results present the chilling and maintenance periods, Figure 3a and 3b.

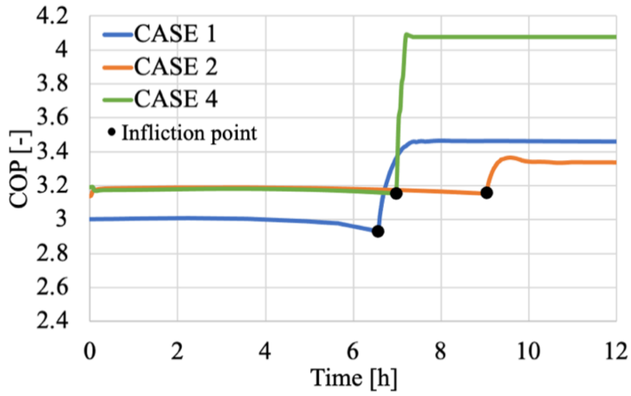


Figure 3a: Seawater temperature at 18 °C.

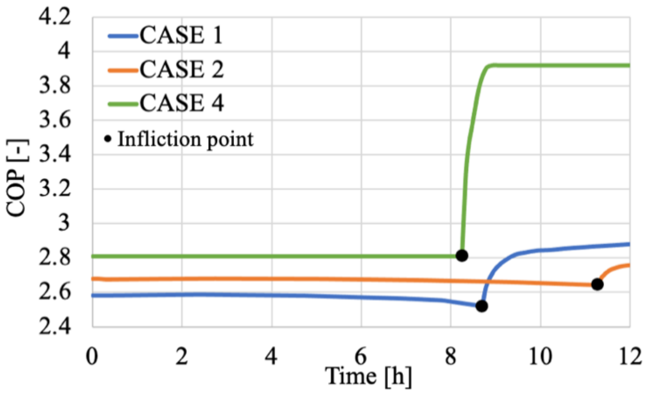


Figure 3b: Seawater temperature at 28 °C.

Figure 3: Systems COP during chilling and maintenance period

The maintenance period started when the set temperature was reached in the RSW tank of seawater and fish mixture (-0.5 °C), visualized as the infliction point in Figure 3. CASE 2 and CASE 4 yielded a higher system COP during the chilling period, both in Figure 3a and 3b. This occurred because CASE 4 and CASE 2 used an auxiliary compressor after the first expansion. However, the chilling time was longer at seawater at 18 °C when compared to CASE 1. This was due to a higher chilling capacity of one stage compression, which occurs at low seawater temperatures. A significant increase in systems COP was observed after the infliction point. The increase of COP resulted from suction pressure increase during the maintenance period, due to the chosen control strategy at part load operation. Figure 3 presents a more considerable increase for CASE 4 because the ejector provided the requested maintenance cooling without using the RSW compressors.

Figure 4 shows the share of power consumption during the chilling and maintenance period. Less energy demand throughout the trip resulted in less fuel consumption, since the electric power onboard fishing vessels was provided by petrol engine. As shown, the length of maintenance period largely effects overall energy demand. As a result of higher system COP during maintenance, shown in Figure 3, CASE 4 had lower hourly energy demand when compared to CASE 2, shown in Figure 4a. The difference is visualised by a smaller gradient for CASE 4 line, in comparison with CASE 2 and CASE 1 line in Figure 4a. Accordingly, after 42 hours of operation time, CASE 4 achieved the lowest share of energy demand. The difference was 9% and 13%, respectively.

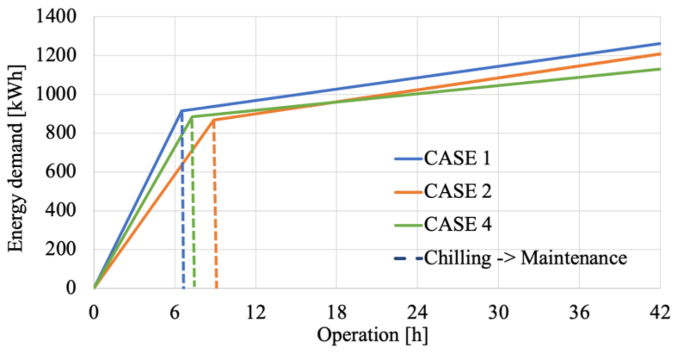


Figure 4a: Seawater temperature at 18 °C.

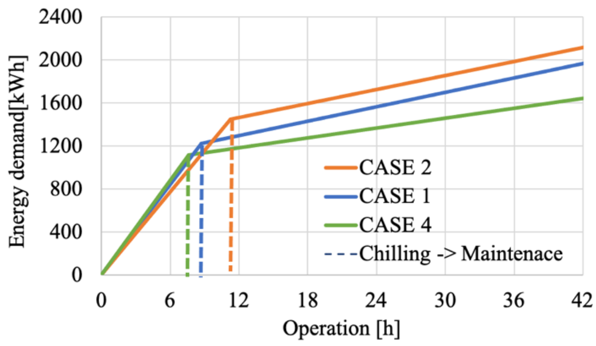


Figure 4b: Seawater temperature at 28 °C.

Figure 4: Energy demand during chilling and maintenance period

Figure 4b presents the overall energy demand, at seawater temperature of 28°C. High seawater temperature decreased the systems cooling effect and increased the transmission losses at RSW tank. CASE 4 resulted in shorter chilling time when compared to CASE 1 and CASE 2, shown in Figure 4b. The difference was 14% and 50%, respectively. A fast chilling can be dominant factor for the high seawater temperatures. Additionally, CASE 4 resulted in less hourly energy demand when compared with CASE 1 and CASE 2. The results were as expected, due to reduction in energy consumption utilizing an ejector for warmer climates. Additionally, CASE 4 delivered the requested cooling for RSW, applying the auxiliary compressor throughout the maintenance period. After 42 hours of operation, the difference in energy demand was 380 kWh comparing CASE 4 with CASE 1 , and 525 kWh comparing CASE 4 with CASE 2. The low performance of CASE 2 when compared to CASE 1 is explained by refrigeration provided at two temperature levels (AC and RSW).

4. CONCLUSIONS

In this work, the performance of an industrial CO_2 transcritical system was analysed in a simulation model. Four cases were included, with different system configurations and considering the influence of heat exchangers and gas cooler pressure. The cases were RSW chilling (CASE 1), AC and RSW chilling (CASE 2), AC, RSW chilling and LT cooling (CASE 3), and RSW chilling with parallel compression ejector supported (CASE 4).

The influence of the efficiency of the internal heat exchanger (IHX) on refrigeration capacity depended on the selected system configuration. For CASE 1, the change was minor: at a low seawater temperature of 15 °C with 90% effectiveness, the value of refrigeration capacity was 3% less when compared with the value at 10% effectiveness. For CASE 4, the change was 15%. However, one trend was noticed throughout all depicted operation modes. The RSW refrigeration capacity was larger when utilizing a lower efficient IHX at temperatures below the critical temperature of CO_2.

According to the simulation predictions, there is an optimal discharge pressure for a maximum COP. The systems COP increased steeply with increasing the gas cooler pressure to the highest value and then slide decrease is observed with the further increasing of pressure. The slide decrease can be beneficial, as the lower gradient represents a plateau and makes the system’s performance less sensitive to the adjustment and control of the high-pressure level. The predictions also showed that the system COP at CASE 4 was the highest, due to the expansion work utilization with ejectors. A lower energy demand was also predicted at CASE 4 throughout the chilling and temperature maintenance period. However, CASE 2 has a longer chilling time and higher energy demand throughout the chosen time interval, as less capacity can be applied for the RSW cooling when also AC is provided with the same unit.

Based on this work, it can be said that the system benefits from the ejectors. However, further work is needed to enhance the developed simulation model. Additionally, a system will be built to conduct experiments considering the predictions.

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NOMENCLATURE

RSW	Refrigerated sea water	DMH	Domestic hot water
AC	Air condition	SH	Space heating
COP	Coefficient of performance	EES	Engineering Equation Solver
T_{SW}	Seawater temperature (°C)	LT	Low temperature
η_{IHX}	Effectiveness	IHX	Internal heat exchangers
$\dot{Q}_{RSW}$	Refrigeration capacity at RSW		

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