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Innovative refrigeration concept for passenger ships - combining CO_2 refrigerant, cold recovery and cold storage

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ABSTRACT

More stringent international regulations on ship's emissions require a shift towards more climate friendly fuels, such as liquefied natural gas (LNG). On LNG-driven ships, the fuel is stored onboard at cryogenic temperature. The fuel must be vaporised before injected into the engine, implying a potential for cold recovery. Today, concepts are commercially available for utilising this surplus cold in conventional AC chiller system. This paper proposes an innovative concept where the LNG cold recovery system is integrated with a provision refrigeration system based on a CO_2 booster unit and a cold thermal storage (CTES) due to the dynamic nature of loads and cold-recovery availability. The CTES is based on phase change materials (PCM) which, together with the choice of CO_2 as refrigerant, ensures a compact system. The results show a potential for significant reduction in power consumption of the refrigeration systems and thereby contributing to reduced GHG emissions.

Keywords: cruise ship, CO_2, R744, refrigeration, cold recovery, cold thermal storage

1. INTRODUCTION

Cruise ships have been described as "energivorous" systems (Barone, Buonomano et al. 2020), i.e., they require enormous amounts of energy for all the needs on board. In addition to propulsion, the hotel facilities require energy in different forms and for different purposes, constituting about 50 % of the ship's total energy usage (IMO 2020). Thus, even when docked or cruising at a low speed, the energy use is significant.

Most of a cruise ship's energy supply, both for propulsion and hotel facilities, is based on fossil fuelled engines. This leads to a high carbon footprint per passenger, in addition to other harmful emissions. The thermal energy demands occur at different temperature levels, including hot water heating, freshwater production, space heating and cooling, and provision refrigeration. These are preferably supplied with waste heat from the engines. Cooling demands are supplied by vapour-compression systems, mostly applying high-GWP refrigerants, and driven with electricity produced onboard.

International Maritime Organization (IMO) has agreed to reduce carbon intensity from shipping with at least 40 % by 2030 and greenhouse gas (GHG) emissions with 50 % by 2050, compared to 2008. The Norwegian Parliament has decided that from 2026, only zero-emission cruise ships will be allowed to operate in fjords featured on UNESCO’s World Heritage List. These regulations, in addition to requirements on low-sulphur fuel and increased public awareness of energy use and emissions, force the shipping sector to adopt new fuels and/or power supply systems. State-of-the-art cruise ships use LNG as main energy carrier, instead of fuel oil. To enable zero-emission power supply in port or during shorter sailing periods, shore power connection and battery package(s) are implemented.

Changing to hybrid power supply and new fuels, such as LNG, also influences the thermal energy systems onboard. Thus, it requires and enables development of innovative concepts for supplying the heating and cooling demands. In addition to being environmental-friendly and energy efficient, important considerations for such concepts include weight and space utilization, cost efficiency and safety aspects. Also, to achieve their full potential the operational profile of the ship, i.e., dynamic behaviour of energy demand and supply, must be considered.

This paper delves into opportunities for waste energy recovery on LNG-driven fuel ships: utilization of "waste cold" and implementation of thermal energy storage using phase change materials (PCM) to decouple availability of heat sink (LNG fuel) and demand (provision freezing and cooling). A case ship was defined based on the operational profile and energy demands for a real ship. A thorough evaluation using dynamic tools are given, the results are presented, including a pre-dimensioning of a PCM-based thermal energy storage (PCM-TES) using pillow-plates as heat exchanger.

2. CASE DESCRIPTION AND COLD RECOVERY CONCEPT

2.1. The case ship

A conventional diesel-electric cruise ship with a capacity of 3800 passengers was chosen as case ship. The case definition is based on data from an 8-day Mediterranean cruise in August 2018, with 35 % of time spent in port. The actual ship operates on fuel oil, but for this study a case with dual-fuel engines was constructed, assuming the ship operates solely on LNG fuel. For power supply in port, two scenarios were considered; power supplied by LNG engine or power supplied with shore power.

The largest part of refrigeration demand on cruise ships sailing in warm climates is for air conditioning (AC) purposes. However, provision freezing (LT) and provision cooling (MT) take a non-negligible part of the electricity use. Based on information about the case ship, the electricity demand for LT cooling was set constant at 100 kW, while for MT cooling it was set to 350 kW in port and 300 kW at sea.

2.2. LNG surplus cold and provision refrigeration demand

Before injected into engines, the LNG fuel must be vaporised at a temperature of around -130 °C and superheated to 25 °C or higher. Usual heat sources for this vaporization are sea water or low-temperature engine cooling water, but LNG has a great potential for cold recovery. Today, concepts are commercially available for utilising this surplus cold in AC chillers. However, since provision refrigeration are the cooling demands with lowest temperature in a cruise ship, it is more thermodynamically efficient to utilize the heat sink with lowest temperature, LNG, to satisfy them.

The mass flow of LNG was calculated based on the total power demand onboard, the electrical efficiency of the engines and the lower heating value (LHV) of LNG. The electrical efficiency was assumed to be the same as for the real ship, which was calculated from the specific fuel oil consumption (SFOC) and the LHV for diesel. The LNG mass flow rate during the case-study period is represented in Figure 1. The green line shows the mass flow rate that would be available with a current scenario, i.e., no limitation on using LNG engines in port. On the other hand, the purple profile depicts a hypothetical, but probable, future scenario where no LNG could be used at port, and the cruise ship is to be connected to shore power. As can be observed in Figure 1, the ship is at port whenever the LNG mass flow rate is below 0.5 kg/s.

Based on the compressor power load (section 2.1) and considering the refrigeration system being a conventional solution for cruise ship installations, a cold room calculator from a well-known manufacturer (INTARCON 2012) was used to estimate the EER (Energy Efficiency Ratio) and thereby the actual cooling demands. The conventional systems were defined as independent R404A cooling and freezing systems condensing against LT engine cooling water, with a return temperature of 35 °C. The calculated EERs for the MT and LT systems were 1.85 and 1.3, respectively, resulting in the cooling and freezing demand profiles as shown in Figure 1 Figure 8.

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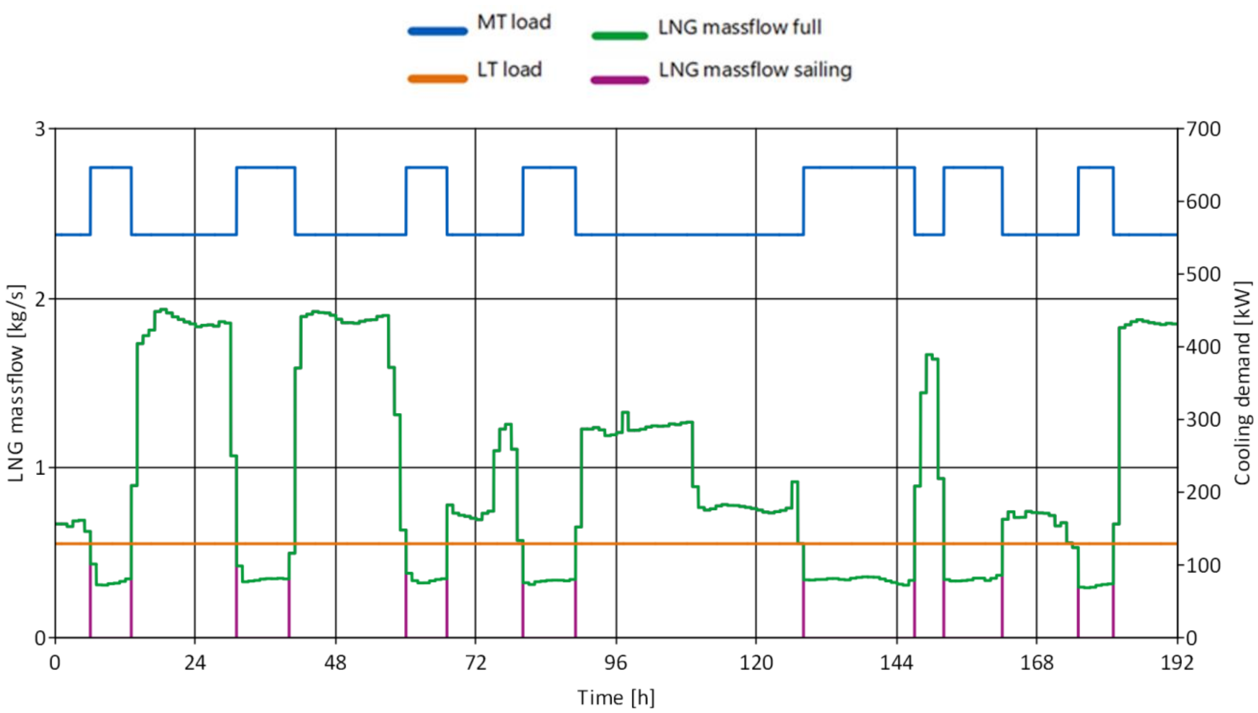

Figure 1: LNG mass flow and cooling (MT) and freezing (LT) demands during the case study period (8 days).

2.3. The innovative concept

The alternative solution for provision refrigeration (MT and LT) suggested in this work (see **Error! Reference source not found.**), is based on three concepts:

Heat exchanger between LNG and CO2 ($HX_{LNG-CO2}$) based on natural circulation of CO_2. Alternatively, a heat exchanger (coil) could be introduced in the receiver to condense CO2 directly there.

Thermal storage unit with CO_2 and PCM (PCM_TES), based on a solution with pillow-plates as heat exchanger (Selvnes, Allouche et al. 2021), which would have evaporation of CO_2 during PCM charging, and CO_2 condensation during discharging.

A CO_2 booster refrigeration unit with fluctuating saturation temperature in the receiver. The system should work with as high saturation temperature in the receiver as possible when compressor-driven, but with lower values when supported by LNG, to allow for PCM charging with the CO_2 loop. On the other hand, evaporators should not be highly affected by the receiver saturation temperature. For this purpose, a pumped loop with back pressure valve for the evaporators is a proper solution to get the requested evaporation temperature even if the receiver saturation temperature is reduced to allow charging the PCM-TES.

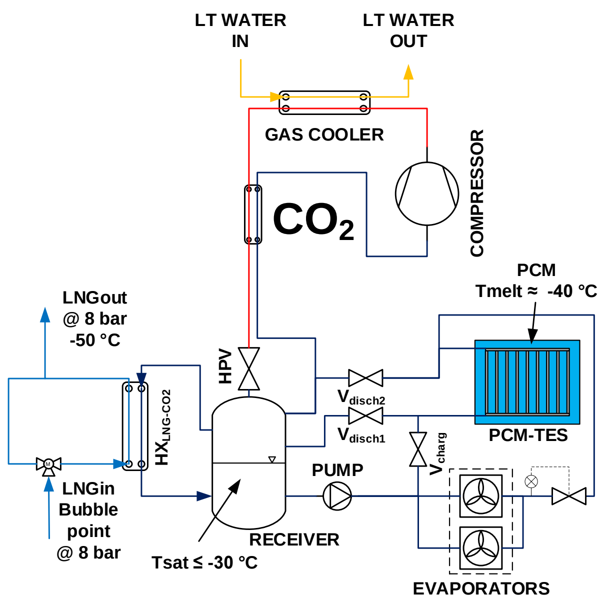

Figure 2: Proposal for integration of LNG cold recovery and utilisation in a CO_2 refrigeration system with a PCM-TES

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3. DYNAMIC SIMULATION MODEL

The solution presented in **Error! Reference source not found.** of utilization of LNG vaporization as heat sink for freezing and cooling processes was adapted to a booster system solution, i.e., integrating both demands in the same unit, and with the possibility of PCM-TES for LT (freezing). Then it was simulated with a dynamic model built in Dymola-Modelica and using TIL 3.11 libraries (https://www.tlk-thermo.com/index.php/en/til-suite) and demand profiles and LNG availability as described in section 2.2. A view of the model can be seen in Figure 3. Some particularities or features of the model are summarised in the following bullet points:

- Fluids from TIL Media 3.11: TILMedia_CO2, TILMedia_MoistAir (air loops at evaporators), TILMedia_Water (LT Water at gas cooler), Refprop_NaturalGasMixture[1] (for LNG calculations).

- Compressors. Simple definitions as one compressor per temperature level, LT and MT compressor, with constant isentropic efficiency at 0.8. Compressor capacity can be adjusted to maintain the saturation temperature in the LT and MT receiver at -30 °C and -4 °C, respectively. Compressors are activated/deactivated by a hysteresis block in the region of the saturation temperature setpoint to improve model stability and to reproduce better the behaviour or control strategy in a real system.

- LNG-CO_2 HXs. Modelled as a heat boundary in each receiver to avoid additional complexity in the dynamic model. The heat boundary input (heat sink capacity) was a function of the LNG mass flow rate at 8 bar(a) and around -127 °C temperature (bubble point) and the receiver saturation temperature measured at each time (assuming 3 K temperature approach between the maximum LNG temperature and the saturation temperature at each receiver). The LNG mass flow rate was mathematically adjusted to control the saturation temperatures in the LT and MT receivers at -45 °C and -15 °C, emulating the bypass function with the three-way valve shown in **Error! Reference source not found.**. These temperatures would be low enough to allow PCM-TES charging, but not as low as to risk the system operation.

- MT and LT evaporators were modelled by a fin-and-tube heat exchanger for each level, with suitable dimensions for the existing loads.

    - Refrigerant side. Pumped loops with back pressure valves to adjust the evaporation temperature at the setpoints of -30 °C or -4 °C while making this parameter as independent as possible of the saturation temperatures at the receiver. The pumps were controlled according to the quality at the outlet of the evaporators.

    - Air side. Each evaporator has a closed loop with a volume (representing the cold or freezing room), a boundary port to simulate the existing cooling or freezing loads according to the profile, and a fan, which speed could be adjusted to reach air temperatures at the LT and MT evaporator inlet equal to -22 °C or 2 °C, respectively.

- Gas cooler. A plate heat exchanger was defined in the model to get a low enough temperature approach between the LT water (inlet conditions at 35 °C) and the CO_2 flow at times with compressor(s) in operation. The LT water mass flow is controlled to maintain 45 °C at the water leaving the gas cooler. The CO_2 high pressure was adjusted by a PI controller with floating setpoint calculated by an optimizing function according to the gas cooler outlet conditions.

- MT and LT receiver control. The migration of charge happens from LT receiver to MT receiver when the LT and MT compressors are in operation. To guarantee that no receiver runs empty of liquid CO_2, a valve connecting them is opened or closed according to a hysteresis block which uses as input the

[1] Typical Natural Gas as defined in REFPROP contains mostly methane (88.8 mass%), but also carbon dioxide (6.5 mass%) or ethane (3.2 mass%), among others.

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LT receiver filling level. This implementation in the dynamic model emulates the control with liquid switches that could be expected in a real system.

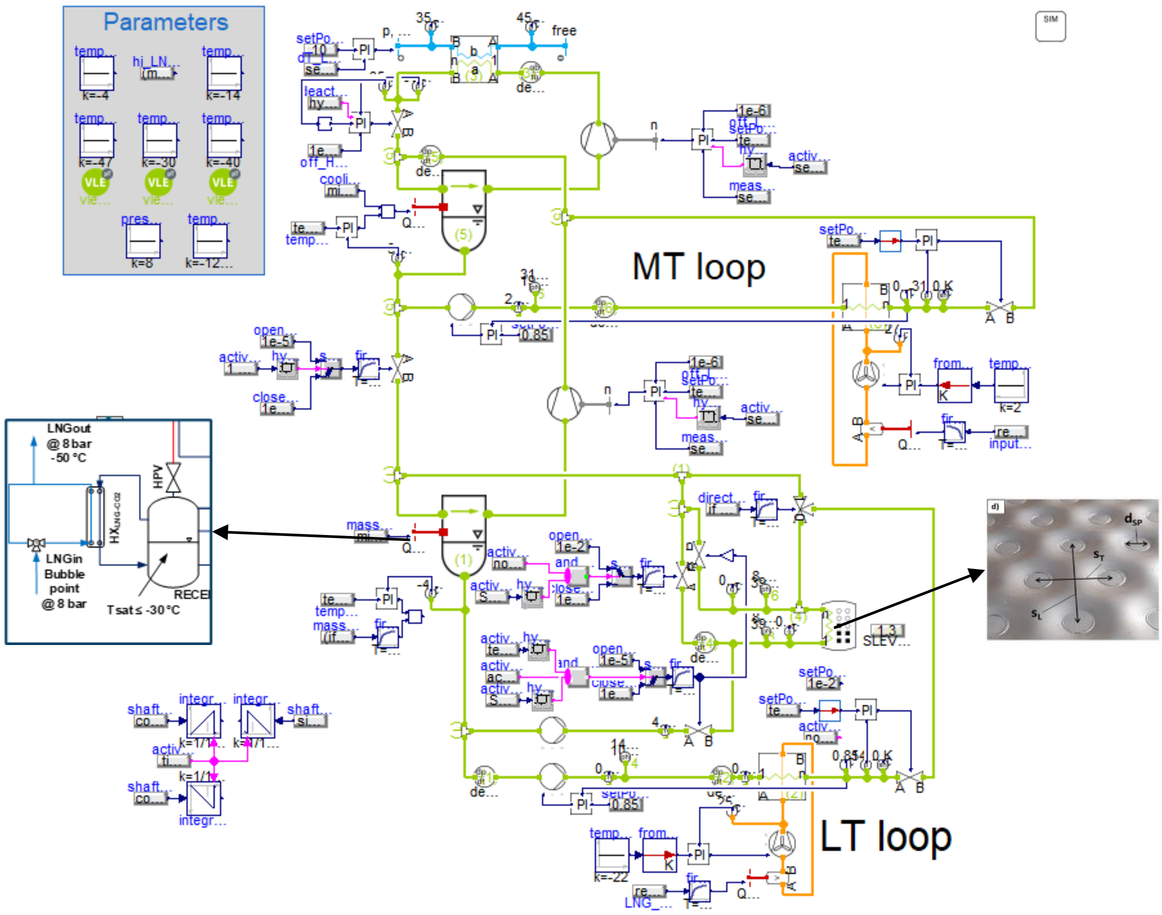

Figure 3: View of the model developed in Dymola-Modelica of the booster solution for utilization of LNG vaporization as heat sink for freezing and cooling loads and integration of PCM-TES.

At this stage of the dynamic modelling, the PCM-TES implementation was restricted to the freezing demands. Information about the definition and modelling of the LT PCM-TES is listed below:

- The PCM chosen was PureTemp @-37 °C, with latent heat 145 kJ/kg. The PCM was specifically modelled extending BaseSLEMedium model from TILMedia, being the most important parameters as in Table 1.

Table 1. Properties of selected PCM PureTemp-37.

	c_p [kJ/kg K]	T_{PCM} [°C]	Density [kg/m^3]	Therm. conduct. [W/m K]
Liquid/melting	1.99	-36	880	0.15
Solid/freezing	1.39	-38	970	0.25

- The PCM-TES storage would be intended to cover as much as possible of the demand when the cruise ship is at harbour. According to the case ship data, the engines are always in operation, which in our adaption to LNG means that there would always be heat sink available to meet the freezing demand at the ship. However, a short-term scenario for cruise ships is that no fossil fuel consumption should happen at port and the cruise ship should be connected to shore power. In case ship for the 8-day period analysed, the average time at port would be 9.71 hours (the maximum stop being equal to 19 hours). A rough calculation of the needed PCM mass and volume to meet 10 hours of 130 kW freezing load indicates that more than 32 t PCM and 36.7 m^3 (considering the liquid density of 880 kg/m^3 and

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disregarding space for heat exchanger), which would be slightly higher than the volume of a 20-ft container. Half this period (5 hours) could be considered as a more realistic case, with 650 kWh, 16 t PCM in a way that heat exchanger and PCM would easily fit in a containerized solution.

- Pillow plate model. A partially validated model of PCM-TES based on pillow plates as heat exchanger between CO_2 and the PCM was utilized (Figure 3) (Försterling, Selvnes et al. 2021). The number of plates, their size and spacing between them were estimated aiming at i) having a discharge rate (heat flow) like the freezing load, and ii) the heat exchanger should fit a 20-ft container.

- Control strategy, activation of charging and discharging. A combination of Boolean signals is responsible of opening the valve used for PCM-TES charging: i) the CO_2 saturation temperature should be low enough compared to the freezing temperature, and ii) the tank should have a level of charge below 60 % before starting another charging cycle. For discharging: i) the controller should check that the tank is not in the middle of a charging process, and ii) the tank should have at least a 50 % of charge before starting a discharging process. These values should be optimized and analyzed further to maximize the utilization of the PCM-TES.

The cases simulated and analysed for this deliverable where:

- <u>No LNG</u>: Reference case, where no LNG is available to meet the cooling and freezing demands, meaning that LT and MT compressors should be in operation.

- <u>LNG Full</u>: The LNG availability follows the green profile in Figure 1, i.e., without limitations in the use of the engines at port.

- <u>LNG Sailing</u>: The LNG availability follows the purple profile in Figure 1, i.e., no engine use at port.

- <u>LNG Sailing + Storage</u>: Same as the previous one, but the PCM-TES for freezing needs is implemented.

4. RESULTS

The dynamic model allows for investigating the effect of various degrees of LNG cold recovery on the operation of the CO_2 booster refrigeration system providing provision cooling (MT) and freezing (LT) onboard the ship. The refrigeration system is operated with refrigerant pumps and back-pressure valves to control the evaporation temperature in the LT and MT evaporators independently of the receiver pressure, as described in section 3. To prove the operation of this configuration, the LT storage room temperature, receiver (saturation) temperature and evaporation temperature in the LT evaporator over the 8-day scenario are presented in Figure 4.

It can be observed that the target air temperature of -22 °C for the LT freezing room is met throughout period. Furthermore, it can be observed that the receiver saturation temperature is reduced to the allowed lower limit of -45 °C when there is available LNG for cold recovery in the system. The refrigerant pump and back-pressure valve operates to keep the evaporation temperature nearly constant at -30 °C, ensuring stable conditions in the freezing rooms.

The overall aim of LNG cold recovery into the refrigeration system is to unload the MT and LT compressors to ultimately reduce the electric power consumption of the system. The LT compressor power consumption for the four case studies is presented in Figure 5. The purple line, representing the scenario with no LNG cold recovery, shows that the power consumption for the scenario where the LT compressors must serve the entire LT load is around 22 kW. For the LNG full scenario, LNG cold recovery is available also during the period when the ship is at port. It is observed that for this scenario the entire LT freezing load can be covered by recovering the LNG cold into the refrigeration system. For the case where the ship engines are switched off during the stay at port (green line) the LNG cold recovery can cover the LT load when the ship is sailing, while the LT compressors are activated during the stay at port to meet the demand. The red line demonstrates the effect of introducing the PCM-TES on the LT power consumption into the refrigeration system. During port

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stay, the PCM-TES covers some of the LT load to partly unload the LT compressors before these are activated to cover the remaining load. It is observed that the LT compressor power in the "LNG sailing+storage case" (red line) is gradually increasing during the port stay, indicating a shift of load coverage from the PCM-TES to the LT compressors during the discharging cycle of the storage.

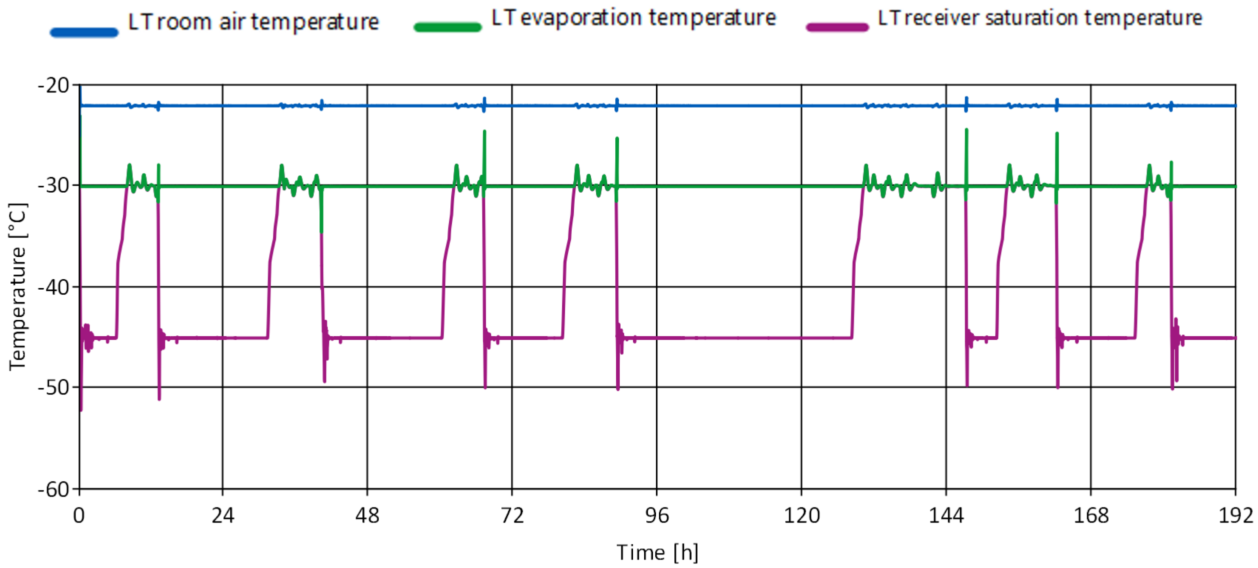

Figure 4: Operation of the LT evaporator with pump and back-pressure valve.

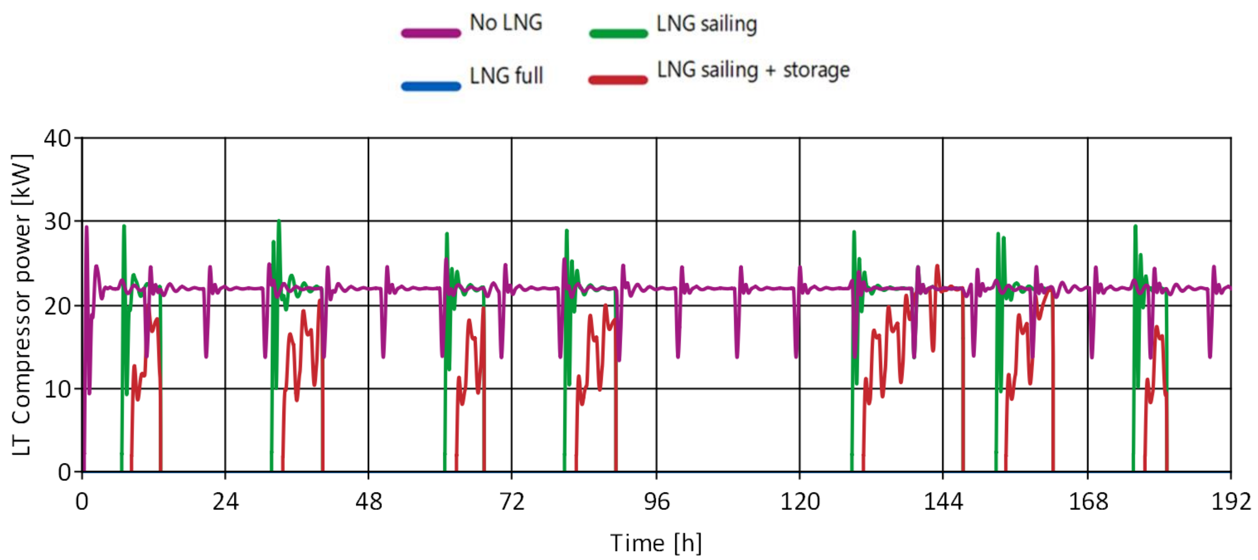

Figure 5: LT compressor power consumption for the four case studies.

The comparison of the MT compressor power consumption between the four case studies over the 8-day period is presented in Figure 6. For the "No LNG" scenario, the MT compressor power consumption follows the MT cooling load pattern with a power consumption ranging from 320 kW to 380 kW (purple line, neglecting fluctuations). For all three scenarios involving LNG cold recovery, the MT load is mostly covered when the ship is sailing and LNG is available, particularly for the period 39-45.000 seconds. Observing the LNG mass flow profile in **Error! Reference source not found.** for this period shows a combination of lower LNG consumption in the engines and higher MT cooling load, which makes it necessary to activate the MT compressors. Furthermore, the MT compressor power consumption is reduced during the stay at port for the "LNG full" scenario compared to the "LNG sailing" and "LNG sailing + storage" scenarios due to available LNG cold recovery. The power consumption to the MT compressors is very similar for the "LNG sailing" and

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"LNG sailing + storage" scenarios during the stay at port, proving that there is no negative effect for the MT compressors when PCM-TES is integrated at LT level.

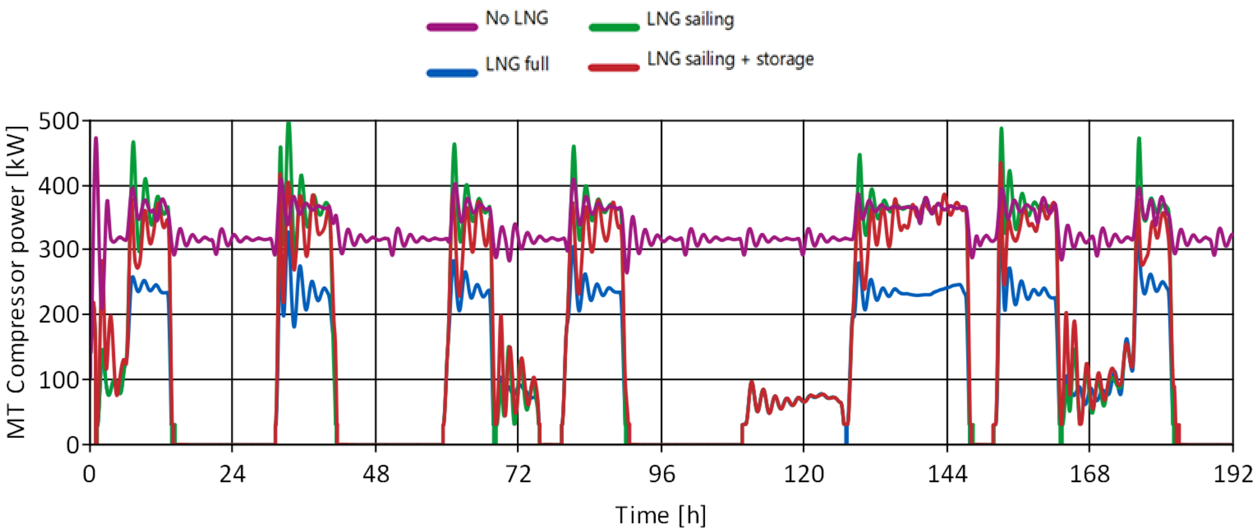

Figure 6: MT compressor power consumption for the four case studies.

To further understand the combined effort of the LT compressor and PCM-TES to cover the LT load in the storage scenario, the heat rate from/to the PCM-TES and the operating period of the LT compressors are presented in Figure 7. The negative heat rate is indicating that the PCM-TES is in charging mode, while a positive heat rate is indicating discharging mode. It is shown that the recovered LNG cold during sailing is sufficient to cover both the LT freezing load and to charge the PCM-TES. When the ship is docked, the storage is initially discharging before the LT compressors are activated to support the PCM-TES. The heat transfer to the storage is decreasing over the discharging cycle mainly due to the increasing PCM liquid fraction. During the initial phase of the discharging period the solid PCM is adjacent to the pillow plates inside the PCM-HEX, providing sufficient heat transfer. As the discharging process continues, the melted PCM forms a liquid layer that the heat needs to be conducted through. Due to the low thermal conductivity of the liquid PCM, the heat transfer rate from the CO_2 to the PCM decreases as the liquid layer increases in thickness. Furthermore, it can be observed that as soon there is available LNG cold recovery again (ship is sailing) the charging process of the PCM-TES is initiated.

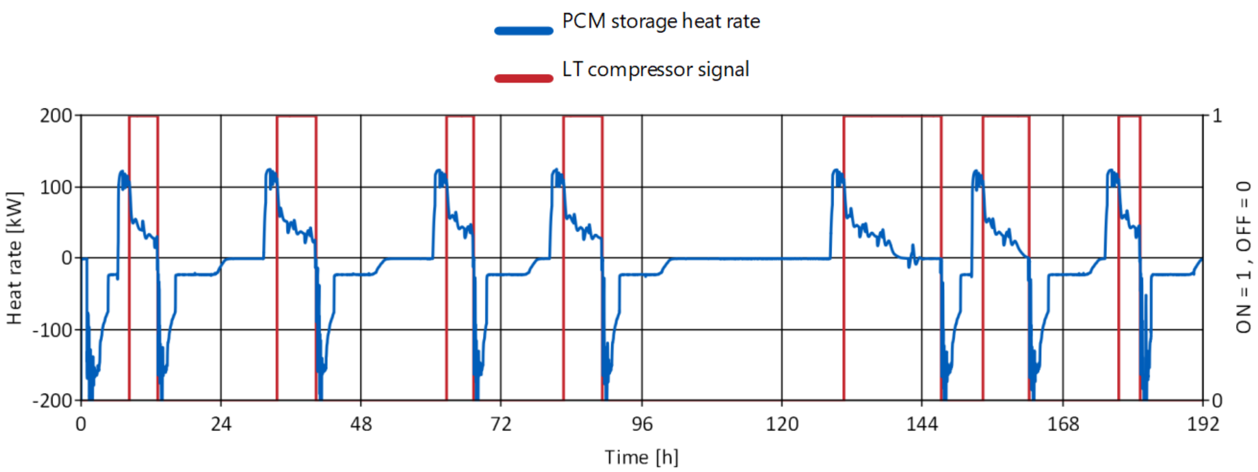

Figure 7: Heat rate to/from the PCM-TES and operating period of LT compressors in the LNG sailing + storage scenario.

The change in the PCM-TES charging level through the 8-day scenario is shown in Figure 8. A charge level of 100 % corresponds the maximum latent heat in the storage, meaning all the PCM is solidified. It can be

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observed that the charging level exceeds 100 % during the charging period in the simulations. This additional stored energy is the contribution from the sensible heat storage of the PCM. This is the energy stored by changing the temperature of the solid PCM from the phase change temperature (around -37 °C) to the final temperature (approaching minimum CO_2 LT temperature of -45 °C) after solidification. Furthermore, it can be observed that the charging level of the PCM-TES is mostly above 25 % after a discharging period. The reason for this is partly the short duration of the period where the storage is needed to cover some of the LT load (stay at port). However, after a long stay at port (after approximately 500 000 s), the remaining capacity is still around 15-20 %. Comparing with the heat rate to the storage in Figure 7 for the same period, it is close to zero. This observation indicates that the distance between the pillow plates inside the PCM-TES unit is too large for the considered PCM and should be optimised to increase the utilisation of the storage capacity. A shorter distance between the plates would require more plates per unit volume, increasing the cost. On the other hand, the heat rate to the storage is expected to increase with more plates, ensuring that a larger fraction of the LT demand can be covered by the storage alone.

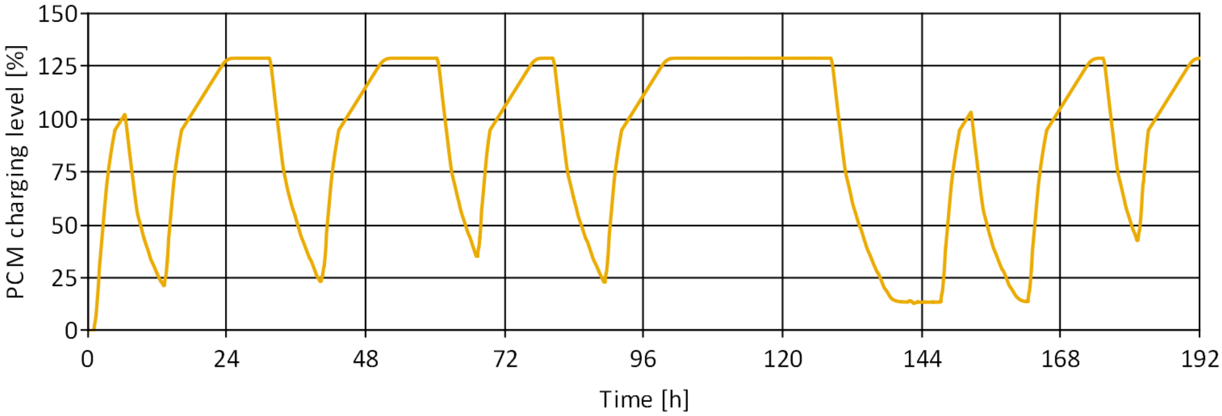

Figure 8: Charging level of the PCM-TES during the LNG sailing + storage scenario.

To compare the electricity consumption between the four different cases, the LT and MT compressor power consumption are integrated over the 8-day period and summarised in Table 2. The reference case (No LNG) confirms that the electric energy requirement for refrigeration purposes onboard these ships are significant, adding up to just over 68 MWh over the 8 days when only the compressor consumption is considered. If this demand profile is extended to the rest of the year, the annual electricity consumption for refrigeration exceeds 3 GWh. By integrating LNG cold recovery into the CO_2 refrigeration system this energy consumption can be reduced substantially. In particular, the "LNG full" scenario reduces the energy consumption for the MT compressors by more than 2/3 and removes the LT compressor energy consumption entirely. In the future scenario where cruise ships need to operate without fossil energy sources during the stay at port, it is worth pointing out that more than half of the MT compressor energy consumption can still be eliminated by LNG cold recovery. For the "LNG sailing" scenario, the LT energy consumption can be reduced by up to 67 % with LNG cold recovery. By integrating a PCM-TES into the LT system, a further reduction to about 80 % can be achieved.

Table 2: Comparison of LT and MT compressor energy consumption for the four cases over the 8 days.

Case	LT compressor [MWh]	Reduction [%]	MT compressor [MWh]	Reduction [%]
Reference: No LNG	4.14	-	63.93	-
LNG full	0	100 %	20.31	68.2 %
LNG sailing	1.38	66.8 %	29.04	54.6 %
LNG sailing + LT storage	0.80	80.7 %	27.70	56.7 %

As a reference and for comparison of the results from the dynamic model, a simple dimensioning of the reference system (no LNG use or availability) was performed with Bitzer's calculation tool (Bitzer 2022). The resulting combined freezing and cooling COP from the Bitzer software is 1.76. On the other hand, from the

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dynamic simulation, the combined cooling and freezing COP calculated for the 8-day period was around 2. The most probable cause of discrepancy is a too optimistic definition of the compressors in the dynamic model, with isentropic efficiencies higher than those defined in Bitzer's calculation tool. The dynamic model should be refined in a later phase to account for more realistic behaviour of compressors.

Table 3 show the dimensions and characteristics of the selected PCM-TES unit used in the simulation model. The proposed dimensions of the PCM-TES that was implemented into the dynamic model was based on previous experience at SINTEF/NTNU with similar storages based on pillow plate heat exchangers. A factor often used both for performance characterisation and dimensioning of PCM-TES storages are the ratio of provided heat rate during the discharge process to the total storage capacity of the storage. In this study, the purpose of the PCM-TES was to cover the LT refrigeration load of 130 kW for about 5 hours. This gives a heat rate to storage capacity ratio of 0.2 (130 kW / 650 kWh = 0.2). This indicates that the storage needs to provide relatively high storage capacity, which directed the authors to choose a distance between the pillow plates (plate pitch) towards the upper limit of what has been tested in the laboratory. The chosen plate pitch of 40 mm and 40 plates in total provided a total heat transfer area of 760 m^2. For reference, providing the equivalent heat transfer area with 20 mm pipes would require a total length of 12 km. The current study has proven the principles of operation for CTES integration at the LT evaporation level, indicating that a similar setup could be implemented at the MT stage.

Table 3: Dimensions and characteristics of the PCM-TES unit in the dynamic model

Parameter	Value
Pillow plate length	5 m
Pillow plate height	2 m
Number of plates	40
Plate pitch	40 mm
Heat transfer area	760 m^2
Weight PCM	12.5 ton
Weight HX	7 ton
Total volume	16 m^3
Latent/total storage capacity	511/660 kWh = 77 %
Heat rate to storage capacity ratio	130 kW / 650 kWh = 0.2

5. CONCLUSIONS

State-of-the art cruise ships and larger passenger boats use LNG as fuel in order to comply with stricter emission requirements in the maritime sector. Cold recovery by vaporization of the LNG stream is promising for refrigeration purposes in the ship. To maximise the utilisation of the LNG cold recovery, a PCM-TES solution is essential to decouple the demand for refrigeration and the availability of LNG cold recovery. A system architecture was proposed in this paper to integrate cold recovery, a CO_2 booster system, and a PCM-TES unit (pillow-plate heat exchanger between CO_2 and PCM). This solution was evaluated with a dynamic system model, based on data from a case ship (8-day cruise) and under different working scenarios.

The results from the model show that the system structure is operating in a satisfying manner, providing the necessary LT freezing and MT cooling demand. The system uses a combination of refrigerant pumps and back-pressure valves to decouple the receiver pressure and the pressure in the LT and MT evaporators, allowing for a decrease in receiver pressure during LNG cold recovery. Furthermore, it was found that implementing LNG cold recovery could reduce the power consumption of the MT compressors up to 68 % for the LNG full scenario and between 54-57 % for the LNG sailing and storage scenario. Moreover, all of the LT load could be covered in the LNG full scenario, while a reduction in LT power consumption of 67 % could be achieved in the LNG sailing scenario. Using the PCM-TES in the system could increase the energy saving with up to 80 % for the LT compressors.

Even though the suggested concept shows a large potential for energy savings, future work is needed before an implementation can be realised. Further research include characterisation of suitable PCM materials and

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optimisation of the geometrical configuration of the PCM-TES. Other important aspect to consider is the economic feasibility and the technical requirement from classification societies.

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