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Waste to Energy Onboard Cruise Ships: A New Paradigm for Sustainable Cruising

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Abstract: The newest cruise ships can guest a constantly increasing number of passengers and concentrate their environmental impact on the limited areas interested by their path. The generated solid waste contributes significantly to this impact; therefore, we propose an innovative solution for recovering embedded energy from that garbage. In more detail, we study the feasibility of an absorption plant able to exploit the residual energy of the flue gas of the ship's incinerator. No payload space shall be sacrificed to install the considered absorption plant. Furthermore, it can be integrated with the existing plants providing for a limited number of heat exchangers. The recovered energy can be used to control the temperature of the refrigerated storerooms; operating simultaneously with, or in place of the existing compression vapors system already installed; it allows a reduction of the CO_2 emissions and of fuel consumption. We show that the proposed approach can be applied to a variety of cruise ships, independently of their tonnage or passenger capacity.

Keywords: onboard incinerator; absorption refrigeration; ammonia; energy rcovery; waste management; cruise ship; sustainability; environmental friendliness

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1. Introduction

The significant growth of the cruise market in the past few years requires particular attention from the point of view of environmental pollution and economic development, due to the high number of passengers and crew members involved [1–3]. Furthermore, to cope with the growing competition within the sector, shipowners request new cruise ships of always growing size and tonnage, which in turn generate increasing emissions of pollutants such as CO_2 and larger amounts of waste. The International Convention for the Prevention of Pollution from Ships (MARPOL 73/78) [4] and local regulations, emanated by the nations visited during the cruise, prescribe the good design and operation practice for reducing the environmental impact of ships. One of the most pressing issues concerns the management of the waste produced on board, which consists of wastewater from different origins, such as wastewater from sinks, showers and laundries (gray water), hazardous waste, solid waste, oily bilge water and ballast water. Various published studies focus on the treatment of different kinds of wastewater onboard ships, due to their impact on the sea environment. Batista et al. [5] analyze different Ballast Water Management Systems, whose main goal is to reduce the *invasion and establishment of non-native species* transferred with the ballast water itself. Several approaches are available to this end: nevertheless, as mentioned in [5], there are still concerns about their *practicality, efficiency, and possible environmental impacts upon discharge*. Jang et al. [6] highlight how the ballast water treatment performance depends on the concentration of suspended solids found in port water. Lu et al. [7] suggest the implementation of onboard produced drinking water tanks, in order to compensate for the weight changes due to fuel consumption and to avoid seawater use. In the past few years, there has been increasing interest in the treatment of the oily wastewater. Abuhasel et al. [8] compare several methods, which already are or can be used onboard, concluding that the best practice will be combining more of them to attain better reliability. Capodici et al. [9] provide a detailed investigation of Membrane Bioreactor Systems, revealing their remarkable performance in the treatment of hydrocarbon-contaminated wastewater, provided it undergoes specific pre-treatment stages such as oil and grease removal. Öz Ç. and Çetin E. [10] concentrate on bilge water and its treatment by Fenton Oxidation followed by Granular Activated Adsorption: best operational conditions are identified revealing a good economical potential for the proposed system. Mustapha et al. [11] propose a novel method, i.e., Fractional Freezing Process, which shows to have good efficiency in oily water separation. Guilbaud et al. [12,13] concentrate on laundry water implementing the recycling of 80% of it onboard ship thanks to nanofiltration.

As shown by Sanches et al. [14], there is limited scientific literature regarding the treatment or the management of the solid/domestic waste produced onboard ships, although the trend is increasing. Strazza et al. [15] explore some good practices for paper consumption reduction, while recently Toneatti et al. [16] explored the whole solid waste management system, highlighting how even a partial energy recovery could provide remarkable economic, environmental and logistic advantages in the management of cruise ships.

According to the current regulations, each ship designed to carry more than 15 people with a gross tonnage exceeding 100 tons, as well as fixed and floating platforms, must have a waste management plan with specific procedures, which involve four waste management phases: collection, treatment (compaction, shredding, pulping, sewage treatment, drying and incineration), storage and unloading [1,3,16,17]. To reduce the quantity of stored waste to be conveyed to the reception port, currently food waste is mainly dumped outboard after a preliminary treatment and respecting the limits prescribed by the MARPOL Annex V. The part of solid waste that can be safely burned, i.e., dry combustible waste (paper, wood, fabrics, rags, etc.), some medical waste and used oils, is incinerated in small onboard plants (less than 2 MW), without performing any energy recovery. This plant system is usually divided into two incineration lines, sized for a daily operating time of slightly less than 12 h, as so to reduce the size of the line equipment and to better use technical compartments, as well as to ensure a certain level of redundancy [16,18]. It is common practice to use the components of one of the lines as spare parts. Even in cruise ships with high attention to the environmental problem and that adopt the current *green* practices, a percentage of the solid waste is incinerated onboard [19]. Table 1 reports the solid waste production onboard a cruise ship with gross tonnage of 141,000 tons, carrying up to 5400 persons [16,18,20,21], which is considered as reference vessel throughout this work.

Table 1. Production of solid waste on board the reference vessel (5400 people on board).

Type of Waste	Total Mass Daily Production [kg/day]	Lower Heating Value [MJ/kg]	Total Recoverable Energy [MJ/day]
Plastic	1188	36	42,768
Paper and cardboard	5360	14.3	76,707
Food waste	10,800	5.7	61,535
Glass	3672	-	-
Aluminum	108	-	-
Total	21,128	-	181,010

The total energy consumption of cruise ships has significant economic and environmental impacts: cruise ships accommodate thousands of passengers and to satisfy all the high-level services they need a large amount of power, both thermal and electric. Consequently, the improvement of the energy efficiency of onboard plants is a key issue. The propulsion accounts for 40–60% of the total energy consumption and its power prediction plays a central role. To this end, Gonsalves et al. [22] proposed advanced Machine Learning

Techniques. Wei et al. [23] focus on the energy management system, investigating the operation of different generators (diesel engines and photovoltaic panels), the strategy of fuel purchase, the controllable and uncontrollable loads and the performance of a battery storage system, in order to minimize total costs, lower GHG emissions and reduce the travel time in the case of an emergency that require the maximum power achievable. The proposed methodology shows different scenarios, which optimize the economic and/or environmental impact. Firouzmakan et al. [24] consider combined heat and power units and electrical energy storage systems, to cope with the challenge of improving the performance in all-electric ships. Their proposed optimal power management results in reduced operating costs. Fang et al. [25] concentrate on heterogeneous energy storage, including thermal tanks and batteries. Their two-stage operating framework reveals good energy efficiency. Other authors study main engine waste heat recovery systems, as Zhang et al. [26]. Their proposal, which consists of the combination of a tCO_2 Rankine power cycle and an ejector refrigeration cycle, shows advantages, both of performance and operational flexibility. Gnes et al. [27] study different configurations of the onboard plants, considering organic Rankine cycle/Stirling units, absorption cycles and water boilers. Though an overall reduction of emissions and fuel consumption is obtained by the proposed approach, the authors point out the different requests between winter and summer cruises. Focusing on the waste heat recovery from the main engine to improve the overall efficiency of the system, Alklaibi and Lior [28] show the good performance of a synergistic system based on ORC, ARS and ACS systems to generate electricity power and provide refrigeration and cooling. Ouyang et al. [29] explore the increasing power output obtainable by supercritical carbon dioxide Brayton cycle, double-effect absorption refrigeration system and Kalina cycle. Ouyang et al. [30] optimize the performance of a dual-pressure organic Rankine cycle system, attaining a remarkable exergy efficiency using cyclopentane as working fluid.

In this work, we explore if and how it could be possible to recover the energy of the flue gases, analyzing in-depth the technical aspects of the exploitation of the energy recoverable from the incinerator plant. We also focus on the compatibility of the proposed system with the spaces and the plants already available onboard. The research uses approaches applied in other fields and combines them for the first time in the maritime sector to find feasible solutions to the problem of waste management and exploitation onboard, based on the evidence that it is not currently possible to recycle all the waste produced onboard. A feasible solution is proposed, which could represent a step forward towards the conversion to *green* cruise ships.

2. Materials and Methods

The proposed plant modification relies on the synergic operation of the incineration plant (IP) and the absorption refrigeration system (ARS), which exploits the thermal energy recovered by the IP to provide refrigerated water for different usages. Consequently, the size of the ASR depends on the recoverable thermal power.

2.1. Incineration System

The potential of an incinerator can be defined either as the maximum waste rate that the plant can burn or as the maximum thermal power that it can produce. In order to provide the best energetic and environmental performance (minimization of dioxin and volatile organic compounds (VOC) emissions), incinerators must be designed and built to operate at different rates, while always maintaining the temperature of the outgoing fumes between 850 °C and 1200 °C. To reach these goals and to fit within the spaces available onboard a cruise ship, the components of a typical incineration plant are piled up vertically (Figure 1).

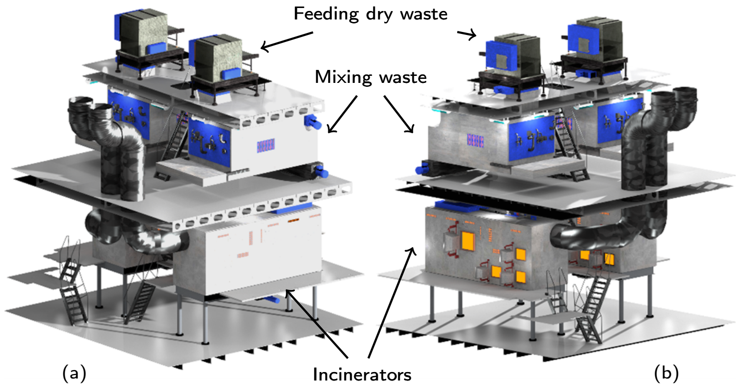

Figure 1. Incineration system, composed of two twin plant. (**a**) front view, (**b**) rear view.

For the reference cruise ship of the present study, whose main data are reported in Table 1, the incineration system occupies 4 ship decks and is approximately 14 m high. Each single incinerator has a footprint of about 30 square meters, including the spaces devoted to standard and extraordinary maintenance and to ancillary service plants. The temperature of the exhaust fumes is controlled by fresh air fed to the chimney. The combustion start-up phase and the control of the process are supported by extra fuel feeding. Another important parameter in the design of incinerators is the transit time of the waste through the combustion chambers, which impacts on their size and must be larger than 2 s, to guarantee the right percentage of oxygen in the flue gases [20].

The incinerator includes three chambers, as reported below:

- a primary combustion chamber, in which a preliminary semi-pyrolytic decomposition is achieved in oxygen deficiency and in a low turbulence atmosphere, to contain the entrainment of dust. A thin layer of solid waste is deposited on a grid, which moves it horizontally and slowly towards the discharge area, where it is finally disposed of as ash with low organic content. The chamber is completed with a burner that allows us to reach the operating temperature (950 °C) during the start-up phases, and to sustain the combustion during ordinary operation. Furthermore, an atomizing nozzle introduces into the chamber the exhausted oil, which must be disposed of;
- a secondary post-combustion chamber, designed to achieve the complete oxidation of the unburnt gas with a residence time of the fumes of at least 2 s; fresh air with high turbulence is admitted into the chamber to ensure optimal mixing. To maintain the temperature above the 850 °C required by the regulations, a burner fed with used fuel oil is installed;
- the third chamber allows us to attain the prescribed minimum transit time of the fumes and to achieve the sedimentation of coarse dust, as a consequence of the sudden impact of the fumes with a solid surface, caused by abrupt path deviation.

The outlet temperature t_f (°C) of the fumes from the combustion chamber can be calculated from the following expression [20]

$$t_f = \frac{\eta_F \cdot H_l}{V_G \cdot C_G} + t_A \tag{1}$$

where:

η_F efficiency of the combustion chamber (taking into account heat losses through isolation, $\eta_F \sim 0.97$);
H_L lower calorific value of combustion waste materials (kJ/kg);
V_G quantity of exhaust gases per 1 kg of waste burned (Nm^3/kg);
C_G specific heat power of the exhaust gas (kJ/Nm^3 °C);
t_A temperature of the air entering the combustion chamber (°C).

Currently, to maintain the temperature of the fumes released in the atmosphere between 200 °C and 350 °C, fresh air is introduced downstream the last chamber.

2.2. Absorption Cycles

Vapor absorption refrigeration systems are vapor-compression cycles, where the refrigerant fluid is absorbed into a liquid mixture, compressed in liquid phase and ultimately released from the mixture by exploiting the heat supplied from low-temperature sources (e.g., hot water or flue gas) rather than using an electrically-driven compressor [31].

Figure 2 shows the simplified scheme of an absorption cycle, highlighting the main heat fluxes exchanged with the surrounding environment. A simplified description of the cycle involves the following stages:

1. In the generator, ammonia separates from a lean (i.e., with reduced concentration in ammonia) water-ammonia solution. The reaction is endothermic: the required heat is transferred from the exhaust gas leaving the incinerator, through an intermediate pressurized-water heat transfer circuit (IPWHTC). Leaving the generator, the evaporated, rich (in ammonia) solution enters a distillation column, where it is further concentrated via an endothermic process, still deriving the necessary heat from the exhaust gas of the incinerator. The concentrated vapor leaving the distillation column is referred to as *ammonia vapor*.
2. The ammonia vapor returns to the state of saturated liquid in the condenser, transferring heat (Q_I) to a flow of seawater through an intermediate pressurized-water heat transfer circuit.;
3. The condensed ammonia is forced through a lamination valve, which lowers the pressure. Part of the ammonia evaporates, resulting in a liquid-vapor mixture of ammonia with a high liquid fraction.
4. In the evaporator, the mixture absorbs heat (Q_L) from the ambient to be refrigerated via an IPWHTC and turns to saturated or superheated vapor.
5. The vapor enters the absorber and is absorbed by the concentrated (in NH_3) water-ammonia solution residing in it. The dissolution process is exothermic: the rejected heat is transferred to the seawater via an IPWHTC.
6. A pump transfers the concentrated solution from the absorber to the generator. In the process, the concentrated solution is pre-heated in a heat exchanger by the warm flow of lean solution being transferred from the generator to the absorber.

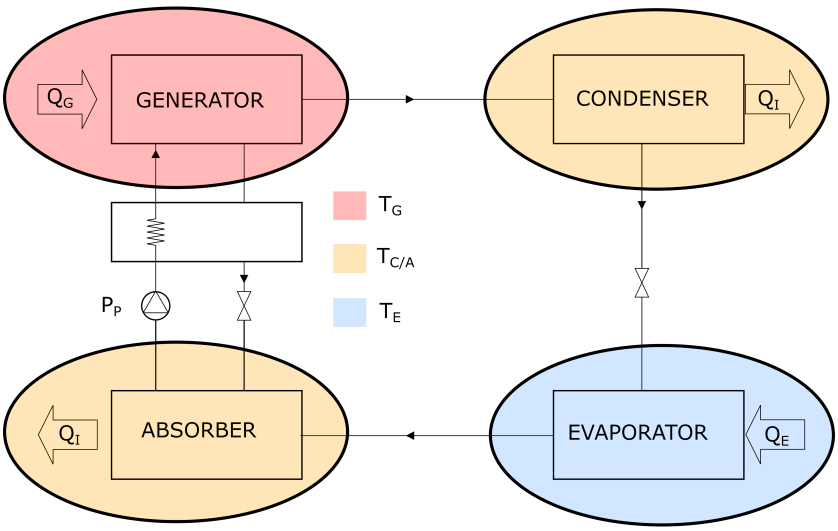

Figure 2. Schematic representation of an absorption cycle.

The electric power required to operate the pump amounts to a minor contribution compared to the other involved energy fluxes. The energy efficiency of the cycle is expressed by the coefficient of performance ϵ_T, defined in Equation (2):

$$\epsilon_T = \frac{Q_E}{Q_G+|P_P|} \backsimeq \frac{Q_E}{QG} \tag{2}$$

Q_E heat absorbed in the evaporator;
Q_G heat supplied to the generator;
P_P mechanical energy supplied to the pump.

ϵ_T turnsout to be generally lower than the corresponding efficiency of cycles operating with vapor compression between the same condensation and evaporation temperatures. The cycle can be considered as a reverse three-temperature cycle, exchanging heat with sources/sinks at mainly three temperature levels:

- the temperature of the condenser and absorber, $T_{C/A}$, which reject heat to the seawater via an IPWHTC;
- the temperature of the generator, T_G, imposed by the energy that can be extracted from the exhaust gases leaving the incinerator at a given temperature;
- the evaporator temperature, T_E, set by the end user according to the needs (cold cell, freezing department, etc.).

The COP decreases with the reduction of the evaporation temperature and with the change from a vapor compression machine to an absorption machine, attaining values of the order of 0.4 for evaporation temperatures around 0 °C [32]. The COP for absorption cycles depends strongly on the attainment of highly concentrated refrigerant vapor at the exit of the distillation column, which is therefore a key component of the system. More advanced absorption cycles attain better performances, optimizing the energy recovery [33].

2.3. Possible Energy Exploitation

The energy exploitation of the fumes produced by the incinerator as a thermal source appears to be a promising solution, with significant overall efficiency. Supporting the existing heating, air conditioning or refrigeration systems requires installing dedicated heat exchangers together with some further minor system modifications. The energy demand for heating of cabins and shared areas is usually satisfied by the primary engine, with the exception of rare cases during stops in port in the winter season. Therefore, the strategy of committing the energy recovery from the incinerators to support the cabin heating system would not induce a reduction of the fuel consumption. On the other hand, recovered thermal energy can contribute to support the air conditioning systems (HVAC) and the refrigeration systems. The refrigeration systems regulate the temperature inside two different types of stocking areas: one chilled at a temperature above 0 °C, the other at sub-zero temperatures, for frozen food. The aforementioned plants have different power requirements: the cooling power required for air conditioning exceeds by an order of magnitude that generated by the incineration system. The cooling power required by the refrigeration plants is comparable with the one generated by the incinerator. These qualitative arguments hold true for arbitrary cruise ships, independently from their size. This is exemplified in Figure 3, in which the power generation/needs of the three categories of power plants are reported for 14 different cruise ships (identified by different code numbers), all having different tonnages and person capacities. It is evident that the proposed absorption plant can support the refrigeration of the food conserving areas, on any of the considered cruise ships, independently from their sizes and person capacity (*vide infra*).

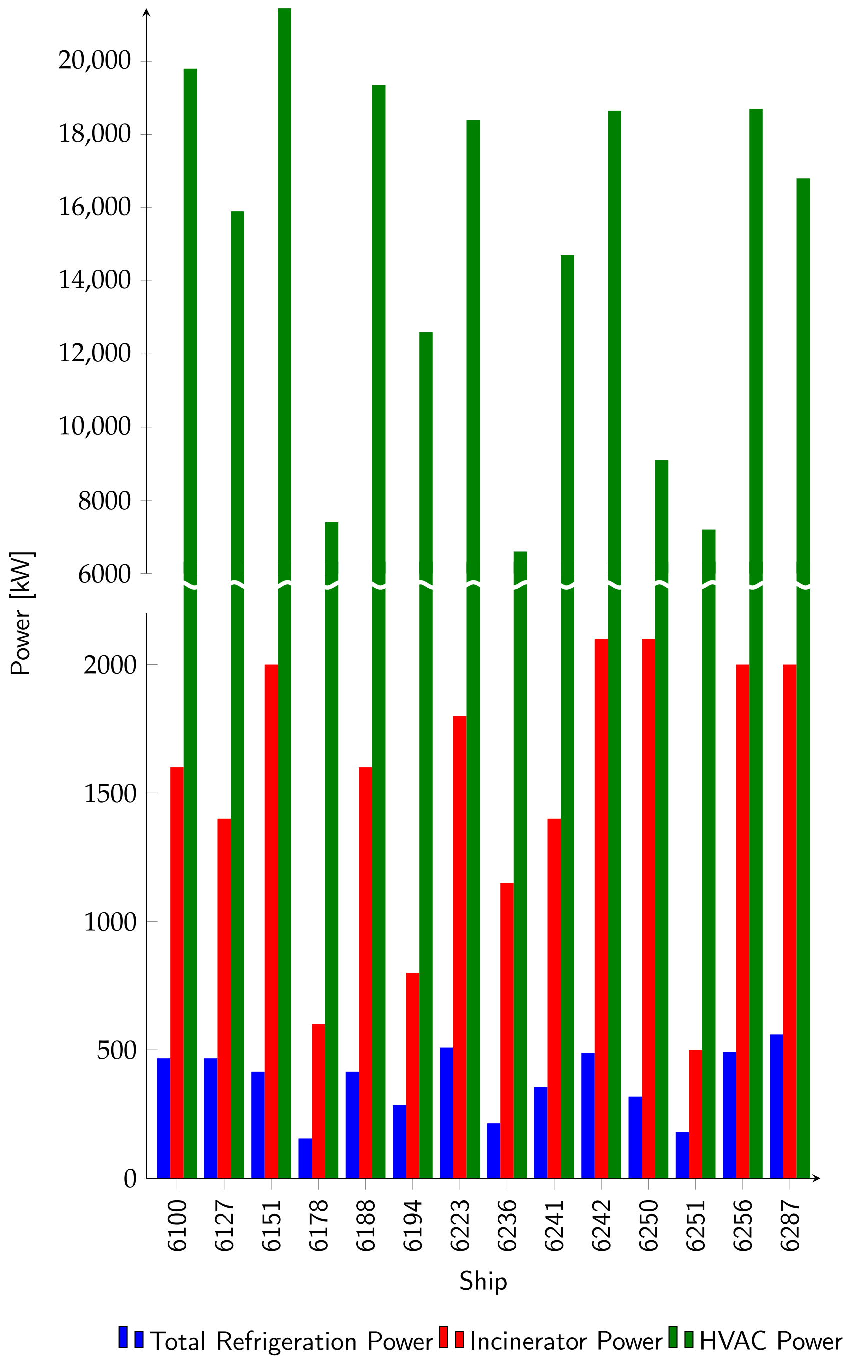

Figure 3. Power of thermal plants onboard different cruise ships, carrying amounts of passengers/crew ranging from 800 to 6592 people. The numeric codes in the abscissa axis are internal reference numbers of Fincantieri. for already built and operating cruise ships.

3. Results

3.1. Correlation between Generated/Absorbed Power, Ship Tonnage and Person Capacity

A correct design of the absorption plant requires the precise knowledge of both the generated and the absorbed power levels. To identify the parameters needed to better describe the different sizing choices, at first we related the involved powers with both the gross tonnage (GRT) of the vessel (Figure 4) and the number of people on board (Total Person Capacity, TPC) (Figure 5), for different currently operating ships.

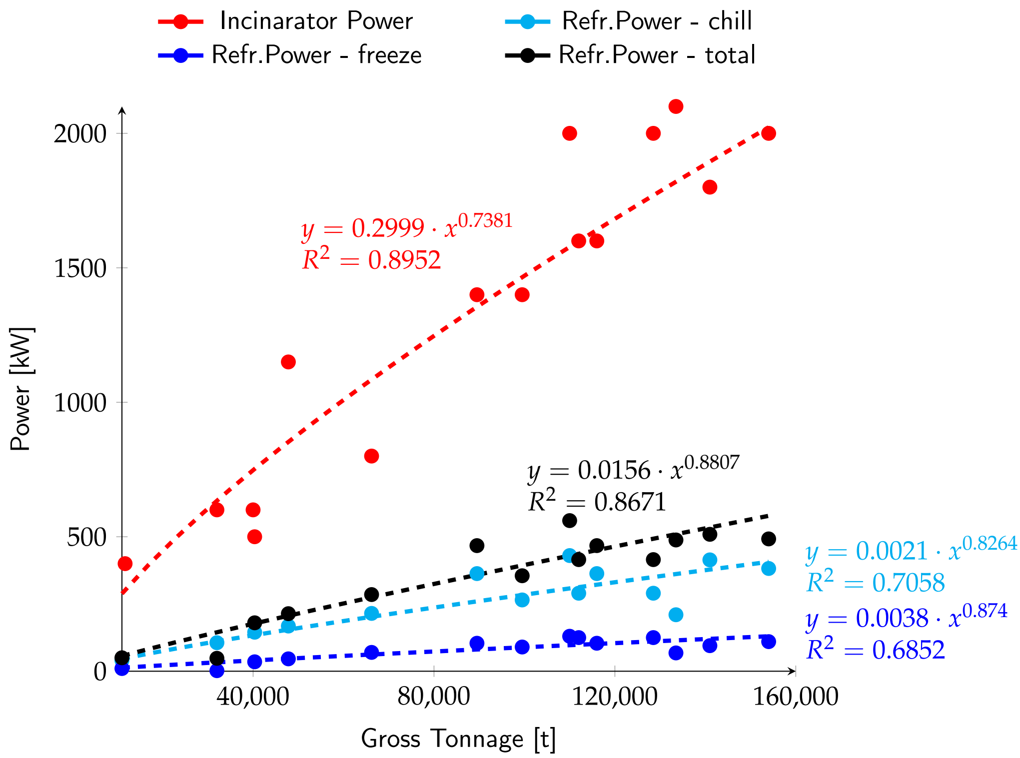

Figure 4. Power of one of the incinerators and the refrigeration plants compared with GRT of cruise ships (DATA: courtesy of Fincantieri jsc).

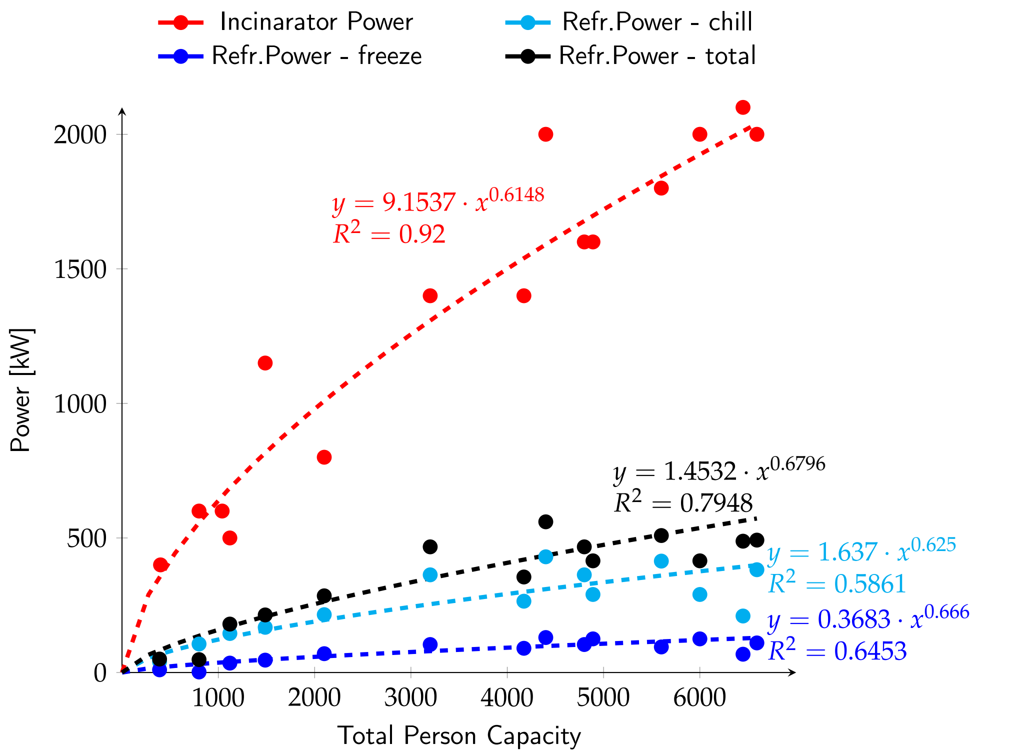

Figure 5. Power of the incinerator and the refrigeration plants compared to TPC (DATA: courtesy of Fincantieri jsc).

It can be readily recognized that there is a clear link between the GRT of a ship and the power of its installed incineration plant (Figure 4). A remarkable correlation also exists between the power of the incinerator and the TPC (Figure 5). The relation between the GRT and the power P of one of the installed incinerators is described by a simple exponential fit (Equation (3)):

$$P = 0.2999 \cdot GRT^{0.7391} \tag{3}$$

with R^2 (coefficient of determination) equal to 0.8732. This relation allows us to size the incinerator for new plants according to the gross tonnage of the new ship.

An analogous analysis conducted about the two independent refrigeration systems (one dedicated to storage at temperatures above zero—CHILL, the other to freeze—FREEZE), revealed a good correlation between the total power of refrigeration $P_{ref,tot}$ and GRT, as for Equation (4):

$$P_{ref,tot} = 0.0156 \cdot GRT^{0.8807} \tag{4}$$

with R^2 equal to 0.8671. The low values of the R^2 parameter for the correlations between the single refrigeration plant and the proposed independent variables can be ascribed to the different types of ships and their hotelier levels. Furthermore, though the volumes of the refrigerated storerooms are expected to increase linearly with the number of passengers and the amount of food, their refrigeration powers follow more complex functions, since they are related to the heat transfer through their surrounding walls.

3.2. General Considerations About the Sizing of the Absorption Plant

The absorption plant is designed for the considered reference ship, characterized by 141,000 GRT and 5600 persons onboard (including passengers and crew). The existing vapor compression refrigeration plants feature the following performance (courtesy of Fincantieri):

- CHILL: cooling power 414 kW, mechanical power 195.5 kW and COP 2.12;
- FREEZE: cooling power 95 kW, mechanical power 106 kW and COP 0.90.

These values suggest that the absorption cycle can support, or even substitute, the currently used ship main engine power to operate the refrigerating systems. The additional mechanical energy required to power the absorption refrigeration cycle is negligible, being essentially the one needed for the pump. The above reported figures highlight that there is no need to push the energy recovery plant towards very high efficiencies, since already exploiting *standard* ARC cycles the required cooling capacities are met and the overall *yield* of the ship system is increased. This simple approach allows us to avoid onerous choices in terms of investments for the upgrade of plants.

The present work focuses on the integration of the energy recovery system with the existing refrigeration system, managing the exploitation of the two energy sources in terms of their alternate or combined use, depending on the energy requirements of the refrigeration system and on the geographic position of the cruise ship with respect to the coastline (i.e., on the possibility to run the incinerator or not). Given the two required temperature levels, it is necessary to foresee different exchanger systems, so to properly feed the cooling and freezing systems. On the other hand, particular attention must be paid to the management of the fluids used for condenser and evaporator (i.e., superheated water, ammonia, hydrofluorocarbons and sea water), since they present peculiar characteristics requiring proper handling and safety measures.

The refrigeration plant coping the needs of the Chill circuit requires a power input higher than that of the Freeze plant. The present investigation focuses on the Chill plant, considering that the available space allows for only one system, even if future studies could involve a more complex system able to provide the needed refrigeration effect for both plants. The proposed configuration optimizes the interaction with the existing systems allowing an instant transition from one to the other, as well as the possibility of a combined operation in the event of particular power requests. The hydraulic independence and the synergistic thermal operation of the vapor-compression and of the absorption refrigeration plants can be attained by interposing either a couple of heat exchangers or a single hydraulic separator [34] between the IPWHTC connected with the evaporators and the IPWHTC connected with the storerooms.

Figure 6 shows the proposed solution for the cooling of the refrigerated storeroom. In this configuration a mixture of water and ethylene glycol (highlighted with blue characters) transfers the thermal power to the seawater via the heat exchanger IPWHTC. The cooling power, on the other hand, is exchanged with the refrigerated storeroom through another IPWHTC operating with water and ethylene glycol, to avoid any possible direct contact between the ammonia and the cooled cells, which would result in safety problems and/or possible food contamination. This part of the heat exchanging system can be realized using aluminum bronze, which has a higher thermal conductivity value with respect to standard stainless steel. For example, the AISI 316 Stainless Steel, widely used in the marine environment thanks to its resistance to corrosion that the presence of Molybdenum imparts to the alloy, has a thermal conductivity λ of about 12.1 W/(mK) at 20 °C, while a standard (nickel) aluminum bronze such as the C63000 has a λ of 39.1 W/(mK) at 20 °C, which means that it exchanges heat more than three times more effectively than the AISI 316. This characteristic allows us to lower substantially the overall weight of the exchanger, while keeping extremely good heat exchanging properties. In turn, this helps to achieve a reduction in overall equipment weight, which is important onboard cruise ships (even though any reduction in plant weight is minimal with respect to the considered overall weight of the vessel). Moreover, the C63000 have a good oxidation resistance at high temperatures, up to about 650 °C, thanks to the formation of compact oxide films. Finally, it is very resistant to stress corrosion cracking, which, together with its aforementioned other properties, makes it particularly well suited for applications in high T, vapor-saturated heat exchangers operating in marine environment. Despite its slightly higher material cost (about 6000–9000 USD/ton vs the 1500–7000 of AISI 316 pipes, depending on the size, finishing and other features), the C63000 compares favorably to AISI 316 when it comes to use the material in a highly demanding environment, such as a heat exchanger to be used at temperatures around 300 °C in a marine environment. It is important to notice that the same C63000, though important for heat exchanging systems, cannot be used in the other heat exchanger due to its known sensitivity to corrosion in ammoniacal environment.

A superheated water circuit transfers the necessary power from the fumes to the water-ammonia solution present in the generator (green characters).

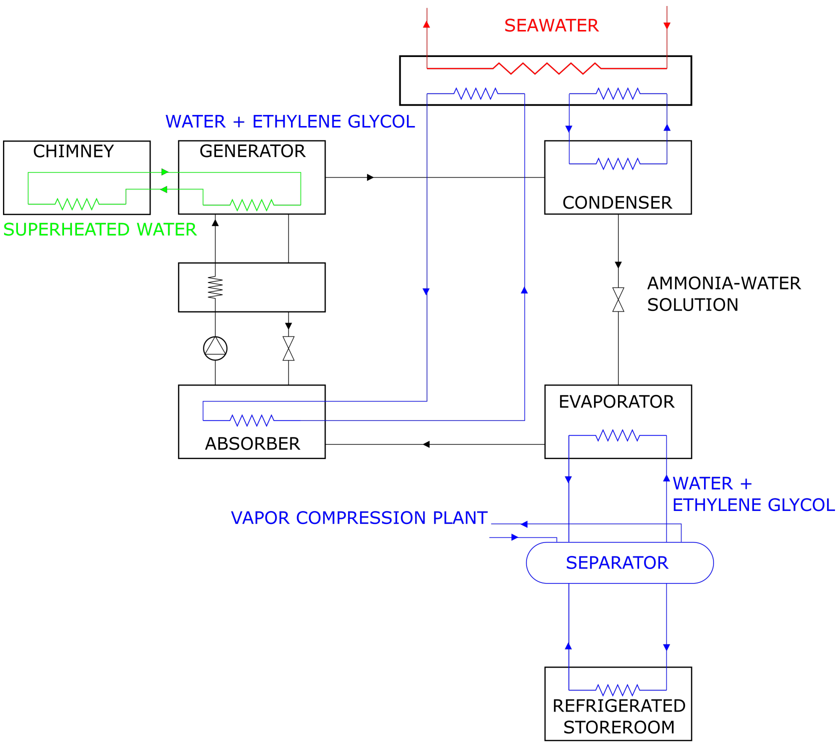

Figure 6. Scheme of the refrigeration plant.

The ARC-Chill plant requires a cooling power of about 414 kW, to be compared with the installed power of 1800 kW of the incinerator. The provisions of Annex VI of the MARPOL convention prescribe that the temperature of the fumes leaving the combustion chamber must be between 850 °C and 1200 °C. To proceed with the approximate sizing of the plants, it is hence necessary to carefully evaluate the energy sources that will be used both to supply heat to the new system and to allow its cooling, as well as the temperatures at which these sources are available. The fumes coming out of the incinerator provide the heating source; they have a high thermal power and are available at high temperatures (at a maximum of about 1200 °C by regulations). The energy exchange between the hot fumes of the incinerator and the water/ammonia mixture inside the generator of the ARC is performed by a circuit operating with superheated water circulating through two heat exchangers. In our simulation, this part of the plant is disregarded as it does not impact on the power availability. The heat extracted from both the condenser and the absorber is rejected to the seawater (even if not directly), whose temperature depends on day/night condition, season, geographical location and atmospheric weather. The maximum temperatures recorded for seawater are around 35 °C and this is, therefore, the value that we will consider in the plant sizing phase [35]. The approximate sizing of the proposed system takes into account the required cooling power and the temperature of the different sources used to heat or cool down the operating fluids. The other relevant parameters are mainly the thermodynamic properties of the different fluids that flow in the different circuits of the plant: the ammonia-water solution, the water-ethylene glycol solution and the seawater. Appendix A outlines the major design stages for the considered ARC. Further details are provided in the Supplementary Materials. The major outcomes of the design process for the CHILL plant are reported in Table 2.

Table 2. Major outcomes of the design of the ARC (CHILL).

Parameter	Value		Description
q_{ev}	414	kW	Cooling power exchanged to the evaporator
q_g	917	kW	Thermal power required from the generator
$q_a + q_{cond}$	1329	kW	Thermal power rejected to the seawater
COP	0.45		Coefficient of performance
$\dot{m}_{w,gl}$	22.7	kg/s	Mass flow rate of the water-ethylene glycol solution
$\dot{m}_{NH_3}$	0.41	kg/s	Mass flow rate of the ammonia
q_a	899	kW	Thermal power exchanged by the absorber
x_p	0.237	kg_{NH_3}/kg_{sol}	Titer of the poor concentration solution
x_r	0.318	kg_{NH_3}/kg_{sol}	Titer of the rich concentration solution

The major operating parameters of the ARC-Chill system derived from the proposed design process are compared with the corresponding values provided by a company specialized in refrigeration systems for land plants, since, to the knowledge of the authors, this technology is not yet implemented onboard (Table 3). In both cases the same design parameters are considered, i.e., refrigeration power and operational temperatures of the different sections to validate the results.

Table 3. Theoretical and real plant values.

Parameter	Value			Description
	Theoretical	Company		
q_{ev}	414	414	kW	Cooling power exchanged to the evaporator
q_g	917	880	kW	Thermal power required from the generator
$q_a + q_{cond}$	1329	1315	kW	Thermal power to be dissipated in the sea
COP	0.45	0.47		Coefficient of performance

The favorable comparison shows that the proposed approximate design approach is reliable and can be used for a technical and economic feasibility analysis of the proposed integrated plant. The thermal power required reaches 917 kW and therefore falls within the exploitable potential of the incinerator.

The overall dimensions of the commercial ARC system mentioned in Table 3 are estimated by the manufacturing company as 6 m × 2.5 m × 6 m. It appears that it would fit in the space that could be made available by removing one the of the two incinerators currently installed onboard the reference ship. Another crucial aspect of the proposed solution consists in its feasible coupling with the existing plant, in order to operate under all possible conditions imposed by the different phases of the navigation. To implement the new plant together with the existing one, the exploitation of a simple but well-functioning device such as the thermal-hydraulic separator allows us to combine different plants with a reliable solution with high hydraulic and thermal performances (Figure 6) [34].

3.3. Expected Energy Savings

A preliminary estimate of the amount of the energy savings induced by the installation of the proposed Chill system starts from assuming the conversion factors that involve the energy chain that leads from the chemical energy contained in the fuel to the electricity generated by the installed alternators. The considered cruise ship is equipped with two engines of 16,800 kW each and two engines of 14,400 kW each, for a total power of 62,400 kW. The mechanical power generated is converted into electrical power with the help of alternators. The consumption declared by the producer for both engines is 178.7 g/kWh.

Assuming the operation of both engines and of the refrigeration plants in nominal conditions, the use of the new absorption system save 196 kW of electric power originally absorbed by the compressor of the Chill plant. The data of the cruise ship, in operating conditions, show a generation of electricity equal to 4.79 kWh per kg of fuel used. Thus, approximately 41 kg of fuel is saved for every hour of operation of the new refrigeration system.

Considering the emission factor from combustion of marine diesel oil of 645 gCO_2/kWh [36], it is possible to evaluate the CO_2 emission savings achievable by the proposed approach. It can work only when the ship is far away from the coast more than 12 miles and its working load depends on several factors such as the climate conditions and how often the doors of the refrigerated rooms are opened. In the absence of pertinent statistical data, we introduced a utilization factor, which represents the ratio between the effective operation time of the ARC-Chill plant and the whole time during which the ARC-Chill system must control the temperature. Table 4 shows the achievable CO_2 emission savings for three different values of this factor (25%, 50% and 75%) as well as the corresponding fuel savings.

Table 4. Savings achievable operating the proposed plant when beyond 12 miles from the coast.

Utilization Factor	Fuel Savings [kg/h of Navigation]	CO_2 Savings [kg/day of Cruise]	CO_2 Savings [tonnes/year (305 Cruise Days)]
25%	10.25	348	106
50%	20.50	695	212
75%	30.75	1043	318

4. Discussion

Currently, waste management onboard modern cruise ships prescribes many treatments for solid garbage, which end with the incineration of allowed ones. This process is used to reduce the amount of waste to be stored on the cruise ships before delivery to the receiving ports. However, it produces ashes and releases flue gas into the atmosphere. To help a more sustainable approach to waste management, it was decided to study the feasibility of the energy recovery from the incineration plant. Two alternative solutions are considered to achieve the desired energy recovery from incineration:

1. conversion of thermal energy into electricity;
2. direct or semi-direct exploitation of thermal energy.

The first solution seems hardly feasible, due to the system complications that it would entail. In more detail, the need for a dedicated system operating with the fumes and the consequent difficulties in interfacing it with the already existing electric generation system would impose costs difficult to recover with the normal ship operation (due to naval regulations the incinerator plant is operational only for few hours per day, during navigation and never along the coast or on the quay). On the other hand, the energy exploitation of the fumes produced by the incinerator without the conversion to electrical energy appears to be of greater overall efficiency.

To proceed with the evaluation, it is not necessary a fine sizing of the ARC cycle, as this could be attained by modeling the flow and heat/mass exchange processes occurring within each component of the plant. Rather, the individual components are dealt with as *black boxes*, whose inputs and outputs are related by assumed efficiency parameters. For instance, a 100% separation efficiency is assumed for the distillation column, which supplies pure ammonia vapor to the condenser.

Comparing the power level of the different plants onboard and the available spaces, the research shows that the alternative supply to the cooling system of the refrigerated storeroom is a viable solution for the available power exploitation. This new approach allows a reduction of the CO_2 emissions and represents a "ready to implement" solution. Indeed, the new plant can be installed instead of one of the two incinerators normally installed for redundancy and can easily interface with the current operating refrigeration plant. This first configuration, which concentrates on one of the refrigeration plants, can be further evolved by exploiting advanced absorption cycles to manage both the conserving temperatures required by CHILL and FREEZE systems. The results also show that there is no need for further optimization of the design of the ARC plant to cope with the goal requested. The fuel savings are limited, due to the low power level of the plants if compared with that of the main engines, and this influences the economic advantage of the solution. Similar solutions realized in onland facilities can count on public incentives (green and white certificate) that could play a significant role also in this sector if they will be introduced.

5. Conclusions

The need for better efficiency in energy exploitation onboard cruise ships pushes the research to involve all the onboard plants in integrated management systems. The greener approach of cruise ships will reduce the amount of unrecyclable onboard waste, but a fraction will still remain and in the meanwhile, the current waste management system onboard cruise ships, which involves as the final phase the incineration of the solid garbage, does not provide any exploitation of the thermal energy embedded in the flue gas. Waste management is not considered by the studies about energy efficiency, except marginally, as for the work of Schumüller et al. [37], who investigate the exploitation of organic waste to biogas production but without the integration with the main energy system.

The proposed absorption plant fed by the energy carried by the flue gases of the incinerator is able to control the temperature of the refrigerated storerooms avoiding the use of electric energy and thus implying CO_2 emissions reduction and fuel savings. This research focuses on incinerator and refrigeration plants while it does not take into account any possible integration with the other onboard energy systems, whose feasibility should be assessed in further studies.

Supplementary Materials: The following supporting information can be downloaded at: https://www.mdpi.com/article/10.3390/jmse10040480/s1, Figure S1: Schematic representation of the ARC system, Figure S2: Dependence of COP on t_{GEN}, Table S1. Condensation and evaporation temperatures in the two considered operation modes.

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Abbreviations

The following abbreviations are used in this manuscript:

ABS	absorber
ARS	absorption refrigeration system
CND	condenser
COP	coefficient of performance
DC	distillation column
DE	direct expansion
EV	evaporator
GEN	generator
GRT	gross tonnage
HEX	heat exchange circuits
HVAC	heating, ventilation and air conditioning
IE	indirect expansion
IP	incineration plant
IPWHTC	intermediate pressurized-water heat transfer circuit
TPC	total person capacity
VOC	volatile organic compounds
IHX	recuperative heat exchanger
TV	throttling valve

Appendix A. Description of the Absorption Refrigeration Cycle (ARC)

Appendix A.1. System Structure

The proposed ARC system is schematically represented in Figure A1: it exploits water (H_2O) and ammonia (NH_3) as the absorbent (high vapor saturation pressure) and refrigerant fluid (low vapor saturation pressure), respectively. In the following, we will refer to a solution rich in NH_3 as *concentrated* solution. The thermodynamic properties of the water-ammonia mixture are evaluated according to the method proposed by Patek et al. [38].

A concentrated solution in state 2 enters the Generator (GEN). The solution residing in the generator is heated by exchanging heat with the exhaust gases produced in the incinerator: as a consequence, the highly volatile solute (ammonia) evaporates, increasing the concentration of NH_3 within the vapor phase of the mixture in the GEN. The concentrated vapor in state 8 leaves the GEN towards the distillation column (DC), where it is further concentrated. The distillation column must reduce the water content in the vapors produced by the generator to avoid that liquid water accumulates in the evaporator, causing a sensible reduction of the COP [39,40]. Furthermore, the vaporization temperature increases at higher concentrations of water in the vapor mixture (*temperature glide*): this, in turn, implies that a lower evaporation pressure must be attained to realize the required cooling effect [40]. It is assumed that the refrigerant vapor leaves the DC with an ammonia concentration of 100%: this limiting condition is not significantly far from concentrations obtained with conventional distillation towers (see, e.g., Clerx and Trezek [41]). White and O'Neill [42] suggest that an ammonia concentration not lower than 99.9% wt is required for a proper ARC operation. The distillation tower and the generator can be integrated in a single component, as is the case for sieve-tray DC's [40], where a reservoir of liquid solution at the base of the column acts as generator. The refrigerant vapor leaves the DC in state 9, flowing towards the condenser (CND), where it is condensed to saturation conditions by releasing heat to a stream of water-glycol solution. The condensed ammonia leaves the condenser in state 10 and flows towards the evaporator (EV) passing through a throttling valve (TV1). In flowing through the TV the pressure and the temperature of the refrigerant are reduced to evaporation conditions (state 11) and part of the liquid evaporates. The liquid-vapor mixture of ammonia enters the evaporator in state 11: flowing through the evaporator the refrigerant fluid exchanges heat from a stream of warmer water-glycol solution, which, in turn, subtracts heat from the storage room, realizing the desired *cooling effect*. The intermediate water-glycol heat exchange circuits HEX1 and HEX2 could be eliminated, in principle, reducing the gap between the condensation, absorption and evaporation temperatures. The refrigerant fluid leaves the evaporator as a saturated vapor (state 12) and enters the absorber (ABS), where it is *absorbed* by the water-ammonia mixture therein. Absorption is an exo-thermal process: the excess heat is transferred to a water-glycol mixture and eventually to seawater through the heat exchanger HEX3. The concentrated, liquid solution leaves the absorber in state 0 and is conveyed to the GEN by a pump. Along the way, the concentrated mixture absorbs heat from the DC flowing through the recuperative heat exchanger IHX2 reaching state 3, then passes through the *recuperative* heat exchanger IHX1 (states 3→2), where it receives heat from the stream 4→5 of lean mixture hailing from the GEN and eventually enters the GEN. Two more streams of mixture interact with the GEN: the flow of highly diluted liquid mixture transferred from the DC to the GEN and the flow of lean liquid mixture leaving the GEN towards IHX1. The latter flow leaves IHX1 in state 5 and successively passes through the throttling valve TV2, where the pressure of the mixture is reduced to the evaporation pressure.

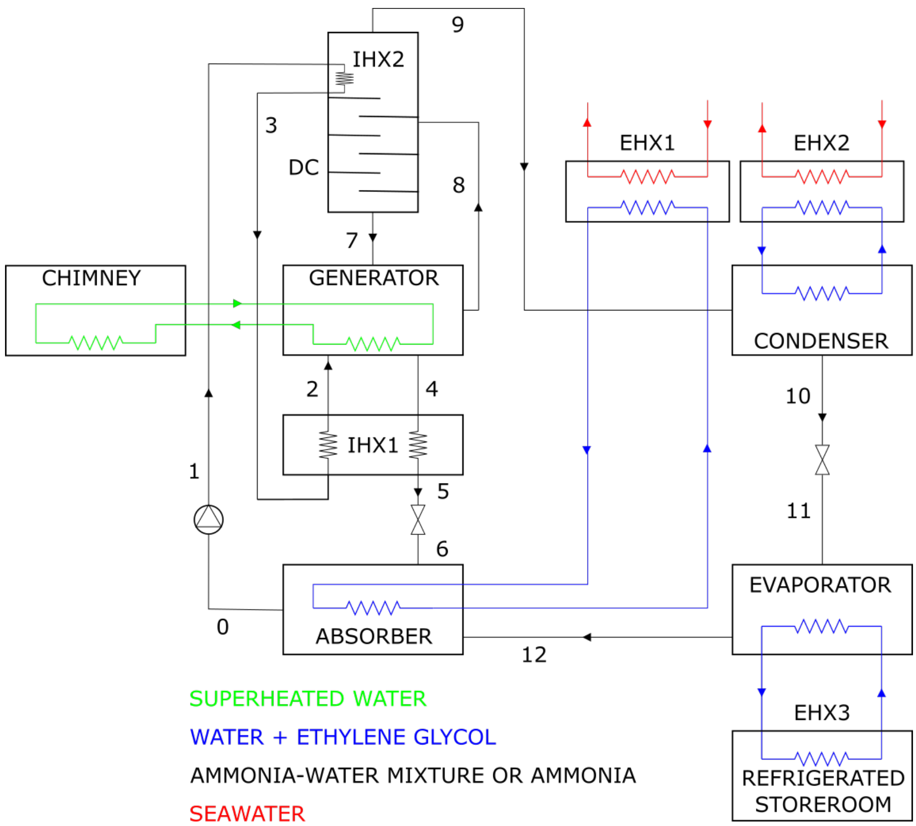

Figure A1. Schematic representation of the ARC system.

Appendix A.2. Assumptions and Design Parameters

The proposed design strategy of the ARC is based on a set of (quite reasonable) assumptions:

- The system runs under steady-state conditions, and changes in the kinetic and potential energies are ignored.

- The flow is adiabatic through the pump PU and iso-enthalpic through the throttling valves TV1 and TV2.
- The flow through the connecting pipes and through the components of the ARC is free of frictional losses and of the ensuing pressure drops.
- The refrigerants at the EV and CND outlets are at the saturated vapor state and saturated liquid state, respectively. The vapor at the DC outlet contains only ammonia.
- The liquid stream of lean mixture transferred from GEN to ABS enters the ABS in saturated conditions at the evaporation pressure (which is present also in the ABS). Notice that the concentration of NH_3 in the aforementioned stream remains constant as it is transferred (in liquid state) from GEN to ABS.
- Only the liquid phase of the water-ammonia mixture is withdrawn from the ABS and transferred to the GEN.
- Only the vapor phase of the water-ammonia mixture is withdrawn from the GEN and transferred to the DC.
- The stream of condensed water-ammonia mixture leaving the DC is saturated at the (p, T) conditions present in the GEN.
- The inlet temperature of the water-glycol mixture in EHX1 is $t_{s.r.} - 8\,\mathrm{K}$ and $t_{evap} + 7\,\mathrm{K}$, while its temperature is increased by 5 K flowing through EHX1.
- The temperature of seawater, used as a refrigerant in the EHX2 and EHX3 heat exchangers, is raised by 5 K when flowing through this components. The outlet temperature of the water-glycol solution from EHX2 and EHX3 is 5 k above the maximum seawater temperature and its temperature drop is 5 k.
- The hydraulic efficiency of the pump is 0.7.

The following design parameters are assigned:

- The desired refrigerating power $\dot{q}^{+}_{EV}$, transferred from the cold room to the refrigerant fluid in the evaporator.
- The temperature of the mixture within the generator: t_{GEN}.
- The temperature of the storage room, the temperature difference between the storage room and the water-glycol mixture in the intermediate heat exchange circuit, the temperature difference between the evaporating refrigerant fluid and the water-glycol mixture in the intermediate heat exchange circuit. Using these input parameters, the evaporation temperature t_{EV} can be derived.
- The seawater temperature, the admissible temperature increase for seawater, the temperature differences between the seawater and the water-glycol mixture in the intermediate heat exchange circuits. Using these input parameters, the condensation temperature t_{COND} and the temperature within the absorber t_{ABS} can be calculated.

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