Marine refrigeration plants for passenger ships: Low-GWP refrigerants and strategies to reduce environmental impact

Lorenzo Pigani [a,*], *Marco Boscolo* [a], *Nicola Pagan* [b]

[a] *Department of Engineering and Architecture, University of Trieste, Via A. Valerio 10, 34127 Trieste, Italy*
[b] *H.V.A.C. System and Ventilation Design, Fincantieri S.p.A., Passeggio Sant'Andrea 6/A, 34123 Trieste, Italy*

A R T I C L E I N F O

Accepted 16 January 2016

Keywords:
Marine refrigeration
Low GWP
Energy efficiency
TEWI

A B S T R A C T

This paper is devoted to the evaluation of the use of low-GWP refrigerants in marine provision plants for cruise ships. We present the state of the art of current refrigeration plants, and we identify ammonia (NH_3), carbon dioxide (CO_2), and the HFOs R1234yf and R1234ze(E) as the most promising low-GWP refrigerants adequate for the marine refrigeration systems considered in the paper. Single-stage, two-stage and cascade plant configurations are examined, and the performances of the different alternatives are evaluated through simulations. The results are analyzed, and the performances are compared with those of the current systems with R407f: in the comparisons we consider COP, volumetric capacity, safety and environmental impact. We conclude that switching from current technologies to systems using low-GWP refrigerants entails a worsening of the performances in at least one of the areas considered. Moreover, we observe that the reduction of the GWP value of the refrigerants is not an effective strategy to diminish the total environmental impact of the refrigeration systems considered.

Installations frigorifiques maritimes pour les navires à passagers: Frigorigènes à faible GWP et stratégies pour réduire l'impact environnemental

Mots clés : Froid maritime ; Faible GWP ; Efficacité énergétique ; TEWI

* *Corresponding author.* Department of Engineering and Architecture, University of Trieste, Via A. Valerio 10, 34127 Trieste, Italy. Tel.: +39 3339899984.
E-mail address: lorenzo.pigani@gmail.com (L. Pigani).

Nomenclature

E	annual energy consumption
h	specific enthalpy [kJ kg^{-1}]
L	annual refrigerant leakage rate [kg year^{-1}]
m	total refrigerant charge [kg]
$\dot{m}$	mass flow rate [kg s^{-1}]
n	life span of the refrigeration system [years]
p	pressure [bar]
P	power [kW]
Q	cooling capacity [kW]
Q_{vol}	volumetric capacity [MJ m^{-3}]
r_p	compression ratio
T	temperature [°C or K]
α	recycling factor of the refrigerant
β	indirect emission factor [kgCO$_2$ kWh^{-1}]
η	compressor isentropic efficiency
LFL	lower flammability limit [% volume in air]
UFL	upper flammability limit [% volume in air]
HOC	heat of combustion [MJ kg^{-1}]

Subscripts

c	chilling
cond	condensation
eco	economizer
ev	evaporation
f	freezing
hp	high pressure circuit
hvac	HVAC chillers
lp	low pressure circuit
sw	sea water

1. Introduction

Environmental laws are imposing increasing constraints regarding the use of refrigerant fluids, in order to reduce the environmental impact caused by their emission in the atmosphere.

The first types of refrigerants to be regulated were the ones whose emission in the atmosphere causes depletion of the ozone layer. The phenomenon is explained in detail by Molina and Rowland (1974). The Montreal Protocol (1987) (United Nations (UN), 1987) and the following regulations (United Nations Environment Programme (UNEP), 2007) established the progressive phase out of ozone depleting refrigerants such as CFCs and HCFCs. The use of CFCs has been prohibited in developed nations since 1997, and the use of HCFCs will be prohibited from 2020.

Following these restrictions, new types of refrigerants that do not deplete the ozone layer, such as HFCs, became increasingly common. Later studies highlighted that numerous HFCs are greenhouse gasses (GHG), i.e. their direct emission in the atmosphere contributes to causing global warming. The Kyoto Protocol (1997) (United Nations (UN), 1997) set targets for the reduction of the use of greenhouse gases. In 2014, the European Union developed a regulation (The European Parliament, 2014) that will progressively reduce the use of refrigerants that have global warming potential (GWP) higher than 150 between 2015 and 2022. Some states and cities in the United States are also proposing regulations designed to reduce GHG emissions (Calm, 2008), such as the ones enforced in California (California Environmental Protection Agency, 2014). The United States and China started a joint program in 2014 in order to reduce GHG emissions (The White House, Office of the Press Secretary, 2014) that commits its parties to a reduction of HFC emissions. Moreover, the use of HFCs is already heavily taxed in many European countries and in Australia (McLinden et al., 2014).

The development of more restrictive environmental standards imposed by governments stimulated researchers and the industry to investigate new environment friendly refrigerants. Several studies such as Bolaji and Huan (2013), Calm (2008), Mohanraj et al. (2009) and Sarbu (2014) have been done to identify low-GWP refrigerants to be used as alternatives to halogenated refrigerants. The studies identified hydrofluoroolefins (HFOs) and natural refrigerants as the best current alternatives to replace HFCs, and pointed out that regulatory requirements impose severe limitation to the choice of refrigerants, leading to limitations of the performances of the systems (Mota-Babiloni et al., 2015).

Low-GWP refrigerants have been analyzed extensively in literature for industrial and commercial applications (Cecchinato and Corradi, 2011; Mota-Babiloni et al., 2015; Sharma et al., 2014). The performances of HFOs have been analyzed mainly as replacements for R134a (Mota-Babiloni et al., 2016; Navarro-Esbrí et al., 2013; Yataganbaba et al., 2015), identifying them as promising alternatives having the main drawback of mild flammability. Numerous numerical and experimental studies in literature investigated the properties of natural refrigerants as replacement for HFCs (Bolaji and Huan, 2013), focusing in particular on carbon dioxide and ammonia (Pérez-García et al., 2013; Rigola et al., 2005; Sánchez et al., 2014) as replacements for halogenated refrigerants. The results highlighted that carbon dioxide systems are affected by low efficiencies: COPs can be improved using systems with internal heat exchangers, even if the performances remain poor compared to HFCs since CO_2 systems usually operate with transcritical cycles. Ammonia systems are efficient; however, high quantities of ammonia stored in the plants may constitute risks for the safety of the systems (ASHRAE, 2010). Cascade configurations using ammonia and carbon dioxide represent a solution to use natural refrigerants with a limited quantity of ammonia (Aminyavari et al., 2014; Bingming et al., 2009; Dopazo and Fernández-Seara, 2011), and to reach efficiencies similar to those of the systems with HFCs (ASHRAE, 2010).

Other studies focused on identifying new substances to be used as refrigerants (McLinden et al., 2014), and to assess limits and trade-offs of refrigerants from a thermodynamical standpoint (Domanski et al., 2014). These studies concluded that among the currently known chemical compounds, there are very few substances that have promising characteristics to be new generation refrigerants. There are correlations between thermodynamical, environmental and safety properties of fluids, and from the studies, it appears that there is always a trade off to be made between different requirements.

Marine refrigeration constitutes a very particular field of study where requirements concerning safety, weight, and dimensions are particularly strict compared to civil and industrial plants. Therefore, several solutions that are adequate for mainland applications may not be adequate for plants mounted onboard ships.

The main regulatory institutions for marine plants are the International Maritime Organization and naval registers. The international treatise MARPOL in annex VI (International Maritime Organization, 1997) prohibits the use of refrigerants that deplete the ozone layer in marine applications.

Currently, the GWP of refrigerants is not constrained by mandatory requirements in marine applications. Some naval registers propose voluntary class notations for refrigeration systems having low environmental impact: the *Lloyd's Register (LR)* proposes the ECO class notation, which limits the GWP of the refrigerants to a maximum of 1950 (Lloyd's Naval Register, 2014), the *Registro Navale Italiano (RINA)* proposes the CLEAN-AIR class notation, which limits the GWP of the refrigerants to a maximum of 2000 (Registro Navale Italiano (RINA), 2014), and the *Bureau Veritas (BV)* proposes the CLEAN-SHIP class notation, which limits the GWP of the refrigerants to a maximum of 2000 (Bureau Veritas, 2014).

Since numerous international treatises and national laws are imposing restrictions for the use of GHG as high-GWP refrigerants, it is reasonable to expect progressive limitations regarding the GWP of the refrigerants in marine applications in the future, as it is currently happening with the current regulations for domestic and civil plants. Moreover, the use of high-GWP refrigerants already requires complicated recovery operations, and the price of the refrigerants is often raised by environmental taxes. For these reasons, shipowner companies are interested in using low-GWP refrigerants.

The novelty of this paper is to analyze the issues of switching to low-GWP refrigerants for marine refrigeration plants, focusing on provision refrigeration systems for cruise ships. The typical system architectures are discussed and several plant layouts and fluids are analyzed. The safety and regulatory requirements and the technical limitations specific for marine application are described and considered as constraints during the analysis. Our goal is to ascertain the effect of switching to low-GWP refrigerants on system efficiencies, volumetric capacities and total environmental impact.

2. Fluids and systems considered

In this section, we describe and analyze current refrigeration plants. We select a set of low-GWP refrigerants that is adequate to be used in marine applications. Then, we present different plant layouts that we use to evaluate the performances of the refrigerants, and we describe the mathematical models used.

2.1. Current refrigeration plants for cruise ships

There are several plant layouts currently used in refrigeration systems for passenger ships. In this paper, we consider the system schematized in Fig. 1 as reference for the performances of current refrigeration plants. The system is composed by two different refrigeration plants, both having configurations with single-stage screw compressors and economizer:

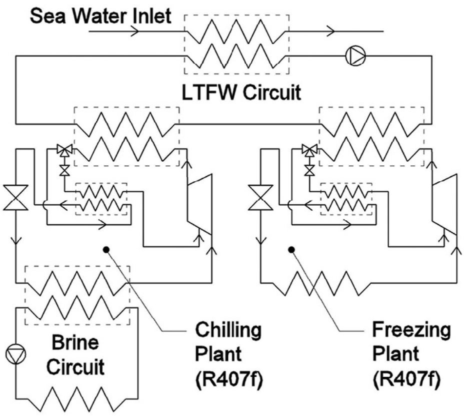


Fig. 1 – Current refrigeration plants.

- The *chilling plant* is an indirect expansion system with brine circuit that generates 75% of the total cooling capacity, and it operates having evaporation temperature $T_{ev,c} = -14.5°C$.
- The *freezing plant* is a direct expansion system that generates 25% of the total cooling capacity, and it operates having $T_{ev,f} = -34°C$.

Both systems use the fluid R407f as refrigerant, which is a mixture of R32, R125 and R134a in the mass percentages of 30.0%, 30.0% and 40.0%, respectively.

The *Low Temperature Fresh Water* (LTFW) system is a circuit in which fresh water flows, subtracting heat from the condenser of the chilling and freezing plants and rejecting it into the sea water. It is used to prevent corrosion problems that would arise using directly sea water in the condensers of the chilling and freezing plants. The LTFW system layout and the mass flow rate regulation considered in this paper are designed in order to maintain a constant value of

$$T_{cond} - T_{sw} = 13.5\ K \tag{1}$$

for both the chilling and freezing plants. Several refrigeration systems currently adopted are designed to maintain the LTFW at constant temperature; we consider a variable temperature LTFW system in order to increase system efficiency for low sea water temperatures.

Both the chilling and the freezing plants have a 100% redundancy, in order to guarantee high reliability.

In the present paper, we neglect the work required by pumps in the LTFW and brine circuits. The LTFW plant is modeled through Eq. (1), which is used to calculate the condensing temperatures given the sea water conditions. The model used to simulate the chilling and freezing plants is described in Sec. 2.3 and 2.5.

2.2. Fluids considered

The fluids considered for the analyses have been selected from the set of refrigerants defined in the Standards ASHRAE 34–2013 (American Society of Heating Refrigerating and Air Conditioning Engineers, 2013) and ISO 817:2014 (ISO, 2014). We impose the following constraints in order to discard the fluids that are not adequate for marine refrigeration:

– Null ozone depletion potential (ODP) and maximum GWP of 150.
– Maximum flammability class 2 L according to ASHRAE Standard 34–2013. Fire safety is a theme of major importance on marine applications, and the use of flammable or toxic refrigerants is discouraged by naval registers (Registro Navale Italiano (RINA), 2014). Highly flammable refrigerants, such as ethane, ethylene and other substances having lower flammability limit lower than 3.5% are prohibited. The use of ammonia as refrigerant is allowed under strict and precise safety requirements (Bureau Veritas, 2014; Lloyd's Naval Register, 2014; Registro Navale Italiano (RINA), 2014).
– We exclude chemically unstable fluids and refrigerants whose chemical or physical properties are currently not known from experimental studies, and the fluids whose physical properties are not adequate to work at the temperature and pressures required on the systems considered.

The fluids selected for the analysis are ammonia (NH_3), carbon dioxide (CO_2), R1234yf and R1234ze(E). The relevant properties of the refrigerants are listed in Table 1. We include the fluid R407f for comparison with current refrigeration technologies.

Ammonia is a natural refrigerant having excellent physical, transport and environmental properties. It has high latent heat and low specific volume compared to the other fluids considered (see Table 1). It is available in large quantities and at very low price. The main drawbacks to the use of ammonia in marine applications are linked to its toxicity. Ammonia leaks are easily identifiable due to the distinct odor of the gas: in industrial plants it is considered an advantage, but in passenger ships, it constitutes an additional risk since the strong ammonia odor may cause panic among passengers.

Carbon dioxide is a natural refrigerant which, among the low-GWP fluids considered, is the sole one having A1 safety class. Carbon dioxide refrigeration systems work at high pressure, which are one order of magnitude higher than those required by the other refrigerants considered (see Table 1), leading to low specific volumes and therefore to compact systems. In the temperature range required by marine applications the plants operate using transcritical cycles, which are discussed in (ASHRAE, 2010; Kim et al., 2004).

The HFOs R1234yf and R1234ze(E) are recently developed refrigerants, which are commercialized by few companies and are available at high price. They have physical properties similar to those of R134a, and they are designed mainly to work in chilling plants with positive evaporation temperatures as substitutes for R134a. Compared to the other fluids in Table 1, they have low latent heat and high specific volume at the temperatures considered. They are mildly flammable.

From the analysis of the data of Table 1 it follows that all the low-GWP refrigerants considered are adequate for chilling plants, but only carbon dioxide is adequate for freezing applications. In fact the evaporation pressure of the other low-GWP refrigerants at freezing evaporation temperatures ($T_{ev,f}$) is lower than atmospheric pressure: in case of leaks this could cause severe problems due to the entrance of air in the refrigerant circuit. Moreover they have low volumetric capacity, requiring large and expensive equipment.

The fluid R407f is a fluorinated gas with GWP two orders of magnitude higher than the other fluids considered, and it is used for comparison with current refrigeration technologies. It can be used on both chilling and freezing plants and it is neither flammable nor toxic. It is a zeotropic mixture, i.e. its components maintain different evaporation and condensation temperatures in the mixture. A detailed description of the peculiarities of zeotropic mixtures is provided in (Mohanraj et al., 2011; Rajapaksha, 2007).

We consider six different plant configurations using low-GWP refrigerants in the analyses. The systems are illustrated in Fig. 2.

2.3. *Simple refrigeration cycle (System A, Fig. 2)*

The system is constituted by four components: compressor, condenser or gas cooler, expansion valve and evaporator. Pressure losses in these components and in the circuits are neglected.

The compression work w is calculated through the equation

$$w = \frac{h_{2s} - h_1}{\eta} \tag{2}$$

where point 2s has the same entropy of point 1 and pressure equal to the condenser or gas cooler pressure. The compressor isentropic efficiency η is calculated for screw compressors as

Table 1 – Properties of the refrigerants considered in the analyses.

	NH_3	CO_2	R1234yf	R1234ze(E)	R407f
ODP	0	0	0	0	0
$GWP_{100\ years}$	0	1	4	6	1824
Critical temperature [°C]	132.3	31.0	94.7	109.4	82.7
Normal boiling point [°C]	−33.3	−56.6	−29.4	−19.0	−45.5
Vapor pressure at −15°C [MPa]	0.236	2.29	0.184	0.120	0.301
Latent heat at −15°C [kJ/kg]	1313	270.9	172.4	193.3	228.3
Vapor specific volume at −15°C [m^3/kg]	0.5086	0.0165	0.0953	0.1493	0.0796
Liquid specific volume at 40°C [m^3/kg]	$17.25 \cdot 10^{-4}$	n.d.	$9.67 \cdot 10^{-4}$	$8.99 \cdot 10^{-4}$	$9.57 \cdot 10^{-4}$
Safety group (ASHRAE 34–2013)	B2L	A1	A2L	A2L	A1
ACGIH TLV-TWA [ppm]	25	5000	500	800	1000
LFL-UFL [% vol.]	16–25	n.d.	6–12.3	n.d.	n.d.
HOC [MJ kg^{-1}]	22.5	n.d.	10.7	n.d.	n.d.

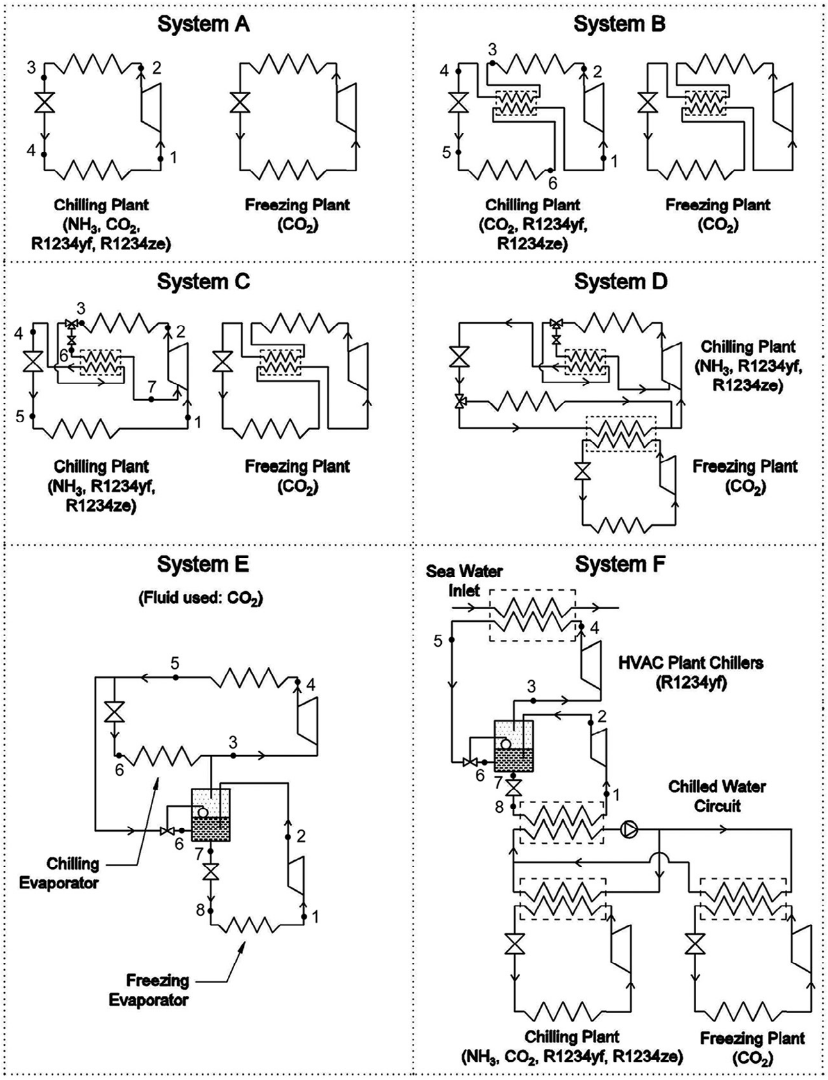


Fig. 2 – Systems considered.

$$\eta = -13.44 r_p^2 + 80.64 r_p - 39.96 \quad \text{for} \quad r_p \in [1.5; 3] \tag{3}$$

$$\eta = -0.375 r_p^2 + 2.25 r_p - 77.63 \quad \text{for} \quad r_p \in [3; 7] \tag{4}$$

$$\eta = -2.50 r_p + 92.50 \quad \text{for} \quad r_p \in [7; 15] \tag{5}$$

We use this model since in this paper we analyze systems with compressors working for very different compression ratios, and it is fundamental to have a reliable estimate of the order of magnitude of η to determine the performances of the systems. The model is based on the following observations (Stoecker, 2004): for intermediate compression ratios ($r_p \in [3;7]$) it is possible to select compressors having different volumetric compression ratios, and in the model we considered the envelope of their efficiency curves assuming to use always the compressor guaranteeing the maximum efficiency for every given r_p; for low compression ratios ($r_p \in [1.5; 3]$) the efficiency of screw compressors drops consistently, while for compression ratios above 7 the efficiency drop is approximately linear. In the first two ranges of compression ratios the efficiency is modeled with parabolic functions, and in the third one with a linear function. The coefficients have been derived in order to have a continuous efficiency function approximating the experimental efficiency data for screw compressors (Hanlon, 2001; Stoecker, 2004).

Carbon dioxide systems use small and compact reciprocating compressors specifically designed for refrigeration systems. Their efficiency is calculated as

$$\eta = -15.55 r_p^2 + 77.76 r_p - 24.21 \quad \text{for} \quad r_p \in [1.5; 2.5] \tag{6}$$

$$\eta = -4.29 r_p + 83.71 \quad \text{for} \quad r_p \in [2.5; 9] \tag{7}$$

The model approximates the efficiency with a linear function for compression ratios above 2.5, and the performance drop for r_p below 2.5 is modeled by a parabolic function. The coefficients have been derived in order to have a continuous efficiency function approximating the experimental efficiency data for reciprocating compressors for CO_2 (Javerschek

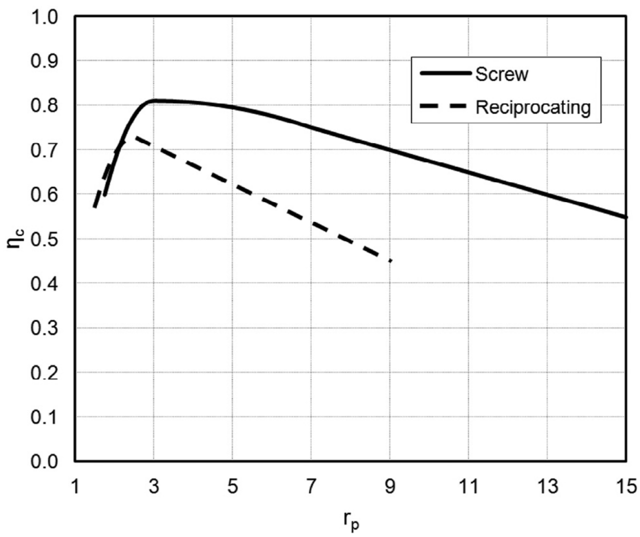


Fig. 3 – Compression isentropic efficiency.

and Dittrich, 2009; Kim et al., 2004; Ma et al., 2012). Isentropic efficiencies for screw and reciprocating compressors are illustrated in Fig. 3.

We consider a constant difference of 13.5 K between the sea water temperature and the condensation temperatures of chilling and freezing plants, as for current refrigeration systems (see Eq. (1)). For transcritical carbon dioxide systems we consider a temperature difference of 13.5 K between the sea water temperature and the temperature of the fluid exiting from the gas cooler.

The analyses are performed for three sea water temperature levels, i.e. 32°C, 25°C and 18°C. The first temperature level represents the maximum sea water temperature considered in design conditions. The other two levels are representatives of different environmental conditions which are typical for the areas where cruise ships operate.

The evaporation temperatures for chilling plants is −14.5°C, corresponding to an approach of 4.5°C with the brine circuit. The evaporation temperature of freezing plants with R407f is −34°C, corresponding to an approach of 9°C. Freezing plants operate with higher approaches since they are direct expansion systems. For carbon dioxide freezing plants, we consider evaporating temperatures of −32°C, corresponding to an approach of 7°C. As done in other studies (Llopis et al., 2015), it is reasonable to consider lower approaches for carbon dioxide systems, due to the excellent heat transfer properties and to the low specific volume of CO_2.

The refrigerant R407f is a mixture that exhibits a significant temperature glide. To evaluate evaporation and condensation temperatures, we use the model proposed by (Buffalo Research Laboratory, Honeywell, 2014; Mota-Babiloni et al., 2014), which approximate them as

$$T_{ev} = \frac{1}{3}T_{Bubble} + \frac{2}{3}T_{Dew} \tag{8}$$

$$T_{cond} = \frac{1}{2}T_{Bubble} + \frac{1}{2}T_{Dew} \tag{9}$$

and we use an iterative procedure to estimate evaporation and condensation pressures.

For transcritical carbon dioxide systems the gas cooler pressure is not univocally determined given the gas outlet temperature. The efficiency of the systems is dependent on the gas cooler pressure, as represented in Fig. 4. We consider the gas cooler pressure, which maximizes the efficiency of the systems under the constraint of $T_2 \le 160°C$. The constraint on the discharge temperature is due to technological requirements for reciprocating compressors. The optimal pressure is identified by means of an iterative procedure.

To determine the energetic efficiency of the chilling and freezing plants we calculate the coefficient of performances

$$COP_x = \frac{h_1 - h_4}{h_2 - h_1} \quad \text{where} \quad x = c, f \tag{10}$$

The volumetric capacity of refrigeration plants is defined as

$$Q_{vol} = \frac{h_1 - h_4}{v_1} \tag{11}$$

For a given cooling capacity, the volumetric capacity is inversely proportional to the volumetric flow rate. Therefore, systems with high volumetric capacity require more compact and lightweight equipment.

We define the overall coefficient of performances of single stage refrigeration systems

$$COP = \frac{Q_c + Q_f}{P_c + P_f} \tag{12}$$

where, for single stage systems

$$P_f = \frac{Q_f}{COP_f} \qquad P_c = \frac{Q_c}{COP_c} \tag{13}$$

The COPs of the chilling and freezing plants are calculated through Eq. (10). The overall COP takes into account the

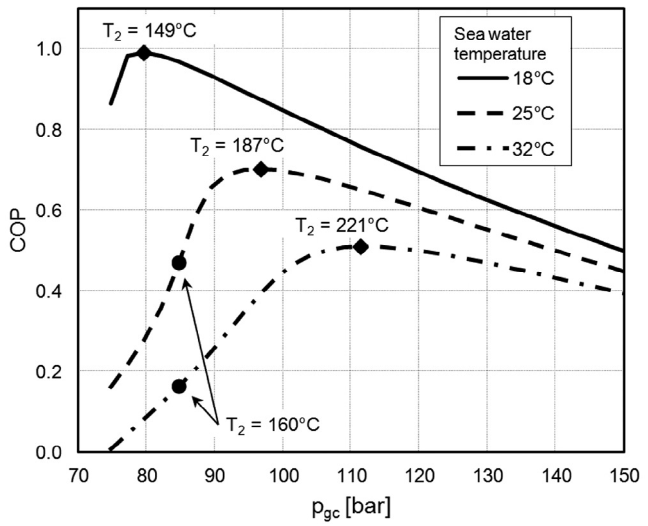


Fig. 4 – Transcritical systems COP and gas cooler pressure.

efficiency of both chilling and freezing plants, and it serves to compare the efficiencies of single stage systems (systems A, B and C in Fig. 2) with the ones of multistage systems (systems D, E and F in Fig. 2).

The simulations are implemented using Matlab. The refrigerant properties are evaluated using the database Refprop (Lemmon et al., 2014). For all the system considered in the paper, otherwise specified differently, we always consider the same hypotheses and parameters described in this section.

2.4. *System with IHX (System B, Fig. 2)*

In this system, an internal heat exchanger (IHX) is used to subcool the fluid exiting from the condenser or gas cooler. The heat is transferred to the fluid exiting from the evaporator. The conditions of points 1 and 4 are calculated by means of the heat exchanger efficiency, defined as

$$\varepsilon = \frac{T_1 - T_6}{T_3 - T_6} \tag{14}$$

We consider $\varepsilon \in [0; 0.7]$ and using an iterative procedure, we select the value of ε, which maximizes the efficiency of the system under the constraint of limited compressor discharge temperature. The maximum discharge temperature is 120°C for screw compressors, and 160°C for reciprocating compressors for CO_2; these are typical values of acceptable discharge temperatures for commercial screw and reciprocating compressors (GEA Bock GmbH, 2015). For transcritical carbon dioxide systems, we optimize concurrently the parameter ε and the gas cooler pressure using a genetic algorithm.

Ammonia systems with IHX are not considered since the discharge temperatures of these systems are always above the constraints considered in this paper.

2.5. *System with economizer (System C, Fig. 2)*

In these systems, the chilling plant includes an economizer in order to increase the efficiency and the cooling capacity of the system. The economizer is not used with CO_2 systems, since they employ reciprocating compressors, which are not technologically adequate to be equipped with an economizer port.

We consider a decrease in efficiency due to the presence of the economizer flow inlet. The efficiency is calculated as

$$\eta_{eco} = \eta \cdot 0.99 \tag{15}$$

where η is the efficiency of screw compressors calculated through Eqs. (3), (4) and (5).

To calculate the efficiency of the system, we consider unitary mass flow rate in the evaporator, and the mass flow rate m_{eco} in the economizer (point 6). We always consider the economizer mass flow rate m_{eco} such as $T_4 - T_6 = 5$ K. The efficiency of the chilling plant is calculated as

$$COP_c = \frac{(h_1 - h_5)}{(h_7 - h_1) + (1 + m_{eco}) \cdot (h_2 - h_7)} \tag{16}$$

and using an iterative procedure, we consider the economizer pressure, which maximizes the COP.

2.6. *Cascade system with economizer (System D, Fig. 2)*

In these systems, the freezing and chilling plants work in cascade, as represented in Fig. 2. They are connected by means of a cascade heat exchanger: we consider a temperature difference of 4.5 K in the heat exchanger, which leads to a condensation temperature of −10°C for the freezing plants. With this configuration, it is possible to have subcritical carbon dioxide freezing plants.

The coefficients of performances of the chilling and freezing plants are calculated separately as discussed in Sec. 2.3 and 2.5. The overall COP of the system is defined as

$$COP = \frac{Q_c + Q_f}{P_c + P_f} \tag{17}$$

where for cascade systems

$$P_f = \frac{Q_f}{COP_f} \qquad P_c = \frac{Q_c + Q_f + P_f}{COP_c} \tag{18}$$

2.7. *Two stage system with flash tank (System E, Fig. 2)*

This system has a single refrigeration circuit with two evaporators working at different temperatures for the chilling and freezing plants.

Given the total refrigeration capacity required, it is possible to calculate the mass flow rate in the freezing evaporator as

$$\dot{m}_f = \frac{Q_f}{h_1 - h_8} \tag{19}$$

The mass flow rate in the high pressure circuit is calculated as

$$\dot{m}_c = \frac{\dot{m}_f \cdot (h_7 - h_2) - Q_c}{h_5 - h_3} \tag{20}$$

The overall efficiency of the system is evaluated from the equation

$$COP = \frac{Q_f + Q_c}{\dot{m}_f \cdot (h_2 - h_1) + \dot{m}_c \cdot (h_4 - h_3)} \tag{21}$$

2.8. *Cascade system using HVAC chillers (System F, Fig. 2)*

Passenger ships HVAC systems are usually separated from provision refrigeration plants. Usually, HVAC systems use chillers to produce water at a temperature of 7°C, which is used in the air handling units.

In this section, we consider chilling and freezing plants, which works as the ones of system A. The sole difference is that, instead of rejecting the heat of condensation in the LTFW circuit, they use condensing water at 7°C precooled by the chillers of the HVAC system. With this configuration, the CO_2 freezing and chilling plants work in subcritical conditions.

In order to evaluate the power required to precool the condensing water, it is necessary to simulate the behavior of the HVAC plant chillers. They work with a two stage system with a flash tank. The temperature difference between condensation temperature and sea water temperature is 7.5 K. The fluid exiting from the evaporator is superheated by 3 K.

The mass flow rate $\dot{m}_{lp}$ in the low pressure compressor is evaluated through Eq. (19). The mass flow rate in the high pressure compressor is calculated by means of the equation

$$\dot{m}_{hp} = \frac{\dot{m}_{lp} \cdot (h_2 - h_7)}{h_3 - h_6} \tag{22}$$

The intermediate pressure is fixed such as the two centrifugal compressors have the same compression ratio, therefore

$$p_{int} = \sqrt{p_4 \cdot p_1} \tag{23}$$

The compression isentropic efficiency is maximum when T_{sw} is 32°C, and it is equal to 0.75. For sea water temperatures T_{sw} of 25°C and 18°C we consider efficiencies of 0.70 and 0.60, respectively.

The efficiency of HVAC chillers is calculated using the equation

$$COP = \frac{\dot{m}_{lp} \cdot (h_1 - h_8)}{\dot{m}_{lp} \cdot (h_2 - h_1) + \dot{m}_{hp} \cdot (h_4 - h_3)} \tag{24}$$

and for sea water temperatures of 32°C, 25°C and 18°C the COP of HVAC chillers are equal to 5.26, 6.34 and 7.46, respectively.

The overall efficiency of the system is calculated taking into account the power absorbed by the HVAC chillers, by means of the equation

$$COP = \left[\frac{q_c}{COP_c} + \frac{q_f}{COP_f} + \frac{q_c(1 + 1/COP_c)}{COP_{hvac}} + \frac{q_f(1 + 1/COP_f)}{COP_{hvac}} \right]^{-1} \tag{25}$$

where

$$q_f = \frac{Q_f}{Q_f + Q_c} \qquad q_c = \frac{Q_c}{Q_f + Q_c} \tag{26}$$

2.9. *Environmental impact analyses*

The GWP value is a parameter proportional to the warming impact due to direct emission of a refrigerant in the atmosphere.

In order to assess the total warming impact of the different systems, we consider the *total environmental warming impact* (TEWI) (ISO, 2014) of the refrigeration systems. The TEWI takes into account the global warming impact due to direct emissions of a refrigerant in the atmosphere; in addition, it takes also into account indirect emissions, which are generated to produce the electricity used by the system. The TEWI is defined as

$$TEWI = [GWP \cdot L \cdot n] + [GWP \cdot m \cdot (1 - \alpha)] + [n \cdot E \cdot \beta] \tag{27}$$

We considered a refrigerant mass m of 200 kg for the chilling plants, and 300 kg for the freezing plants (which are direct expansion systems). In system F, the refrigerant mass considered for the HVAC chillers is 100 kg. The leakage rate is assumed to be 10% per year, which is the maximum allowed for systems classified as CLEAN-AIR. According to the guidelines of The Australian Institute of Refrigerating, Air Conditioning and Heating (2012), we consider a refrigerant recovery rate α of 90%. The life span n of the refrigeration systems is 30 years. To calculate the total energy consumption, we consider the systems working continuously employing on average 70% of the nominal power. The indirect emission factor β considered is 0.8 $kgCO_2$/kWh for electricity generated by marine engines.

3. Results

In the present section, we analyze and discuss the results of the simulations on the systems illustrated in Fig. 2. The results are compared with the performances of current refrigeration systems, illustrated in Fig. 1.

3.1. *Efficiency analyses*

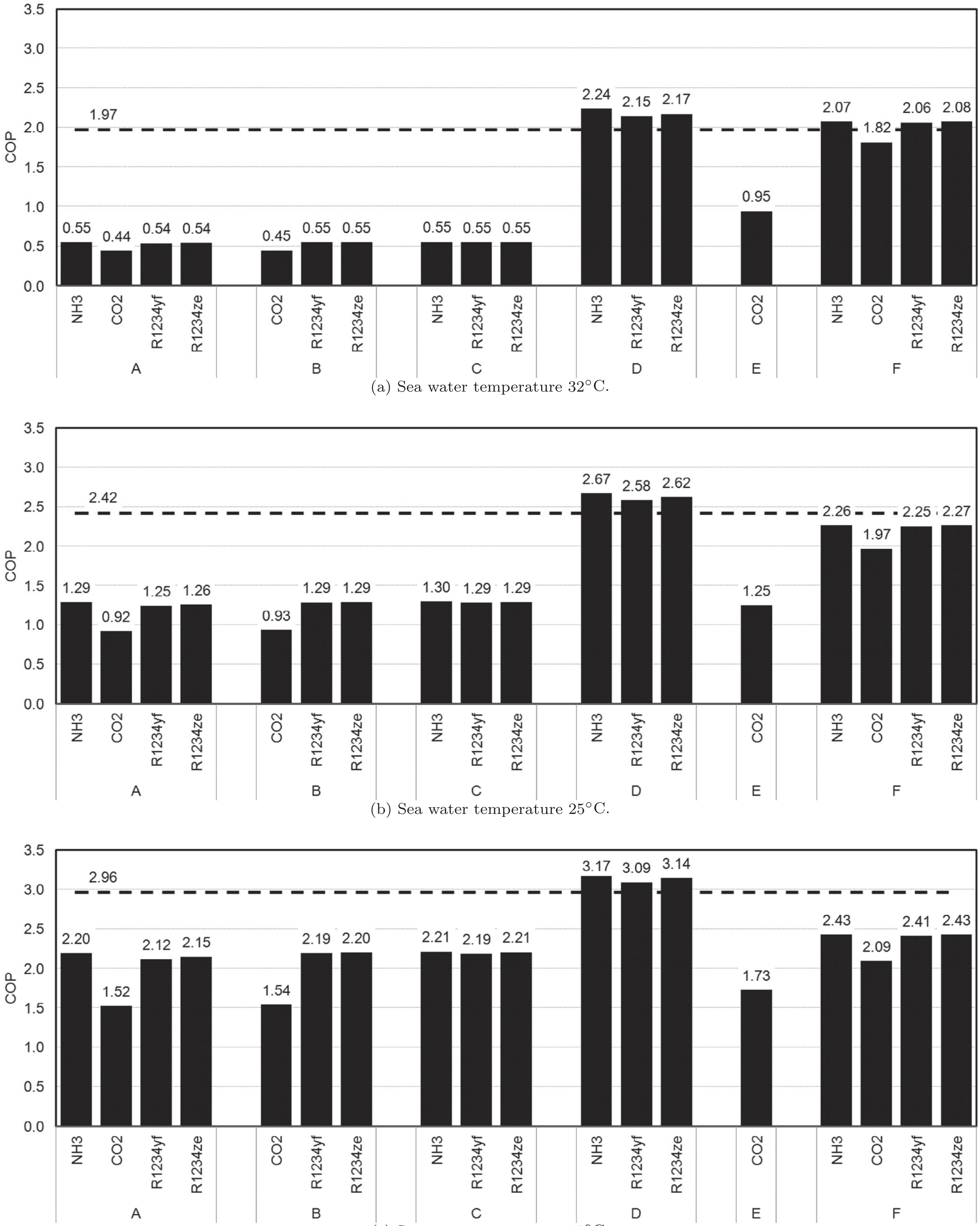
We calculated the efficiencies of the refrigeration systems for three temperature levels of the sea water, i.e. 32°C, 25°C and 18°C. These three temperature levels are representative of the typical range of working conditions for marine refrigeration plants for cruise ships. The results are reported in Fig. 5. The horizontal dashed line in the figure represents the COP of current refrigeration plants. For the temperature level of 32°C the complete simulation data are reported in Table 2.

For every refrigerant considered, the efficiency of systems A is always lower than the one of current refrigeration systems. The difference is larger for high sea water temperature, as reported in Fig. 5. For every sea water temperature level considered, the COPs of the chilling and freezing plants are lower than those of current refrigeration plants. In particular, the systems with low-GWP fluids use transcritical carbon dioxide freezing plants, having extremely low COPs (see Table 2). This penalizes the overall efficiency of the refrigeration systems calculated through Eq. (17).

The presence of internal heat exchangers in system B has a positive impact on the efficiency of chilling plants. In fact, the performances of HFOs are improved by 10% compared to system A. The performance improvement for transcritical systems is very limited for chilling plants, and absent for freezing plants (see Table 2). This is due to the constraints on the maximum compressor discharge temperature, which prevents the exploitation of the performance improvement offered by internal heat exchangers. Due to the low performances of freezing plants, the overall COPs of the systems is extremely low compared to current refrigeration plants.

Economizers in system C guarantee better performances than systems A and B, and lower discharge temperatures compared to system B. However, due to the low efficiencies of transcritical freezing plants, the overall efficiency of the system are poor with every low-GWP refrigerant considered.

System D is the sole one having higher efficiency than current refrigeration systems for every sea water temperature level considered, and for every refrigerant considered. The CO_2 freezing plants work in subcritical conditions with this



(a) Sea water temperature 32°C.

(b) Sea water temperature 25°C.

(c) Sea water temperature 18°C.

Fig. 5 – Efficiencies of the systems for different sea water temperatures.

Table 2 – Simulation results for sea water temperature of 32°C.

System	Chilling Fluid	COP_c [–]	$Q_{vol,c}$ [MJ/m³]	$T_{d,c}$ [°C]	COP_f [–]	$Q_{vol,f}$ [MJ/m³]	$T_{d,f}$ [°C]	COP [–]	$TEWI_{dir}$ [kgCO₂]	$TEWI_{ind}$ [kgCO₂]
Current (Fig. 1)	R407f	2.49	2.41	81.3	1.21	1.09	128	1.97	$1.70 \cdot 10^6$	$2.99 \cdot 10^7$
System A (Fig. 2)	NH_3	2.47	2.06	178*	0.17	0.89	160	0.55	$6.20 \cdot 10^2$	$1.07 \cdot 10^8$
	CO_2	0.99	7.16	142	0.17	0.89	160	0.44	$9.30 \cdot 10^2$	$1.33 \cdot 10^8$
	R1234yf	2.16	0.97	50.3	0.17	0.89	160	0.54	$1.86 \cdot 10^3$	$1.09 \cdot 10^8$
	R1234ze	2.24	0.76	55.5	0.17	0.89	160	0.54	$2.48 \cdot 10^3$	$1.09 \cdot 10^8$
System B (Fig. 2)	CO_2	1.03	6.97	160	0.17	0.89	160	0.45	$9.30 \cdot 10^2$	$1.32 \cdot 10^8$
	R1234yf	2.48	1.14	91.9	0.17	0.89	160	0.55	$1.86 \cdot 10^3$	$1.07 \cdot 10^8$
	R1234ze	2.47	0.85	99.5	0.17	0.89	160	0.55	$2.48 \cdot 10^3$	$1.07 \cdot 10^8$
System C (Fig. 2)	NH_3	2.56	2.28	180*	0.17	0.89	160	0.55	$6.20 \cdot 10^2$	$1.06 \cdot 10^8$
	R1234yf	2.44	1.29	50.7	0.17	0.89	160	0.55	$1.86 \cdot 10^3$	$1.07 \cdot 10^8$
	R1234ze	2.47	0.96	55.9	0.17	0.89	160	0.55	$2.48 \cdot 10^3$	$1.07 \cdot 10^8$
System D (Fig. 2)	NH_3	2.56	2.28	180*	6.29	9.01	24.3	2.24	$6.20 \cdot 10^2$	$2.63 \cdot 10^7$
	R1234yf	2.44	1.29	50.7	6.29	9.01	24.3	2.15	$1.86 \cdot 10^3$	$2.74 \cdot 10^7$
	R1234ze	2.47	0.96	55.9	6.29	9.01	24.3	2.17	$2.48 \cdot 10^3$	$2.71 \cdot 10^7$
System E (Fig. 2)	CO_2	n.d.	7.16	142	n.d.	9.36	15.2	0.95	$9.30 \cdot 10^2$	$6.23 \cdot 10^7$
System F (Fig. 2)	NH_3	5.99	2.45	80.3	2.08	6.65	90.7	2.07	$1.83 \cdot 10^3$	$2.84 \cdot 10^7$
	CO_2	4.11	11.8	53.6	6.29	6.65	90.7	1.82	$2.14 \cdot 10^3$	$3.18 \cdot 10^7$
	R1234yf	5.90	1.57	19.8	6.29	6.65	90.7	2.06	$3.07 \cdot 10^3$	$2.85 \cdot 10^7$
	R1234ze	6.02	1.14	21.8	6.29	6.65	90.7	2.08	$3.69 \cdot 10^3$	$2.84 \cdot 10^7$

* NH_3 systems having T_d higher than 120°C.

cascade configuration, and it is possible to achieve high overall efficiencies.

System E performs poorly compared to system D. The low overall COP is due to the unsatisfactory performances of the transcritical high-pressure circuit.

In system F, the chilling and freezing plants operate at low condensation temperatures, since they reject the heat in water precooled by the HVAC chillers. The chilling and freezing plants have high COPs, and carbon dioxide systems work in subcritical conditions. This system configuration is justified for high sea water temperature, since it grants better performances than current refrigeration systems (as in Fig. 5a). For lower sea water temperatures, as in Fig. 5b and 5c, the overall COP of the system is lower than in current refrigeration systems.

Ammonia systems operate with high discharge temperatures for several system configurations. We considered a maximum discharge temperature of 120°C for screw compressors, and the systems where the discharge temperature is higher than this limit are indicated with an asterisk in Table 2. Special compressors are required to guarantee normal operation for those systems.

From the data of Fig. 5 it emerges that the use of different refrigerants have a slight impact on the system efficiency. The most important element determining systems efficiencies is the plant configuration.

3.2. *Volumetric capacity*

We analyzed the volumetric capacity of the systems considered in Fig. 2, for the same three sea temperature levels considered in the previous section. We report in Table 2 the results for systems operating at the sea water temperature of 32°C. The other results are omitted since they do not differ substantially from the ones presented, and the conclusions that can be drawn are the same.

The volumetric capacity of carbon dioxide systems is always higher than the one of current refrigeration systems. This is due to the high pressures employed in CO^2 plants. In this paper, carbon dioxide is the sole low-GWP refrigerant considered for freezing plants; therefore, all the freezing plants working with low-GWP refrigerants have higher volumetric capacity than current refrigeration systems (see Table 2).

The values of volumetric capacity for chilling plants and for the sea water temperature of 32°C are presented in Table 2 and Fig. 6. The value of volumetric capacity for current refrigeration plants is indicated in the figure with a dashed horizontal line.

The volumetric capacity depends mainly on the refrigerant selected, and it is not strongly affected by the system configuration. Ammonia systems have slightly lower volumetric capacity than current refrigeration plants. The systems with HFOs present volumetric capacity substantially lower compared to current refrigeration systems. In fact, the refrigerants R1234yf and R1234ze have been developed to be used in systems having positive or slightly negative evaporation temperatures, substituting R134a. Marine chilling plants operate with evaporation temperatures of –14.5°C, which lead to very low evaporation pressures for these refrigerants, and therefore high specific volumes. Volumetric capacity drops of approximately 40–60% compared to current refrigeration systems; the fluid R1234yf performs better than R1234ze.

3.3. *Total environmental warming impact*

We analyzed the total environmental warming impact for all the systems Fig. 2, and for the sea temperature levels of 32°C, 25°C and 18°C. We report in Fig. 7 the results for the sea water temperature level of 32°C.

Current refrigeration systems operate with high-GWP refrigerants, and they are the systems having the highest direct

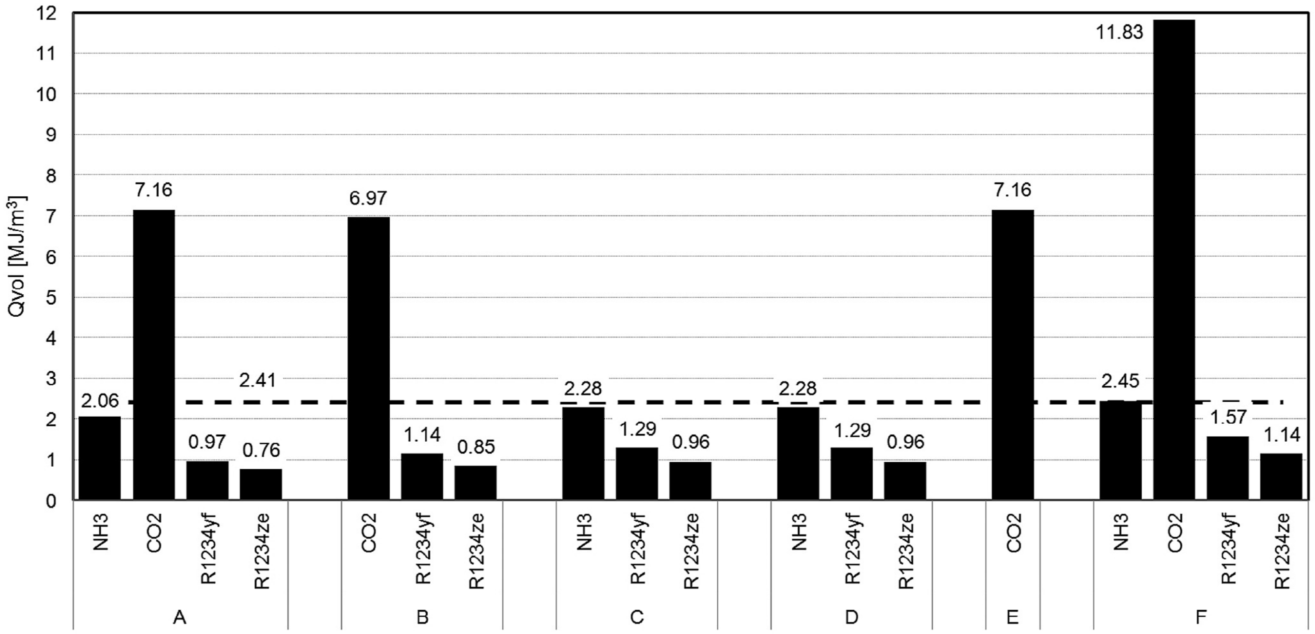


Fig. 6 – Volumetric capacity of chilling plants for sea water temperature of 32°C.

emissions among those considered. The use of low-GWP refrigerants reduce the direct emissions of three orders of magnitude (see Table 2).

Nonetheless, most of the emissions of marine refrigeration plants are indirect, i.e. they are due to the warming impact generated to produce the electric power absorbed by the refrigeration systems. In fact, electric power in passenger ships is produced with diesel generators, mainly alimented using heavy fuel oils. The operation of these generators produces a consistent global warming impact.

Direct emissions amount to less than 10% of the total emissions for current refrigeration plants, which use R407f having a GWP of 1824. Direct emissions are considerably less significant for the systems with low-GWP refrigerants, amounting to less than 1% of the total emissions.

The total emissions of systems A, B, C and E with low-GWP refrigerants are two to four times higher than those of current refrigeration systems, as shown in Fig. 7. This is due to the poor efficiencies achievable by the plants with low-GWP refrigerants. Systems with low COPs require higher power to generate a given cooling capacity, and therefore they generate higher indirect emissions. The reduction of direct emissions consequent to the use of low-GWP refrigerants is negligible compared to the increase in the indirect emissions; this results in an overall increase of the total environmental warming impact of the refrigeration systems.

The sole systems having TEWI lower than current systems are system D and system F (except with carbon dioxide as refrigerant). The cascade configurations allow the use of low-GWP refrigerants, and to have higher COPs than current

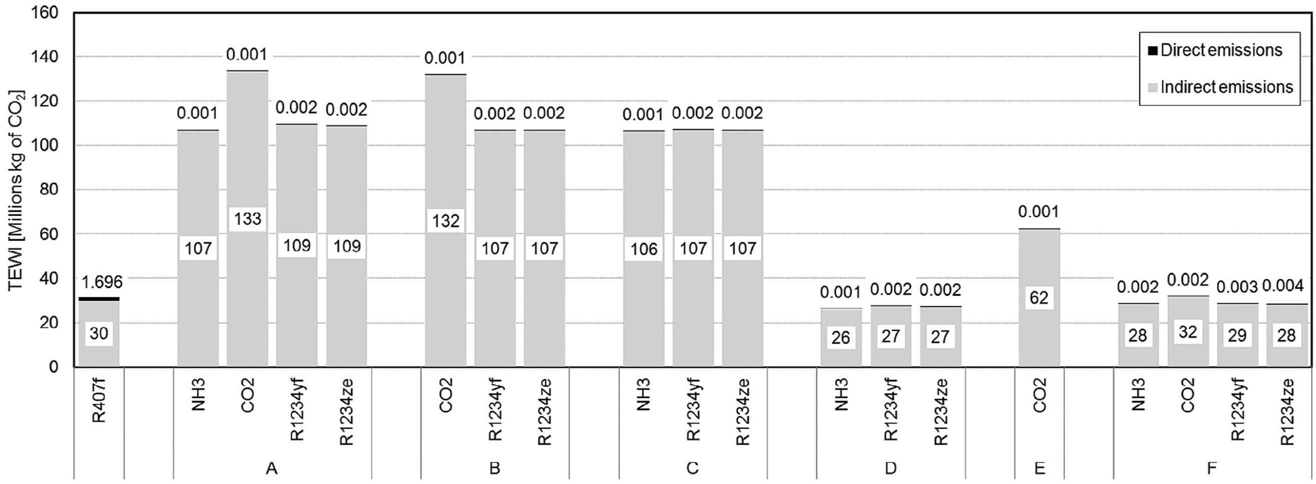


Fig. 7 – TEWI for sea water temperature of 32°C.

systems. This allows the reduction of both direct and indirect emissions.

For all the systems considered in this paper, the total environmental warming impact depends mainly on the efficiency of the refrigeration systems. The importance of the GWP of the refrigerants used in the systems is secondary.

4. Conclusions

In the paper, we analyzed the performances of marine refrigeration systems for passenger ships operating with low-GWP refrigerants. We evaluated through simulations the performances of the most promising low-GWP refrigerants currently available, and we compared them with those of current refrigeration plants considering several system configurations. We examined safety, efficiency, volumetric capacity and total environmental warming impact of the different solutions.

From the analyses, it can be concluded that none of the refrigerants examined is adequate to maintain or improve the levels of safety, efficiency, volumetric capacity and TEWI compared to current refrigeration systems. These conclusions are clear from the data of Table 2 considering for every refrigerant the system having maximum overall efficiency, and comparing the overall COP, TEWI and chilling plant volumetric capacity with the performances of current refrigeration systems. Ammonia is the sole refrigerant having high efficiency (+14%), similar volumetric capacity (–5%), and low TEWI (–17%) compared to current refrigerants: nonetheless it is a toxic substance, which entails severe problems from the safety standpoint. HFOs are acceptable from the efficiency (+9%) and environmental point of view (TEWI –14%), but they are slightly flammable and they have low volumetric capacity (–46%). Carbon dioxide systems are characterized by low efficiencies (–8%) and consequently high TEWI (+1%), with excellent volumetric capacity (three times higher than that of systems with R407f). With the current technologies, the refrigerant R407f, used in current plants, represents an excellent compromise of efficiency, volumetric cooling capacity, TEWI and safety.

We can also conclude that, in the refrigeration systems analyzed in this paper, the adoption of low-GWP refrigerants is not an effective solution to reduce the total environmental impact. For the systems analyzed, it is more effective to improve the plants' efficiencies in order to reduce the environmental impact (see systems D and F). The TEWI is more representative than the GWP to determine the total warming impact of refrigeration systems, since a reduction of the GWP of the refrigerant can lead to an overall increase of the warming impact of refrigeration systems.

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