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Gullo, Paride

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Innovative fully integrated transcritical R744 refrigeration systems for a HFC-free future of supermarkets in warm and hot climates

Paride Gullo

*DTU Technical University of Denmark, Department of Mechanical Engineering, Nils Koppels Allé,
Building 403, 2800 Kgs. Lyngby (Denmark)*

Corresponding author:

- *Paride Gullo*
- Telephone number: +45 45 25 41 21
- Fax number: +45 45 25 19 61
- Email address: parigul@dtu.dk

Abstract:

The regulatory pressures on environmentally deleterious refrigerants have been growing more intense, prompting CO₂ to draw ever-widening attraction as the sole working fluid (R744) in supermarket applications. Direct space heating and cooling is considered as one of the most appealing techniques to promote the diffusion of “CO₂ only” systems in warm and hot climates. However, to the best of the author’s knowledge, the energy and environmental benefits related to its adoption have not been estimated yet.

The results of this work revealed that “CO₂ only” units implementing direct space heating and cooling offer an annual energy saving by up to 33.3% as well as a reduction in carbon footprint by up to 89.5% compared to the systems using hydrofluorocarbons (HFCs) in high ambient temperature countries. It could be concluded that direct space heating and cooling is a very good prospect to lead to a HFC-free future of supermarkets, even in hot climates.

Keywords:

CO₂ refrigeration system; HFC refrigeration system; Direct space heating and cooling; Overfed evaporator; System integration; Two-phase ejector.

Nomenclature

Symbols, abbreviations and subscripts/superscripts

AC	Air conditioning
AUX	Auxiliary (or parallel) compressor(s)
AYT	Average yearly temperature [°C]
CC	Cascade condenser
CD	Air-cooled condenser
compr	Compressor(s)
COP	Coefficient of Performance [-]
CR	Circulation ratio of pump [-]
DHW	Domestic hot water
DS	De-superheater
DXS	Two R404A direct expansion supermarket refrigeration units, one providing MT load and the other satisfying LT load
E	Annual energy consumption [kWh]
EES	Engineering Equation Solver
evap	Evaporator(s)
exp	Expansion valve
GC	Air-cooled R744 gas cooler/condenser
GWP	Global Warming Potential [$\text{kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$]
HFC	Hydrofluorocarbon
HP	High pressure [bar]
HTC	High temperature circuit
HVAC	Heating, ventilation and air conditioning
IHX	Internal heat exchanger
IND	R1234ze(E)/R744 indirect supermarket refrigeration system with MT and LT flooded evaporators
IP	Intermediate pressure [bar]
L	Annual leakage rate [$\text{kg} \cdot \text{year}^{-1}$]
LEJ	Liquid ejectors
LP	Low pressure [bar]
LT	Low temperature
LTC	Low temperature circuit
m	Refrigerant charge [kg]
MEJ	Multi-ejector block
MP	Medium pressure [bar]
MT	Medium temperature [°C]
MTC	Medium temperature circuit
n	System operating life time [year]
out	Outlet
p	Pressure [bar]

PC	Transcritical R744 supermarket refrigeration system relying on direct space heating and cooling and equipped with parallel compression
PC+LEJ	Transcritical R744 supermarket refrigeration system relying on direct space heating and cooling and equipped with parallel compression and MT and LT overfed evaporators
PC+MEJ	Transcritical R744 supermarket refrigeration system relying on direct space heating and cooling and equipped with parallel compression and multi-ejector block
R134a-CS	R134a/R744 cascade supermarket refrigeration system
R448A-CS	R448A/R744 cascade supermarket refrigeration system
R513A-CS	R513A/R744 cascade supermarket refrigeration system
RC	Liquid receiver
RHP	Reversible heat pump unit
SH	Degree of superheating [K]
t	Temperature [°C]
T	Temperature [K]
TEWI	Total Equivalent Warming Impact [$\text{ton}_{\text{CO}_2, \text{equivalent}}$]
tot	Total
VB	Vapour by-pass valve
VEJ	Vapour ejectors
$\dot{W}$	Power input [kW]
x	Quality of refrigerant
3W-1	3-way valve located downstream of the medium temperature compressor rack of the investigated transcritical R744 supermarket refrigeration systems
3W-2	3-way valve located upstream of the air-cooled R744 gas cooler/condenser of the investigated transcritical R744 supermarket refrigeration systems

Greek symbols

α	Recycling factor [%]
β	Indirect emission factor [$\text{kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kWh}^{-1}$]
Δ	Difference
η	Efficiency [-]

1. Introduction

The ongoing transition to a low-carbon economy cannot be performed without the adoption of energy-efficient supermarket heating, cooling and refrigeration solutions. In fact, large refrigeration systems such as those in supermarkets, are highly energy-intensive and accountable for about 40% of total electricity use in a food retail store (Energy Star, 2003). Since these units operate 24/7, year-round, identifying ways to improve energy efficiency will result in a meaningful environmental impact reduction as well as in substantial cost saving. The energy requirements of commercial refrigeration applications are further aggravated by the need for heating, ventilation and air conditioning (HVAC), which is responsible for approximately 30% of total electricity demand (Energy Star, 2003). In addition, the use of low Global Warming Potential (GWP) working fluids in supermarket heating, cooling and refrigeration units has become mandatory due to the implementation of the EU F-Gas Regulation 517/2014 (European Commission, 2014). This legislative act is currently in full force and targets a HFC phase down by about 80% by 2030, leading low-GWP refrigerants to gain substantial momentum in the commercial refrigeration sector. Also, a $\text{GWP}_{100 \text{ years}} \cdot \text{kg}_{\text{CO}_2, \text{equivalent}}^{-1} \cdot \text{kg}_{\text{refrigerant}}^{-1}$ limit of 1500 $\text{kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$ to the refrigerant employed in multipack centralized refrigeration plants as well as of 1500 $\text{kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$ to that used in the primary circuit of cascade/indirect arrangements with rated capacity more than 40 kW were imposed since 2022 (European Commission, 2014). As a consequence, the use of today's most employed refrigerant in commercial refrigeration systems, i.e. R404A ($\text{GWP}_{100 \text{ years}} = 3700 \text{ kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$), will no longer be allowed. Additionally, the adoption of HFCs is being discouraged on a global basis too. In fact, under the Kigali Amendment to the Montreal Protocol, which entered into force in January 2019, both developed and developing countries were committed to phasing down HFCs by more than 80% over the next 30 years. As a consequence of the aforementioned measures, among others, R410A ($\text{GWP}_{100 \text{ years}} = 1924 \text{ kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$), being the predominant refrigerant choice for supermarket space heating and cooling, is also under pressure. The global commitment to climate-friendly refrigerants has raised questions of what the best alternatives to both R404A and R410A are in food retail stores. This holds particularly true for high ambient temperature countries, where the climate context is a considerable challenge for refrigeration and air conditioning (AC) systems. Without actions, in fact, cooling emissions could double by 2030 (Isaac and van Vuuren, 2009). Besides the direct contribution to global warming and energetically fit, safety (refrigerant flammability and toxicity) is the major issue for approval or feasibility of supermarket refrigeration systems. Indirect arrangements allow the use of flammable/toxic working fluids, such as R290 ($\text{GWP}_{100 \text{ years}} = 3 \text{ kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$, non-toxic, highly flammable) and R1234ze(E) ($\text{GWP}_{100 \text{ years}} < 1 \text{ kg}_{\text{CO}_2, \text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$, non-toxic, slightly flammable) in the high temperature circuit (HTC), since these are completely confined in the machinery room. As for the medium temperature (MTC) and the low temperature circuit (LTC), CO₂ is generally employed, as it is one of the most appropriate secondary coolants (Inlow and Groll, 1996; Hesse, 1996; Bansal, 2012). Gullo et al. (2018c) and Sánchez et al. (2019) showed that R1234ze(E)/CO₂ indirect units are more energetically beneficial than R290/CO₂ indirect systems in warm countries, while Gullo and Cortella (2016) evaluated that R1234ze(E)/CO₂ indirect solutions are appropriate for hot locations (e.g. Phoenix, USA). The study by Purohit et al. (2017) brought to light that these arrangements perform better than transcritical R744 systems with both parallel compression and dedicated mechanical subcooling at external temperatures above 23 °C. The 24-hour experimental tests implemented by Sánchez et al. (2018) revealed that the conversion of a R134a unit into a R1234ze(E)-based indirect arrangement implies an increase in total energy consumption by 27.2%. The assessment was carried out by considering a medium temperature (MT) application and three different heat rejection levels (i.e. 43.6 °C, 32.8 °C, 23.3 °C). Llopis et al. (2018) carried out a 24-hour experimental assessment proving that a R1234ze(E)/CO₂ indirect unit presents from 6.9% to 8.2% higher energy consumption

than a R134a/CO₂ cascade arrangement at the selected heat rejection levels (i.e. 18.8 °C, 31.6 °C, 42.2 °C).

However, the safety concerns caused by the use of flammable/toxic working fluids have promoted interest in cascade solutions, in which various non-flammable as well as non-toxic refrigerants complying with the EU F-Gas Regulation 517/2014 can be utilized in their primary circuit (e.g. R134a, R448A, R450A, R513A). R134a/CO₂ cascade solutions are the most preferred HFC-based substitutes for R404A supermarket refrigeration units in warm climates. Sanz-Kock et al. (2014) experimentally found that a R134a/CO₂ cascade arrangement achieves Coefficients of Performance (COPs) between 1.05 and 1.65 at evaporator temperatures between -40 °C and -30 °C and condenser temperatures between 30 °C and 50 °C. According to Souza et al. (2015) and Sanz-Kock et al. (2014), the COPs of such a solution go up with increase in CO₂ compressor operating frequency. Recently R448A as well as two eco-friendlier R134a-like alternatives (i.e. R450A and R513A) have been launched onto the market. R448A is a non-azeotropic mixture of R32 (26%), R125 (26%), R1234yf (20%), R134a (21%) and R1234ze(E) (7%) having a GWP_{100 years} value of 1235 kg_{CO₂,equivalent} · kg_{refrigerant}⁻¹. The theoretical investigation by Beshr et al. (2015) recommended the use of a R448A/R744 cascade solution as an appropriate alternative to a multiplex direct system in Chicago (USA). Makhnatch et al. (2018) suggested the replacement of R134a with R513A and R450A in cascade units for commercial refrigeration applications. R513A is an azeotropic mixture consisting of R134a (44%) and R1234ze(E) (56%), leading to a GWP_{100 years} value of 573 kg_{CO₂,equivalent} · kg_{refrigerant}⁻¹. The performance of the R450A-based system was not investigated in this work, as it was already recently showed that a R450A/CO₂ cascade unit cannot outperform a R513A/CO₂ cascade solution in warm climates (Gullo et al., 2018c). The results obtained by Catalán-Gil et al. (2018) revealed that a R513A/CO₂ cascade system is preferable over R513A/R744 cascade booster solutions in locations with an average yearly temperature (AYT) above 13 °C (e.g. Barcelona, Spain). Carbon dioxide as the sole working fluid (R744) can be used in transcritical supermarket refrigeration systems. R744 is an ultra-low GWP (GWP_{100 years} = 1 kg_{CO₂,equivalent} · kg_{refrigerant}⁻¹) as well as low-cost refrigerant, featuring beneficial thermo-physical characteristics (Kim et al., 2004). Also, this working fluid will never be vulnerable to future regulatory tightening, being non-flammable and non-toxic. Northern and Central Europe have been particularly fertile ground for basic transcritical R744 refrigeration systems thanks to their favourable energy (and environmental) performance compared to HFC-based solutions (Sawalha, 2008b; Sawalha et al., 2017; Finckh et al., 2011). However, under the advent of the EU F-Gas Regulation 517/2014, transcritical R744 supermarket refrigeration solutions have benefited from a number of recent design enhancements suited to warm climate contexts (Gullo et al., 2018a). Consequently, these have led “CO₂ only” units to be on the rise in hot areas too (Blust et al., 2018; Singh et al., 2018), being R744 put at the centre of food retailers’ strategy worldwide. Parallel compression (Gullo et al., 2016b) represents the first step towards the use of such units in high ambient temperature locations. Its adoption offers an energy consumption being 5% lower over a conventional R744 unit (Tsamos et al., 2017b), 4.3% lower compared to a R404A unit (Gullo et al., 2016d) and similar to that of a R134a/R744 cascade system (Gullo et al., 2016a) in Athens (Greece). Overfed evaporators allow the evaporators of “CO₂ only” systems to perform at a higher operating temperature compared to conventional heat exchangers (Minetto et al., 2014; Kriezi et al., 2018). Gullo et al. (2016c) estimated that the implementation of this measure permits energy savings between 4.4% and 9.6% in relation to a R404A-based solution in warm climates. Compared to the same system, the combination of parallel compression and MT overfed evaporators leads to a decrease in yearly energy consumption by 12.4% in Athens. Further energy savings can be achieved by including two-phase ejectors for vapour pre-compression, i.e. by adopting the multi-ejector concept (Hafner et al., 2014a). A multi-ejector based solution allows reducing the annual electricity by from about 20% to 25% compared to a R404A unit in Athens (Gullo et al., 2017b). Madsen and Kriezi (2018) recently showed that the adoption of vapour ejectors with high pressure lift on the part of parallel compression-based R744 systems is economically favourable for supermarkets located in cities with AYT above 15 °C and having refrigeration loads over 75 kW. Pisano (2018) claimed that

a R744 unit relying on multi-ejector concept presents an energy saving by 27.9% as well as a return of investments of 2 years in comparison with a basic R744 booster system in a food retail store located in Southern Italy. Gullo et al. (2019a) recently showed that a R744 multi-ejector enhanced parallel compression system offers about 39% less total inefficiencies over a typical booster unit at the outdoor temperature of 40 °C. Also, further energy benefits could be promoted by adopting cold thermal energy storages (Manescu et al., 2017; Polzot et al., 2016a; Karampour et al., 2019) as well as enhancing the gas cooler performance (Ge et al., 2015; Tsamos et al., 2017a), among others (Gullo et al., 2018a; Karampour and Sawalha, 2018). Furthermore, heat recapture opportunity offered by transcritical R744 systems is considerable (Polzot et al., 2016a) thanks to the high grade of available heat. This permits further reducing their total energy consumption (Polzot et al., 2016b; Sawalha, 2013), enhancing their carbon footprint (Ge and Tassou, 2014) as well as attaining satisfactory payback times (Reinholdt and Madsen, 2010). Being capable of significantly reducing the total investment, maintenance and running costs (Hafner et al., 2014b), the fully integrated (or all-in-one) “CO₂ only” supermarket refrigeration systems are in the limelight (Karampour and Sawalha, 2015). Additional advantageous offered by such solutions are compactness (Karampour and Sawalha, 2017), greater ease in communications among all various people in charge of service and operation of the various units (Karampour and Sawalha, 2017) as well as no need to find an appropriate alternative to R410A. Fully integrated solutions, in fact, are tailor-made and versatile units encompassing refrigeration, AC, domestic heating water (DHW) and heating in one individual unit (Hafner et al., 2015). To the best of the author’s knowledge, only 3 works on fully integrated units (i.e. considering both heating and AC) are currently available. The theoretical results obtained by Karampour and Sawalha (2015) revealed that a parallel compression-based unit is a suitable eco-friendly solution for moderate/cold climates. At a later time, the authors confirmed this conclusion with the aid of field measurements (Karampour and Sawalha, 2017). Gullo et al. (2018b) recently estimated that a “CO₂ only” system equipped with parallel compression and both MT and low temperature (LT) overfed evaporators requires from 20.9% to 0.9% less electricity than HFC-based units in locations having an AYT between 15.4 °C and 21 °C. However, the use of water as the energy carrier between the refrigerating unit and the HVAC installation implies many drawbacks, e.g. need for many heat exchangers, significant investment cost, a further energy transfer level (i.e. significant penalization in energy performance), corrosiveness (Hafner, 2017). Considering the advantageous properties of R744, recently Girotto (2016) recommended the use of this working fluid in direct cooling and heating fan coils and air curtains installed inside the building. The removal of the secondary fluid penalisation along with the simultaneous adoption of one or more of the aforementioned measures (e.g. parallel compression, two-phase ejectors) are expected to provide enormous energy and environmental benefits (Gullo et al., 2018a), even in hot climate contexts. According to Hafner (2017), direct space cooling and heating allows also decreasing the cost and the installation time for the HVAC equipment.

Despite the considerable potential of “CO₂ only” solutions relying on direct space heating and cooling, no works on this topic have been carried out so far, to the best of the author’s knowledge. The present investigation is then intended to bridge this knowledge gap by exhaustively evaluating the energy and environmental performance of three transcritical R744 supermarket refrigeration systems with direct space cooling and heating, i.e. a unit relying on parallel compression, a system combining parallel compression and overfed evaporators and a solution equipped with multi-ejector block. For the multi-ejector based solution, three scenarios involving two different temperatures of the AC evaporators as well as two different values of the approach temperature of the gas cooler (i.e. difference between the temperature of the refrigerant at the outlet and that of the air at the inlet of the heat exchanger) have also been investigated. The performance of these solutions has been compared to that of various HFC-based systems, i.e. a R404A solution (“old” baseline), a R134a/R744 cascade arrangement (“current” baseline), a R448A/R744 cascade unit, a R513A/R744 cascade system and a R1234ze(E)/R744 indirect arrangement. For the solution using R404A, a R410A reversible heat pump unit (RHP) for heating&AC purposes has been adopted, whereas a R1234ze(E) RHP has been selected for the cascade and indirect arrangements. To consider realistic operating conditions, these

have been taken from the field measurements available in the literature (Karampour and Sawalha, 2017). Also, 7 warm locations (AYTs between 15.4 °C and 21 °C) have been identified, i.e. Barcelona (Spain), Lisbon (Portugal), Athens (Greece), Seville (Spain), Tunis (Tunisia), Palermo (Italy) and Tenerife (Spain). Finally, 5 hot cities, i.e. Guangzhou (China), Phoenix (USA), Rio de Janeiro (Brazil), New Delhi (India) and Mombasa (Kenya) (AYTs between 22.3 °C and 25.9 °C), have also been taken into account. In addition, Sydney (Australia) (AYT of 17.9 °C, i.e. warm climate) has been considered in order to include a location from Oceania too. In Section 2 the investigated systems are described in detail, while the studied operation conditions, the evaluated simulation models as well as the implemented assessments are presented in Section 3. The outcomes obtained are summarized and discussed in Section 4. At last, the main conclusions are stated in Section 5.

2. Description of the investigated systems

2.1. R404A direct expansion supermarket refrigeration system and R410A reversible heat pump unit for heating&AC (DXS+R410A RHP) (“old” baseline)

Two R404A direct expansion supermarket systems (DXS) to respectively satisfy the MT and LT refrigeration load and separately performing with a R410A reversible heat pump unit (R410A RHP) for heating&AC purposes were selected as the “old” baseline (DXS+R410A RHP). Despite the several legislative acts in favour of the environment preservation in force, DXS+R410A RHP is still the most widely installed solution in supermarket applications worldwide.

2.2. Cascade supermarket refrigeration arrangements and R1234ze(E) reversible heat pump unit for heating&AC (CS+R1234ze(E) RHP)

Cascade refrigeration arrangements are characterized by two temperature circuits (i.e. LTC and HTC) (Figure 1a). The evaporator of HTC, in which in this investigation R134a (R134a-CS), R513A (R513A-CS) or R448A (R448A-CS) flows, and the condenser of LTH, in which in this study R744 is employed, are thermally coupled through the cascade condenser (CC). As regards supermarket applications, many end-users from high ambient temperature countries are familiar with R134a/R744 cascade systems, leading these solutions to be the most preferred HFC-based alternative to R404A units. Therefore, such a solution was selected as the “current” baseline. The considered cascade units were evaluated to separately operate with a R1234ze(E) reversible heat pump unit (R1234ze(E) RHP) for heating&AC purposes (R134a-CS+R1234ze(E) RHP, R448A-CS+R1234ze(E) RHP and R513A-CS+R1234ze(E) RHP)

2.3. R1234ze(E)/R744 indirect supermarket refrigeration arrangement and R1234ze(E) reversible heat pump unit for heating&AC (IND+R1234ze(E) RHP)

The most energetically beneficial indirect supermarket refrigeration arrangement (Gullo and Cortella, 2016) was considered. Such a solution (IND) consists of three circuits (Figure 1b), i.e. HTC, MTC and LTC. The cascade condenser (CC) allows HTC to be thermally coupled with both MTC and LTC, being employed as an evaporator for the former and as a condenser for the latter. In this work, R1234ze(E) was selected as the refrigerant for HTC. As regards both the MTC and the LTC, R744 was adopted as the working fluid. Also, MT and LT evaporators were supposed to operate in flooded regime in the evaluated indirect system, i.e. these heat exchangers perform with no degrees of superheating and thus at a higher operating temperature in comparison with dry-expansion evaporators. The evaluated refrigerating plant was supposed to separately perform with a R1234ze(E) reversible heat pump unit (R1234ze(E) RHP) aimed at heating&AC purposes (IND+R1234ze(E) RHP).

2.4. Fully integrated transcritical R744 supermarket refrigeration systems

This study was based on the investigation of various fully integrated transcritical R744 supermarket refrigeration systems, all relying on direct space heating and cooling:

- a solution employing parallel compression (PC) (see Subsection 2.4.1);
- a unit presenting parallel compression and MT and LT overfed evaporators (with the aid of liquid ejectors, indicated as LEJ) (PC+LEJ) (see Subsection 2.4.2);
- a multi-ejector based system, i.e. a solution having parallel compression, MT and LT overfed evaporators (with the aid of liquid ejectors, indicated as LEJ) and two-phase ejectors for vapour pre-compression (see Subsection 2.4.3). As for this unit, three scenarios based on different approach temperatures of the gas cooler as well as different AC evaporating temperatures were considered (i.e. PC+MEJ, PC+MEJ_Warm_Climates, PC+MEJ_Hot_Climates).

2.4.1. Fully integrated transcritical R744 supermarket refrigeration system with parallel compression (PC)

The schematic of the transcritical R744 supermarket refrigeration solution outfitted with parallel compression (PC) is showed in Figure 2a (Giroto, 2016). The heating and AC modes can be explained with the aid of Figure 2b and Figure 2c (Giroto, 2016), respectively. In heating mode (Figure 2a and Figure 2b), the refrigerant is drawn by the MT compressors (MT compr), leading to an increase in its pressure from medium (MP) to high pressure (HP). Thus, the 3-way valve indicated as 3-W1 leads R744 to the rooftop units (Figure 2b) in which the heat is directly rejected into the supermarket to satisfy its heating need. As suggested by Sawalha (2013), a 3-way valve (3-W2) needs to be installed to allow possibly bypassing the gas cooler/condenser (GC), depending on the running mode. Consequently, the system can simultaneously offer the highest energy saving possible and the satisfaction of the heating reclaim. Since the energy benefits related to parallel compressors (AUX) can be mainly appreciate as AC mode occurs (Karampour and Sawalha, 2018), AUX are replaced with a vapour by-pass valve (VB) in heating mode. In AC mode (Figure 2a and Figure 2c), the working fluid leaving MT compr is forced by 3-W1 to take the opposite path than in heating mode, whereas 3-W2 leads R744 to GC. At this point, the R744 mass flow rate is split into 2 parts: (i) the expansion devices located upstream of the fan coils and air curtains guarantee a sufficient amount of refrigerant to the units, which are operating as evaporators (Hafner, 2017); (ii) the remaining part is isenthalpically expanded by the high pressure expansion valve (HP exp). Finally, the aforementioned 2 streams reunite before entering the liquid receiver (RC). Furthermore, such a solution can take advantage of parallel compressors in AC mode and, in particular, at outdoor temperatures greater than or equal to 13 °C (Karampour and Sawalha, 2018). If neither AC nor heating is required, CO₂ coming out of MT compr is addressed towards the same path as in AC mode by 3W-1 and then towards GC by 3W-2, before being isenthalpically expanded by HP exp. Furthermore, it is worth noticing that all the investigated transcritical R744 solutions as well as the cascade/indirect arrangements featured the presence of an internal heat exchanger (IHX) to “protect” LT compr as well as that of a de-superheater (DS) downstream of LT compr.

2.4.2. Fully integrated transcritical R744 supermarket refrigeration system with parallel compression and MT and LT overfed evaporators (PC+LEJ)

The simplified schematics of the transcritical R744 refrigeration system with parallel compression and MT and LT overfed evaporators (PC+LEJ) in heating and air conditioning mode are presented in Figure 3a and Figure 3b, respectively. Figure 2b and Figure 2c can be utilized to represent their simplified thermodynamic cycle too. Compared to the previous solution, it is possible to notice two main differences: (i) the presence of an additional receiver (RC(MP)); (ii) the replacement of HP exp with LEJ. As experimentally showed by Minetto et al. (2014), LEJ can be suitably used for pumping some liquid from the receiver at MP (RC(MP)) to that at the intermediate pressure (RC(IP)) and,

simultaneously, providing the appropriate high pressure. As a result of the presence of LEJ, the evaporation process in the MT evaporators (MT evap) features the disappearance of the superheating region, enabling an increase in MT (i.e. overfed evaporators), similarly to flooded evaporators. Besides “protecting” LT compr, the adoption of IHX in this solution allows the LT evaporators (LT evap) to achieve the overfeeding conditions too (Minetto et al., 2015). As regards RC(MP), this component is employed for accumulating the liquid coming out of MT evap being later pumped by LEJ as well as for forcing MT compr to draw only vapour (Gullo et al., 2019b).

2.4.3. Fully integrated transcritical R744 supermarket refrigeration system with parallel compression and multi-ejector block (PC+MEJ, PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates)

The simplified schematics of the transcritical R744 refrigeration system equipped with parallel compression, overfed evaporators (i.e. liquid ejectors) and vapour ejectors (PC+MEJ, PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates) in heating and air conditioning mode are showed in Figure 4a and Figure 4b, respectively. Their simplified thermodynamic cycle can be depicted with the aid of Figure 2b and Figure 2c. This solution relies on a multi-ejector block (MEJ) hosting various fixed geometry ejector cartridges of different size and settled in parallel (Hafner et al., 2014a), i.e. generally 2 LEJ and 4-6 vapour ejectors (VEJ) (Gullo et al., 2019b). The refrigeration load along with the high pressure control are constantly and effectively satisfied by changing the combination of VEJ (Banasiak et al., 2015).

The three considered scenarios involving the multi-ejector based systems presented differences in terms of the approach temperature of GC as well as of the AC evaporating temperature (see Subsection 3.2). Two values of the approach temperature of GC were assessed:

- i. 5 K in any running mode, which was selected in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017);
- ii. the previously mentioned value was reduced down to 2 K in transcritical regime, which was employed being typical for installations in high ambient temperature countries (Gullo et al., 2016a; Purohit et al., 2017).

As regards the AC evaporating temperature, two different values were evaluated:

- i. 8 °C, as suggested by Hafner et al. (2015);
- ii. 12.6 °C, as recommended by Kvalsvik et al. (2017) in the Indian climate context (i.e. hot climates).

The influence of both the approach temperature of GC and the AC evaporating temperature on the energy and environmental performance was investigated only for the ones based on the multi-ejector concept, being these the most promising solutions (Gullo et al., 2018a).

3. Investigated running modes, simulation models and implemented evaluations

3.1. Investigated running modes

The assumptions made in common among all the investigated solutions are listed in Table 1. An average-size supermarket defined in accordance with the study by Karampour and Sawalha (2017) was investigated. This work was strongly based on field measurements from a real supermarket.

3.2. Simulation models

The assumptions made to carry out the implemented simulation models of the all the HFC-based solutions are summarized in Table 2, whereas the ones related to the studied transcritical R744 units are listed in Table 3. No degree of subcooling as well as no degree of external superheating for none

of the investigated solutions were assumed (Karampour and Sawalha, 2017). Also, in this study the DHW reclaim was not considered, being negligible in comparison with the other thermal needs (Karampour and Sawalha, 2017). Furthermore, all the components were assumed to be well-insulated (Gullo et al., 2017a) as well as pressure drop was neglected (Gullo et al., 2017a). All the simulation models were carried out with the aid of Engineering Equation Solver (EES) (F-Chart Software, 2019a) and considering steady running modes.

As presented in Table 2, a minimum condensing temperature ($t_{\text{minimum condensing}}$) of 20 °C and an approach temperature of CD ($\Delta T_{\text{approach,CD}}$) of 10 K were chosen for DXS, as these values are typical for such “old” solutions (Karampour and Sawalha, 2017). It is worth remarking that the condensing temperature for all the cascade and indirect solutions as well as for the RHPs operating in AC mode was computed via either Eq. (1) or Eq. (2):

$$\text{if } t_{\text{minimum condensing}} \geq (\Delta T_{\text{approach,CD}} + t_{\text{outdoor}}): t_{\text{condensing}} = t_{\text{minimum condensing}} \quad (1)$$

$$\text{if } t_{\text{minimum condensing}} < (\Delta T_{\text{approach,CD}} + t_{\text{outdoor}}): t_{\text{condensing}} = \Delta T_{\text{approach,CD}} + t_{\text{outdoor}} \quad (2)$$

However, more recent alternatives, e.g. R134a-CS, R448A-CS, R513A-CS, feature lower values of both $t_{\text{minimum condensing}}$ and $\Delta T_{\text{approach,CD}}$, as showed in Table 2. All the evaluated systems were assumed to use semi-hermetic reciprocating compressors, whose performance was computed with the aid of the correlations summed up in Table 4 and derived from some manufacturers’ software. All the recommended technological constraints (e.g. minimum suction pressure) were respected. As for IND, the suction pressure for HTC compr was taken to correspond to the saturation temperature ($t_{\text{evaporating,HTC}}$ in Eq. (3)) of -15 °C, i.e. equal to the minimum suction pressure allowed by the used manufacturer’s software (Fracold, 2019). This value simultaneously permitted: (i) satisfying all the suggested technological constraints of HTC compr (Fracold, 2019); (ii) reaching a condensing temperature of R744 flowing within CC (t_{MT} in Eq. (3)) equal to -4.3 °C (see Table 1); (iii) reaching a pinch point temperature (i.e. minimum temperature difference between the working fluids flowing in CC) of CC ($\Delta T_{\text{pinch point,CC}}$ in Eq. (3)) of 5 K (Karampour and Sawalha, 2018); (iv) reaching a reasonable degree of superheating ($\Delta T_{\text{SH,HTC evap}}$ in Eq. (3)) in CC, being (Eq. (3)):

$$t_{\text{evaporating,HTC}} = t_{\text{MT}} - \Delta T_{\text{pinch point,CC}} - \Delta T_{\text{SH,HTC evap}} \quad (3)$$

At the suction pressure coinciding with -15 °C, the minimum discharge pressure corresponding to the saturation temperature of 20 °C was suggested by the used manufacturers’ software (Fracold, 2019). As far as the transcritical R744 systems are concerned, it is well-known that an optimal high pressure with respect to the temperature of R744 leaving GC needs to be evaluated to minimize their total power input in transcritical regime (i.e. at $t_{\text{outdoor}} > 25$ °C for PC, PC+LEJ and PC+MEJ and at $t_{\text{outdoor}} > 28$ °C for PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates) (Lorentzen, 1994; Sawalha, 2008a; Ge and Tassou, 2011). This was evaluated by implementing the Golden Section search Method (F-Chart Software, 2019b) in EES. Also, the control technique in heating mode recommended by Sawalha (2013) was implemented in this study, i.e.:

- at the beginning, GC was supposed to operate at fully capacity to reach as large degree of subcooling as possible;
- then the high pressure was fixed to a maxim optimum value and the capacity of GC was supposed to be reduced by decreasing the speed of fans, thus these were considered to be switched off and ultimately GC was supposed to be by-passed with the aid of 3W-2.

3.3. Implemented evaluations

3.3.1. Energy evaluations

The annual energy consumption for each of the considered systems was assessed by multiplying the bin hours for the evaluated temperature bin by the corresponding value of power input and thus adding up the computed values of energy of each bin. Firstly, this was calculated for 7 warm cities (AYTs between 15.4 °C and 21 °C) having a maximum outdoor temperature of 41 °C. The temperature bins for these locations, i.e. Barcelona, Lisbon, Athens, Seville, Tunis, Palermo and Tenerife, are presented in Figure 5a. Two multi-ejector based solutions were considered: PC+MEJ, and PC+MEJ_Warm_Climates. At a later time, 5 hot locations (AYTs between 22.3 °C and 25.9 °C) featuring a maximum external temperature of 46 °C were also considered. The temperature bins for these cities, i.e. Guangzhou, Phoenix, Rio de Janeiro, New Delhi and Mombasa, are showed in Figure 5b. Finally, Sydney (AYT of 17.9 °C, i.e. warm weather) was also taken into account to encompass a city from Oceania as well (Figure 5b). Three multi-ejector based solutions were taken into account: PC+MEJ, PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates.

3.3.2. Total equivalent warming impact assessment

The computation of the total equivalent warming impact (TEWI) allows evaluating the total carbon footprint of the studied solutions (AIRAH, 2012), being equal to (Eq. 4):

$$TEWI = TEWI_{indirect} + TEWI_{direct} \quad (4)$$

The indirect contribution to global warming (i.e. $TEWI_{indirect}$) of the evaluated units (calculated via Eq. 5) is caused by the combustion of fossil fuels to generate the electricity necessary to run the refrigeration and reversible heat pump systems. The direct environmental impact (i.e. $TEWI_{direct}$) of these units (calculated via Eq. 6) is owing to the refrigerant leaks into the atmosphere.

$$TEWI_{indirect} = E \cdot \beta \cdot n \quad (5)$$

$$TEWI_{direct} = GWP \cdot L \cdot n + GWP \cdot m \cdot (1 - \alpha) \quad (6)$$

In this work, the contribution to $TEWI_{direct}$ on the part of the selected ultra-low GWP refrigerants, i.e. R744 and R1234ze(E), was neglected (Gullo and Cortella, 2016). As a result, $TEWI_{direct}$ of IND, PC, PC+LEJ, PC+MEJ, PC+MEJ_Warm_Climates, PC+MEJ_Hot_Climates was not calculated.

4. Results and discussion

4.1. Comparison in warm climates

4.1.1. Comparison in terms of annual energy consumption

The annual energy consumption, split into its refrigeration, heating and AC contribution, of all the investigated solutions for the selected warm cities is listed in Table 6. As regards the “CO₂ only” systems, the electricity demand related to heating and AC was calculated as a difference between that required by the solutions described in Subsection 2.4 and the one of the corresponding unit satisfying neither heating nor AC need. Also, the resulting differences in energy consumption compared to the “old” (i.e. DXS+R410A RHP) and the “current” baseline (i.e. R134a-CS+R1234ze(E)) are showed in Figure 6a and Figure 6b, respectively.

Taking DXS+ R410A RHP into account (Figure 6a), in the evaluated warm locations it was found that:

- R134a-CS+R1234ze(E) RHP and R448A-CS+R1234ze(E) RHP feature energy savings from 2.4% in Barcelona (AYT = 15.4 °C) to 7.1% in Tenerife (AYT = 21.0 °C), whereas R513A-CS+R1234ze(E) RHP presents similar energy performance;
- IND and PC are not suitable replacements (from an energy viewpoint). In fact, IND needs from 3.7% in Tenerife (AYT = 21.0 °C) to 6.9% in Barcelona (AYT = 15.4 °C) more electricity. As regards PC, this solution requires from 1.5% in Lisbon (AYT = 16.4 °C) to 5.8% in Seville (AYT = 18.3 °C) more electricity, being this about equal in Barcelona (AYT = 15.4 °C);
- the overfeeding of the evaporators of PC, i.e. PC+LEJ, leads to a decrease in annual energy consumption between 4.4% in Seville (AYT = 18.3 °C) and 10.3% in Barcelona (AYT = 15.4 °C) and equal to 7.4% in Tenerife (AYT = 21.0 °C);
- the adoption of PC+MEJ permits annual energy benefits ranging from 8.0% in Seville (AYT = 18.3 °C) to 11.4% in Barcelona (AYT = 15.4 °C) and equal to 9.5% in Tenerife (AYT = 21.0 °C);
- PC+MEJ_Warm_Climates enables achieving energy savings between 25.0% in Seville (AYT = 18.3 °C) and 32.1% in Tenerife (AYT = 21.0 °C) and equal to 27.7% in Barcelona (AYT = 15.4 °C).

As showed in Figure 6b, R448A-CS+R1234ze(E) RHP had to be preferred to R134a-CS+R1234ze(E) RHP, as the former required from 2.1% in Barcelona (AYT = 15.4 °C) and 2.6% in Tenerife (AYT = 21.0 °C) less electricity to be run. Also, R513A-CS+R1234ze(E) RHP, IND and PC were not energetically appropriate substitutes for R134a-CS+R1234ze(E) RHP, leading to an increase in annual energy consumption up to 3.3% (in Tenerife, AYT = 21 °C), 9.5% (in Seville, AYT = 18.3 °C) and 10.8% (in Seville, AYT = 18.3 °C), respectively. Finally, PC+MEJ and PC+MEJ_Warm_Climates permitted energy savings from 3.6% in Seville (AYT = 18.3 °C) to 9.2% in Barcelona (AYT = 15.4 °C) and from 21.5% in Seville (AYT = 18.3 °C) to 28.8% in Tenerife (AYT = 21.0 °C), respectively. It is worth remarking that IND was substantially penalized by the low suction pressure of HTC compr caused by their compressor operating envelope. As regards PC, this solution was found not to be energetically competitive with the evaluated baselines due to the large amount of flash-gas generated with rise in outdoor temperature (Gullo et al., 2016a).

Furthermore, Table 6 reveals that R1234ze(E) RHP performed similarly to R410A RHP in heating mode. On the contrary, transcritical R744 units were found to provide a significant energy saving over R410A RHP. It was estimated that PC needs from 1.8% in Barcelona (AYT = 15.4 °C) to 23.7% in Palermo (AYT = 18.9 °C) less electricity than HFC-based RHPs. Also, the electricity demand could be further reduced by from 10.5% in Barcelona (AYT = 15.4 °C) to 36.8% in Palermo (AYT = 18.9 °C) by adopting PC+LEJ, PC+MEJ and PC+MEJ_Warm_Climates. These results were due to the fact that the investigated “CO₂ only” solutions present a higher heat source temperature in relation to RHPs. At last, the outcomes listed in Table 6 brought to light that R1234ze(E) RHP has similar electricity demands to R410A RHP in AC mode as well. As regards PC, the high outdoor temperatures and thus the resulting large amount of flash-gas generated significantly penalized this solution, leading it to energy consumptions between 3.3% in Tenerife (AYT = 21.0 °C) and 11.6% in Seville (AYT = 18.3 °C) higher than those of HFC-based RHPs. However, the overfeeding of the evaporators (i.e. PC+LEJ) permitted reducing by from 1.5% in Seville (AYT = 18.3 °C) to 16.1% in Barcelona (AYT = 15.4 °C) the electricity need in comparison with the HFC-based RHPs, being the energy saving equal up to 12.6% in Tenerife (AYT = 21.0 °C). Also, PC+MEJ and PC+MEJ_Warm_Climates respectively led to energy savings ranging between 7.4% in Seville (AYT = 18.3 °C) to 18.3% in Barcelona (AYT = 15.4 °C) and between 20.0% in Seville (AYT = 18.3 °C) to 28.6% in Barcelona (AYT = 15.4 °C). In particular, a reduction in energy consumption equal up to 15.4% for PC+MEJ and up to 26.9% for PC+MEJ_Warm_Climates was calculated in Tenerife (AYT = 21.0 °C).

4.1.2. Comparison in terms of TEWI

All the outcomes in terms of TEWI related to the selected warm locations are summarized in Table 7, while the differences in TEWI compared to the “old” and “current” baselines are respectively showed with the aid of Figure 7a and Figure 7b. It was found that all the investigated solutions offer better environmental performance than DXS+R410A RHP. The evaluated cascade arrangements could reduce the carbon footprint by from 21.0% in Athens (AYT = 17.9 °C) to 59.4% in Barcelona (AYT = 15.4 °C) as well as by up to 55.2% in Tenerife (AYT = 21.0 °C) over DXS+R410A RHP. It is worth remarking that, although R513A-CS+R1234ze(E) RHP showed the poorest energy performance, it was found to be the best candidate from an environmental viewpoint among the assessed cascade solutions in almost all the selected warm locations. This was due to the fact that TEWI_{direct} can be reduced by about 75% by adopting both R134a-CS+R1234ze(E) RHP and R448A-CS+R1234ze(E) RHP and by around 89% with the aid of R513A-CS+R1234ze(E) RHP compared to the “old” baseline. In relation to R134a-CS+R1234ze(E) RHP, R513A-CS+R1234ze(E) RHP allowed bringing TEWI_{direct} down to roughly 56%. Also, the use of ultra-low GWP refrigerants compensates the unfavourable energy performance of both IND and PC compared to DXS+R410A RHP in all the selected locations. On the other hand, they both were found not to be suitable alternatives to R134a-CS+R1234ze(E) RHP in Athens (AYT = 17.9 °C) owing to its very high indirect emission factor and less high GWP of R134a compared to R404A. The great energy efficiencies along with the use of R744 as the unique working fluid permitted PC+LEJ, PC+MEJ and PC+MEJ_Warm_Climates to be characterized by the lowest environmental impact. In fact, PC+LEJ featured TEWI values from 27.5% in Athens (AYT = 17.9 °C) to 69.9% in Barcelona (AYT = 15.4 °C) lower than DXS+R410A RHP and from 8.2% in Athens (AYT = 17.9 °C) to 38.2% in Barcelona (AYT = 15.4 °C) lower than R134a-CS+R1234ze(E) RHP, respectively. As regards PC+MEJ, this solution was able to bring TEWI down from 30.0% in Athens (AYT = 17.9 °C) to 70.3% in Barcelona (AYT = 15.4 °C) compared to the “old” baseline and from 11.4% in Athens (AYT = 17.9 °C) to 38.9% in Barcelona (AYT = 15.4 °C) compared to the “current” baseline. Finally, TEWI of PC+MEJ_Warm_Climates was calculated to be from 44.0% in Athens (AYT = 17.9 °C) to 75.8% in Barcelona (AYT = 15.4 °C) over DXS+R410A RHP and from 29.1% in Athens (AYT = 17.9 °C) to 50.2% in Barcelona (AYT = 15.4 °C) in relation to R134a-CS+R1234ze(E) RHP.

4.2. Comparison in hot climates

4.2.1. Comparison in terms of annual energy consumption

The annual energy consumption, divided into its refrigeration, heating and AC contribution, of all the investigated systems for the evaluated hot locations is listed in Table 8. Furthermore, the resulting differences in energy consumption in relation to the “old” (i.e. DXS+R410A RHP) and the “current” baseline (i.e. R134a-CS+R1234ze(E)) are depicted in Figure 8a and Figure 8b, respectively.

Considering DXS+ R410A RHP (Figure 8a), the main findings could be summarized as follows:

- R134a-CS+R1234ze(E) RHP enables energy savings from 6.0% in Guangzhou (AYT = 22.3 °C) to 8.0% in New Delhi (AYT = 24.5 °C) and equal to 3.2% in Sydney (AYT = 17.9 °C);
- PC+MEJ needs from 2.0% in Phoenix (AYT = 22.5 °C) to 6.3% less electricity in Rio de Janeiro (AYT = 23.6 °C) and 11.2% less electricity in Sydney (AYT = 17.9 °C);
- PC+MEJ_Warm_Climates features decreases in energy consumption between 18.4% in Phoenix (AYT = 22.5 °C) and 30.3% in Rio de Janeiro (AYT = 23.6 °C) and equal to 30.5% in Sydney (AYT = 17.9 °C);
- PC+MEJ_Hot_Climates accomplishes energy savings ranging from 21.3% in Phoenix (AYT = 22.5 °C) to 33.3% in Rio de Janeiro (AYT = 23.6 °C) and equal to 32.2% in Sydney (AYT = 17.9 °C).

Figure 8b reveals that PC+MEJ was not an appropriate replacement for R134a-CS+R1234ze(E) in the evaluated hot locations, leading to increases in energy consumption up to 6.5% in New Delhi (AYT = 24.5 °C). However, it was also found that PC+MEJ_Warm_Climates and

PC+MEJ_Hot_Climates respectively enable energy savings between 11.3% in Phoenix (AYT = 22.5 °C) and 25.8% in Rio de Janeiro (AYT = 23.6 °C) and between 14.5% in Phoenix (AYT = 22.5 °C) and 28.9% in Rio de Janeiro (AYT = 23.6 °C) compared to the same baseline.

It is noteworthy to highlight that in the comparison in hot climates only the well-established DXS+R410A RHP and R134a-CS+R1234ze(E) were evaluated as HFC-based solutions, being the transitions towards new synthetic refrigerants occurring slowly. Also, the reason why only multi-ejector based solutions were considered was due to their more advantageous energy and environmental performance in warm cities (owing to the very high annual yearly temperatures of the assessed locations) compared to the other “CO₂ only” solutions showed above. However, the author estimated that R448A-CS+R1234ze(E) would have led to energy savings from 8.4% in Guangzhou (AYT = 22.3 °C) to 10.4% in New Delhi (AYT = 24.5 °C) compared to DXS+R410A RHP and from 2.5% in Phoenix (AYT = 22.5 °C) to 2.7% in Mombasa (AYT = 25.9 °C) compared to R134a-CS+R1234ze(E). Also, it was computed that PC+MEJ_Hot_Climates would have permitted reducing the annual electricity demand by from 27.1% in Seville (AYT = 18.3 °C) to 34.5% in Tenerife (AYT = 21.0 °C) over DXS+R410A RHP and from 23.6% in Seville (AYT = 18.3 °C) to 31.3% in Tenerife (AYT = 21.0 °C) in comparison with R134a-CS+R1234ze(E).

In addition, as showed in Table 8, R410A RHP and R1234ze(E) RHP offered similar energy performance in heating mode. As for the multi-ejector based “CO₂ only” systems, it was computed that energy savings up to 30.2% in Guangzhou (AYT = 22.3 °C) can be attained compared to HFC-base units in the evaluated hot locations. It is important to remark that PC+MEJ_Hot_Climates was evaluated to perform in the same way as PC+MEJ and PC+MEJ_Warm_Climates in heating mode, i.e. the intermediate pressure was taken as 42.83 bar in heating mode, being the heating demand very low in the selected cities. The results listed in Table 8 suggest that R410A RHP and R1234ze(E) RHP featured similar annual energy consumption in AC mode too. The adoption of PC+MEJ enabled reducing the electricity demand by 6.3% over R410A RHP and by 5.6% over R1234ze(E) RHP in Guangzhou (AYT = 22.3 °C), by 9.4% compared to R410A RHP and by 9.0% compared to R1234ze(E) RHP in Rio de Janeiro (AYT = 23.6 °C), by 5.2% in relation to R410A RHP and by 4.7% in relation to R1234ze(E) RHP in Mombasa (AYT = 25.9 °C) and by 18.4% over R410A RHP and by 17.9% over R1234ze(E) RHP in Sydney (AYT = 17.9 °C). However, such a solution was found not to be able to outperform the HFC-based RHPs in Phoenix (AYT = 22.5 °C) and New Delhi (AYT = 24.5 °C), leading to increases in energy consumption up to 5.3%. As for PC+MEJ_Warm_Climates, such a system required from 9.7% in Phoenix (AYT = 22.5 °C) to 23.5% in Rio de Janeiro (AYT = 23.6 °C) less electricity compared to HFC-based RHPs, leading to energy savings up to 28.1% in Sydney (AYT = 17.9 °C). Furthermore, PC+MEJ_Hot_Climates was found to bring the annual energy consumption from 22.6% in Phoenix (AYT = 22.5 °C) down to 36.2% in Rio de Janeiro (AYT = 23.6 °C), being this energy saving equal up to 41.9% in Sydney (AYT = 17.9 °C). Finally, it was estimated that PC+MEJ_Hot_Climates would have reduced the electricity demand by from 33.3% in Seville (AYT = 18.3 °C) to 42.5% in Barcelona (AYT = 15.4 °C) compared to the HFC-based RHPs in AC mode.

4.2.2. Comparison in terms of TEWI

All the results associated with the TEWI assessments for the evaluated hot locations are presented in Table 9, while the differences in TEWI over the “old” and “current” baselines are respectively depicted in Figure 9a and Figure 9b. The outcomes obtained suggest that R134a-CS+R1234ze(E) RHP could reduce TEWI by from 24.8% in New Delhi (AYT = 24.5 °C) to 64.8% in Rio de Janeiro (AYT = 23.6 °C) compared to DXS+R410A RHP, being this decrease equal to 32.4% in Sydney (AYT = 17.9 °C). Also, PC+MEJ featured a carbon footprint, being from 26.3% in New Delhi (AYT = 24.5 °C) to 85.2% in Rio de Janeiro (AYT = 23.6 °C) over DXS+R410A RHP and from 2.0% in New Delhi (AYT = 24.5 °C) to 57.9% in Rio de Janeiro (AYT = 23.6 °C) over R134a-CS+R1234ze(E) RHP lower. Its negligible reduction in TEWI in New Delhi compared to the “current” baseline could be justified by considering the very high indirect emission factor and less

high GWP of R134a compared to R404A. Furthermore, the decrement in TEWI for this solution was equal to 46.9% over DXS+R410A RHP and to 21.4% over R134a-CS+R1234ze(E) RHP in Sydney (AYT = 17.9 °C). As regards PC+MEJ_Warm_Climates, this system allowed bringing TEWI from 39.5% in New Delhi (AYT = 24.5 °C) down to 89.0% in Rio de Janeiro (AYT = 23.6 °C) in relation to DXS+R410A RHP and from 19.5% in New Delhi (AYT = 24.5 °C) down to 68.7% in Rio de Janeiro (AYT = 23.6 °C) in relation to R134a-CS+R1234ze(E). This decrease was evaluated to be equal to 58.4% over the “old” baseline and to 38.5% over the “current” baseline in Sydney (AYT = 17.9 °C). More environmental benefits were offered by PC+MEJ_Hot_Climates, which was found to reduce the environmental impact by from 41.9% in New Delhi (AYT = 24.5 °C) to 89.5% in Rio de Janeiro (AYT = 23.6 °C) over DXS+R410A RHP and from 22.7% in New Delhi (AYT = 24.5 °C) to 70.0% in Rio de Janeiro (AYT = 23.6 °C) over R134a-CS+R1234ze(E). As for Sydney (AYT = 17.9 °C), the reduction in TEWI was found to be equal to 59.4% over DXS+R410A RHP and to 40.0% over R134a-CS+R1234ze(E).

In addition, the author estimated that the implementation of PC+MEJ_Hot_Climates in the selected warm locations would have led to reductions in carbon footprint by from 45.5% in Athens (AYT = 17.9 °C) and 76.2% in Barcelona (AYT = 15.4 °C) over DXS+R410A RHP and from 31.0% in Athens (AYT = 17.9 °C) and 51.1% in Barcelona (AYT = 15.4 °C) over R134a-CS+R1234ze(E), respectively. Finally, it was also found that the adoption of R448A-CS+R1234ze(E) in the evaluated hot locations would have permitted decreases in TEWI by from 26.9% in New Delhi (AYT = 24.5 °C) and 66.3% in Rio de Janeiro (AYT = 23.6 °C) over DXS+R410A RHP and from 2.8% in New Delhi (AYT = 24.5 °C) and 4.0% in Rio de Janeiro (AYT = 23.6 °C) over R134a-CS+R1234ze(E), respectively.

5. Conclusions

A severe mechanism to reduce the adoption of HFCs is being put in place worldwide, leading CO₂ to emerge as one of the most promising long-term working fluids in commercial refrigeration sector. State-of-the-art “CO₂ only” refrigerating systems, such as the ones based on the innovative concept of direct space heating and cooling, are expected to contribute to take a giant step forward towards a HFC-free future in supermarket applications. However, to the best of the author’s knowledge, this topic has not been investigated yet. In order to fill this knowledge gap, the energy and environmental performance of various “CO₂ only” refrigeration solutions relying on direct space heating and cooling, i.e. a unit employing parallel compression, a system combining parallel compression and MT and LT overfed evaporators and a solution equipped with multi-ejector block, has been in-depth investigated. As for the multi-ejector based unit, three scenarios based on two different temperatures of the AC evaporators as well as two different values of the approach temperature of the gas cooler have also been considered. The aforementioned performance has been compared with that of several HFC-based system in different warm and hot locations. The HFC-based units taken into account have been a R404A solution (“old” baseline), a R134a/R744 cascade unit (“current” baseline), a R448A/R744 cascade arrangement, a R513A/R744 cascade solution and a R1234ze(E)/R744 indirect arrangement. The R404A solution has been supposed to separately perform along with a R410A reversible heat pump unit for heating&AC purposes, whereas a R1234ze(E) reversible heat pump system has been evaluated for the others. The running modes of the investigated solutions have been derived from the field measurements available in the literature. Furthermore, 7 warm cities (AYTs between 15.4 °C and 21 °C), i.e. Barcelona (Spain), Lisbon (Portugal), Athens (Greece), Seville (Spain), Tunis (Tunisia), Palermo (Italy) and Tenerife (Spain) as well as 5 hot locations, i.e. Guangzhou (China), Phoenix (USA), Rio de Janeiro (Brazil), New Delhi (India) and Mombasa (Kenya) (AYTs between 22.3 °C and 25.9 °C), have been considered.

The results obtained have revealed that “CO₂ only” refrigerating systems relying on both multi-ejector concept and direct space heating and cooling feature reductions in annual energy saving and environmental impact respectively by up to 32.1% and 75.8% compared to the HFC-based solutions

in the selected warm locations. As for the investigated hot cities, energy savings and decreases in carbon footprint respectively by up to 33.3% and 89.5% have been estimated in relation to the HFC-based units. Also, in very high ambient temperature locations, a low value of the approach temperature of the gas cooler is compulsory in transcritical running modes.

The outcomes of the present work will help to potentially put an end to the debate about the most appropriate alternative to R404A (and to R410A) in the selected warm and hot locations and by extension in similar climate contexts in the rest of the world.

To conclude, transcritical R744 supermarket refrigeration systems implementing direct space heating and cooling represent a noteworthy leap forward in the highly performing technology available to fulfil a HFC-free commercial refrigeration sector, even in hot areas (e.g. India). However, it is important to remark that close attention needs to be devoted to the installation of the CO₂ pipes belonging to direct space heating and cooling solutions. Also, the heat exchanger coils require to be designed for dual operation.

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HIGHLIGHTS

- Performance of various “CO₂ only” supermarket refrigerating solutions was evaluated
- Energy and environmental benefits of direct space heating and cooling were assessed
- Several warm and hot locations were considered
- Proposed solutions consume up to 33.3% less electricity than HFC-based units
- Proposed solutions reduce TEWI by up to 89.5% compared to HFC-based units

Table 1 – Assumptions in common among all the investigated solutions.

Boundary condition	Assumed value	Notes/References
MT	-8 °C for DXS, R134a-CS, R513A-CS, R448A-CS, PC	Taken in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)
	-4.3 °C for IND, PC+LEJ, PC+MEJ, PC+MEJ_Warm_Climates, PC+MEJ_Hot_Climates	Karampour and Sawalha (2018)
MT load	Assumed to equal 100 kW at the outdoor temperature of 10 °C and increase linearly to reach 200 kW at the outdoor temperature of 35 °C. At outdoor temperatures below 10 °C, the MT load was kept constant and equal to 100 kW	Karampour and Sawalha (2017)
LT	-32 °C for DXS, R134a-CS, R513A-CS, R448A-CS, PC	Taken in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)
	-29 °C for IND, PC+LEJ, PC+MEJ, PC+MEJ_Warm_Climates, PC+MEJ_Hot_Climates	Karampour and Sawalha (2018)
LT load	Assumed to be constant (Sawalha et al., 2017) and equal to 35 kW	Karampour and Sawalha (2017)

Heating load	Assumed to equal 40 kW at the outdoor temperature of 10 °C and increase linearly up to 190 kW at the outdoor temperature of -20 °C	Karampour and Sawalha (2017)
AC load	Assumed to equal 0 kW at the outdoor temperature of 10 °C and increase linearly to reach 100 kW at the outdoor temperature of 32 °C	Karampour and Sawalha (2017)
Fan power	3% of the heat transfer rate associated with GC and CD	Karampour and Sawalha (2017)
De-superheater (DS)	35 °C	Karampour and Sawalha (2018)
Effectiveness of internal heat exchanger (IHX)	0.65	Llopis et al. (2016)

Table 2 – Assumptions made for the investigated HFC-based solutions.

R404A direct expansion refrigeration systems		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature of CD ($\Delta T_{\text{approach,CD}}$)	10 K	Gullo et al. (2017a)
Minimum condensing temperature of CD ($t_{\text{minimum condensing}}$)	20 °C	Karampour and Sawalha (2017)
Degree of internal (i.e. useful) superheating (ΔT_{SH})	10 K	Taken in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)
Cascade refrigeration arrangements		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature of CD ($\Delta T_{\text{approach,CD}}$)	7 K	Karampour and Sawalha (2018)
Minimum condensing temperature of CD ($t_{\text{minimum condensing}}$)	15 °C	Due to the technological constraints imposed by BITZER (2019)
Pinch point temperature of CC ($\Delta T_{\text{pinch point,CC}}$)	5 K	Karampour and Sawalha (2018)
Degree of internal (i.e. useful) superheating (ΔT_{SH})	10 K	Taken in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)
Indirect refrigeration arrangement		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature of CD ($\Delta T_{\text{approach,CD}}$)	7 K	Karampour and Sawalha (2018)

Minimum condensing temperature of CD ($t_{\text{minimum condensing}}$)	20 °C	Due to the technological constraints imposed by Frascold (2019)
Pinch point temperature of CC ($\Delta T_{\text{pinch point,CC}}$)	5 K	Karampour and Sawalha (2018)
Degree of internal (i.e. useful) superheating ($\Delta T_{\text{SH,HTC evap}}$)	5.7 K for CC	To allow R744 flowing within CC to reach the temperature of -4.3 °C
	0 K for MT and LT flooded evaporators. The operating condition of R1234ze(E) at the outlet of the evaporators was evaluated through the calculation of its quality as follows: $CR_{\text{pump,MT evap}} = \frac{1}{x_{R744,\text{out_MTevap}}} = 1.5$ $CR_{\text{pump,LT evap}} = \frac{1}{x_{R744,\text{out_LTEvap}}} = 2.5$	Gullo and Cortella (2016)
Power input of pumps	$\dot{W}_{\text{pump}} = 0.01 \cdot \dot{W}_{\text{tot,compr}}$	Gullo and Cortella (2016)
Reversible heat pump units (RHPs)		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Heating mode	The condensing temperature ($t_{\text{condensing,CD}}$) was set to 40 °C	Gullo et al. (2017a)
	The approach temperature of EV was assumed as 5 K	Taken in accordance with the field measurements presented by Karampour and Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)

AC mode	The approach temperature of CD ($\Delta T_{\text{approach,CD}}$) was assumed as 10 K	Gullo et al. (2017a)
	The evaporating temperature was taken as 0 °C	Karampour and Sawalha (2017)
Degree of internal (i.e. useful) superheating	10 K	Karampour and Sawalha (2018)

Table 3 – Assumptions made for the investigated transcritical R744 solutions.

Assumptions in common among all the investigated solutions		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Heating mode	The temperature of R744 was assumed to be equal to the sum between the supermarket ambient temperature (i.e. 21 °C) and the selected approach temperature of the rooftop units (i.e. 5 K). Also, the same heat recovery strategy suggested by Sawalha (2013) was adopted	Personal communication with a company specialized in the design, production and installation of such solutions
Parallel compressors (AUX)	They were assumed to be put in operation at $t_{\text{outdoor}} \geq 13 \text{ }^{\circ}\text{C}$ for all the investigated transcritical R744 refrigeration solutions	Karampour and Sawalha (2018)
PC		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature and running mode of GC	5 K for GC and see Heating mode at $t_{\text{outdoor}} \leq 10 \text{ }^{\circ}\text{C}$ (Subcritical regime)	Karampour and Sawalha (2017)
	5 K for GC and $t_{\text{condensing}}$ computed via Eq. (2) at $10 \text{ }^{\circ}\text{C} < t_{\text{outdoor}} \leq 25 \text{ }^{\circ}\text{C}$ (Transition regime)	Karampour and Sawalha (2017)
	5 K for GC and high pressure optimized at $t_{\text{outdoor}} > 25 \text{ }^{\circ}\text{C}$ (Transcritical regime)	Karampour and Sawalha (2017)
Intermediate pressure (i.e. at RC(IP))	Corresponding to the saturation temperature of 8 °C (i.e. 42.83 bar), coinciding with the AC evaporating temperature in AC mode	Hafner et al. (2015)
Degree of internal (i.e. useful) superheating	10 K	Taken in accordance with the field measurements presented by Karampour and

		Sawalha (2017), Sawalha et al. (2015) and Sawalha et al. (2017)
PC+LEJ		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature and running mode of GC	5 K for GC and see Heating mode at $t_{\text{outdoor}} \leq 10$ °C (Subcritical regime)	Karampour and Sawalha (2017)
	5 K for GC and $t_{\text{condensing}}$ computed via Eq. (2) at 10 °C $< t_{\text{outdoor}} \leq 25$ °C (Transition regime)	Karampour and Sawalha (2017)
	5 K for GC and high pressure optimized at $t_{\text{outdoor}} > 25$ °C (Transcritical regime)	Karampour and Sawalha (2017)
Intermediate pressure (i.e. at RC(IP))	Corresponding to the saturation temperature of 8 °C (i.e. 42.83 bar), coinciding with the AC evaporating temperature in AC mode	Hafner et al. (2015)
Degree of internal (i.e. useful) superheating	0 K (i.e. quality of R744 exiting the evaporators = 1)	Karampour and Sawalha (2018)
Liquid ejectors (LEJ)	They were not simulated, i.e. PC+LEJ was evaluated to perform in the same way as PC, but with a higher value of MT (and so was LT thanks to IHX)	<p>This assumption was taken from Gullo et al. (2017a) and can be justified as follows:</p> <ul style="list-style-type: none"> • LEJ do not permanently work (unlike VEJ), but these come into operation as soon as the liquid indicator of RC(MP) exceeds a specific threshold • their primary energy advantage is represented by the increase in MT (Minetto et al., 2014). This result was supposed to be obtained on a yearly basis (Hafner and Banasiak, 2016)

PC+MEJ		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature and running mode of GC	5 K for GC and see Heating mode at $t_{\text{outdoor}} \leq 10$ °C (Subcritical regime)	Karampour and Sawalha (2017)
	5 K for GC and $t_{\text{condensing}}$ computed via Eq. (2) at 10 °C $< t_{\text{outdoor}} \leq 25$ °C (Transition regime)	Karampour and Sawalha (2017)
	5 K for GC and high pressure optimized at $t_{\text{outdoor}} > 25$ °C (Transcritical regime)	Karampour and Sawalha (2017)
Intermediate pressure (i.e. at RC(IP))	Corresponding to the saturation temperature of 8 °C (i.e. 42.83 bar), coinciding with the AC evaporating temperature in AC mode	Hafner et al. (2015)
Degree of internal (i.e. useful) superheating	0 K (i.e. quality of R744 exiting the evaporators = 1)	Karampour and Sawalha (2018)
Liquid ejectors (LEJ) and vapour ejectors (MEJ)	<p>LEJ were not simulated, i.e.:</p> <ul style="list-style-type: none"> • PC+MEJ was evaluated to perform exactly in the same way as PC+LEJ at $t_{\text{outdoor}} \leq 25$ °C • besides the increase in MT (and in LT as well), the energy benefits related to MEJ were assessed too at $t_{\text{outdoor}} > 25$ °C. Their performance was evaluated taking the suggested pressure lift (i.e. difference between AC evaporating pressure and MT evaporating pressure) into account 	<p>The assumption related to LEJ was taken from Gullo et al. (2017a) and can be justified as follows:</p> <ul style="list-style-type: none"> • LEJ do not permanently work (unlike VEJ), but these come into operation as soon as the liquid indicator of RC(MP) exceeds a specific threshold • their primary energy advantage is represented by the increase in MT (Minetto et al., 2014). This result was supposed to be obtained on a yearly basis (Hafner and Banasiak, 2016) <p>The energy advantages associated with VEJ were supposed to be taken at $t_{\text{outdoor}} > 25$ °C, as</p>

		no noteworthy increases in energy efficiency can be experienced below these values (Pardiñas et al., 2018)
PC+MEJ_Warm_Climates		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature and running mode of GC ($\Delta T_{\text{approach,GC}}$)	5 K for GC and see Heating mode at $t_{\text{outdoor}} \leq 10$ °C (Subcritical regime)	Karampour and Sawalha (2017)
	The approach temperature of GC was computed by setting a transition zone between subcritical and transcritical regime at 10 °C $< t_{\text{outdoor}} \leq 28$ °C, i.e.: $\Delta T_{\text{approach,GC}} = -0.1765 \cdot t_{\text{outdoor}} + 6.9412$	As suggested by Cecchinato et al. (2007) and Gullo et al. (2016a), the adoption of a transition zone allows the system to smoothly move from the subcritical to the transcritical regime (and vice versa), i.e. gradually reduce the approach temperature of GC. This assumption was not made for PC, PC+LEJ and PC+MEJ, as the same approach temperature of GC was used in all the running modes
	2 K for GC and high pressure optimized at $t_{\text{outdoor}} > 28$ °C (Transcritical regime)	Gullo et al. (2016a), Purohit et al. (2017)
Intermediate pressure (i.e. at RC(IP))	Corresponding to the saturation temperature of 8 °C (i.e. 42.83 bar), coinciding with the AC evaporating temperature in AC mode	Hafner et al. (2015)
Degree of internal (i.e. useful) superheating	0 K (i.e. quality of R744 exiting the evaporators = 1)	Karampour and Sawalha (2018)

Liquid ejectors (LEJ) and vapour ejectors (MEJ)	<p>LEJ were not simulated, i.e.:</p> <ul style="list-style-type: none"> • PC+MEJ was evaluated to perform exactly in the same way as PC+LEJ at $t_{\text{outdoor}} \leq 25\text{ }^{\circ}\text{C}$ • besides the increase in MT (and in LT as well), the energy benefits related to MEJ were assessed too at $t_{\text{outdoor}} > 25\text{ }^{\circ}\text{C}$. Their performance was evaluated taking the suggested pressure lift (i.e. difference between AC evaporating pressure and MT evaporating pressure) into account • 	<p>The assumption related to LEJ was taken from Gullo et al. (2017a) and can be justified as follows:</p> <ul style="list-style-type: none"> • LEJ do not permanently work (unlike VEJ), but these come into operation as soon as the liquid indicator of RC(MP) exceeds a specific threshold • their primary energy advantage is represented by the increase in MT (Minetto et al., 2014). This result was supposed to be obtained on a yearly basis (Hafner and Banasiak, 2016) <p>The energy advantages associated with VEJ were supposed to be taken at $t_{\text{outdoor}} > 25\text{ }^{\circ}\text{C}$, as no noteworthy increases in energy efficiency can be experienced below these values (Pardiñas et al., 2018)</p>
PC+MEJ_Hot_Climates		
<i>Boundary condition</i>	<i>Assumed value</i>	<i>Notes/References</i>
Approach temperature and running mode of GC ($\Delta T_{\text{approach,GC}}$)	5 K for GC and see Heating mode at $t_{\text{outdoor}} \leq 10\text{ }^{\circ}\text{C}$ (Subcritical regime)	Karampour and Sawalha (2017)
	<p>The approach temperature of GC was computed by setting a transition zone between subcritical and transcritical regime at $10\text{ }^{\circ}\text{C} < t_{\text{outdoor}} \leq 28\text{ }^{\circ}\text{C}$, i.e.:</p> $\Delta T_{\text{approach,GC}} = -0.1765 \cdot t_{\text{outdoor}} + 6.9412$	As suggested by Cecchinato et al. (2007) and Gullo et al. (2016a), the adoption of a transition zone allows the system to smoothly move from the subcritical to the transcritical regime (and vice versa), i.e. gradually reduce the approach temperature of GC. This assumption was not made for PC, PC+LEJ and PC+MEJ, as the same approach temperature of GC was used in any running mode

	2 K for GC and high pressure optimized at $t_{\text{outdoor}} > 28\text{ }^{\circ}\text{C}$ (Transcritical regime)	Gullo et al. (2016a), Purohit et al. (2017)
Intermediate pressure (i.e. at RC(IP))	<ul style="list-style-type: none"> Corresponding to the saturation temperature of $8\text{ }^{\circ}\text{C}$ (i.e. 42.83 bar) in heating mode Corresponding to the saturation temperature of $12.6\text{ }^{\circ}\text{C}$ (i.e. 48.0 bar) in AC mode. The intermediate pressure coincides with the AC evaporating pressure 	Hafner et al. (2015)
Degree of internal (i.e. useful) superheating	0 K (i.e. quality of R744 exiting the evaporators = 1)	Kvalsvik et al. (2017)
Liquid ejectors (LEJ) and vapour ejectors (MEJ)	<p>LEJ were not simulated, i.e.:</p> <ul style="list-style-type: none"> PC+MEJ was evaluated to perform exactly in the same way as PC+LEJ at $t_{\text{outdoor}} \leq 25\text{ }^{\circ}\text{C}$ besides the increase in MT (and in LT as well), the energy benefits related to MEJ were assessed too at $t_{\text{outdoor}} > 25\text{ }^{\circ}\text{C}$. Their performance was evaluated taking the suggested pressure lift (i.e. difference between AC evaporating pressure and MT evaporating pressure) into account 	<p>The assumption related to LEJ was taken from Gullo et al. (2017a) and can be justified as follows:</p> <ul style="list-style-type: none"> LEJ do not permanently work (unlike VEJ), but these come into operation as soon as the liquid indicator of RC(MP) exceeds a specific threshold their primary energy advantage is represented by the increase in MT (Minetto et al., 2014). This result was supposed to be obtained on a yearly basis (Hafner and Banasiak, 2016) <p>The energy advantages associated with VEJ were supposed to be taken at $t_{\text{outdoor}} > 25\text{ }^{\circ}\text{C}$, as no noteworthy increases in energy efficiency can be experienced below these values (Pardiñas et al., 2018)</p>

Table 4 - Global efficiency correlations of all the selected compressors.

Configuration	Global efficiency correlations	Reference
DXS	$\eta_{global,LTC} = -0.0004 \cdot \left(\frac{p_{HP}}{p_{LT}}\right)^2 - 0.0021 \cdot \left(\frac{p_{HP}}{p_{LT}}\right) + 0.6989$ $\eta_{global,MTC} = -0.0075 \cdot \left(\frac{p_{HP}}{p_{MT}}\right)^2 + 0.0652 \cdot \left(\frac{p_{HP}}{p_{MT}}\right) + 0.5609$	Gullo et al. (2017a), BITZER (2019)
R134a-CS	$\eta_{global,R134a} = -0.0053 \cdot \left(\frac{p_{HP,R134a}}{p_{MP,R134a}}\right)^2 + 0.0674 \cdot \left(\frac{p_{HP,R134a}}{p_{MP,R134a}}\right) + 0.4802$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	Gullo et al. (2016c), BITZER (2019)
R448A-CS	$\eta_{global,R448A} = -0.0111 \cdot \left(\frac{p_{HP,R450A}}{p_{MP,R450A}}\right)^2 + 0.1189 \cdot \left(\frac{p_{HP,R450A}}{p_{MP,R450A}}\right) + 0.4262$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	BITZER (2019)
R513A-CS	$\eta_{global,R513A} = -0.0043 \cdot \left(\frac{p_{HP,R513A}}{p_{MP,R513A}}\right)^2 + 0.0533 \cdot \left(\frac{p_{HP,R513A}}{p_{MP,R513A}}\right) + 0.4989$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	Gullo et al. (2018c), BITZER (2019)
IND	$\eta_{global,R1234ze(E)} = -0.0028 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.5305$	Gullo and Cortella (2016), Frascold (2019)

	$\eta_{global,R1234ze(E)_{AUX}} = -0.0028 \cdot \left(\frac{p_{HP}}{p_{AC}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{AC}}\right) + 0.5305$	
R410A RHP	$\eta_{global,R410A} = -0.0293 \cdot \left(\frac{p_{HP}}{p_{LP}}\right)^2 + 0.2008 \cdot \left(\frac{p_{HP}}{p_{LP}}\right) + 0.3549$	Gullo et al. (2017a), BITZER (2019)
R1234ze(E) RHP	$\eta_{global,R1234ze(E)} = -0.0280 \cdot \left(\frac{p_{HP}}{p_{LP}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{LP}}\right) + 0.5305$	Gullo et al. (2018c), Frascold (2019)
PC	$\eta_{global,HS} = -0.0323 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.1577 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.4891 \quad \text{for } t_{R744,out:GC} \leq 30 \text{ }^{\circ}\text{C}$ $\eta_{global,HS} = -0.0156 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0847 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.5407 \quad \text{for } t_{R744,out:GC} > 30 \text{ }^{\circ}\text{C} (75 \text{ bar} \leq p_{HP} \leq 110 \text{ bar})$ $\eta_{global,AUX} = -0.3310 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 1.3131 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) - 0.6074 \quad t_{R744,out:GC} \leq 30 \text{ }^{\circ}\text{C} (t_{outdoor} \geq 13 \text{ }^{\circ}\text{C})$ $\eta_{global,AUX} = -0.0243 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 0.1324 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) + 0.4741 \quad \text{for } t_{R744,out:GC} > 30 \text{ }^{\circ}\text{C} (75 \text{ bar} \leq p_{HP} \leq 110 \text{ bar}; t_{outdoor} \geq 13 \text{ }^{\circ}\text{C})$ $\eta_{global,LS} = -0.0114 \cdot \left(\frac{p_{MT}}{p_{LT}}\right)^2 + 0.0439 \cdot \left(\frac{p_{MT}}{p_{LT}}\right) + 0.5376$	Gullo et al. (2017a), DORIN (2019)
PC+LEJ, PC+MEJ, PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates	$\eta_{global,HS} = -0.1004 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.4359 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.2334 \quad \text{for } t_{R744,out:GC} \leq 30 \text{ }^{\circ}\text{C}$ $\eta_{global,HS} = -0.0032 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0164 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.6472 \quad \text{for } t_{R744,out:GC} > 30 \text{ }^{\circ}\text{C} (75 \text{ bar} \leq p_{HP} \leq 110 \text{ bar})$	Gullo et al. (2017a), DORIN (2019)

	$\eta_{global,AUX} = -0.3432 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 1.3625 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) - 0.6324 \quad \text{for } t_{R744,out:GC} \leq 30 \text{ }^{\circ}\text{C} \text{ (} t_{outdoor} \geq 13 \text{ }^{\circ}\text{C)}$ $\eta_{global,AUX} = -0.0665 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 0.3569 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) + 0.2108 \quad \text{for } t_{R744,out:GC} > 30 \text{ }^{\circ}\text{C} \text{ (} 75 \text{ bar} \leq p_{HP} \leq 110 \text{ bar; } t_{outdoor} \geq 13 \text{ }^{\circ}\text{C)}$ $\eta_{global,LS} = -0.0500 \cdot \left(\frac{p_{MT}}{p_{LT}}\right)^2 + 0.2116 \cdot \left(\frac{p_{MT}}{p_{LT}}\right) + 0.3715$	
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Table 5 – Assumptions made to evaluate TEWI of all the investigated solutions.

Parameter	Assumed value	Reference
GWP [$\text{kg}_{\text{CO}_2,\text{equivalent}} \cdot \text{kg}_{\text{refrigerant}}^{-1}$]	R404A = 3700 R134a = 1300 R448A = 1235 R513A = 573 R410A = 1924	AR5
L [%]	15% for DXS, R134a-CS, R513A-CS, R448A-CS, PC, PC+LEJ, PC+MEJ	Gullo et al. (2016a)
	7% for R410A RHP	AIRAH, (2012)
n [year]	10	Gullo et al. (2016a)
m [kg]	$2 \text{ kg}_{\text{refrigerant}} \cdot \text{kW}_{\text{design cooling capacity}}$ for DXS for MT refrigeration load	Gullo et al. (2016d)
	$4 \text{ kg}_{\text{refrigerant}} \cdot \text{kW}_{\text{design cooling capacity}}$ for DXS for LT refrigeration load	Gullo et al. (2016d)
	$2 \text{ kg}_{\text{refrigerant}} \cdot \text{kW}_{\text{design cooling capacity}}$ for HTC of all cascade arrangements	Gullo et al. (2016a)
α [%]	95%	Gullo et al. (2016a)
β [$\text{kg}_{\text{CO}_2,\text{equivalent}} \cdot \text{kWh}^{-1}$]	Barcelona = 0.34287509	Brander et al. (2011)
	Lisbon = 0.400151316	
	Athens = 1.921092777	
	Seville = 0.34287509	
	Tunis = 0.572169413	
	Palermo = 0.410898038	

	Tenerife = 0.34287509	
	Sydney = 0.991757127	
	Guangzhou = 0.972581723	
	Phoenix = 0.547096737	
	Rio de Janeiro = 0.092643638	
	New Delhi = 1.333174843	
	Mombasa = 0.332297783	

Table 6 – Results in terms of annual energy consumption [MWh] of all the investigated solutions in the selected warm locations.

Solution		Barcelona (AYT = 15.4 °C; Country Code = ES; Latitude = 41.390205; Longitude = 2.154007)	Lisbon (AYT = 16.4 °C; Country Code = PT; Latitude = 38.736946; Longitude = -9.142685)	Athens (AYT = 17.9 °C; Country Code = GR; Latitude = 37.983810; Longitude = 23.727539)	Seville (AYT = 18.3 °C; Country Code = ES; Latitude = 37.392529; Longitude = -5.994072)	Tunis (AYT = 18.8 °C; Country Code = TN; Latitude = 36.81897; Longitude = 10.16579)	Palermo (AYT = 18.9 °C; Country Code = IT; Latitude = 38.13205; Longitude = 13.33561)	Tenerife (AYT = 21.0 °C; Country Code = ES; Latitude = 28.291565; Longitude = -16.629129)
DXS+R410A RHP	Refrigeration	477.4	492.1	541.6	552.9	559.2	549.8	599.4
	Heating	27.9	14.8	18.6	17.0	11.3	3.8	0.0
	AC	59.0	65.7	89.6	95.2	97.6	92.0	114.0
	Total	564.3	572.6	649.8	665.2	668.1	645.7	713.4
R134a-CS+R1234ze(E) RHP	Refrigeration	464.6	478.3	516.3	524.3	531.1	525.9	566.7
	Heating	27.7	14.6	18.4	16.9	11.2	3.8	0.0
	AC	58.6	65.2	88.9	94.1	96.7	91.5	113.4
	Total	551.0	558.2	623.6	635.3	639.0	621.2	680.1
R448A-CS+R1234ze(E) RHP	Refrigeration	452.9	465.9	501.7	509.3	515.8	510.8	549.3
	Heating	27.7	14.6	18.4	16.9	11.2	3.8	0.0
	AC	58.6	65.2	88.9	94.1	96.7	91.5	113.4
	Total	539.2	545.7	609.0	620.4	623.6	606.1	662.7
R513A-CS+R1234ze(E) RHP	Refrigeration	479.1	493.9	535.4	544.3	551.5	545.4	589.2
	Heating	27.7	14.6	18.4	16.9	11.2	3.8	0.0
	AC	58.6	65.2	88.9	94.1	96.7	91.5	113.4
	Total	565.4	573.8	642.7	655.3	659.4	640.7	702.6
IND+R1234ze(E) RHP	Refrigeration	517.0	528.7	575.2	584.7	590.1	580.3	626.1
	Heating	27.7	14.6	18.4	16.9	11.2	3.8	0.0

	AC	58.6	65.2	88.9	94.1	96.7	91.5	113.4
	Total	603.3	608.6	682.5	695.7	698.0	675.6	739.6
PC	Refrigeration	475.1	498.3	568.5	582.9	589.5	569.8	619.7
	Heating	27.2	12.1	17.1	15.9	9.5	2.9	0.0
	AC	62.2	70.6	97.4	105.0	106.3	97.7	117.8
	Total	564.5	581.0	682.9	703.9	705.3	670.4	737.5
PC+LEJ	Refrigeration	432.1	451.7	515.5	528.9	534.1	514.9	560.7
	Heating	24.8	10.6	15.3	14.5	8.4	2.4	0.0
	AC	49.5	57.2	85.2	92.7	92.9	82.4	99.6
	Total	506.4	519.5	616.0	636.1	635.4	599.8	660.3
PC+MEJ	Refrigeration	427.1	443.6	498.8	510.7	516.6	501.8	548.8
	Heating	24.8	10.6	15.3	14.5	8.4	2.4	0.0
	AC	48.2	55.0	80.5	87.1	87.8	79.0	96.5
	Total	500.1	509.1	594.6	612.2	612.8	583.3	645.3
PC+MEJ_Warm_Climates	Refrigeration	340.9	352.2	391.8	408.8	403.9	374.2	401.2
	Heating	24.8	10.6	15.3	14.5	8.4	2.4	0.0
	AC	42.1	47.9	68.6	75.3	75.4	67.2	83.3
	Total	407.9	410.7	475.8	498.6	487.6	443.9	484.5

Table 7 – Results in terms of TEWI [tonCO_{2,equivalent}] of all the investigated solutions in the selected warm locations.

Solution		Barcelona (AYT = 15.4 °C; Country Code = ES; Latitude = 41.390205; Longitude = 2.154007)	Lisbon (AYT = 16.4 °C; Country Code = PT; Latitude = 38.736946; Longitude = -9.142685)	Athens (AYT = 17.9 °C; Country Code = GR; Latitude = 37.983810; Longitude = 23.727539)	Seville (AYT = 18.3 °C; Country Code = ES; Latitude = 37.392529; Longitude = -5.994072)	Tunis (AYT = 18.8 °C; Country Code = TN; Latitude = 36.81897; Longitude = 10.16579)	Palermo (AYT = 18.9 °C; Country Code = IT; Latitude = 38.13205; Longitude = 13.33561)	Tenerife (AYT = 21.0 °C; Country Code = ES; Latitude = 28.291565; Longitude = -16.629129)
DXS+R410A RHP	TEWI _{direct}	3838.0	3838.0	3838.0	3838.0	3838.0	3838.0	3838.0
	TEWI _{indirect}	1934.8	2291.3	12482.8	2280.7	3822.7	2653.2	2446.1
	TEWI	5772.8	6129.3	16320.7	6118.6	7660.6	6491.1	6284.0
R134a- CS+R1234ze(E) RHP	TEWI _{direct}	918.8	918.8	918.8	918.8	918.8	918.8	918.8
	TEWI _{indirect}	1889.1	2233.6	11979.6	2178.2	3656.1	2552.5	2331.9
	TEWI	2807.9	3152.5	12898.4	3097.0	4574.9	3471.4	3250.7
R448A- CS+R1234ze(E) RHP	TEWI _{direct}	872.9	872.9	872.9	872.9	872.9	872.9	872.9
	TEWI _{indirect}	1848.9	2183.8	11699.8	2127.1	3568.3	2490.4	2272.3
	TEWI	2721.8	3056.7	12572.7	3000.0	4441.2	3363.3	3145.2
R513A- CS+R1234ze(E) RHP	TEWI _{direct}	405.0	405.0	405.0	405.0	405.0	405.0	405.0
	TEWI _{indirect}	1938.6	2296.0	12347.7	2246.9	3772.6	2632.5	2409.1
	TEWI	2343.6	2701.0	12752.7	2651.9	4177.6	3037.5	2814.1
IND+R1234ze(E) RHP		2068.7	2435.1	13111.8	2385.4	3993.5	2776.1	2535.8
PC		1935.4	2324.9	13118.9	2413.4	4035.6	2754.6	2528.8
PC+LEJ		1736.5	2078.9	11834.7	2181.0	3635.8	2464.4	2264.0
PC+MEJ		1714.6	2037.3	11423.4	2099.1	3506.2	2396.7	2212.5
PC+MEJ_Warm_Climates		1398.5	1643.2	9140.7	1709.5	2790.0	1823.8	1661.2

Table 8 – Results in terms of annual energy consumption [MWh] of all the investigated solutions in the selected hot locations.

Solution		Sydney (AYT = 17.9 °C; Country Code = AU; Latitude = -33.865143; Longitude = 151.209900)	Guangzhou (AYT = 22.3 °C; Country Code = CN; Latitude = 23.128994, Longitude = 113.253250)	Phoenix (AYT = 22.5 °C; Country Code = USA; Latitude = 33.448376; Longitude = -112.074036)	Rio de Janeiro (AYT = 23.6 °C; Country Code = BR; Latitude = -22.908333; Longitude = -43.196388)	New Delhi (AYT = 24.5 °C; Country Code = IN; Latitude = 28.644800; Longitude = 77.216721)	Mombasa (AYT = 25.9 °C; Country Code = KE; Latitude = -4.036878; Longitude = 39.669571)
DXS+R410A RHP	Refrigeration	523.7	651	697.8	674.5	736.5	744.9
	Heating	8.3	5.3	14.4	0.0	5.7	0.1
	AC	79.4	141.0	167.1	150.4	184.4	184.9
	Total	611.4	797.3	879.3	824.9	926.6	929.9
R134a-CS+R1234ze(E) RHP	Refrigeration	505.0	604.2	632.1	625.0	665.7	679.7
	Heating	8.2	5.3	14.2	0.0	5.7	0.1
	AC	78.9	139.9	163.1	149.6	181.2	183.9
	Total	592.1	749.4	809.4	774.6	852.6	863.7
PC+MEJ	Refrigeration	471.8	615.7	676.4	636.3	716.9	718.0
	Heating	6.4	3.7	13.2	0.0	4.3	0.1
	AC	64.8	132.1	171.8	136.2	186.4	175.2
	Total	543.0	751.5	861.4	772.5	907.6	893.3
PC+MEJ_Warm_Climates	Refrigeration	361.6	465.8	557.4	460.1	581.9	512.3
	Heating	6.4	3.7	13.2	0.0	4.3	0.1
	AC	57.1	110.8	147.2	115.0	159.1	144.1
	Total	425.1	580.2	717.8	575.0	745.2	656.5
PC+MEJ_Hot_Climates	Refrigeration	362.2	461.1	552.3	454.7	575.5	503.4
	Heating	6.4	3.7	13.2	0.0	4.3	0.1
	AC	46.1	92.8	126.3	95.9	135.9	121.4
	Total	414.7	557.6	691.8	550.5	715.7	624.8

Table 9 – Results in terms of TEWI [ton_{CO2,equivalent}] of all the investigated solutions in the selected hot locations.

Solution		Sydney (AYT = 17.9 °C; Country Code = AU; Latitude = - 33.865143; Longitude = 151.209900)	Guangzhou (AYT = 22.3 °C; Country Code = CN; Latitude = 23.128994, Longitude = 113.253250)	Phoenix (AYT = 22.5 °C; Country Code = USA; Latitude = 33.448376; Longitude = - 112.074036)	Rio de Janeiro (AYT = 23.6 °C; Country Code = BR; Latitude = - 22.908333; Longitude = - 43.196388)	New Delhi (AYT = 24.5 °C; Country Code = IN; Latitude = 28.644800; Longitude = 77.216721)	Mombasa (AYT = 25.9 °C; Country Code = KE; Latitude = - 4.036878; Longitude = 39.669571)
DXS+R410A RHP	TEWI _{direct}	4073.7	4073.7	4073.7	4073.7	4073.7	4073.7
	TEWI _{indirect}	6063.8	7754.8	4810.7	764.2	12352.7	3090.1
	TEWI	10137.5	11828.5	8884.4	4838.0	16426.4	7163.8
R134a- CS+R1234ze(E) RHP	TEWI _{direct}	983.3	983.3	983.3	983.3	983.3	983.3
	TEWI _{indirect}	5871.8	7288.6	4428.1	717.6	11366.0	2870.1
	TEWI	6855.1	8271.9	5411.5	1700.9	12349.3	3853.4
PC+MEJ		5385.6	7308.8	4712.5	715.7	12100.5	2968.3
PC+MEJ_Warm_Climates		4216.4	5642.9	3927.2	532.7	9935.3	2181.6
PC+MEJ_Hot_Climates		4113.3	5423.5	3784.7	510.0	9541.7	2076.2

Figure captions

Figure 1a. Schematic of the investigated cascade supermarket refrigeration arrangements.

Figure 1b. Schematic of the investigated R1234ze(E)/R744 indirect supermarket refrigeration arrangement (IND).

Figure 2. Schematic of the investigated transcritical R744 supermarket refrigeration system equipped with parallel compression (PC) (a), its simplified schematic and thermodynamic cycle in heating mode (b) and its simplified schematic and thermodynamic cycle in air conditioning mode (c).

Figure 3. Simplified schematic of the investigated transcritical R744 supermarket refrigeration system equipped with parallel compression and overfed evaporators (PC+LEJ) in heating mode (a) and in air conditioning mode (b).

Figure 4. Simplified schematic of the investigated transcritical R744 supermarket refrigeration system equipped with parallel compression, overfed evaporators and multi-ejector block (PC+MEJ, PC+MEJ_Warm_Climates and PC+MEJ_Hot_Climates) in heating mode (a) and in air conditioning mode (b).

Figure 5a. Temperature bins for the investigated warm cities (EnergyPlus, 2018).

Figure 5b. Temperature bins for the investigated hot cities (EnergyPlus, 2018).

Figure 6a. Comparison in terms of difference in annual energy consumption [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 6b. Comparison in terms of difference in annual energy consumption [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 7a. Comparison in terms of difference in TEWI [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 7b. Comparison in terms of difference in TEWI [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 8a. Comparison in terms of difference in annual energy consumption [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 8b. Comparison in terms of difference in annual energy consumption [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 9a. Comparison in terms of difference in TEWI [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 9b. Comparison in terms of difference in TEWI [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 1a

Figure 1b

Figure 2a

Figure 2b

Figure 2c

Figure 3a

Figure 3b

Figure 4a

Figure 4b

Figure 5a. Temperature bins for the investigated warm cities (EnergyPlus, 2018).

Figure 5b. Temperature bins for the investigated hot cities (EnergyPlus, 2018).

Figure 6a. Comparison in terms of difference in annual energy consumption [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 6b. Comparison in terms of difference in annual energy consumption [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 7a. Comparison in terms of difference in TEWI [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 7b. Comparison in terms of difference in TEWI [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected warm cities.

Figure 8a. Comparison in terms of difference in annual energy consumption [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 8b. Comparison in terms of difference in annual energy consumption [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 9a. Comparison in terms of difference in TEWI [%] (DXS+R410A RHP as the reference) among the investigated solutions in the selected hot cities.

Figure 9b. Comparison in terms of difference in TEWI [%] (R134a-CS+R1234ze(E) RHP as the reference) among the investigated solutions in the selected hot cities.