Crossing CO_2 equator with the aid of multi-ejector concept: A comprehensive energy and environmental comparative study

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HIGHLIGHTS

- Performance of various supermarket refrigeration systems was exhaustively assessed
- Investigation included 12 locations below "CO_2 equator" as well as AC reclaim
- Multi-ejector concept was estimated to reduce energy intake by 26.9% over HFC units
- Multi-ejector concept was found to decrease TEWI by 90.9% over HFC units
- Multi-ejector concept can potentially push "CO_2 equator" below Northern Africa

Abstract:

The ever-stricter regulations put into effect worldwide to significantly decrease the considerable carbon footprint of commercial refrigeration sector have forced the transition to eco-friendlier working fluids (e.g. CO_2, R290, R1234ze(E), R450A, R513A). However, the identification of the most suitable long-term refrigerant is still today's major challenge for supermarkets located in high ambient temperature countries, especially as their air conditioning (AC) need is considered.

The results of this theoretical study revealed that multi-ejector "CO_2 only" systems can outperform R404A-, R290-, R1234ze(E)-, R134a-, R450A- and R513A-based solutions in an average-size supermarket located in various cities below the so-called "CO_2 equator". In fact, energy savings as well as reductions in environmental impact respectively up to 26.9% and 90.9% were estimated over conventional hydrofluorocarbon (HFC)-based solutions for the scenario including the AC demand. Also, the solution using multi-ejector block (in non-optimized operating conditions) enabled reducing the power input up to 50.3% over HFC-based units at outdoor temperatures from -10 °C to 5 °C. Finally, the study demonstrated that transcritical CO_2 multi-ejector systems integrated with the AC unit allow potentially pushing the "CO_2 equator" further South than Northern Africa.

Keywords:

Air conditioning; Supermarket; System integration; TEWI; Transcritical CO_2 refrigeration system; Warm climates.

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
CFC	Chlorofluorocarbon
CH	Chiller
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 systems, one providing MT load and the other satisfying LT load
E	Annual energy consumption [kWh]
EES	Engineering Equation Solver
EJ	Transcritical R744 booster supermarket refrigeration system equipped with multi-ejector block including MT overfed evaporators
EJ_OV	Transcritical R744 booster supermarket refrigeration system equipped with multi-ejector block including MT and LT overfed evaporators
EJ_OV_AC	Transcritical R744 booster supermarket refrigeration system equipped with multi-ejector block including MT and LT overfed evaporators and integrated with air conditioning unit
evap	Evaporator(s)
ext	External
GC	Air-cooled R744 gas cooler/condenser
GHG	Greenhouse gas
GWP	Global Warming Potential [$\mathrm{kg}_{\mathrm{CO}_{2,\mathrm{equivalent}}} \cdot \mathrm{kg}_{\mathrm{refrigerant}}^{-1}$]
h_{fg}	Latent heat of phase change [$\mathrm{kJ \cdot kg^{-1}}$]
HCFC	Hydrochlorofluorocarbon
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HFO-IND	R1234ze(E)/R744 indirect supermarket refrigeration system with MT flooded evaporators
HFO-IND_FL	R1234ze(E)/R744 indirect supermarket refrigeration system with MT and LT flooded evaporators
HFO-IND_FL_AC	R1234ze(E)/R744 indirect supermarket refrigeration system with MT and LT flooded evaporators and integrated with air conditioning unit
HP	High pressure [bar]
HPU	Heat pump unit
HS	High stage
HTC	High temperature circuit
HX_AC	Heat exchanger for air conditioning purposes
IHX	Internal heat exchanger

IP	Intermediate pressure [bar]
L	Annual leakage rate [kg·year^{-1}]
LEJ	Liquid ejectors
LP	Low pressure [bar]
LS	Low stage
LT	Low temperature [°C]
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]
ODP	Ozone Depletion Potential
out	Outlet
p	Pressure [bar]
R290-IND	R290/R744 indirect supermarket refrigeration system with MT flooded evaporators
R290-IND_FL	R290/R744 indirect supermarket refrigeration system with MT and LT flooded evaporators
R290-IND_FL_AC	R290/R744 indirect supermarket refrigeration system with MT and LT flooded evaporators and integrated with air conditioning unit
RC	Liquid receiver
R134a-CS	R134a/R744 cascade supermarket refrigeration system
R450A-CS	R450A/R744 cascade supermarket refrigeration system
R513A-CS	R513A/R744 cascade supermarket refrigeration system
t	Temperature [°C]
TEWI	Total Equivalent Warming Impact [$\mathrm{ton}_{\mathrm{CO_{2,equivalent}}}$]
tot	Total
ν_v	Saturated vapour volume per unit of mass [m^3·kg^{-1}]
VB	Vapour by-pass valve
VEJ	Vapour ejectors
$\dot{W}$	Power input [kW]
x	Quality of the refrigerant [-]

Greek symbols

α	Recycling factor [%]
β	Indirect emission factor [$\mathrm{kg}_{\mathrm{CO_{2,equivalent}}} \cdot \mathrm{kWh}^{-1}$]
η	Efficiency [-]
ω	Entrainment ratio [-]

Other symbols

ΔT_{cc}	Temperature difference in the cascade condenser [K]
ΔT_{SH}	Degree of internal superheating (i.e. within the evaporator) [K]

1. Introduction

Large supermarkets are a vital cornerstone of modern society as these guarantee one of the most fundamental aspects, such as food safety. On the other hand, food retail industry features energy consumptions, being responsible for between 3% and 4% of the annual electricity intake in industrialized countries (Reinholdt and Madsen, 2010; Tassou et al., 2011). Consequently, significant indirect greenhouse gas (GHG) emissions can be ascribable to this sector. Its carbon footprint is further worsened due to the massive use of high Global Warming Potential (GWP) refrigerants, i.e. HFC-404A ($GWP_{100\ years}$ = 3700 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$) in Europe and HCFC-22 ($GWP_{100\ years}$ = 1760 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$) on global perspective. In fact, the estimated average annual leakage rate is respectively about between 15% and 20% for R404A and 30% for R22 of the total charge (Hafner et al., 2014a).

The implementation of the Montreal Protocol in 1987 brought about the phase-out of ozone-depleting refrigerants (i.e. CFCs, HCFCs), giving rise to the wide approval of HFCs. However, the predicted damaging effects for the environment and human life associated with ongoing global warming request the gradual abandon of these refrigerants too. As a remarkable countermeasure against the considerable use of environmentally deleterious working fluids, the European Commission issued the EU F-Gas Regulation 517/2014 (European Commission, 2014). The commencement of this legislative act will lead the EU market to experience a progressive reduction in the supply of HFCs by 79% by 2030 in comparison with the average levels in 2009-2012. Furthermore, the EU F-Gas Regulation 517/2014 also dictated a $GWP_{100\ years}$ limit of 150 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$ as of 2022 to multipack centralized refrigerating units with rated capacity above 40 kW. However, an exception was introduced only for the primary circuit of cascade/indirect arrangements, whose $GWP_{100\ years}$ limit was taken as 1500 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$ (e.g. R134a, R450A, R513A). Further impetus to environmentally benign refrigerants was given during the 28th Meeting of the Parties to the Montreal Protocol in which 197 countries committed to reduce the production and consumption of HFCs by more than 80% over a 30-year period.

Therefore, food retailers and industrial end-users all over the world have been exploring various solutions relying on ultra-low GWP refrigerants to permanently replace HFC-based systems, such as "CO_2 only" refrigeration solutions, R290/R744 indirect arrangements and R1234ze(E)/CO_2 indirect units. The identification of the most suitable solution for supermarket applications will have a key role to play in the context of the ongoing HFC phase-down occurring on global perspectives under the legislative acts in force. This holds true with respect to warm locations, where the climate represents a remarkable challenge for refrigeration units as well as for AC equipment.

Carbon dioxide as a refrigerant (R744) is virtually able to eliminate the direct contribution to global warming on the part of supermarkets. In addition, R744 is a non-toxic, non-flammable, inexpensive and readily available working fluid, besides offering advantageous thermo-physical properties (Kim et al., 2004). The favourable energy performance given by conventional transcritical R744 supermarket refrigeration systems has led these technologies to take root in Northern Europe. However, the low critical temperature of CO_2 (about 31 °C) implies long lasting transcritical operating conditions for such solutions in high ambient temperature countries, causing poorer energy efficiencies compared to HFC-based units (Sawalha, 2008a). According to Sawalha et al. (2017) and Finckh et al. (2011), in fact, a conventional "CO_2 only" supermarket refrigerating system can outperform the solutions relying on man-made refrigerants at outdoor temperatures up to about 25 °C. As a consequence, the "CO_2 equator" concept was coined (Matthiesen et al., 2010) to refer to an imaginary geographical limit splitting Europe into two parts, i.e. Northern and Southern area. The Northern region (i.e. Northern and Central Europe) includes the sites where basic transcritical R744 booster supermarket refrigeration systems are preferred to HFC-based units, being more energy efficient and cost-effective. On the contrary, the Southern area (i.e. Meridional Europe) describes the locations where basic "CO_2 only" supermarket refrigeration plants cannot outperform the solutions relying on other working fluids (e.g. R404A). This led R404A-based units to take root in this region and recently cascade/indirect arrangements to draw attention due to the adoption of the EU F-Gas Regulation 517/2014. The so-called "CO_2 equator" is supposed to currently pass through the Northern shore of the Mediterranean (Gullo et al., 2017b). In order to promote the use of these systems in warm areas, the adoption of some expedients aimed at significantly enhancing their energy efficiency is compulsory. The implementation of parallel compression enable "CO_2 only" units to perform similarly to or slightly better than HFC-based systems (Gullo et al., 2016a, 2016b; Purohit et al., 2017) as well as to achieve modest energy savings over a conventional R744 booster unit (Tsamos et al., 2017) in warm climates. However, Karampour and Sawalha (2017) defined a transcritical R744 system using parallel compression a suitable solution only for cold weathers as the heating and AC equipment is integrated into the refrigerating unit. In order to push the threshold to introduce such solutions considerably further South, the multi-ejector concept was introduced (Hafner et al., 2012, 2014a). Besides parallel compression, in fact, multi-ejector concept involves other two appealing technologies, i.e. two-phase ejectors for expansion work recovery and overfed evaporators (by liquid ejectors). The use of two-phase ejectors permits addressing one of major thermodynamic penalizations occurring in transcritical R744 systems, i.e. significant exergy destruction associated with the expansion valve (Fazelpour and Morosuk, 2014; Cavallini and Zilio, 2007). These pre-compress a large amount of refrigerant for parallel compressors by recovering a part of the available expansion work, delivering greater load to them compared to high stage (HS) compressors. Being the suction pressure of the former higher than that of the latter, considerable energy savings can be attained. Furthermore, two-phase ejectors offer the capability to address two-phase flows with no damage, the absence of moving parts and low price. Additionally, it was found that a two-phase ejector based "CO_2 only" unit is a more cost-effective solution than a transcritical CO_2 system employing parallel compression (Gullo and Cortella, 2016b). The adoption of liquid ejectors allows accomplishing additional energy advantageous, as medium temperature (MT) evaporators can be run in overfed conditions all year round (Hafner and Banasiak, 2016; Hafner et al., 2016). This means that their heat transfer area can be optimally used as the superheated region is prevented as well as the corresponding heat transfer capability penalisation, giving rise to a higher operating temperature in comparison with conventional dry-expansion evaporators (Minetto et al., 2014a; Gullo et al., 2016c). As a reference, the combination of parallel compression, expansion work recovery through two-phase ejectors and overfeeding of evaporators leads to an energy saving of 22.5% over a conventional "CO_2 only" unit in the South of Italy (Minetto et al., 2014b). Also, the peculiar properties of R744 permits effectively implementing heat recovery for space heating and domestic hot water (DHW) purposes (Polzot et al., 2016a). Consequently, great opportunity for further decreasing their consumption (Polzot et al., 2016b; Sawalha, 2013) as well as their environmental impact (Ge and Tassou, 2014) along with satisfactory payback times (Tambovtsev et al., 2011; Reinholdt and Madsen, 2010) can be accomplished. The heat recapture process additionally promotes the use of a multi-ejector block as transcritical operating conditions commonly take place in heating mode (Sawalha, 2013). In addition, promising economic advantages are supposed to be achieved as the AC equipment is integrated into a "CO_2 only" supermarket refrigerating unit (Hafner et al., 2015). This is potentially another argument in favour of multi-ejector concept, since parallel compression and its combination with overfed evaporators lead to poor energy efficiencies with rise in outdoor temperature in AC mode (Karampour and Sawalha, 2015; Gullo et al., 2018b).

Besides the favourable energy efficiencies which can be achieved in high ambient temperature countries (Sawalha, 2008a), cascade/indirect solutions can considerably reduce the direct contribution to global warming on the part of food retail stores. Indirect arrangements, in fact, permits using refrigerants featuring a negligible GWP in the primary circuit, despite their possible flammability/toxicity (e.g. R290, R1234ze(E)). On the other hand, the safety concerns related to the use of these working fluids have aroused interest in cascade solutions. In fact, different working fluids classified as A1 (i.e. non-flammable, non-toxic) and complying with the environmental regulations in force can currently be adopted in their primary circuit, such as R134a ($GWP_{100\ years}$ = 1300 $kg_{CO_{2,equivalent}} \cdot kg^{-1}_{refrigerant}$). In addition, new synthetic refrigerants satisfying the EU F-Gas Regulation 517/2014 have recently emerged. In particular, R450A and R513A have been introduced to appropriately substitute R134a by offering favourable environmental performance ($GWP_{100\ years} \approx 550\ kg_{CO_{2,equivalent}} \cdot kg^{-1}_{refrigerant}$) (Makhnatch et al., 2018). As suggested by Hesse (1996), a further benefit related to indirect/cascade arrangements is that the working fluid charge can be reduced by 95% as the length of pipes can be drastically decreased.

At the present time the identification of the most suitable solution to push the "CO_2 equator" further South and the corresponding obtainable energy and environmental benefits are still a matter of intense debate among researchers, being one of today's key research topics related to commercial refrigeration sector. Despite the great interest in commercial "CO_2 only" systems and to the best of authors' knowledge (see Table 1), in fact, the effect of the adoption of the multi-ejector concept on the current "CO_2 equator" has not exhaustively investigated. In order to suitably bridge this knowledge gap, the energy and environmental performance of R744 multi-ejector enhanced parallel compression systems has been contrasted to that of two conventional solutions, i.e. R404A-based units and a R134a/R744 cascade system. The choice of these can be justified as follows:

- most of the supermarket applications still relies on R404A direct expansion refrigeration systems. Therefore, the selection of a R404A DX unit as the baseline allows the end-users to properly understand the benefits related to the other investigated units compared to the solution with which they are most familiar;
- many end-users from warm countries are uniquely familiar with synthetic refrigerants (e.g. R134a) and R744 operating in subcritical running modes (as R744 acts similarly to conventional working fluids in these operating conditions). Therefore, they are still reluctant when it comes to the adoption of "CO_2 only" systems in such a climate context (Minetto et al., 2018) and more prone to implementing a R134a/R744 cascade arrangement. This unit, in fact, is a well-established solution in warm areas, besides using well-known (and safe) refrigerants complying with the regulations in force.

Also, for the first time ever to the best of authors' knowledge (see Table 1), the comparison has been extended to various cascade/indirect arrangements relying on both ultra low-GWP refrigerants (i.e. R290, R1234ze(E)) and new eco-friendlier synthetic working fluids (i.e. R450A, R513A). An additional scientific merit of the present study is represented by the evaluation of the energy and environmental performance related to the integration with the AC equipment, which is expected to significantly promote the diffusion of "CO_2 only" solutions in warm climates. This concept has also been applied to R290- and R1234ze(E)-based systems for the first time ever to the best of authors' knowledge. The results have been contrasted with those associated with various separated HFC-based systems. Finally, the benefits from the heat recovery implementation in the investigated transcritical R744 system have been evaluated in relation to separated HFC-based alternatives. It is important to remark that, despite the considerable interest drawn by CO_2 ejector supported parallel solutions, a few studies including its performance evaluations in AC and heating modes are still available (see Table 1). All the assessments have been carried out by selecting an average-size supermarket located in various cities located below the current "CO_2 equator".

In Section 2, the adopted refrigerants, the investigated scenarios, the selected systems as well as the implemented assessments are described. The investigated operating conditions and the outcomes obtained are presented in Section 3 and Section 4, respectively. At last, the conclusions and the future developments are stated in Section 5.

2. Selected working fluids, investigated scenarios, system description, and implemented analyses

2.1. Selected working fluids

Due to the current significant need for the environment preservation and the considerable negative contribution to global warming on the part of supermarket applications, the so-called ultra-low GWP refrigerants (Table 2) are in the spotlight to supplant today's employed working fluids (i.e. R404A). These include both natural working fluids, such as R744, R290, and man-made refrigerants, such as R1234ze(E). Carbon dioxide can be employed either as the only refrigerant in transcritical refrigerating plants or as a secondary fluid in cascade/indirect arrangements. On the contrary, R290 and R1234ze(E) can be uniquely used in the primary circuit of indirect solutions owing to their flammability. Despite its high GWP, the European regulation in force allows R134a use in the high temperature circuit belonging to cascade units for large supermarkets. Two alternatives to R134a are currently available on the market, i.e. R450A and R513A (Makhnatch et al., 2018), offering a noteworthy reduction in the direct contribution of cascade arrangements to global warming.

The physical, environmental and safety properties of the selected refrigerants are summarized in Table 3. An in-depth discussion about these are left out of the present work, since many investigations aimed at this purpose are currently available in the open literature. However, it is worth noticing that:

- R744 features a very low critical temperature (i.e. about 31 °C), causing the occurrence of transcritical running modes and thus poor performance with rise in outdoor temperature for conventional R744 booster refrigerating systems;
- apart from R290 and R1234ze(E), all the considered working fluids are classified A1 (i.e. not-flammable, not-toxic). On the contrary, R290 is flammable (A3 ASHRAE classification), whereas R1234ze(E) is slightly flammable (A2L ASHRAE classification), respectively;
- R404A, R134a and R410A present a considerable GWP value, being equal to 3700 $\mathrm{kg_{CO_{2,equivalent}}} \cdot \mathrm{kg_{refrigerant}^{-1}}$, 1300 $\mathrm{kg_{CO_{2,equivalent}}} \cdot \mathrm{kg_{refrigerant}^{-1}}$ and 1924 $\mathrm{kg_{CO_{2,equivalent}}} \cdot \mathrm{kg_{refrigerant}^{-1}}$, respectively. Also, R450A and R513A feature a GWP valued of 547 $\mathrm{kg_{CO_{2,equivalent}}} \cdot \mathrm{kg_{refrigerant}^{-1}}$ and 573 $\mathrm{kg_{CO_{2,equivalent}}} \cdot \mathrm{kg_{refrigerant}^{-1}}$, respectively. The other considered refrigerants offer a negligible direct environmental impact.

2.2. Investigated scenarios

In the present work three scenarios were investigated:

1. scenario aimed at evaluating the energy and environmental performance of the selected systems without considering the AC demand. The investigated units and their corresponding adopted abbreviations are listed in Table 4;
2. scenario to compare the energy and environmental performance of some of the aforementioned systems and integrated with the AC equipment (see Subsection 2.3.5) to that of various conventional solutions (i.e. relying on refrigeration unit for MT and low temperature loads separately performing from a chiller for AC need). The studied units and their corresponding adopted abbreviations are presented in Table 5;
3. scenario aimed at assessing the energy performance of a R744 multi-ejector enhanced parallel compression system in heating mode (see Subsection 2.3.6).

No de-superheater located downstream of the low stage (LS) compressors was considered in all the evaluations implemented in the present work. It is worth remarking that, on the one hand, the adoption of such a component would have led to some energy savings over the year (Karampour and Sawalha, 2018). On the other hand, the total investment cost wold have increased.

2.3. System description

2.3.1. R404A direct expansion refrigeration system (baseline)

Two centralized units separately serve the low (LT) and MT refrigeration loads involving various semi-hermetic reciprocating compressors in the machinery room, a condenser on the roof and the display cabinets and/or cold rooms with their corresponding expansion valves. The long discharge and suction lines represent their most significant drawback, giving rise to enormous leakages of working fluid and thus a considerable charge of refrigerant. A detailed investigation as well as the schematic of such a system can be found in Sharma et al. (2014).

2.3.2. R744 multi-ejector enhanced parallel compression systems

Unlike an individual constant-geometry ejector, the multi-ejector block (MEJ in Fig. 1a and Fig. 1b) can be properly employed for controlling the heat rejection pressure and simultaneously pre-compressing some vapour in commercial "CO_2 only" refrigerating solutions (Banasiak et al., 2015). The module currently available on the market hosts from 4 to 6 vapour ejectors (VEJ in Fig. 1a and Fig. 1b) as well as 2 liquid ejectors (LEJ in Fig. 1a and Fig. 1b) connected in parallel, having all a fixed geometry and different size. The required discharge pressure is permanently maintained by switching the ejector cartridges on/off. The vapour ejectors pre-compress a large amount of refrigerant from the medium (MP) to the intermediate pressure (IP), which in turn implies a considerable unloading of HS compressors to the detriment of auxiliary ones (AUX) (Fig. 1a and Fig. 1b). As suggested by Hafner et al. (2015), IP can be up to 15 bar higher than MP, therefore leading to considerable energy conservations. Further energy savings are offered by the liquid ejectors, which permit avoiding the dry-out region and thus increasing the medium temperature compared to dry-expansion evaporators (Minetto et al., 2014a). The adoption of the internal heat exchanger (IHX) (Fig. 1b) allows also overfeeding the LT evaporators compared to the solution sketched in Fig. 1a (Minetto et al., 2015a; Schönenberger, 2016). The growth in MT and LT, which can be achieved all year round (Hafner and Banasiak, 2016; Hafner et al., 2016), reduces the frost formation and the number of defrost cycles (Hafner and Banasiak, 2016; 2014; Schönenberger, 2016). Also, the considerable usage of AUX decreases their service problems (Minetto et al., 2014b). A simplified p-h diagram of the solution represented in Fig. 1b is depicted in Fig. 1c.

In-depth information on multi-ejector based R744 systems can be found in Gullo et al. (2018a, 2018c). Also, the state-of-the-art commercial "CO_2 only" refrigeration units have been exhaustively examined by Gullo et al. (2018a).

2.3.3. Indirect refrigeration arrangements

Many indirect arrangement layouts for supermarket applications have been suggested (Sawalha, 2008b; Sharma et al., 2014). In this study, the most promising solutions have been selected, in accordance with the outcomes by Gullo and Cortella (2016a). As sketched in Fig. 2, these refrigeration solutions present a high temperature (HTC) and two secondary circuits, i.e. MT loop (MTC) and LT unit (LTC). Being HTC completely confined in the machinery room, flammable/toxic refrigerants, such R290 and R1234ze(E), can be adopted. This shares a cascade condenser (CC) with both MT and LT circuit, which acts as an evaporator for the former and as a condenser for the latter. As for the secondary loop, R744 has been selected as the refrigerant to preserve the products in the display cabinets and cold rooms, being one of the most suitable working fluids for such purposes (Inlow and Groll, 1996; Hesse, 1996; Bansal, 2012). The cascade joint arrangement in Fig. 2 offers a partial de-superheating of R744 discharged by the LS compressors by mixing this part of refrigerant with some saturated vapour coming out of the receiver at medium pressure. As it can be noticed in Fig. 2, MT evaporators in both the investigated systems operate in flooded mode. Similarly to overfed heat exchangers, these do not feature any degrees of superheating at their outlet, promoting the refrigerant-side heat transfer and therefore allowing the increase in their operating temperature compared to dry-expansion evaporators. Unlike the solution presented in Fig. 2a, the arrangement sketched in Fig. 2b also possesses LT flooded evaporators.

Two main drawbacks can be associated with these solutions, i.e. the additional heat transfer level and the need for at least a pump. However, Inlow and Groll (1996) claimed that these can be compensated by adopting an appropriate secondary working fluid as well as by properly designing the system. Wang et al. (2010) exhaustively reviewed refrigeration systems relying on a secondary loop. The researchers claimed that the maintenance of these technologies is easier than that of direct expansion units. Also, the use of plastic pipes and the reduction in the refrigerant charge potentially compensate the further costs owing to the additional heat exchanger, safety devices ascribable to the use of flammable/toxic working fluids and pump. In addition, according to Bansal (2012), the adoption of R744 as the secondary fluid permits reducing the inner diameters by 60%-70% as well as the compressor size.

Mota-Babiloni et al. (2016) recently suggested the use of R1234ze(E) in R744 cascade arrangements, having a lower liquid density and viscosity compared to R134a (i.e. reduced charge as well as pressure drop). It was also reported that this hydrofluoroolefin (HFO) is less dangerous as well as more affordable than HFO-1234yf (Palm, 2011). As a practical reference, a solutions using R1234ze(E) was recently installed in supermarket located in Europe (UNEP, 2014).

2.3.4. Cascade refrigeration arrangements

Cascade refrigeration systems rely on two circuits (i.e. LTC and HTC), which thermally interact by means of the cascade condenser (CC) (Fig. 3). In this heat exchanger the working fluid in the HTC evaporates (i.e. R134a, R450A or R513A in this investigation) in the wake of the condensation of the refrigerant flowing in the low temperature circuit (LTC) (i.e. R744 in this study). The high side shares some further evaporators aimed at cooling down chilled food display cabinets and cold rooms (Fig. 3). A push to lower GWP refrigerant deployment is offered by R134a-like alternatives, such as R450A and R513A. The former is a zeotropic blend of 42% R134a and 58% R1234ze(E) with a $GWP_{100\ years}$ value of 547 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$. The azeotropic mixture based on R134a (44%) and R1234ze(E) (56%) with a $GWP_{100\ years}$ value of 573 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$, i.e. R513A, has also been suggested as an adequate replacement for R134a.

2.3.5. Integration with AC equipment

A “fully integrated” (or “all-in-one”) R744 multi-ejector enhanced parallel compression system (Hafner et al., 2016), which is sketched in Fig. 4, represents a solution being capable of providing most of or even the whole heating reclaim (with the aid of DS2 in Fig. 4) as well as the whole AC (with the aid of HX_AC in Fig. 4), DHW (with the aid of DS1 in Fig. 4) and refrigeration needs of a supermarket. In conventional food retail stores, in fact, the AC demand is satisfied by relying on chiller(s) using high-GWP refrigerants and separately performing from the refrigeration plant (Karampour and Sawalha, 2017). Despite its considerable GWP ($GWP_{100\ years}$ = 1924 $kg_{CO_{2,equivalent}} \cdot kg_{refrigerant}^{-1}$), R410A is still one of the most widely used refrigerants in air conditioning units, including commercial refrigeration sector. On the one hand, the ever-growing pressure towards environment preservation on global perspectives has led this working fluid to be intended to be substituted with a more environmentally acceptable refrigerant. On the other hand, its most suitable replacement is still a matter of intense debate. Firstly, a “fully integrated” solution enables avoiding future complications (e.g. future restrictions to flammability/toxicity refrigerants, more severe GWP limits, identification of an appropriate R410A replacement). Also, the integration of the AC unit into the refrigeration equipment represents an innovative concept, whose aim is to considerably reduce the total investment, maintenance and running costs of transcritical R744 supermarket refrigeration systems (Hafner et al., 2015). These benefits can be achieved by replacing the vapour-compression unit(s) commonly adopted for AC purposes with an additional evaporator, which is specifically employed for satisfying the AC load of the selected supermarket (Hafner et al., 2016), in the transcritical R744 refrigerating system equipped with multi-ejector block (i.e. HX_AC in Fig. 4). One (or more) of the auxiliary compressors belonging to the multi-ejector based CO_2 system is/are then devoted to deal with the refrigerant coming out of HX_AC. According to Karampour and Sawalha (2017), the reduction in the intricacy in terms of communications between various operation and maintenance entities dealing with running all the different units, the compactness of these solutions and the need for a few additional components can be numbered as further benefits. As a consequence, the attention to such an expedient has been intensifying in the last few years, meaning that its adoption is potentially bound to become standard for next generation of transcritical CO_2 supermarket refrigeration systems (Hafner et al., 2015).

The integration of the AC unit into cascade/indirect arrangements could lead to similar benefits to those mentioned above. On the other hand, despite this and to the best of authors' knowledge, no investigations have been found in the literature. The attractiveness of the implementation of such a technique to indirect solutions can be practically revealed by considering that this concept was recently applied to a R1234ze(E)-based system installed in an Italian supermarket (Honeywell, 2016). Similarly to the transcritical R744 system and as suggested by Honeywell (2016), this purpose can be achieved by employing an evaporator located in HTC (i.e. HX_AC in Fig. 5) and outfitted with parallel compression.

In this study the energy and environmental performance of some of the aforementioned systems and integrated with the AC equipment was compared to that of various conventional solutions (i.e. relying on refrigeration unit for MT and LT loads separately performing from a chiller for AC need), as summarized in Table 5. The investigated systems coupled with the AC unit were a R744 multi-ejector enhanced parallel compression system integrated (Fig. 4) and two indirect arrangements (Fig. 5).

2.3.6. Heat recovery implementation

The heat recovery in transcritical R744 supermarket refrigerating solutions can be suitably implemented by adopting two additional heat exchangers (i.e. de-superheaters) located upstream of the conventional gas cooler/condenser (Fig. 4) (Sawalha, 2013; Tambovtsev et al., 2011). The three heat exchangers, which are connected in series, can be completely by-passed by using the corresponding 3-way valve located upstream of each heat exchanger. The first de-superheater (DS1 in Fig. 4) is arranged downstream of the HS compressors so as to produce DHW, followed by a similar heat exchanger (DS2 in Fig. 4) whose purpose is to provide the space heating. The gas cooler/condenser is possibly used for further cooling down R744 coming out of either the first or second de-superheater so as to reduce the overall energy consumption.

This investigation is mainly focused on the evaluation of the energy and environmental benefits related to the integration of the AC unit into the supermarket refrigeration system, being currently one of today's most important key research topics. Also, a few studies on this subject are still available in the open literature, although this will have a crucial importance in an eco-friendlier future for commercial refrigeration sector. Finally, it is worth remarking that for climate reasons, the AC need as well as the refrigeration loads play a pivotal role on economic, energy and environmental perspectives in the selected locations. However, nowadays the implementation of some heat recovery from transcritical R744 supermarket refrigeration systems has become standard. For this reason, a scenario involving the heat recovery implementation in the CO_2 ejector supported parallel solution sketched in Fig. 4 was also presented.

2.4. Implemented analyses

2.4.1. Energy evaluations

The energy evaluations involved the comparison in terms of both Coefficient of Performance (COP) at outdoor temperatures ranging from -2 °C to 42 °C and annual energy consumption in 12 locations below the current "CO_2 equator". Fig. 6 displays the temperature bins (EnergyPlus, 2018) for the twelve selected cities, i.e. Bilbao (Spain), Florence (Italy), Porto (Portugal), Madrid (Spain), Istanbul (Turkey), Marseille (France), Barcelona (Spain), Naples (Italy), Algiers (Algeria), Malaga (Spain), Tunis (Tunisia) and Palermo (Italy). The annual energy consumption for each of the investigated solutions was computed by multiplying the bin hours for the selected temperature bin by the corresponding value of power input and thus summing up the calculated values of energy of each bin.

2.4.2. Total equivalent warming impact assessment

The concept of total equivalent warming impact (TEWI) aims at estimating the total contribution to global warming on the part of the investigated refrigeration equipment (AIRAH, 2012). As indicated by Eq. (1), TEWI calculation involves the computation of its direct and indirect emissions of greenhouse gases.

$$TEWI = TEWI_{direct} + TEWI_{indirect} \qquad (1)$$

The indirect environmental impact of the investigated system, which can be calculated through Eq. (2) (AIRAH, 2012), is due to the CO_2 emissions caused by the combustion of fossil fuels to generate power to run the refrigerating unit. As regards the direct contributions, this is ascribable to the leakages of refrigerants into the atmosphere and can be computed by means of Eq. 3 (AIRAH, 2012).

$$TEWI_{direct} = GWP \cdot L \cdot n + GWP \cdot m \cdot (1 - \alpha) \qquad (2)$$

$$TEWI_{indirect} = E \cdot \beta \cdot n \qquad (3)$$

3. Investigated operating conditions

3.1. Running modes for the scenario with no AC demand

The present study was based on the operation conditions of a typical supermarket (Gullo et al., 2017a). As suggested by Girotto et al. (2004), the design cooling capacities were assumed as 120 kW and 25 kW for the MT and the LT loads, respectively. The deviation from the rated running modes caused by the variations of the boundary conditions was considered with the aid of Eq. (4) (Zhang, 2006):

$$Load\ fraction = \left(1 - (1 - min)\frac{(30 - t_{ext})}{(30 - 5)}\right) \qquad (4)$$

in which *min* refers to the minimum fraction of design load (taken as 0.66 for MT and as 0.8 for LT). According to Zhang (2006), the refrigeration demands range between a minimum value, reached at $t_{ext} \leq 5$ °C, and the design value, obtained at $t_{ext} \geq 30$ °C. This can be ascribable to the adoption of the heating and air conditioning equipment, which maintains the store temperature within certain limits.

All the assumptions needing to reproduce all the simulation models based on the scenario with no AC demand are summarized in Table 4.

The advantages related to overfed evaporators were assessed by increasing MT and LT respectively by 6 K and 8 K, as suggested by Wiedenmann (2015) (with the aid of field measurements) and Gullo et al. (2017a). In order to implement a fair comparison, the same operating temperatures were assumed for the corresponding flooded evaporators belonging to the indirect arrangements. Similarly, the temperature difference in the cascade condensers of all the investigated indirect and cascade units was taken as 2 K, as experimentally verified by Sawalha et al. (2006). All the components were considered well-insulated (Gullo et al., 2017a), while pressure drop was evaluated to be negligible (Gullo et al., 2017a). All the simulation models were implemented by employing Engineering Equation Solver (EES) (F-Chart Software, 2018) and assuming steady state working operations. Furthermore, the temperature of the refrigerant was supposed to go up by 5 K in all the suction lines (Gullo et al., 2017a), except for those related to the indirect arrangements thanks to the reduced length of the pipes (Gullo and Cortella, 2016a). The power input of the fans was taken as 3% of the heat capacity rejected through the corresponding high pressure heat exchanger (Karampour and Sawalha, 2015). Semi-hermetic reciprocating compressors were selected as well as all the recommended technological constraints were respected in all the implemented evaluations. Their performance was assessed by employing the correlations listed in Table 6, which were derived from some manufacturers' software.

3.1.1. Additional details necessary for R744 multi-ejector enhanced parallel compression systems

The simulation models of the investigated R744 multi-ejector enhanced parallel compression systems were based on the ones proposed by Gullo et al. (2017a). The most important assumptions can be summarized as follows:

- the liquid ejectors were not simulated, since their energy advantageous were mainly associated with the growth in the evaporating temperature (Gullo et al., 2017a) and the evaporators were supposed to run in overfed conditions all over the year (Gullo et al., 2017a). Also, the quality of the refrigerant coming out of these heat exchangers was taken as 1. This assumption did not influence the results obtained considerably (Gullo et al., 2017a), since the refrigerant coming out of the overfed evaporators commonly has a quality value of around 0.9 (Minetto et al., 2014a);
- the effectiveness of the internal heat exchanger belonging to EJ_OV was assumed as 0.5 (Gullo et al., 2017a);
- the occurrence of the subcritical, transition and transcritical operating conditions were assumed to occur in the same way as the one suggested by Gullo et al. (2016a, 2017a) (see Table 4). Furthermore, the parallel compressors were assumed to be switched off in subcritical running modes (i.e. $t_{ext} \leq 17$ °C) and replaced with the vapour by-pass valve (VB) (Gullo et al., 2017a). It was assumed, in fact, that the amount of flash gas was lower than the minimum suction volume rate of the smallest parallel compressor in these operating conditions;
- as proposed by Minetto et al. (2015b), the entrainment ratio (ω) of vapour ejectors, i.e. the ratio of the mass flow rate associated with the suction nozzle to the mass flow rate related to the motive nozzle, needs to be evaluated with respect to both the optimum pressure lift (P_{lift}), i.e. the pressure difference between the two receivers, and the gas cooler/condenser exit temperature. This is due to the fact that, at a given running mode, a growth in P_{lift} would cause a reduction in amount of pre-compressed R744 and vice versa. In the present study, ω was computed with the aid of the correlations listed in Table 7 (Gullo et al., 2017a), which were derived from some experimental measurements (Haida et al., 2016; Palacz et al., 2015);
- the same optimization procedures suggested by Gullo et al. (2017a) were adopted in this study. In particular, the required total power input of the solution equipped with the multi-ejector module was minimized with respect to P_{lift} in subcritical and transition operating conditions and with regard to P_{lift} as well as to the discharge pressure in transcritical running modes. As proposed by Gullo et al. (2017), the minimum and maximum values of P_{lift} were taken as 4

bar (Banasiak et al., 2015) so as to guarantee a suitable feeding of the evaporators and as 15 bar (Hafner et al., 2015), respectively.

3.2. Additional assumptions necessary for the scenario including AC demand

The adopted abbreviations and the additional assumptions needing to reproduce all the simulation models based on the scenario including the AC reclaim are listed in Table 5. It is worth remarking that the aforementioned assumptions need to be combined with those presented in Subsection 3.1 and Subsection 3.1.1 to implement all the assessments related to the scenario considering the AC need.

The performance of all the compressors for the scenario including AC reclaim was also evaluated by relying on the correlations listed in Table 6, which were derived from some manufacturers' software.

The AC reclaim was supposed to linearly range from a minimum value (equal to 0 kW at t_{ext} = 5 °C) to the design one (equal to 120 kW at $t_{ext} \geq 30$ °C) (Gullo et al., 2017a). Also, this was assumed to be satisfied by free cooling at $t_{ext} < 24$ °C (Gullo et al., 2017a). Also, all the chillers as well as all the AC evaporator (i.e. HX_AC in Fig. 4 and Fig. 5) were supposed to cool down water from 12 °C to 7 °C for AC purposes (Karampour and Sawalha, 2017).

3.3. Additional assumptions necessary for the scenario including heat recovery implementation

A comparison in terms of power input between the solution sketched in Fig. 4 and a R410A heat pumping unit (i.e. R410A HPU) was performed in heating mode. For all the four investigated scenarios (i.e. outdoor temperature of -10 °C, -5 °C, 0 °C and +5 °C), the evaluation was based on:

- a constant value of heating load equal to 140 kW, since the selected external temperatures had low values (Sawalha, 2013);
- a return temperature of the water as 30 °C (Karampour and Sawalha, 2015);
- a negligible tap water heating load (i.e. reclaim related to DS1 in Fig. 4), since this need is commonly much lower than the other demands (Karampour and Sawalha, 2017).

As regards the "CO_2 only" system, it was assumed that:

- the air-cooled gas cooler/condenser (GC) was by-passed by using a 3-way valve (Fig. 4), as low values of outdoor temperature were selected (Sawalha, 2013);
- the R744 temperature coming out of DS2 was selected equal to 35°C (Karampour and Sawalha, 2015).

With respect to R410A HPU, it was supposed that:

- the selected outdoor temperatures lead to evaporating temperatures of -5 °C at t_{ext} = +5 °C, -10 °C at t_{ext} = 0 °C, -15 °C at t_{ext} = -5 °C and -20 °C at t_{ext} = -10 °C, respectively;
- its condensing temperature and its internal degree of superheating (i.e. within the evaporator) were equal to 37 °C and 5 K, respectively;
- the performance of the compressors was assessed with the aid of the correlations presented in Table 6, which were derived from some manufacturers' software.

3.4. Assumptions necessary for total equivalent warming impact assessment

The TEWI analyses were based on the following assumptions:

- the annual leak rate was supposed to be equal to 15% for DXS, R134a-CS, R450A-CS and CS-513A (Gullo et al., 2016a), whereas it was taken as 7% for R410A CH (AIRAH, 2012);

- the operating life for all the investigated solutions was assumed equal to 10 years (Gullo et al., 2016a);
- the charge of the refrigerant flowing through HTC of all the cascade arrangements as well as the one related to DXS serving MT was set to 2 $kg_{refrigerant} \cdot kW_{design\ cooling\ capacity}$ (Gullo et al., 2016a; 2016b);
- the charge of the refrigerant associated with DXS serving LT was taken as 4 $kg_{refrigerant} \cdot kW_{design\ cooling\ capacity}$ (Gullo et al., 2016b);
- 95% of the working fluid was considered to be recycled (Gullo et al., 2016a);
- the CO_2 emission owing to the electricity generation was chosen as the ones suggested by Brander et al. (2011);
- the direct contribution to global warming related to all the selected ultra-low refrigerants was neglected (Gullo and Cortella, 2016).

4. Results

4.1. Comparison in terms of COP: scenario without the AC demand

In this Subsection, the COP values of all the investigated solutions are presented at outdoor temperatures between -2 °C and 42 °C (scenario without AC demand) and summarized with the aid of Fig. 7. It was possible to notice that both EJ and EJ_OV offer respectively increases in COP by 77.6% and by 96.3% over DXS at outdoor temperatures up to 4 °C as both solutions performed at the minimum condensing temperature (subcritical conditions). At external temperatures between 5 °C and 17 °C, the COP related to EJ and EJ_OV sharply diminished, becoming from 87.8% to 22.8% higher than that of DXS. This was due to the ever-growing amount of flash gas generated in the liquid receiver at IP and, at the same time, the parallel compressors were assumed to be off. As the auxiliary compressors were put in operation (i.e. $t_{ext} \geq 18$ °C), a sudden growth in COP was observable, leading to at worst 23% better performance over DXS at outdoor temperatures up to 27 °C. In transcritical operating conditions (i.e. $t_{ext} > 27$ °C), EJ and EJ_OV showed on average 12.7% and 18.4% higher COPs than DXS at external temperatures between 28 °C and 42 °C.

As regards the indirect and cascade arrangements, the COP values were constant at outdoor temperatures up to 15 °C as all the solutions were supposed to perform at the minimum condensing temperature (Gullo and Cortella, 2016; Gullo et al., 2016a). It was also possible to notice that HFO-IND_FL was the only solution capable of outperforming DXS at these running modes, offering an energy saving on average by 4% at $t_{ext} \leq 15$ °C. At outdoor temperatures between 16 °C and 27 °C, it was found that in relation to DXS:

- HFO-IND was an appropriate alternative only at $t_{ext} \geq 19$ °C, whereas HFO-IND_FL featured reductions in the energy consumption from 4.3% to 11.1%;
- R290-IND was a suitable replacement for DXS only at $t_{ext} \geq 24$ °C, whereas R290-IND_FL brought the energy consumption down to 8.6%;
- the adoption of R134a-CS led to energy conservations between 1.2% and 10.4%, whereas the other two investigated cascade arrangements were energetically competitive at $t_{ext} \geq 22$ °C.

Taking DXS as the term of comparison, the results related to $t_{ext} > 27$ °C revealed that all the investigated cascade and indirect solutions perform better. In particular, it was found that:

- HFO-IND_FL, R290-IND_FL and R134a-CS presented the best energy efficiencies among these technologies, leading to growths in COP values from 11.9% to 25.6%, from 9.6% to 26.6% and from 11.4% and 25.6%, respectively;

- R450A-CS and R513A-CS offered increases in COPs between 4.1% and 17% and between 4.1% and 15.7%, respectively;
- the COPs associated with HFO-IND and R290-IND were from 7% to 20.9% and from 4.8% to 22% greater, respectively.

4.2. Comparison in terms of annual energy consumption: scenario without the AC demand

The results in terms of annual energy consumption (scenario without AC demand) are showed in Fig. 8 for all the investigated solutions in the selected locations, whereas the ones on the basis of the difference in annual energy intake (DXS as the reference) are summarized in Fig. 9. In the latter, the negative values indicate that the selected solution consumes less electricity than DXS in the evaluated location.

In comparison with DXS, it was estimated that for the locations featuring an average yearly temperature (AYT) up to 14.5 °C (i.e. Bilbao, Florence, Porto, Madrid, Istanbul):

- EJ and EJ_OV offered energy conservations from 21.4% (in Porto) to 23.7% (in Istanbul) and from 26.5% (in Porto) to 28.6% (in Istanbul), respectively;
- HFO-IND performed similarly to the baseline in all the investigated cities, whereas HFO-IND_FL presented energy savings from 4.7% (in Porto) to 5.9% (in Florence);
- the additional heat transfer level featuring the indirect arrangements was particularly (energetically) deleterious for the R290-based solutions. In fact, R290-IND consumed between 3.5% (in Florence) and 5.3% (in Porto) more electricity, whereas the most sophisticated solution (i.e. R290_IND_FL) had similar performance to DXS;
- R134a-CS is the only (energetically) acceptable cascade solution in these locations as it had moderate energy savings from 1.9% (in Porto) and 3.5% (in Florence). On the contrary, the other two investigated cascade units presented increases in annual energy consumption from 1.5% (R513A-CS in Florence) to 3.4% (R450A-CS in Porto).

In the cities having an average annual temperature between 14.9 °C and 16.4 °C (i.e. Marseille, Barcelona, Naples) and compared to the same previous baseline, the outcomes obtained revealed that:

- the energy conservations associated with EJ ranged between 21.2% (in Naples) and 22.8% (in Marseille), whereas the ones related to EJ_OV were between 26% (in Naples) and 27.7% (in Marseille);
- HFO-IND offered similar performance, whereas HFO-IND_FL featured reductions in energy consumption from 5.5% (in Barcelona) to 6.2% (in Naples);
- R290-IND is not (energetically) justifiable in the previous mentioned cities either, whereas R290-IND_FL offered modest energy savings up to 2.5% (in Naples);
- R134a-CS was the only cascade arrangement capable of outperforming DXS, presenting decreases in annual energy consumption between 3% (in Barcelona) and 4% (in Naples). On the contrary, the other two investigated cascade systems consumed from 1.5% (R513A-CS in Naples) to 2.4% (R450A-CS in Barcelona) more electricity.

As regards the selected locations presenting an AYT between 17.7 °C and 18.9 °C (i.e. Algiers, Malaga, Tunis, Palermo) and in relation to DXS, it was found that:

- EJ and EJ_OV led to energy conservations from 18.6% (in Palermo) to 19.9% (in Algiers) and from 23.3% (in Palermo) to 24.6% (in Algiers), respectively;
- HFO-IND had slightly lower energy consumptions, whereas HFO-IND_FL offered reductions in energy consumption ranging from 6.4% (in Malaga) to 7.1% (in Tunis);

- R290-IND presented slightly higher energy intakes (up to 2.5% in Malaga), whereas the adoption of R290-IND_FL permitted reducing the energy consumption from 2.7% (in Malaga) to 3.8% (in Tunis);
- R134a-CS reduced the consumptions in electricity between 4.3% (in Malaga) and 5.3% (in Tunis). Also, R450-CS and R513A-CS could perform similarly to the selected baseline in all the cities mentioned above.

4.3. Comparison in terms of TEWI: scenario without the AC demand

Fig. 10 sums up the outcomes related to the comparison in terms of TEWI (scenario without AC demand) for all the investigated solutions in the selected locations. The differences in TEWI (in %) among the investigated solutions and DXS in the chosen cities are presented in Fig. 11, in which the negative values suggest that the selected system had lower TEWI than DXS. It was brought to light that the adoption of all the suggested systems would have led to significant reductions in the carbon footprint of the refrigeration unit over R404A-based units.

Compared to DXS, in the locations having an AYT up to 14.5 °C (i.e. Bilbao, Florence, Porto, Madrid, Istanbul) the outcomes obtained suggested:

- decreases in TEWI from 50.7% (in Istanbul) to 31.9% (in Bilbao) on the part of EJ and from 53.9% (in Istanbul) to 70.7% (in Bilbao) by EJ_OV;
- reductions in TEWI between 35.7% (in Istanbul) to 50.1% (in Bilbao) by HFO-IND and from 39.1% (in Istanbul) to 61.4% (in Bilbao) on the part of HFO-IND_FL;
- drops in TEWI from 33.1% (in Istanbul) to 57.3% (in Bilbao) performed by R290-IND and from 36.6% (in Istanbul) to 59.6% (in Bilbao) on the part of R290-IND_FL;
- decreases in TEWI from 28.8% (in Istanbul) to 45.5% (in Bilbao) by R134a-CS, from 30.4% (in Istanbul) to 51.9% (in Bilbao) achieved by R450A-CS and from 30.5% (in Istanbul) to 51.8% (in Bilbao) on the part of R513A-CS.

In the cities presenting an AYT between 14.9 °C and 16.4 °C (i.e. Marseille, Barcelona, Naples) and in relation to DXS, the results obtained showed:

- drops in TEWI from 62.6% (in Naples) to 89.9% (in Marseille) performed by EJ and from 64.8% (in Naples) to 90.6% (in Marseille) by EJ_OV;
- decreases in TEWI between 52.9% (in Naples) to 87% (in Marseille) on the part of HFO-IND and from 55.4% (in Naples) to 87.7% (in Marseille) by HFO-IND_FL;
- reductions in TEWI from 51.1% (in Naples) to 86.5% (in Marseille) attained by R290-IND and from 53.7% (in Naples) to 87.2% (in Marseille) by R290-IND_FL;
- falls in TEWI from 41.4% (in Naples) to 65.8% (in Marseille) on the part of R134a-CS, from 46.3% (in Naples) to 77.6% (in Marseille) accomplished by R450A-CS and from 46.2% (in Naples) to 77.3% (in Marseille) by R513A-CS.

As for the solutions performing in the locations with an AYT between 17.7 °C and 18.9 °C (i.e. Algiers, Malaga, Tunis, Palermo) and in comparison with DXS, the outcomes obtained led to:

- decreases in TEWI from 51.6% (in Algiers) to 64.6% (in Malaga) achieved by EJ and from 54.5% (in Algiers) to 66.6% (in Malaga) on the part of EJ_OV;
- drops in TEWI between 40.6% (in Algiers) to 56.7% (in Malaga) by HFO-IND and from 43.6% (in Algiers) to 58.9% (in Malaga) performed by HFO-IND_FL;
- decrements in TEWI from 38.4% (in Algiers) to 55.1% (in Malaga) on the part of R290-IND and from 41.5% (in Algiers) to 57.3% (in Malaga) accomplished by R290-IND_FL;

- reductions in TEWI from 32.6% (in Algiers) to 44.1% (in Malaga) by R134a-CS, from 34.9% (in Algiers) to 49.7% (in Malaga) attained by R450A-CS and from 35% (in Algiers) to 49.6% (in Malaga) by R513A-CS.

In Fig. 12 the contribution in terms of $TEWI_{direct}$ and $TEWI_{indirect}$ of DXS (Fig. 12a), R134a-CS (Fig. 12b), R450A-CS (Fig. 12c) and R513A-CS (Fig. 12d) are presented. It was found that R134a-CS, R450A-CS and R513A-CS offered reductions in $TEWI_{direct}$ by 75.2%, 89.6% and 89.1% over DXS, respectively.

4.4. Comparison in terms of annual energy consumption: scenario including the AC demand

This Subsection is devoted to investigate the energy performance of the most promising solutions described above (i.e. EJ_OV, HFO-IND_FL, R290-IND_FL, R134a-CS, R450A-CS and R513A-CS) as the AC demand is also taken into account. However, the multi-ejector block operating conditions were related to EJ_OV_AC performance in Appendix.

Firstly, the power input associated with DXS+R410A CH (Fig. 13a) was contrasted with that of EJ_OV_AC (Fig. 13b) at outdoor temperatures ranging from 25 °C and 40 °C. At a later time, the annual energy consumption of EJ_OV_AC, HFO-IND_FL_AC, R290-IND_FL_AC, HFO-IND_FL+R1234ze(E) CH, R290-IND_FL+R1234ze(E) CH, R134a-CS+R1234ze(E) CH, R450A-CS+R1234ze(E) CH and R513A-CS+R1234ze(E) CH was compared to that of DXS+R410A CH in all the previous mentioned cities.

As regards EJ_OV_AC, the results in AC mode depicted in Fig. 13(b) were computed as a difference between the power input of EJ_OV_AC and that related to EJ_OV at the same external temperatures. It was found that the total required power input of EJ_OV_AC is lower than that of the selected baseline, leading to energy conservations between 18% and 3.1%. On the contrary, the power input of EJ_OV_AC related to the AC demand is similar to that of DXS+R410A CH at the outdoor temperature of 25 °C, becoming from 15.8% to 25% higher at more severe operating conditions.

The results in terms of annual energy consumption and difference in annual electricity intake (DXS+R410A CH as the reference) are respectively summarized with the aid of Fig. 14 and Fig. 15 for all the chosen locations. Taking DXS+R410A CH into account as the baseline, it was found that for the cities having an AYT up to 14.5 °C (i.e. Bilbao, Florence, Porto, Madrid, Istanbul):

- EJ_OV_AC featured energy savings from 25.6% (in Florence) to 26.9% (in Bilbao);
- HFO-IND_FL+R1234ze(E) CH and HFO-IND_FL_AC offered similar reductions in annual energy consumption, ranging from 4.7% (in Porto) to 5.5% (in Florence);
- R290_IND_FL+R1234ze(E) CH and R290_IND_FL_AC performed similarly to the selected baseline;
- R134-CS is the only (energetically) suitable cascade arrangement in the selected locations, offering modest energy conservations between 1.9% (in Porto) and 3.3% (in Florence). On the contrary, the other two studied cascade systems were estimated to consume from 1.1% (R513A-CS in Florence) to 3.2% (R450A-CS in Porto) more electricity.

In the sites presenting an average annual temperature from 14.9 °C to16.4 °C (i.e. Marseille, Barcelona, Naples) and in relation to DXS+R410A CH, it was brought to light that:

- the energy savings related to EJ_OV_AC were between 23.2% (in Naples) and 25.3% (in Barcelona);
- HFO-IND_FL+R1234ze(E) CH and HFO-IND_FL_AC led to about the same energy advantageous, offering reductions in energy consumption from 5.2% (in Barcelona) and 5.7% (in Naples);

- R290-IND_FL+R1234ze(E) CH performed slightly better than R290-IND_FL_AC, offering energy savings from 1.5% (in Barcelona) and 2.3% (in Naples). At the same locations, R290-IND_FL_AC consumed from 1% (in Barcelona) and 1.6% (in Naples) less electricity;
- R134a-CS+R1234ze(E) CH was the only cascade-based solution offering some energy savings compared to the selected baseline, leading to decreases in energy consumption from 3% (in Barcelona) to 3.7% (in Naples). On the contrary, the other two investigated cascade-based arrangements needed from 0.6% (R513A-CS in Naples) to 2% (R450A-CS in Barcelona) more electricity to be run.

As for the selected locations characterized by an average annual temperature from 17.7 °C to 18.9 °C (i.e. Algiers, Malaga, Tunis, Palermo) and compared to DXS+R410A CH, it was revealed that:

- EJ_OV_AC needed from 19.3% (in Tunis) to 21.3% (in Malaga) less electricity to be put in operation;
- HFO-IND_FL+R1234ze(E) CH and HFO-IND_FL_AC performed similarly, presenting drops in energy consumptions between 5.8% (in Malaga) to 6.4% (in Tunis);
- the performance of R290-IND_FL+R1234ze(E) CH was slightly greater than that of R290-IND_FL_AC, leading to energy conservations from 2.5% (in Malaga) and 3.4% (in Tunis). At the same cities, the energy consumption of R290-IND_FL_AC was from 1.8% (in Malaga) and 2.5% (in Tunis) lower than that of the selected baseline;
- R134a-CS+R1234ze(E) CH decreased the electricity consumption from 4% (in Malaga) to 4.7% (in Tunis), whereas R450-CS+R1234ze(E) CH and R513A-CS+R1234ze(E) CH performed similarly to the adopted baseline in all the investigated locations.

4.5. Comparison in terms of TEWI: scenario including the AC demand

The results in terms of TEWI comparison of the investigated solutions in all the selected locations as the AC load is also considered are presented in Fig. 16, whereas the estimated differences in TEWI (DXS+R410A CH as the reference) are summarized with the aid of Fig. 17.

In comparison with DXS+R410A CH and for the locations presenting an AYT up to 14.5 °C (i.e. Bilbao, Florence, Porto, Madrid, Istanbul), it was found that:

- the adoption of EJ_OV_AC would have led to reductions in TEWI from 54% (in Istanbul) to 72.5% (in Bilbao);
- HFO-IND_FL+R1234ze(E) CH and HFO-IND_FL_AC offered from 40.8% (in Istanbul) to 64.2% (in Bilbao) better environmental impact;
- R290-IND_FL+R1234ze(E) CH had from 38.5% (in Istanbul) to 62.7% (in Bilbao) lower TEWI values, whereas the implementation of R290-IND_FL_AC would have enabled decrements in TEWI from 38.1% (in Istanbul) to 62.6% (in Bilbao);
- drops in TEWI from 31.5% (in Istanbul) to 50.1% (in Bilbao) performed by R134a-CS, from 33% (in Istanbul) to 55.8% (in Bilbao) on the part of R450A-CS and from 33.1% (in Istanbul) to 55.8% (in Bilbao) attained by R513A-CS could have been possible.

In the locations characterized by an AYT between 14.9 °C and 16.4 °C (i.e. Marseille, Barcelona, Naples) and compared to DXS+R410A CH, the outcomes obtained suggested that:

- TEWI could be reduced from 64.6% (in Naples) to 90.9% (in Marseille) by adopting EJ_OV_AC;
- reductions in TEWI between 56.5% (in Naples) to 88.5% (in Marseille) could be accomplished by employing either HFO-IND_FL+R1234ze(E) CH or HFO-IND_FL_AC;

- the carbon footprint could be decreased from 54.9% (in Naples) to 88.1% (in Marseille) by R290-IND_FL+R1234ze(E) CH and from 54.6% (in Naples) to 88% (in Marseille) on the part of R290-IND_FL_AC;
- drops in TEWI from 44.2% (in Naples) to 69.7% (in Marseille) attained by R134a-CS, from 48.6% (in Naples) to 79.9% (in Marseille) by R450A-CS and from 48.6% (in Naples) to 79.5% (in Marseille) on the part of R513A-CS could have been accomplished.

The results related to the solutions running in the cities with an AYT between 17.7 °C and 18.9 °C (i.e. Algiers, Malaga, Tunis, Palermo) and in relation to DXS revealed that:

- the use of EJ_OV_AC featured decrements in TEWI from 53.2% (in Algiers) to 66.6% (in Malaga);
- the adoption of either HFO-IND_FL+R1234ze(E) CH or HFO-IND_FL_AC would have permitted bringing TEWI down from 44.2% (in Algiers) to 60% (in Malaga);
- drops in TEWI from 42.3% (in Algiers) to 58.6% (in Malaga) on the part of R290-IND_FL+R1234ze(E) CH and from 41.8% (in Algiers) to 58.3% (in Malaga) attained by R290-IND_FL_AC could have been achieved;
- decreases in TEWI from 34.6% (in Algiers) to 71% (in Malaga) achieved by R134a-CS+R1234ze(E) CH, from 36.7% (in Algiers) to 52.1% (in Malaga) by R450A-CS+R1234ze(E) CH and from 36.8% (in Algiers) to 52% (in Malaga) on the part of R513A-CS+R1234ze(E) CH could have been possible.

The contribution in terms of $TEWI_{direct}$ and $TEWI_{indirect}$ related to DXS+R410A CH (Fig. 18a), R134a-CS+R1234ze(E) CH (Fig. 18b), R450A-CS+R1234ze(E) CH (Fig. 18c) and R513A-CS+R1234ze(E) CH (Fig. 18d) are depicted in Fig. 18. The results obtained suggested that the adoption of R134a-CS+R1234ze(E) CH, R450A-CS+R1234ze(E) CH and R513A-CS+R1234ze(E) CH would have led to decreases in $TEWI_{direct}$ by 78.9%, 91.1% and 90.7% compared to DXS+R410A CH, respectively.

4.6. Comparison in terms of power input in heating mode

In Fig. 19 the outcomes of EJ_OV and those of DXS+R410A HPU (i.e. R404A-based units for refrigeration loads separately performing with a R410A heat pump system aimed at heating purposes) associated with the heating mode are contrasted in terms of power input at outdoor temperatures between -10 °C and 5 °C.

In spite of the significant design value of the heating demand, the "CO_2 only" refrigerating solution with multi-ejector block offered from 36.8% to 50.3% lower need for power input than separated HFC-based units over the investigated range of external temperatures. It was significant to notice that the power input related to the multi-ejector based solution was constant, since the temperature of the cooling medium corresponded with that of the return water (i.e. 30 °C) in all the evaluated operating conditions as well as the gas cooler was assumed to be by-passed.

5. Conclusions and future work

Being R744 a non-toxic, non-flammable and climate-friendly refrigerant, "CO_2 only" units are perceived as appealing solutions in commercial refrigeration sector. Also, these systems have benefited from a number of recent design enhancements, which potentially make them the go-to solutions for supermarket applications worldwide. The implementation of the multi-ejector concept has brought transcritical R744 refrigeration systems to achieve promising energy efficiencies even in warm climates, as theoretically showed in the present investigation.

In this work the energy and environmental performance of various supermarket refrigeration systems has been exhaustively studied. All the assessments have been based on an average-size food retail store located in various warm cities (Bilbao, Florence, Porto, Madrid, Istanbul, Marseille, Barcelona, Naples, Algiers, Malaga, Tunis, Palermo). These feature an average yearly temperature between 14.1 °C and 18.9 °C, besides being positioned below the so-called “CO_2 equator” (Gullo et al., 2017b).

As regards the case excluding the AC demand and compared to R404A direct expansion systems, the main results obtained can be summarized as follows:

- it has been estimated that the implementation of the multi-ejector concept leads transcritical CO_2 supermarket refrigeration systems to energy savings from 18.6% to 28.6% in locations featuring average yearly temperatures between 14.1 °C and 18.9 °C. This gives to rise a drop in TEWI ranging from 50.7% to 90.6%;
- in contrast to the other investigated indirect/cascade arrangements, the R1234ze(E)/R744 indirect solution with MT and LT flooded evaporators and the R134a/R744 cascade unit offer some moderate reductions in energy consumption in cities having average annual temperatures between 14.1 °C and 18.9 °C. The implementation of such arrangements would permit consuming from 4.7% to 7.1% and from 1.9% to 5.3% less electricity as well as reducing from 39.1% to 87.7% and from 28.8% to 65.8% TEWI, respectively.

As for the aforementioned scenario and considering the R134a/R744 cascade arrangement as the baseline, it has been brought to light that the multi-ejector based solutions lead from 14.2% to 26.2% to higher energy conservations as well as from 28% to 72.4% to better carbon footprint in locations characterized by average yearly temperatures between 14.1 °C and 18.9 °C. At the same boundary conditions, the R1234ze(E)/R744 indirect unit with MT and LT flooded evaporators offers from 1.9% to 2.9% better energy performance and from 14.5% to 64% lower TEWI.

The adoption of ever-stricter regulations worldwide with the purpose of environment preservation is making refrigerant selection a challenge for engineers and end users in food retail industry, especially as the AC demand of the selected supermarket is considered. Consequently, the scenario including the AC demand and based on R404A direct expansion systems for refrigeration loads and a R410A chiller for AC purposes as the baseline has also been investigated. The outcomes obtained suggest that for locations having average yearly temperatures between 14.1 °C and 18.9 °C:

- the investigated multi-ejector solution integrated with the AC equipment can achieve energy conservations ranging from 19.3% to 26.9% and, as a consequence, reductions in TEWI from 53.2% to 90.9%;
- the R1234ze(E)-based indirect systems (i.e. with and without coupling with the AC unit) has been found to perform similarly, presenting drops in electricity consumption between 4.7% and 6.4% and thus leading to reductions in TEWI from 40.8% to 88.5%;
- the R134a/R744 cascade system separately working with a R1234ze(E) chiller is the only other solution presenting some energy savings. These range between 1.9% and 4.7%, giving to rise decreases in TEWI from 31.5% to 69.7%.

Considering the latter solution as the baseline, the implementation of the multi-ejector concept can reduce the energy consumption and TEWI respectively from 15.3% to 25.3% and from 27.3% to 70% in the investigated locations. At the same boundary conditions the R1234ze(E)-based indirect arrangements consume from 1.7% to 2.8% less electricity, leading to reductions in TEWI between 13.5% and 88.5%.

Finally, it has been estimated that, despite the considerable selected heating load, a CO_2 ejector supported parallel solution could be able to decrease the required power input from 36.8% to 50.3% compared to separated HFC-based units at outdoor temperatures between -10 °C and 5 °C. It is worth remarking that potentially further energy efficiency enhancements can be achieved by adopting a suitable control strategy for the heat recapture implementation (i.e. possible use of the air-cooled gas cooler/condenser).

To conclude, it can be claimed that the "CO_2 equator" can be potentially pushed further South than the North of Africa, even as the AC reclaim (i.e. integration with the AC equipment) is considered. Efforts are underway to address the persisting non-technological barriers (e.g. lack of trained installers and service technicians, little confidence in these technologies, social and political factors), which represent a considerable obstacle impeding multi-ejector based solutions from an ever wider acceptance worldwide (Minetto et al., 2018). However, it is worth remarking that other appealing solutions complying with the EU F-Gas Regulation 517/2014 (i.e. other "CO_2 only" units as well as other solutions employing R744 as the secondary coolant), which are currently on the market, could outperform the multi-ejector based systems in the investigated climate context, even in the scenario including AC and heating needs.

As future work, it is still essential to cope with:

- the assessment of the corresponding energy benefits for R744 multi-ejector enhanced parallel compression systems in heating mode;
- exhaustive experimental work to accurately evaluate the energy and economic benefits related to the adoption of two different multi-ejector blocks (i.e. one for the AC need and the other for the refrigeration loads), of the direct cooling fan coils and air curtains and of the principle of pivoting (Hafner, 2017);
- in-depth experimental evaluations on "fully integrated" CO_2 multi-ejector enhanced parallel compression systems in hot climates (e.g. India).

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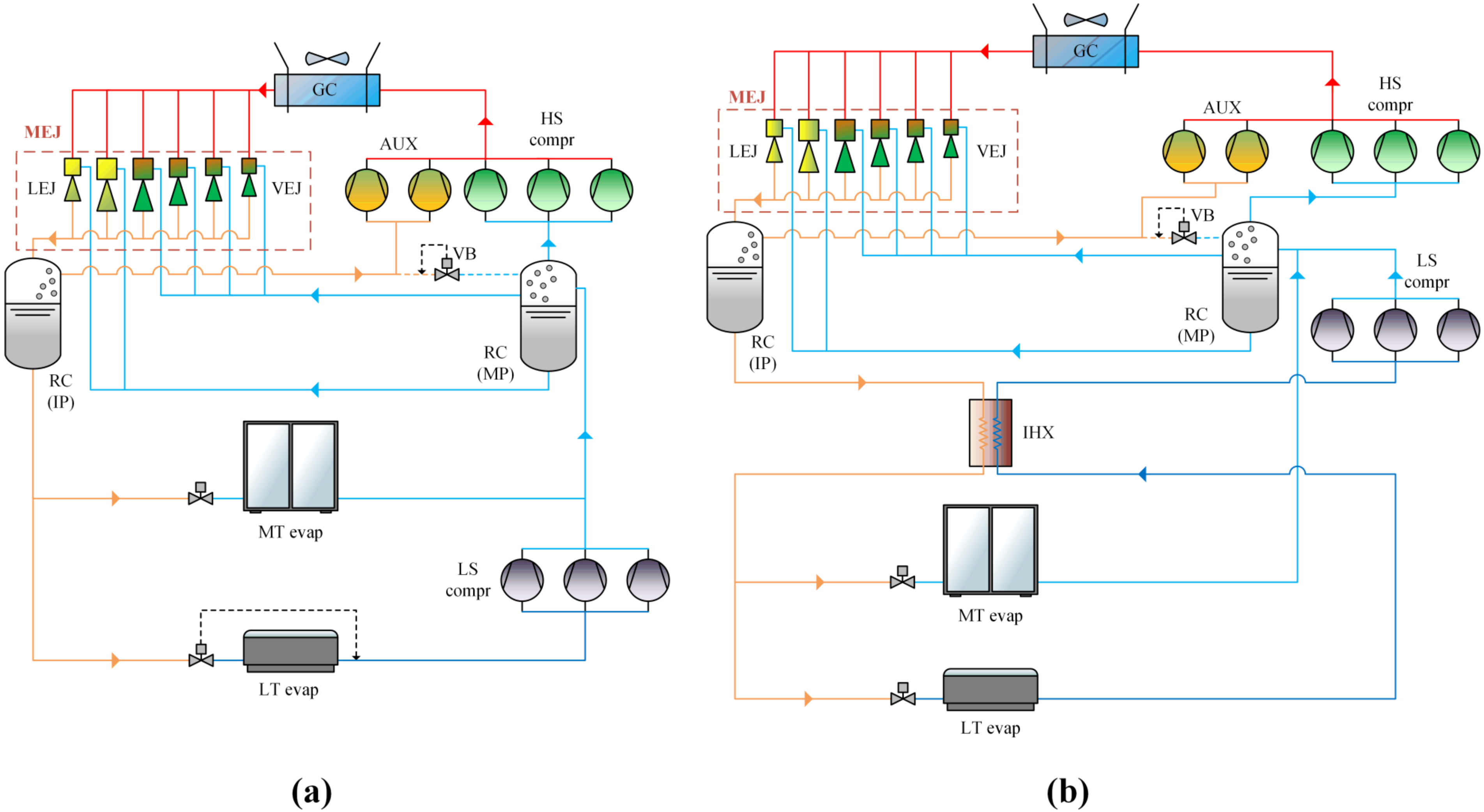
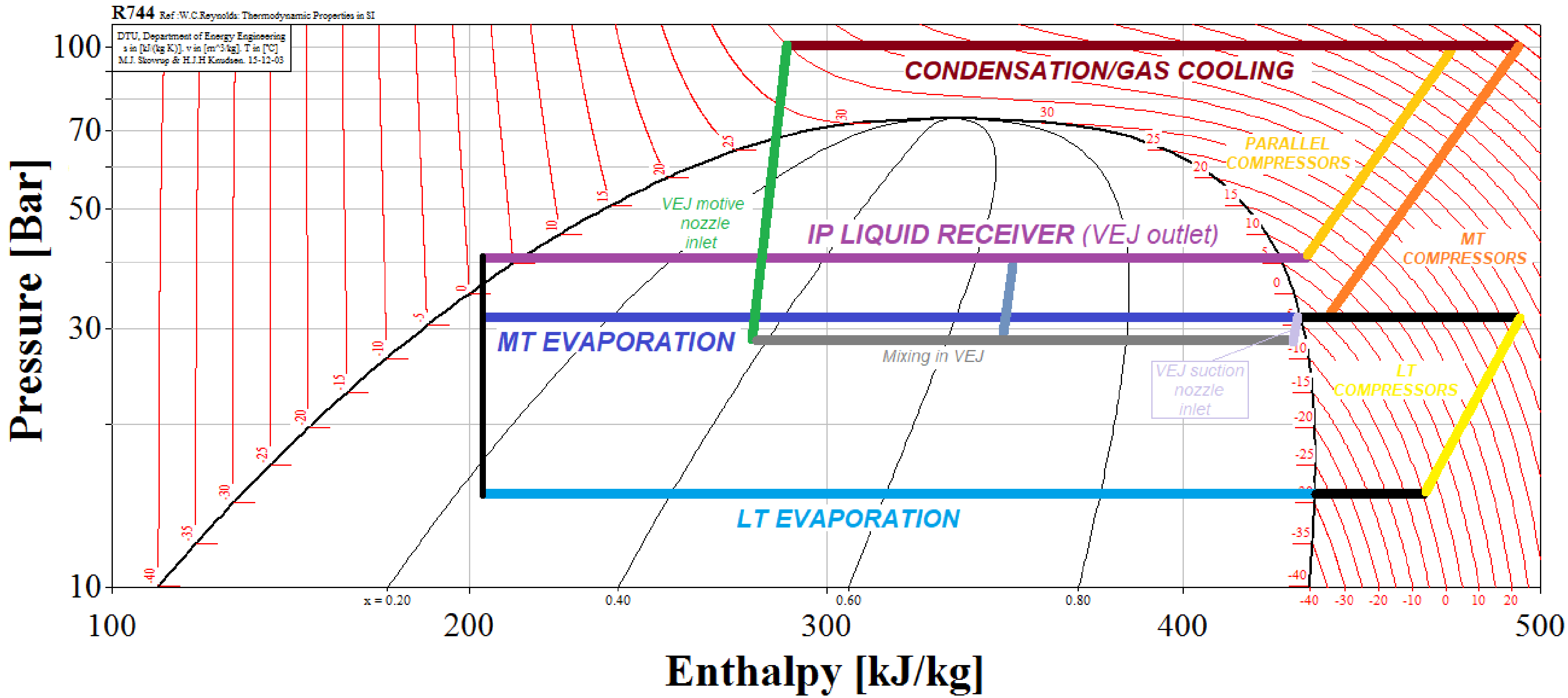
Fig. 1. Schematic of the investigated R744 multi-ejector enhanced parallel compression systems and simplified p-h diagram.



(c)



R744 Ref :W.C.Reynolds: Thermodynamic Properties in SI
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 15-12-03
CONDENSATION/GAS COOLING
PARALLEL COMPRESSORS
MT COMPRESSORS
LT COMPRESSORS
VEJ motive nozzle inlet
IP LIQUID RECEIVER (VEJ outlet)
MT EVAPORATION
Mixing in VEJ
VEJ suction nozzle inlet
LT EVAPORATION
Pressure [Bar]
Enthalpy [kJ/kg]
100
70
50
30
10
100
200
300
400
500
x = 0.20
0.40
0.60
0.80

Fig. 2. Schematic of the investigated indirect supermarket refrigeration systems.

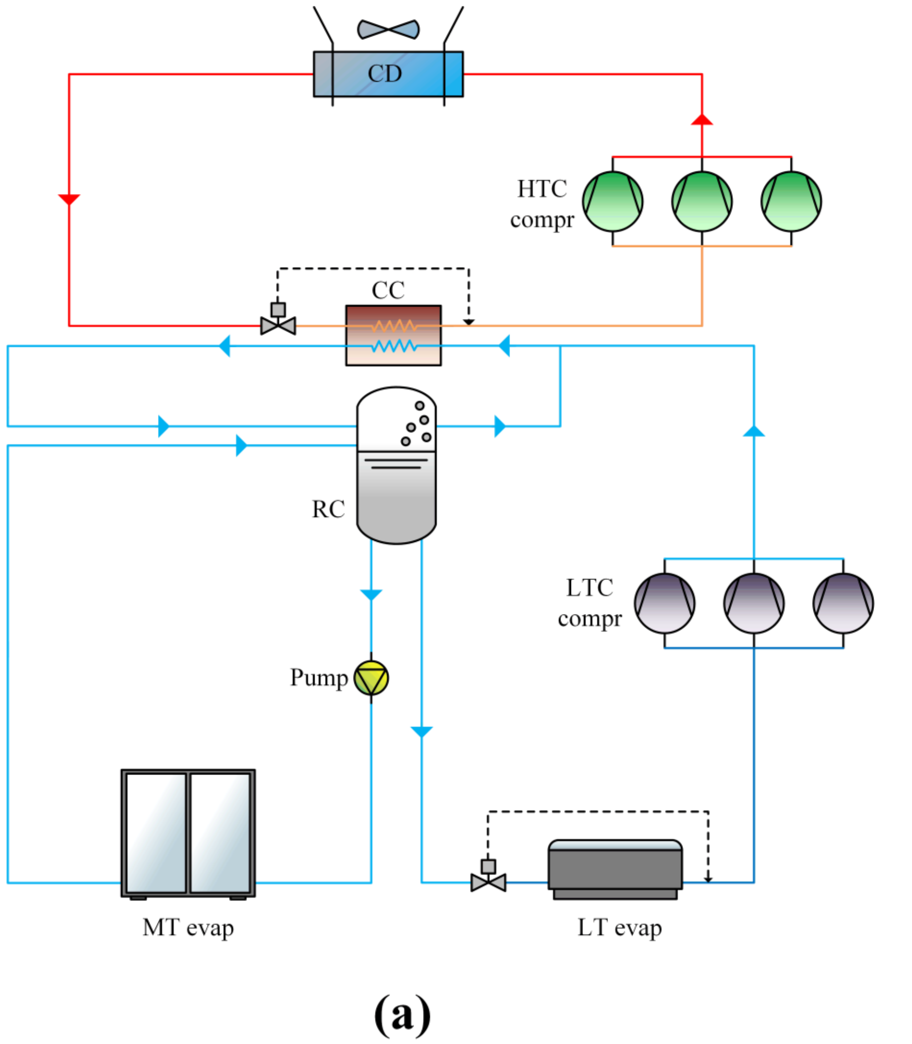


Fig. 3. Schematic of the investigated cascade supermarket refrigeration systems.

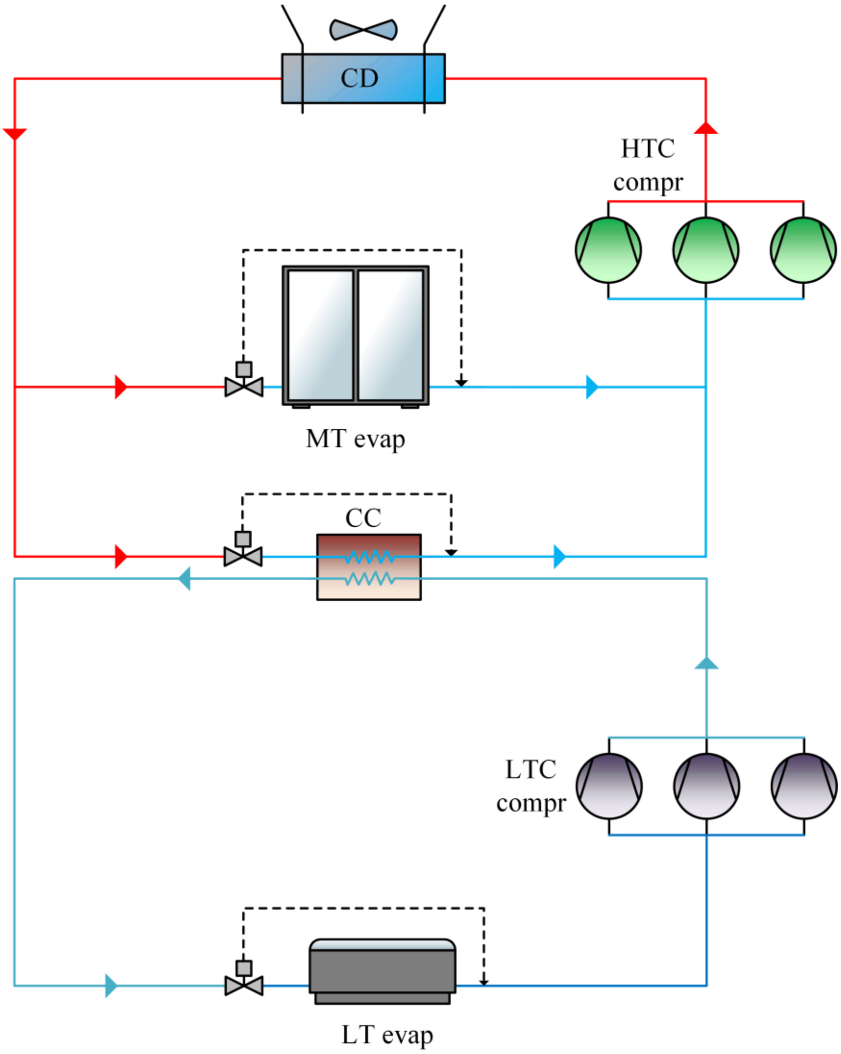


Fig. 4. Schematic of an "all-in-one" R744 multi-ejector enhanced parallel compression system.

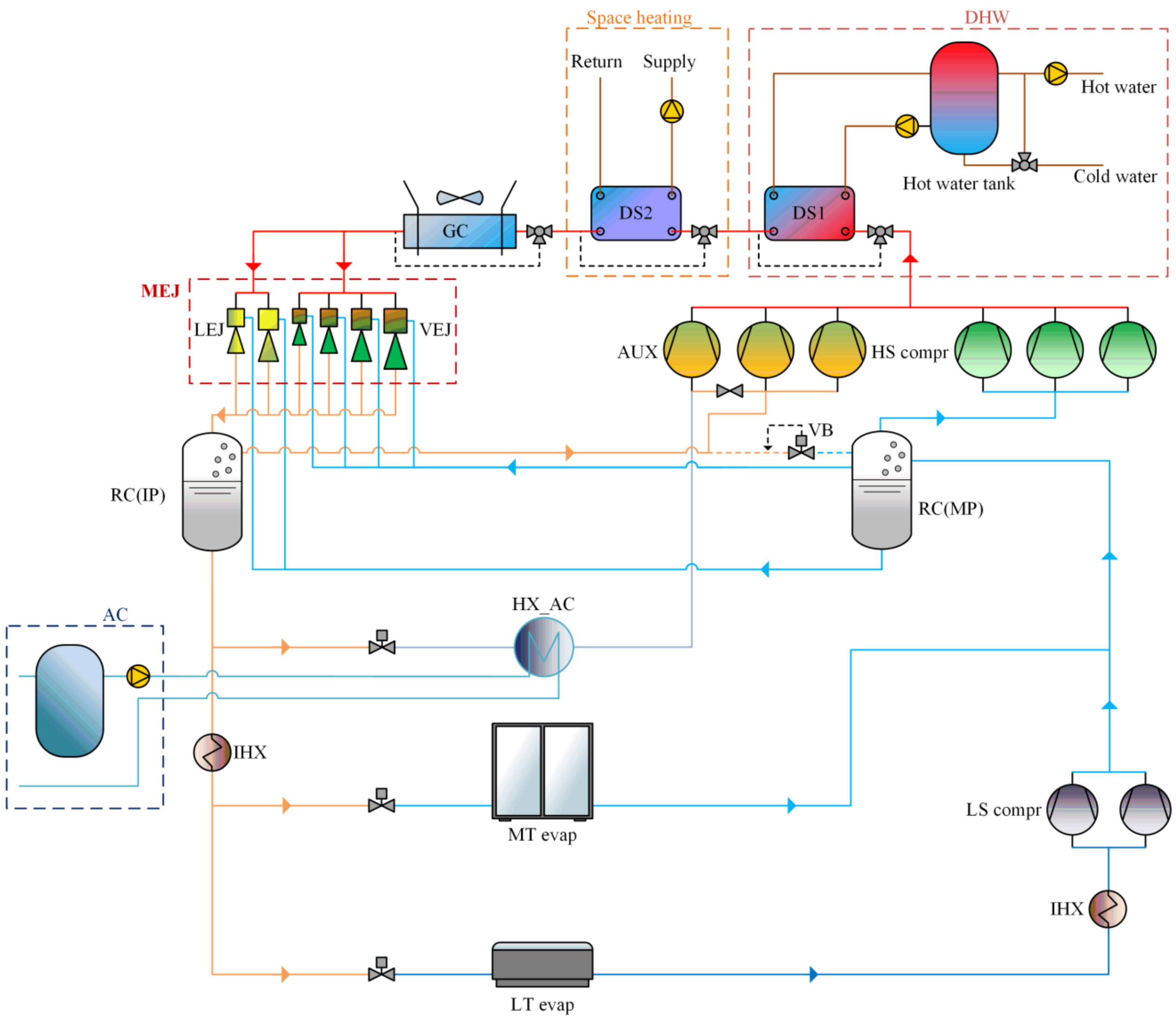


Fig. 5. Schematic of the investigated indirect supermarket refrigeration systems integrated with AC equipment.

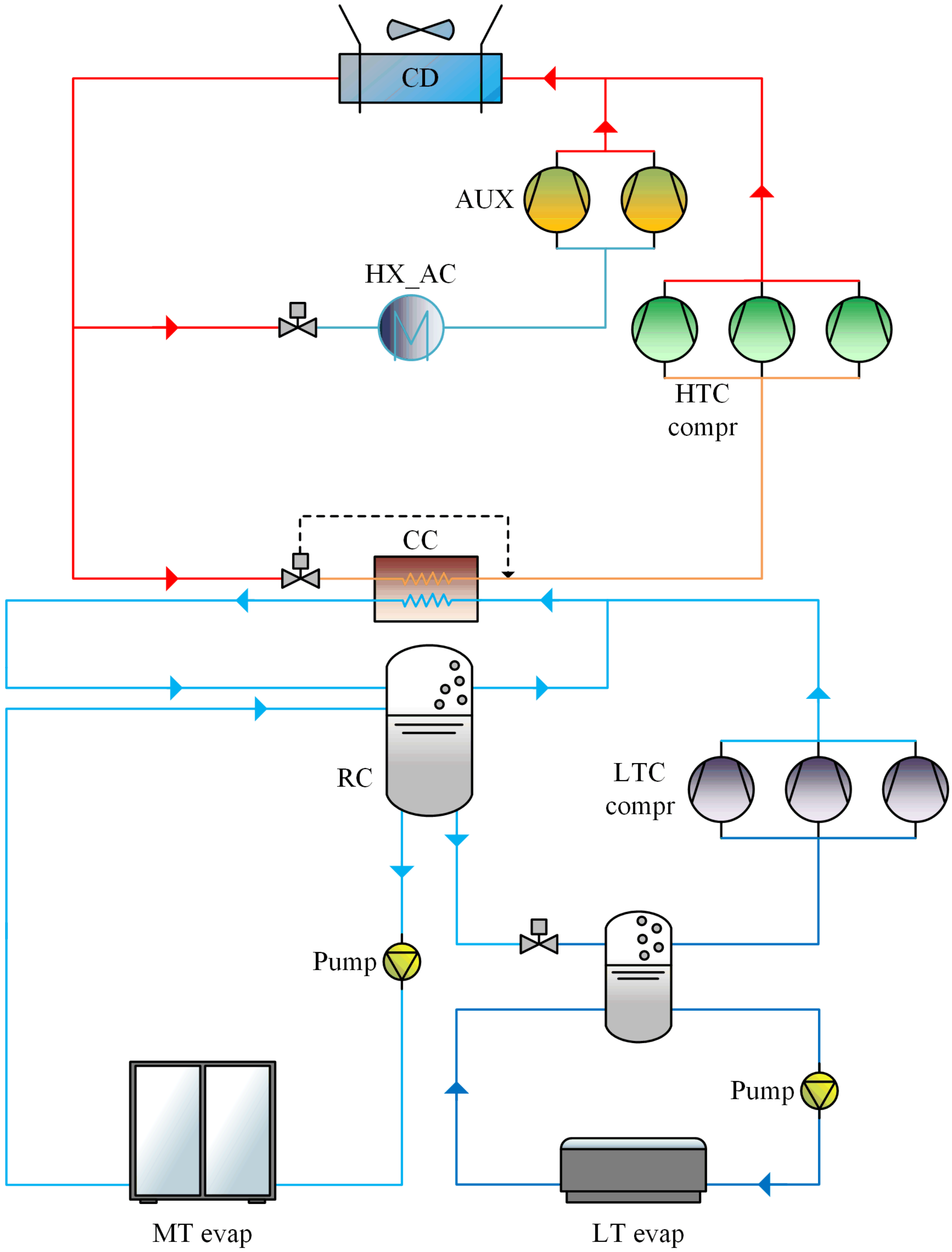


Fig. 6. Temperature bins for the investigated cities (EnergyPlus, 2018).

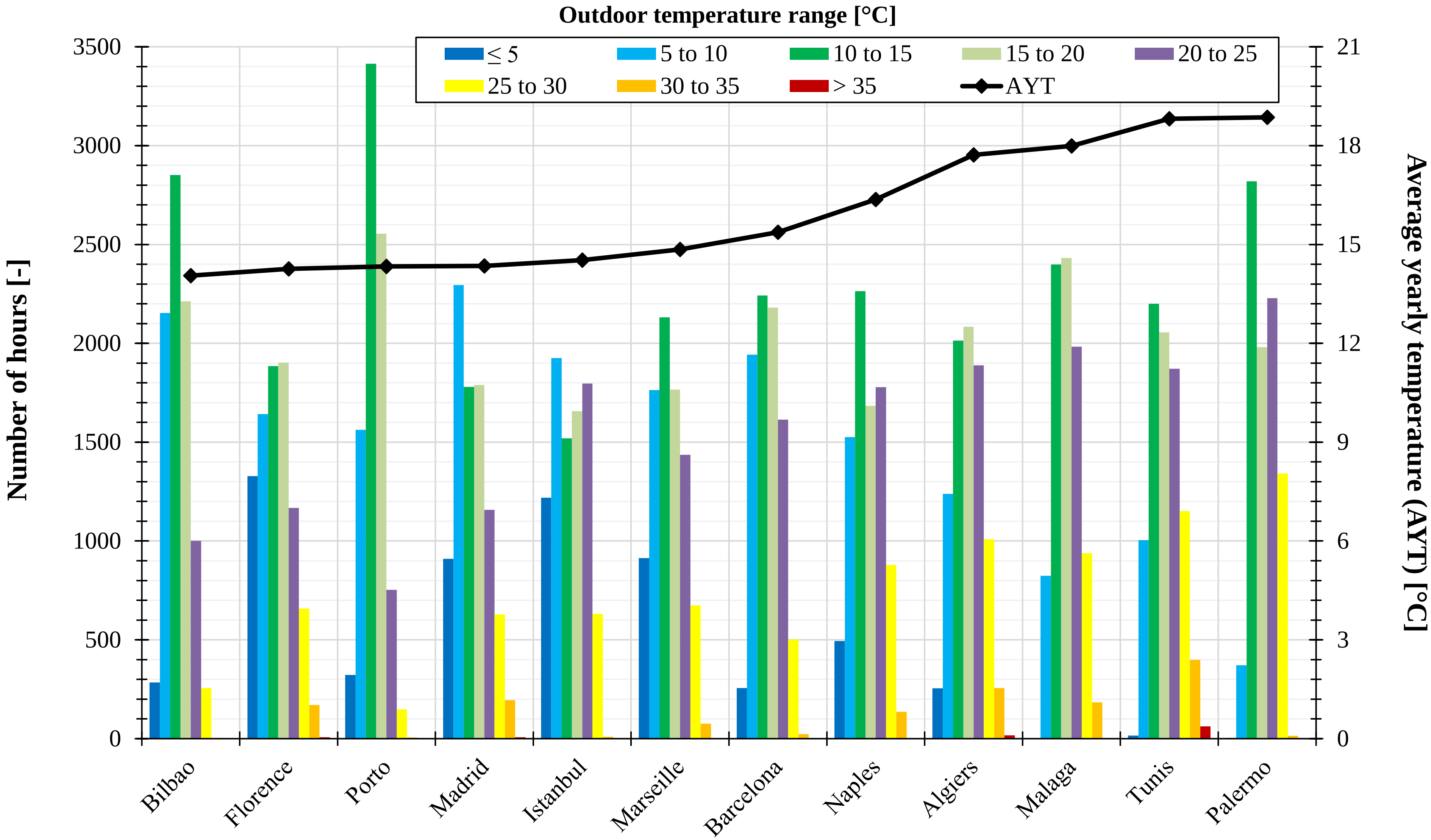


Fig. 7. Comparison in terms of COP among the investigated solutions (scenario with no AC demand).

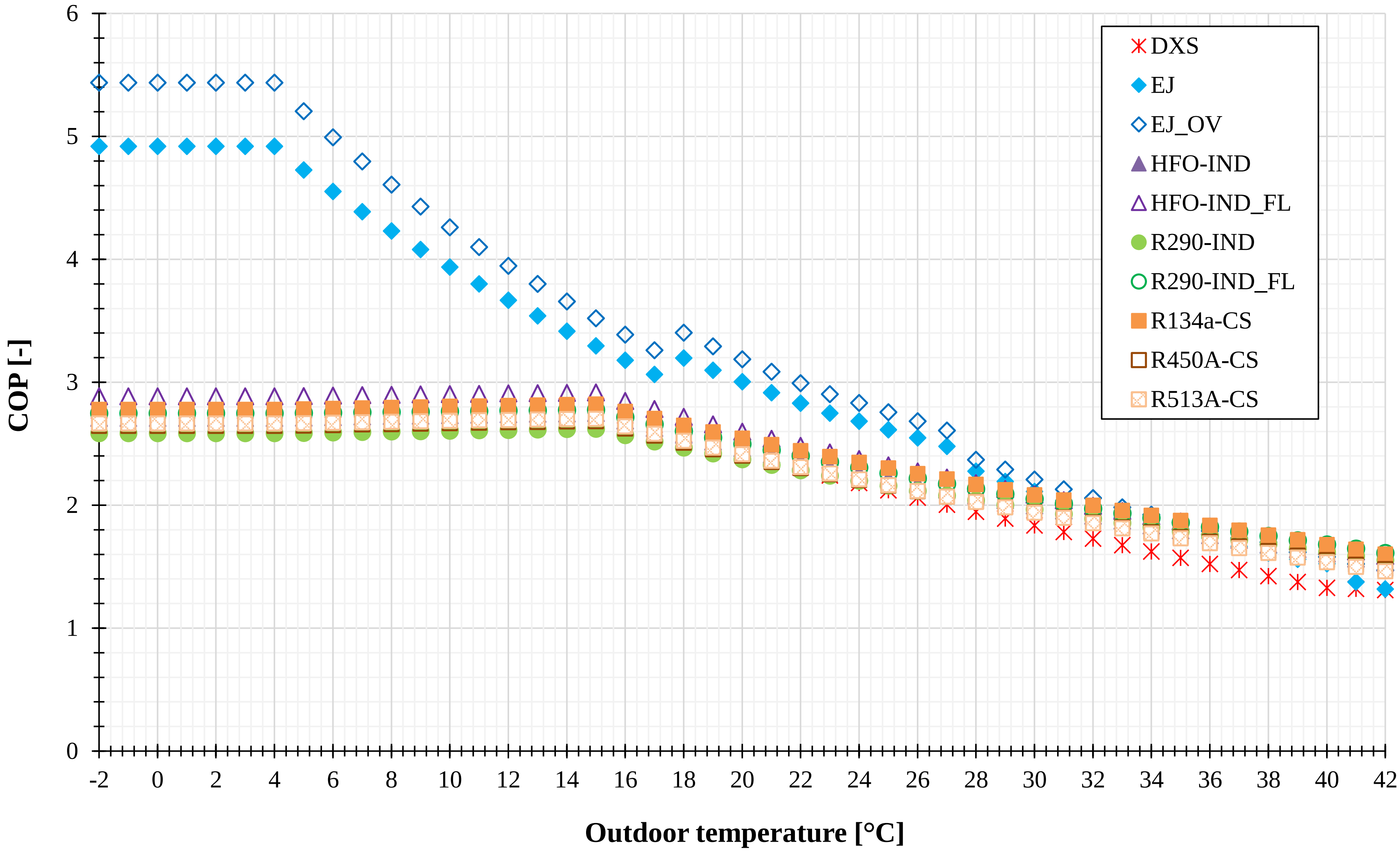


Fig. 8. Comparison in terms of annual energy consumption [MWh] among the investigated solutions (scenario with no AC demand).

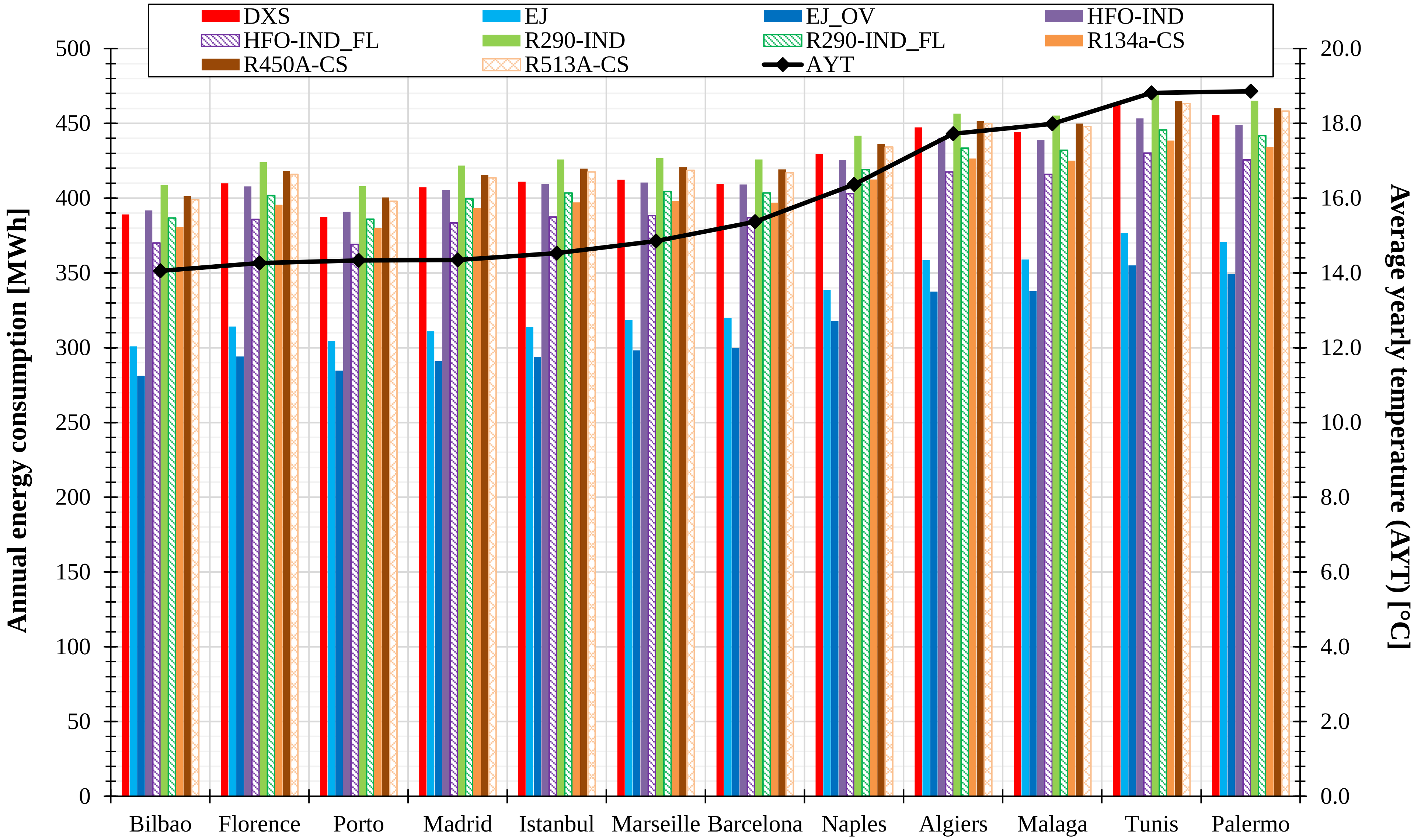


Fig. 9. Comparison in terms of difference in annual energy consumption [%] (DXS as the reference) among the investigated solutions (scenario with no AC demand).

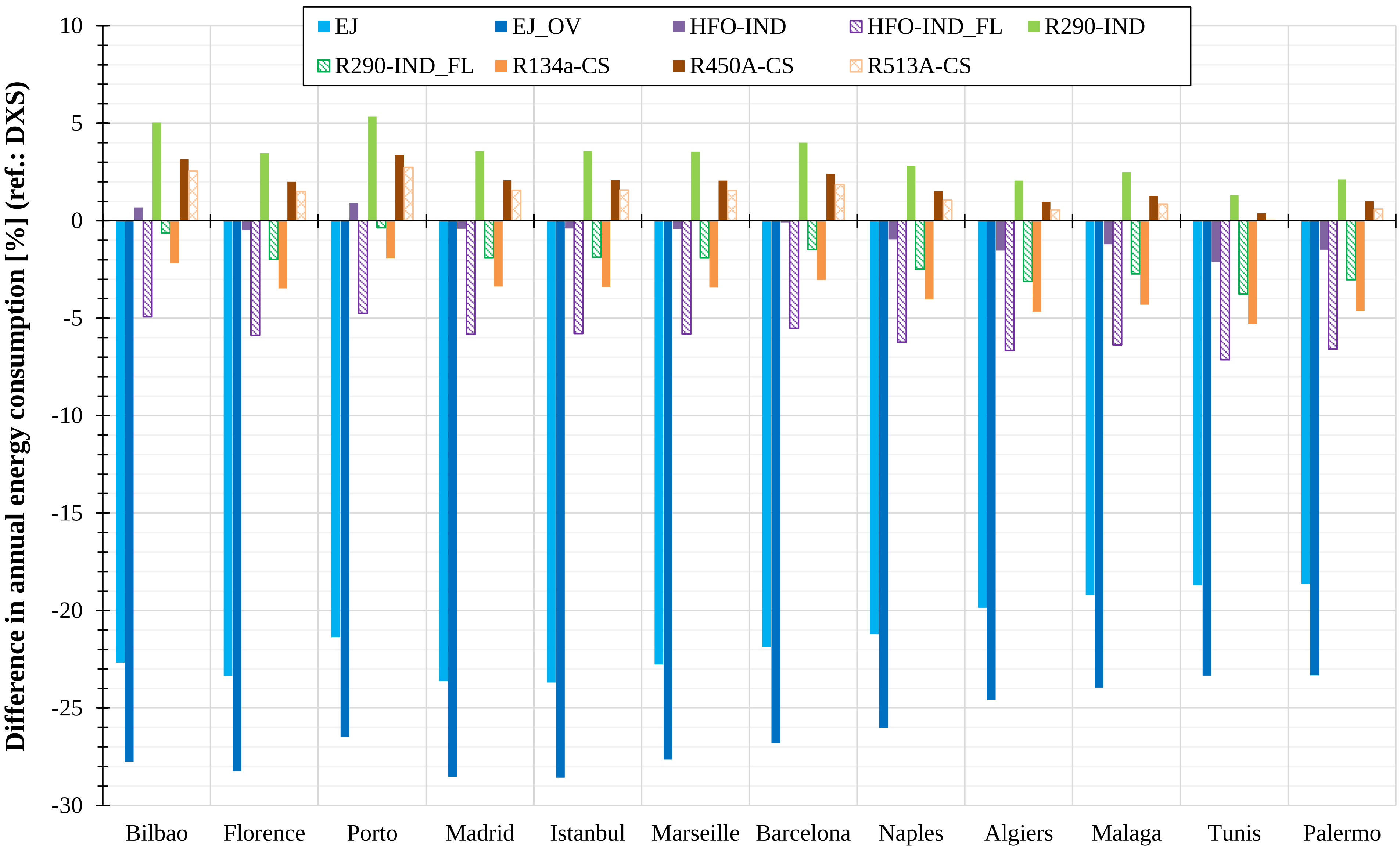


Fig. 10. Comparison in terms of TEWI [$ton_{CO_{2,equivalent}}$] among the investigated solutions (scenario with no AC demand).

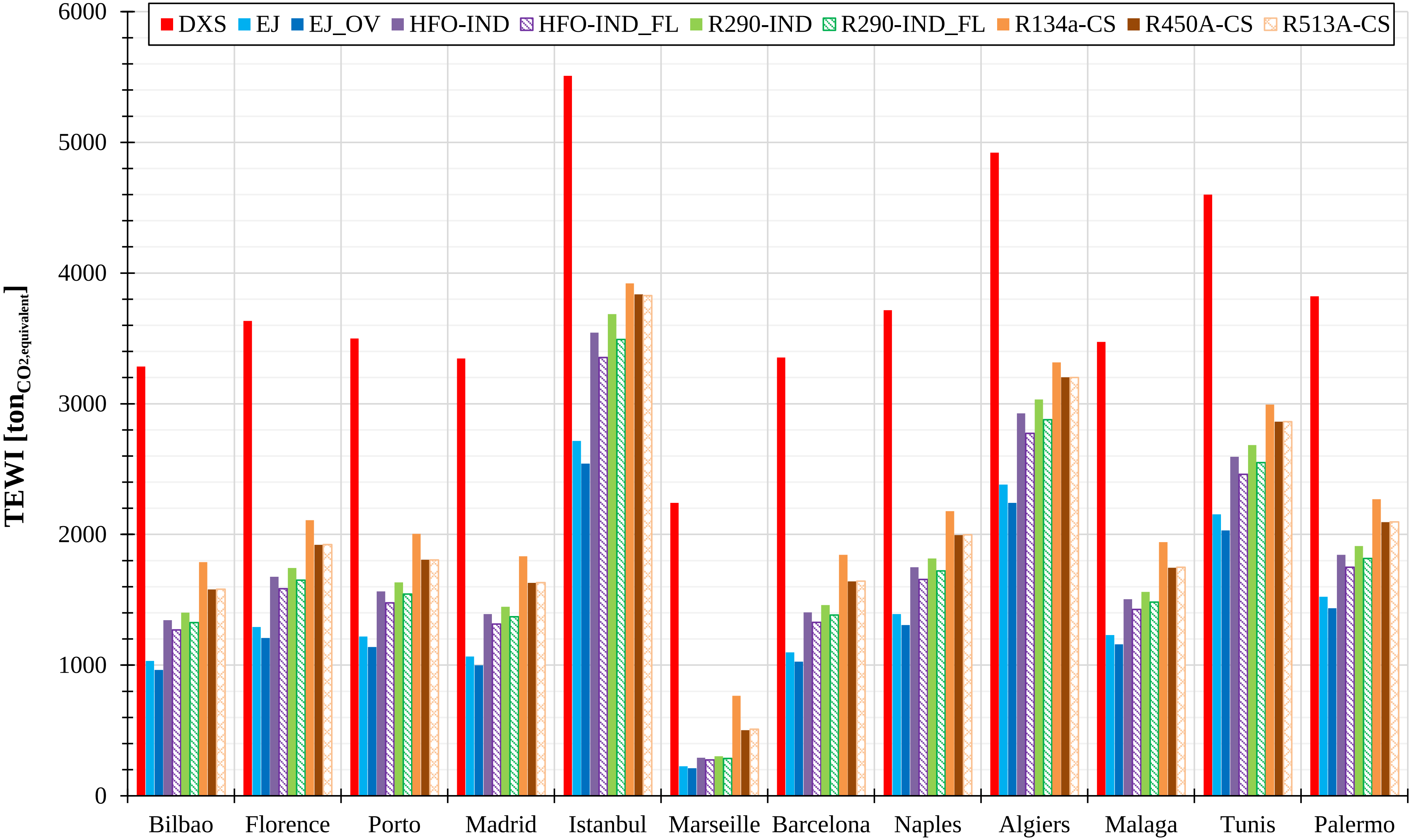


Fig. 11. Comparison in terms of difference in TEWI [%] (DXS as the reference) among the investigated solutions (scenario with no AC demand).

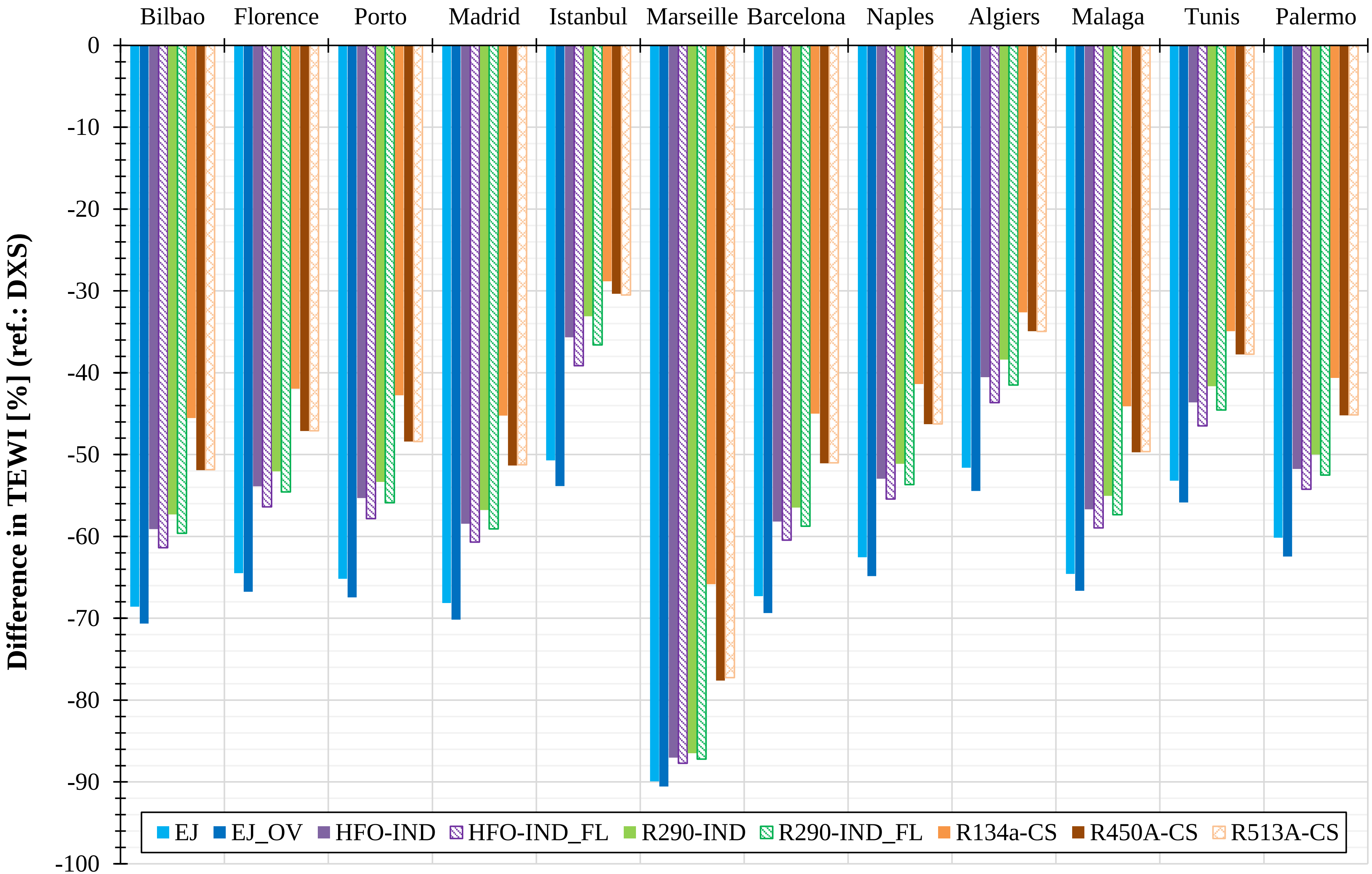


Fig. 12.(a) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with DXS (scenario with no AC demand).

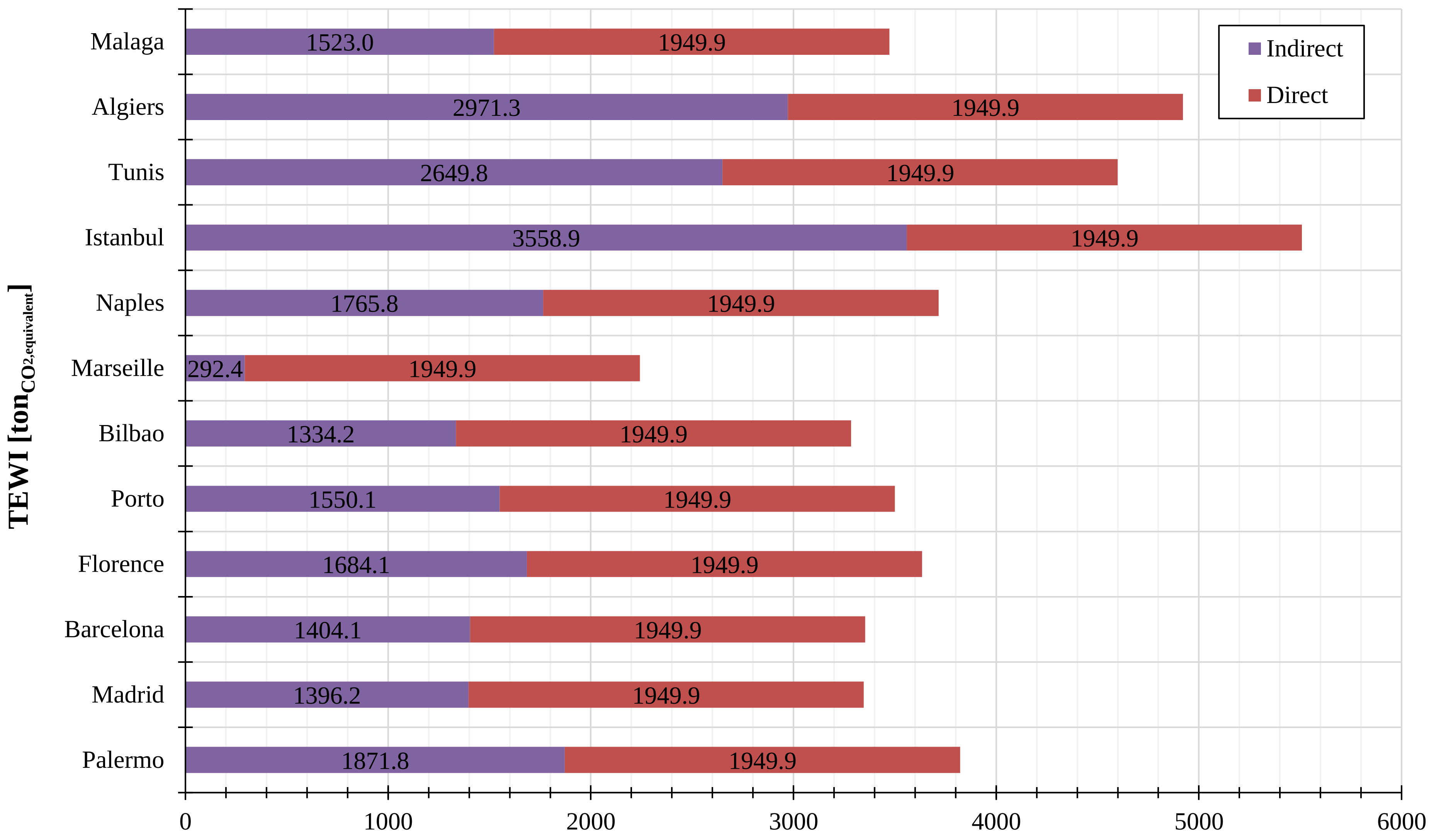


Fig. 12.(b) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with R134a-CS (scenario with no AC demand).

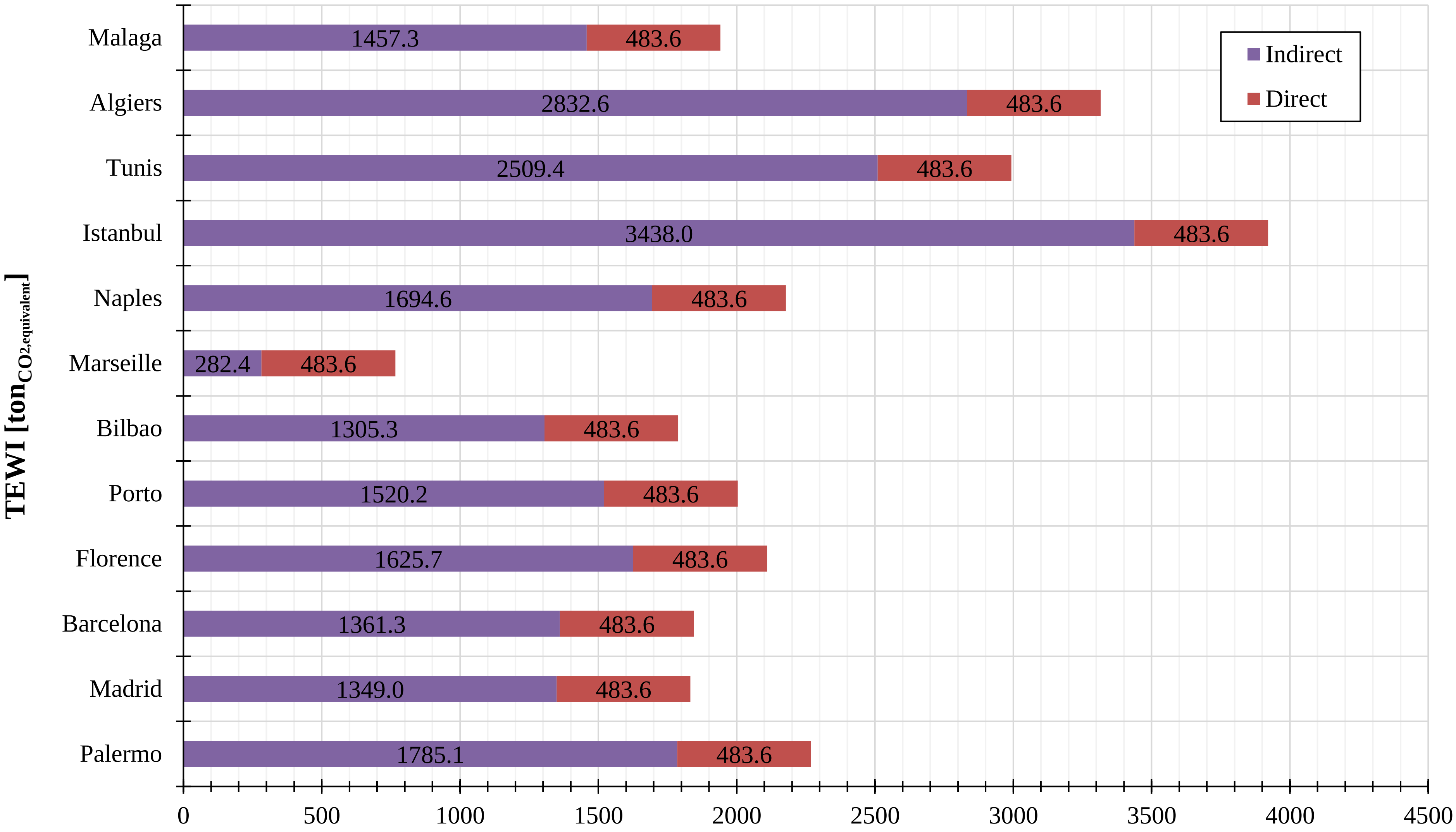


Fig. 12.(c) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with R450A-CS (scenario with no AC demand).

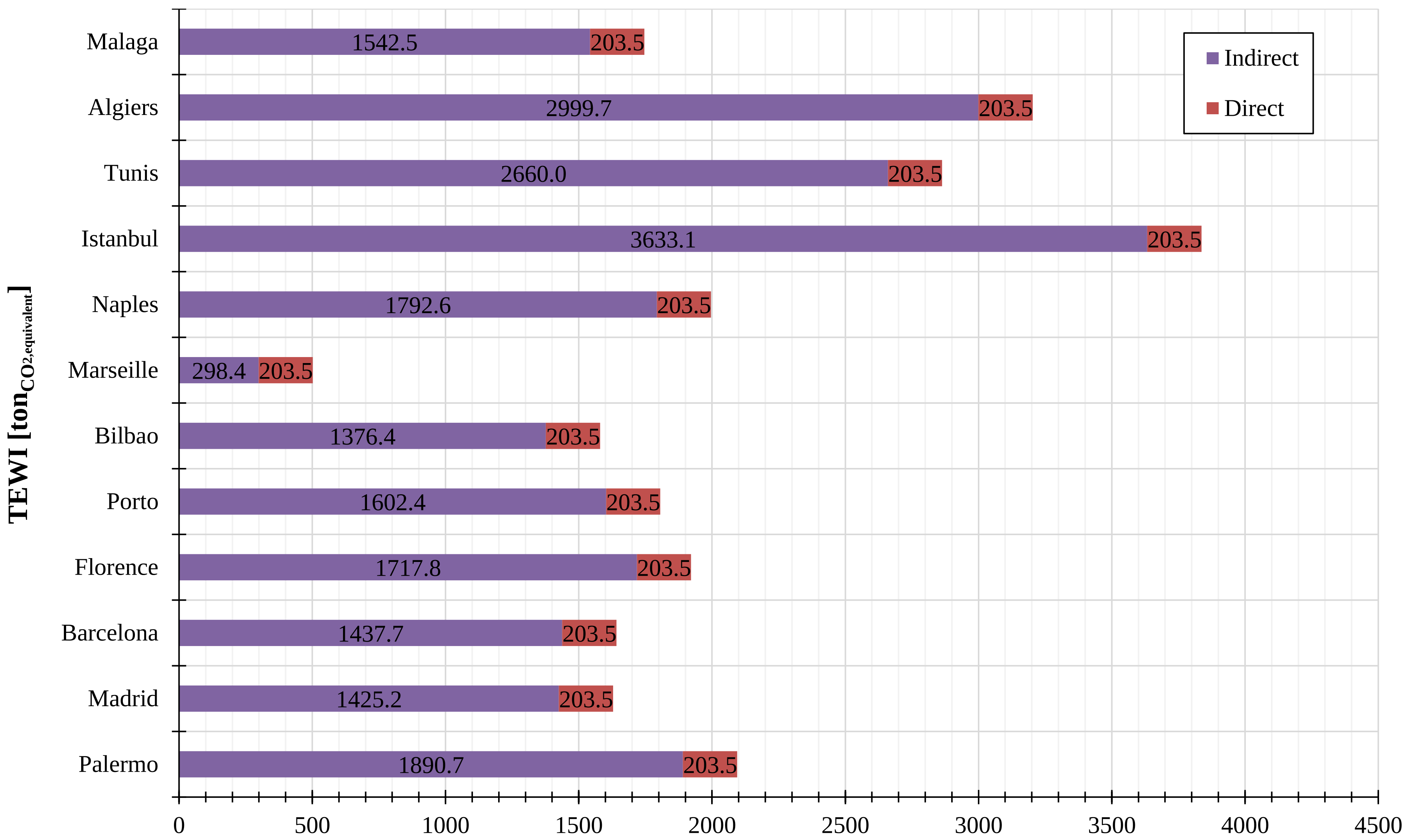


Fig. 12.(d) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with R513A-CS (scenario with no AC demand).

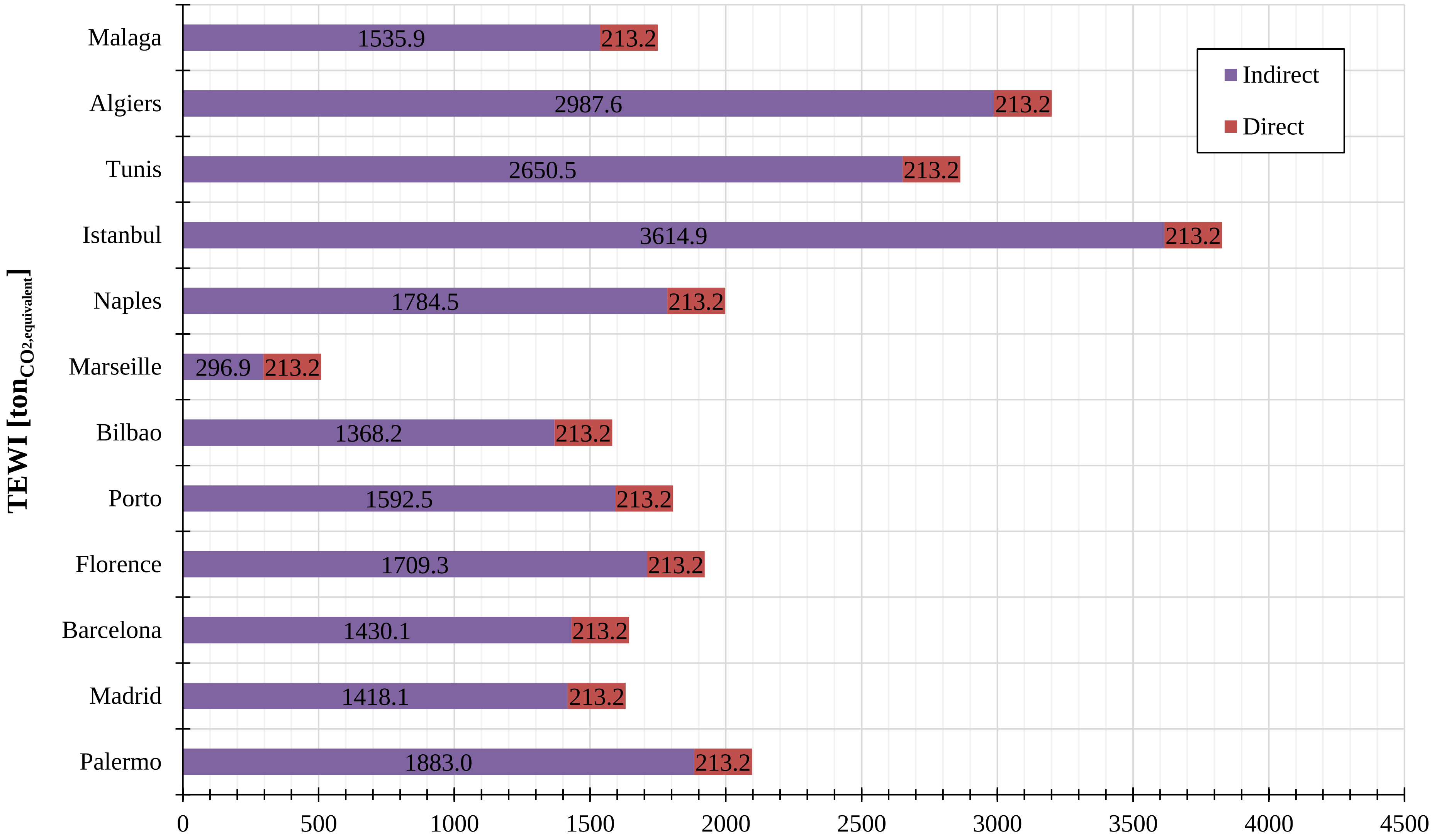


Fig. 13.(a) Power input [kW] of DXS+R410A CH at different outdoor temperatures.

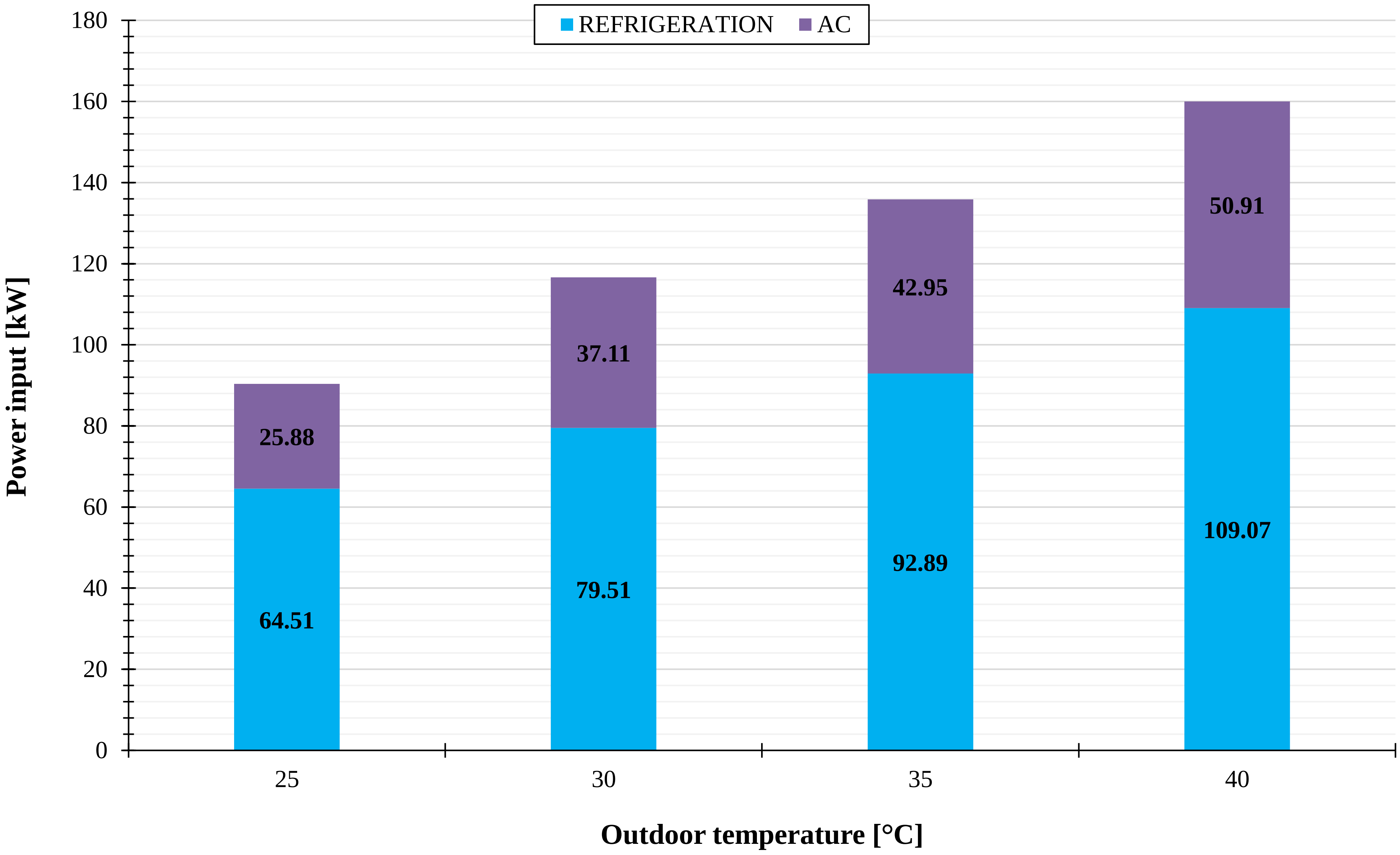


Fig. 13.(b) Power input [kW] of EJ_OV_AC at different outdoor temperatures.

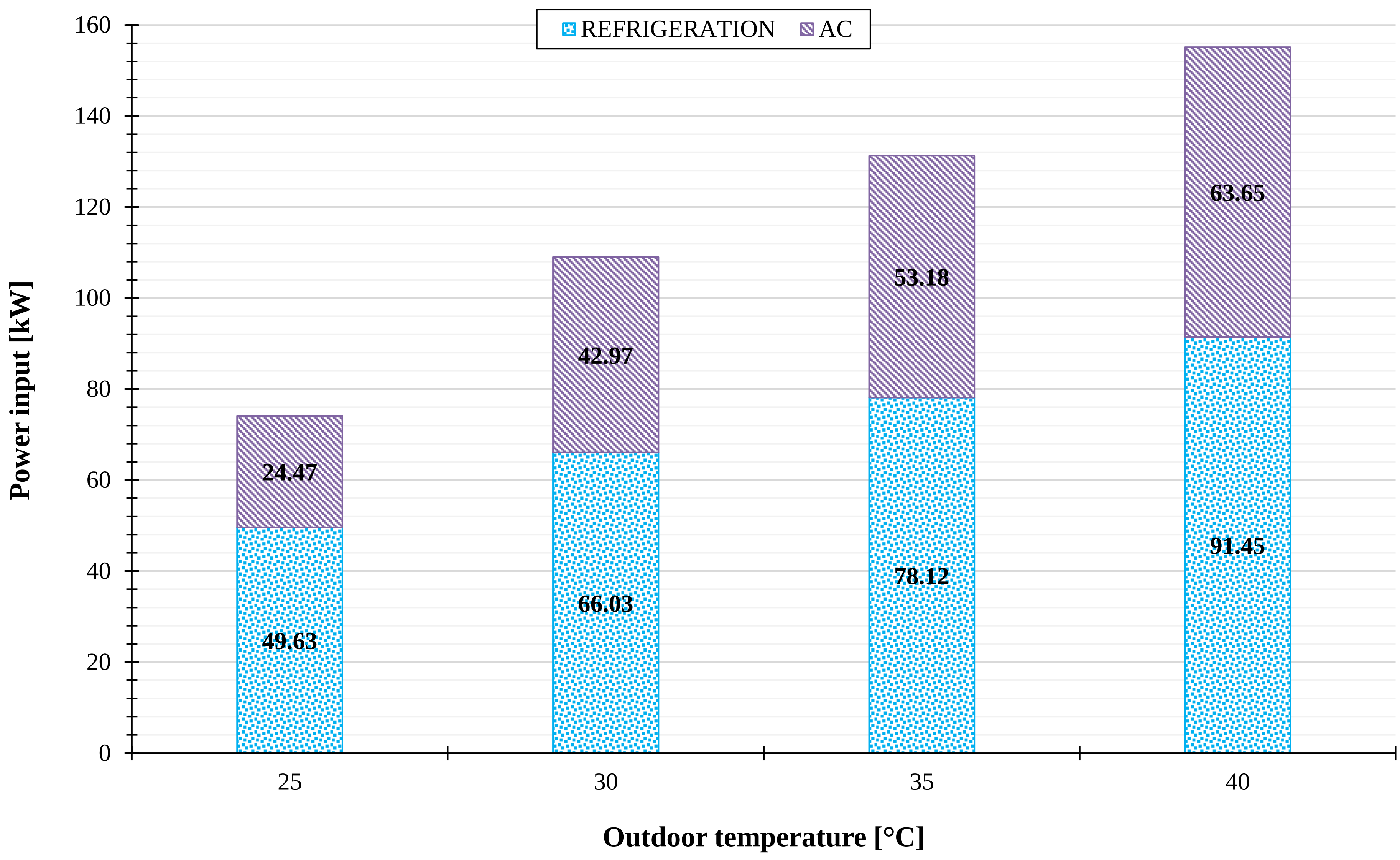


Fig. 14. Comparison in terms of annual energy consumption [MWh] among the investigated solutions (scenario including AC demand).

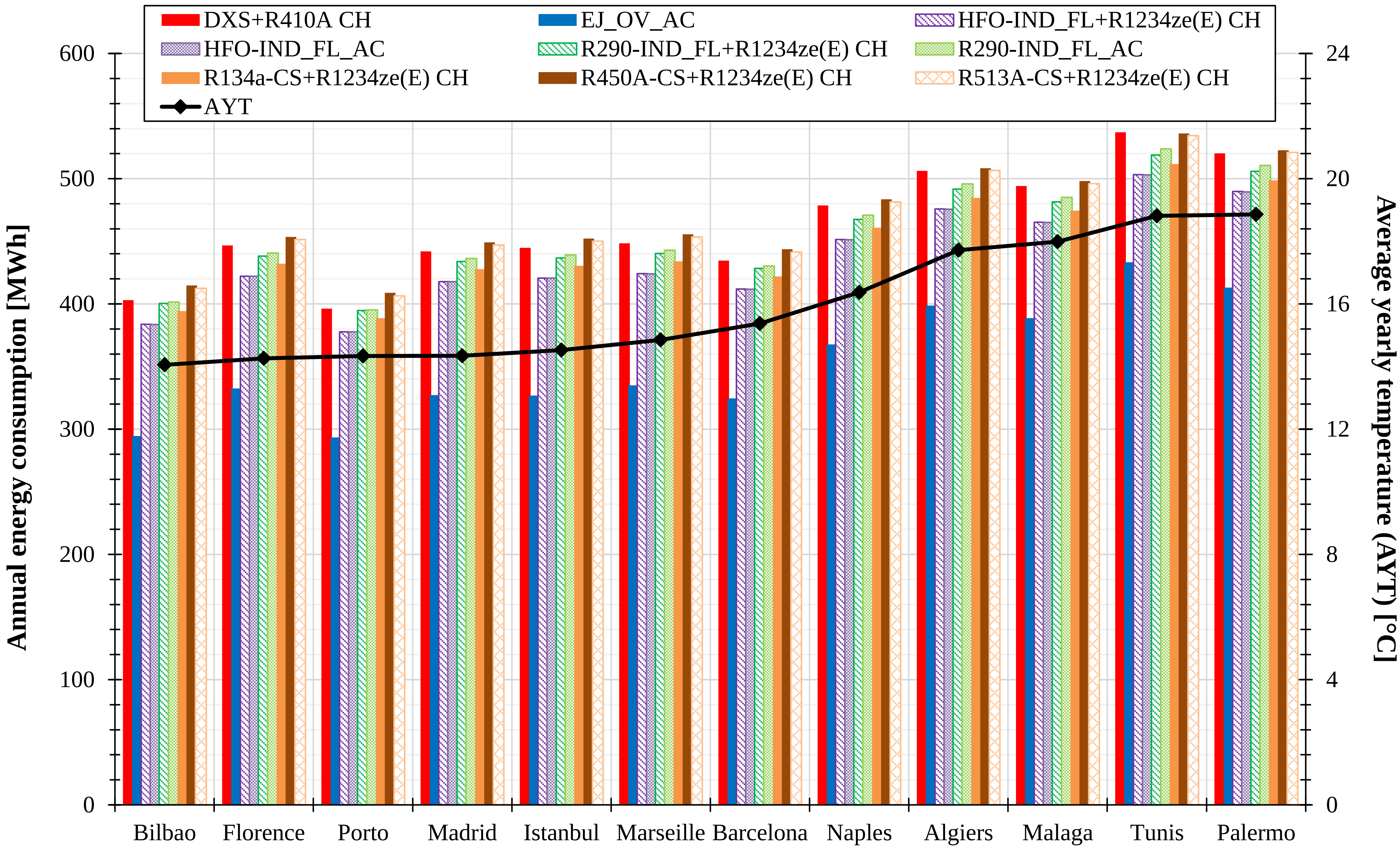


Fig. 15. Comparison in terms of difference in annual energy consumption [%] (DXS+R410A CH as the reference) among the investigated solutions (scenario including AC demand).

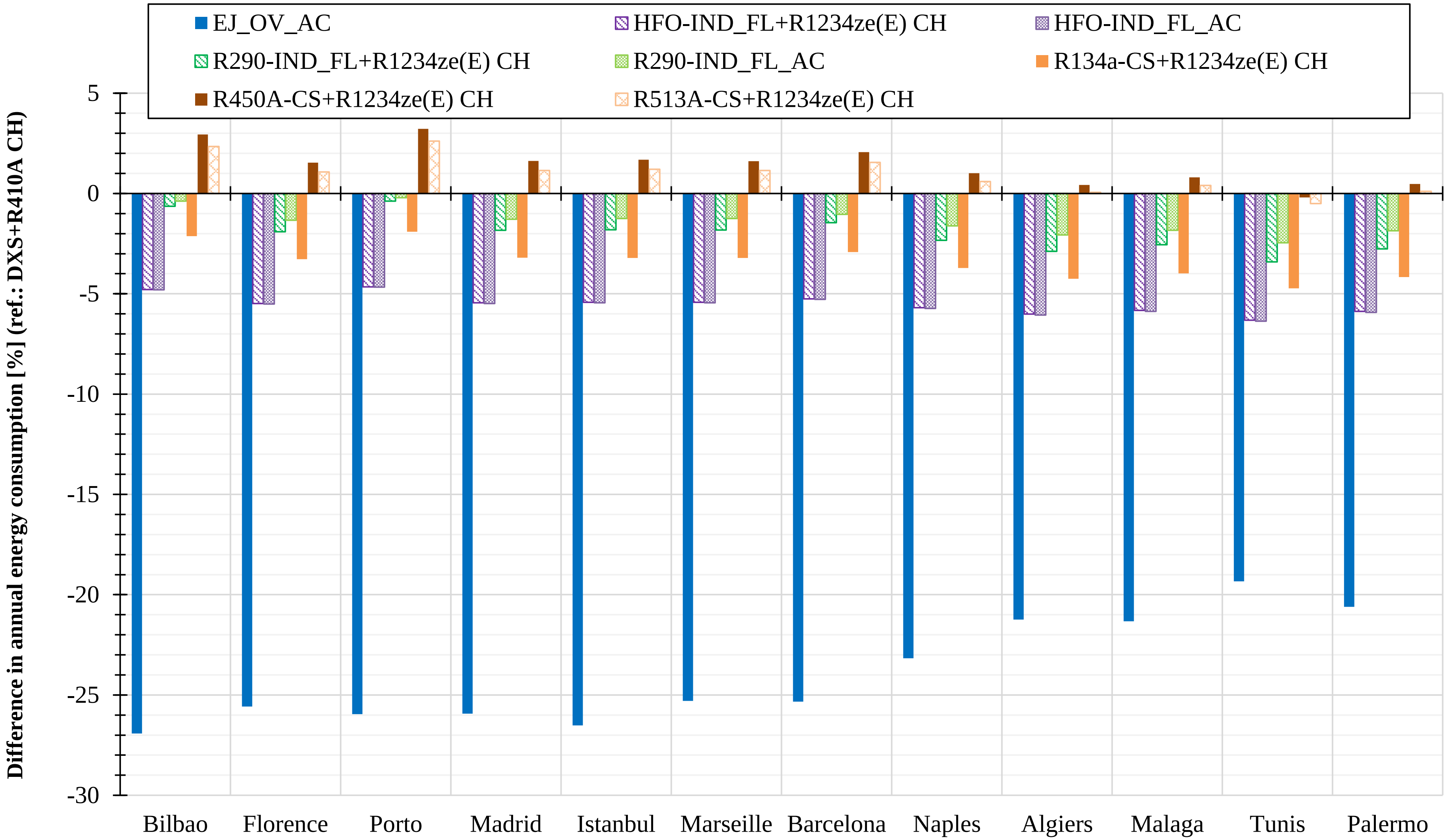


Fig. 16. Comparison in terms of TEWI [$ton_{CO_{2,equivalent}}$] among the investigated solutions (scenario including AC demand).

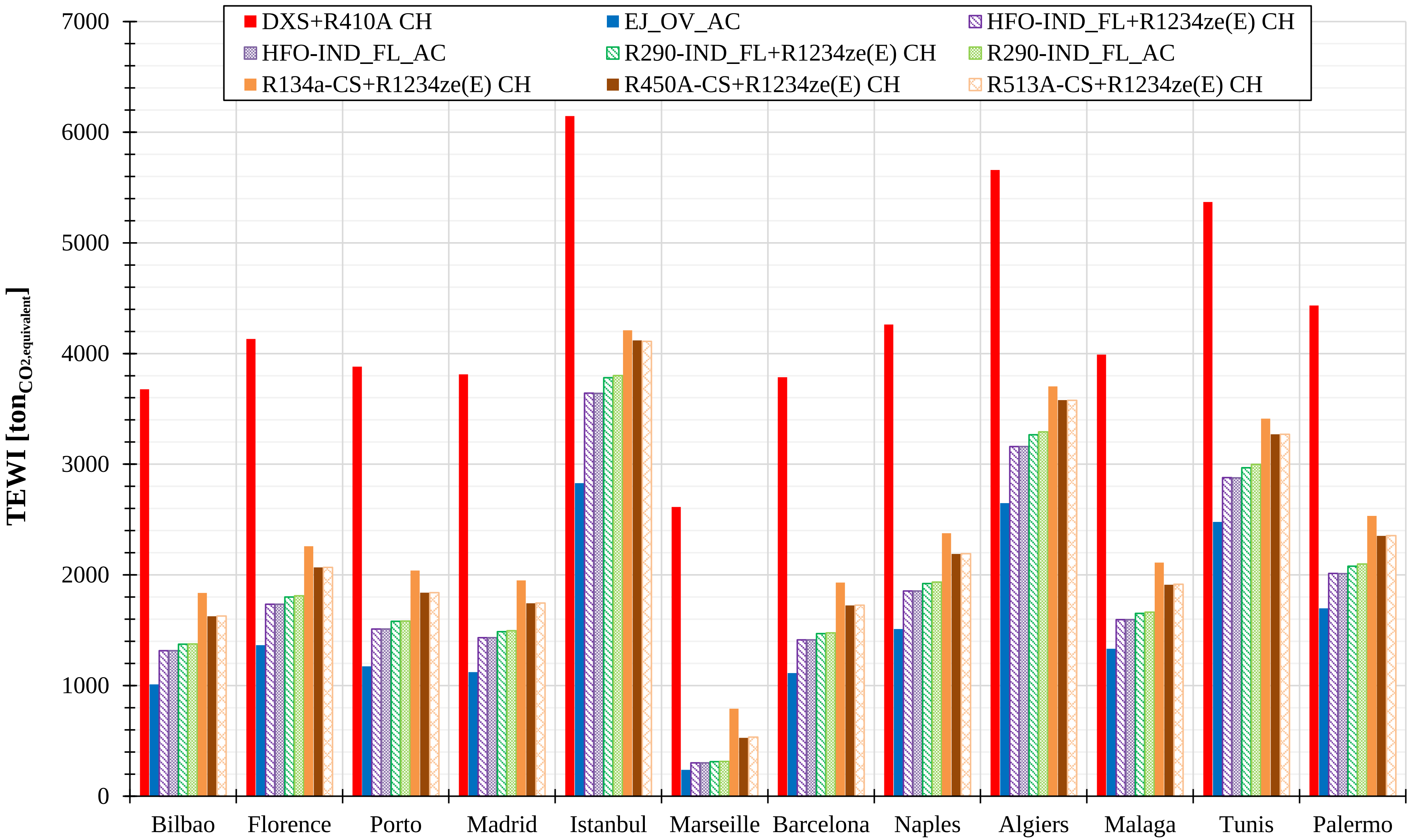


Fig. 17. Comparison in terms of difference in TEWI [%] (DXS+R410A CH as the reference) among the investigated solutions (scenario including AC demand).

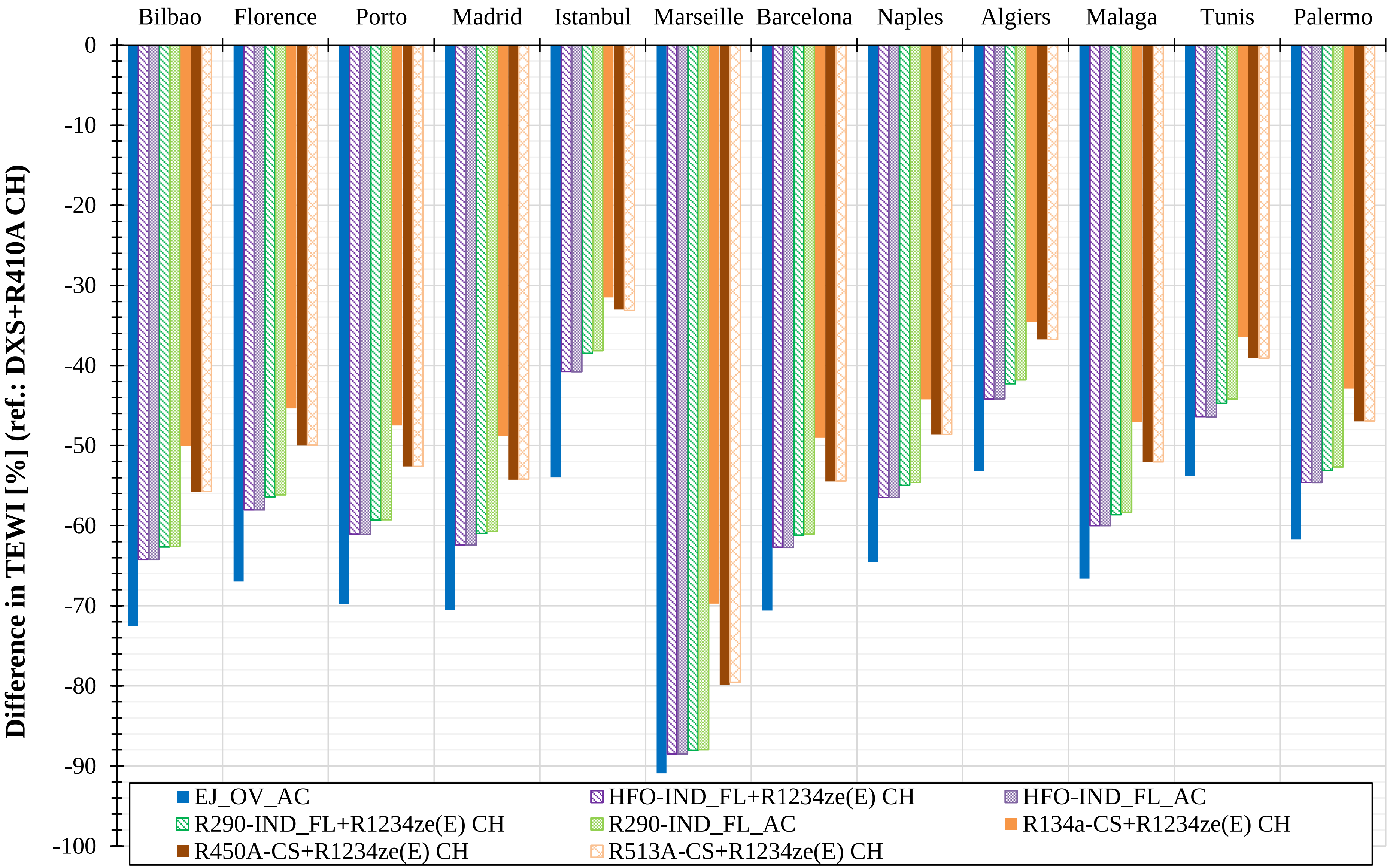


Fig. 18.(a) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with DXS+R410A CH (scenario including AC demand).

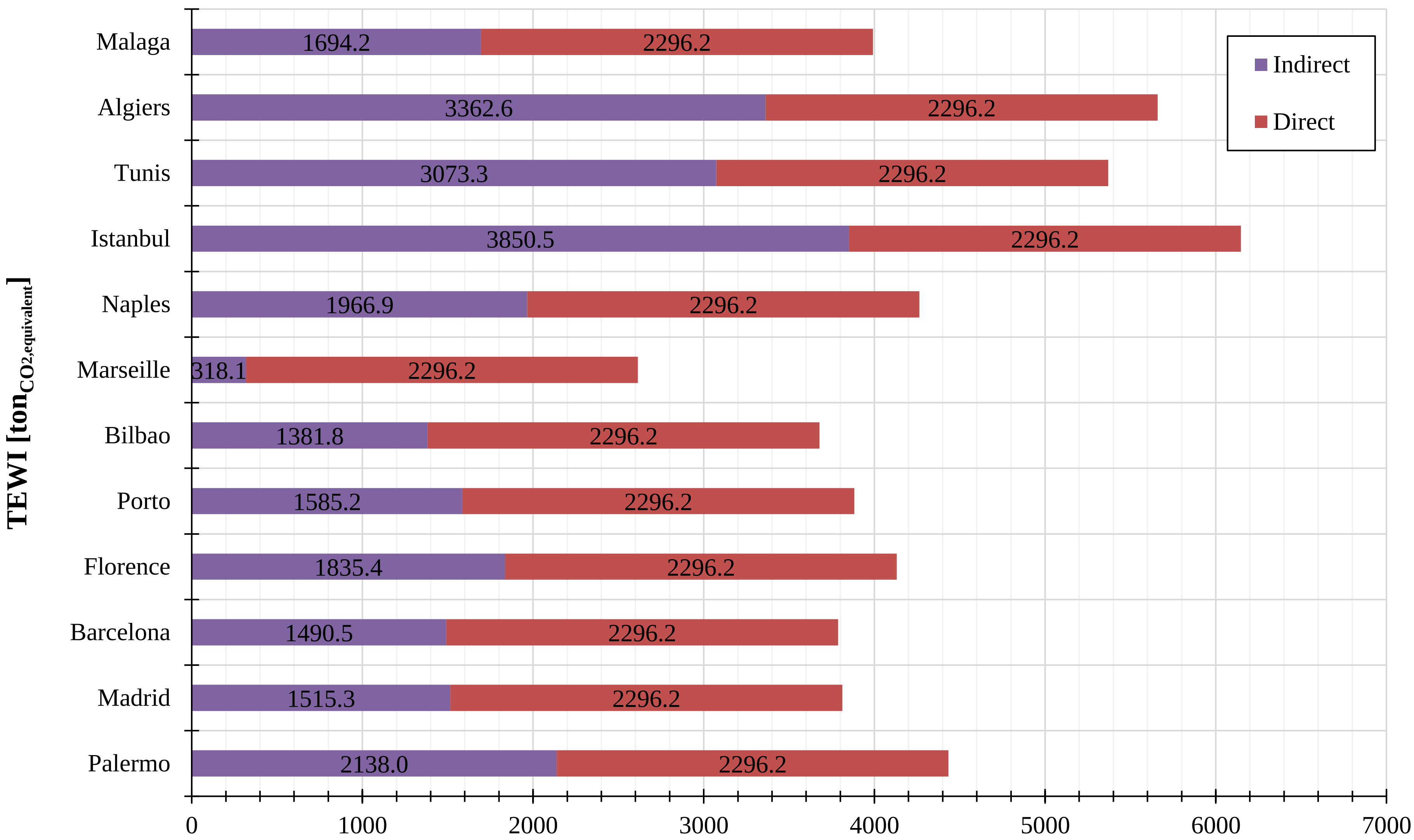


Fig. 18.(b) Direct and indirect TEWI [$ton_{CO_{2,equivalent}}$] associated with R134a-CS+R1234ze(E) CH (scenario including AC demand).

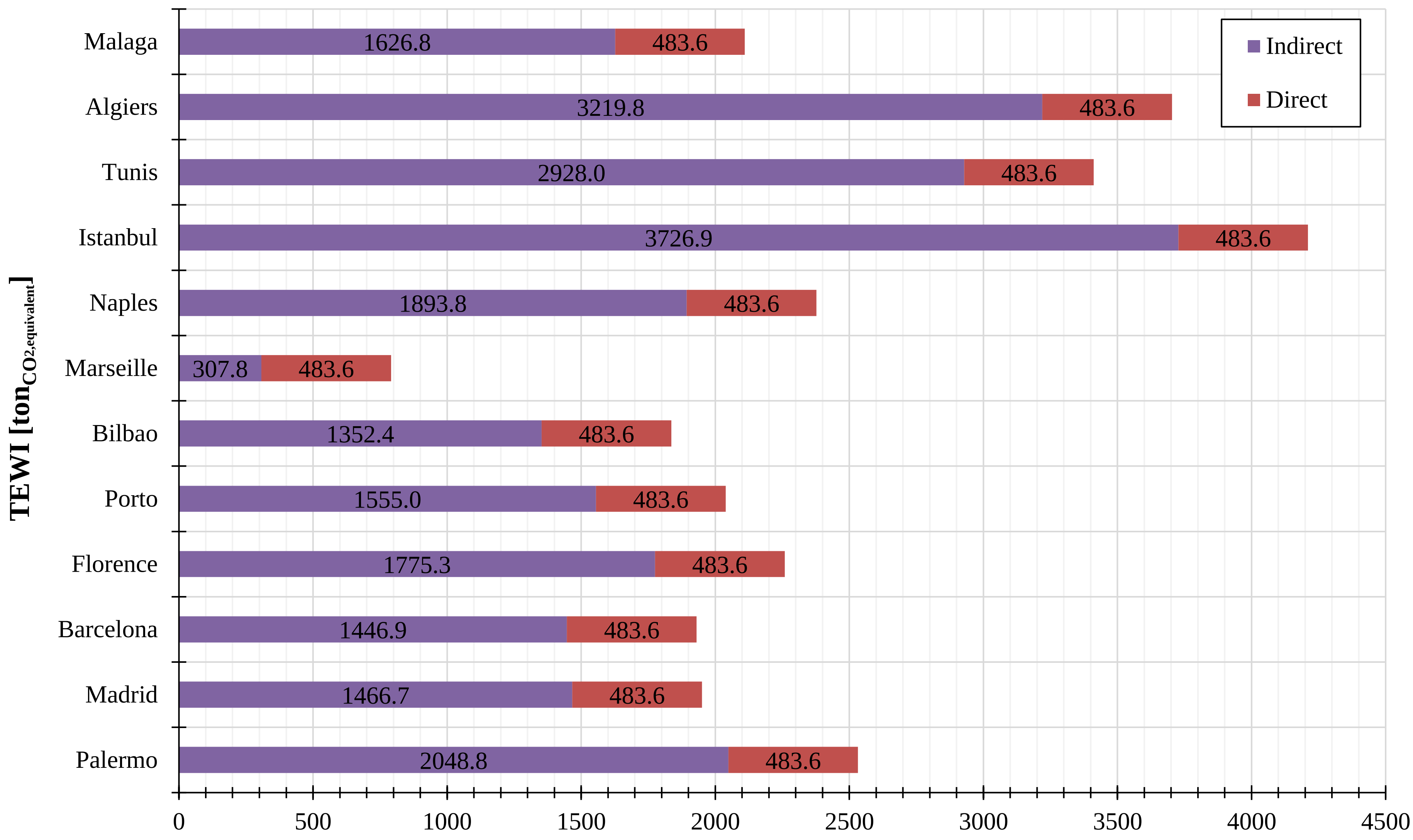


Fig. 18.(c) Direct and indirect TEWI [$ton_{CO_2,equivalent}$] associated with R450A-CS+R1234ze(E) CH (scenario including AC demand).

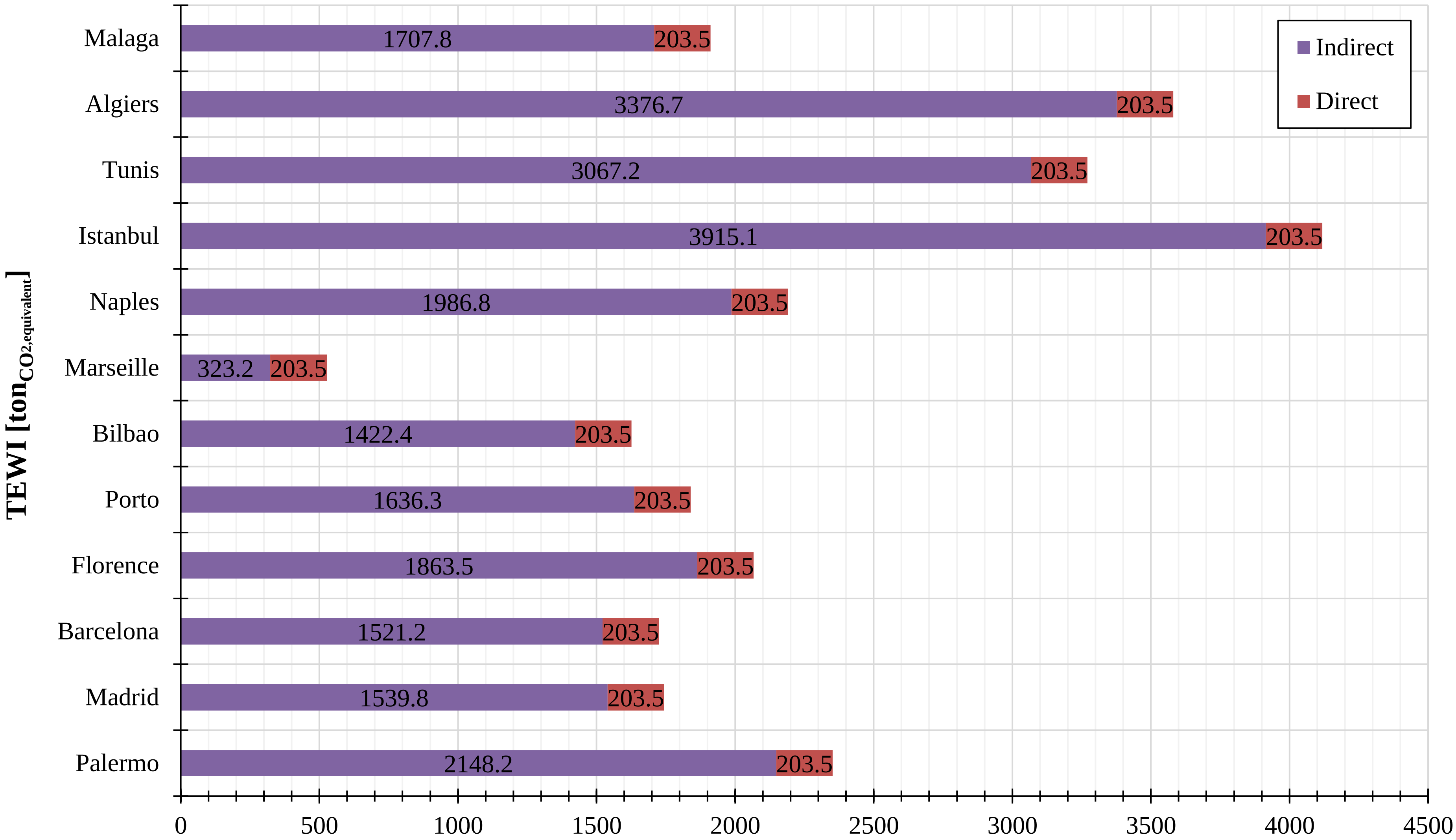


Fig. 18.(d) Direct and indirect TEWI [$ton_{CO_2,equivalent}$] associated with R513A-CS+R1234ze(E) CH (scenario including AC demand).

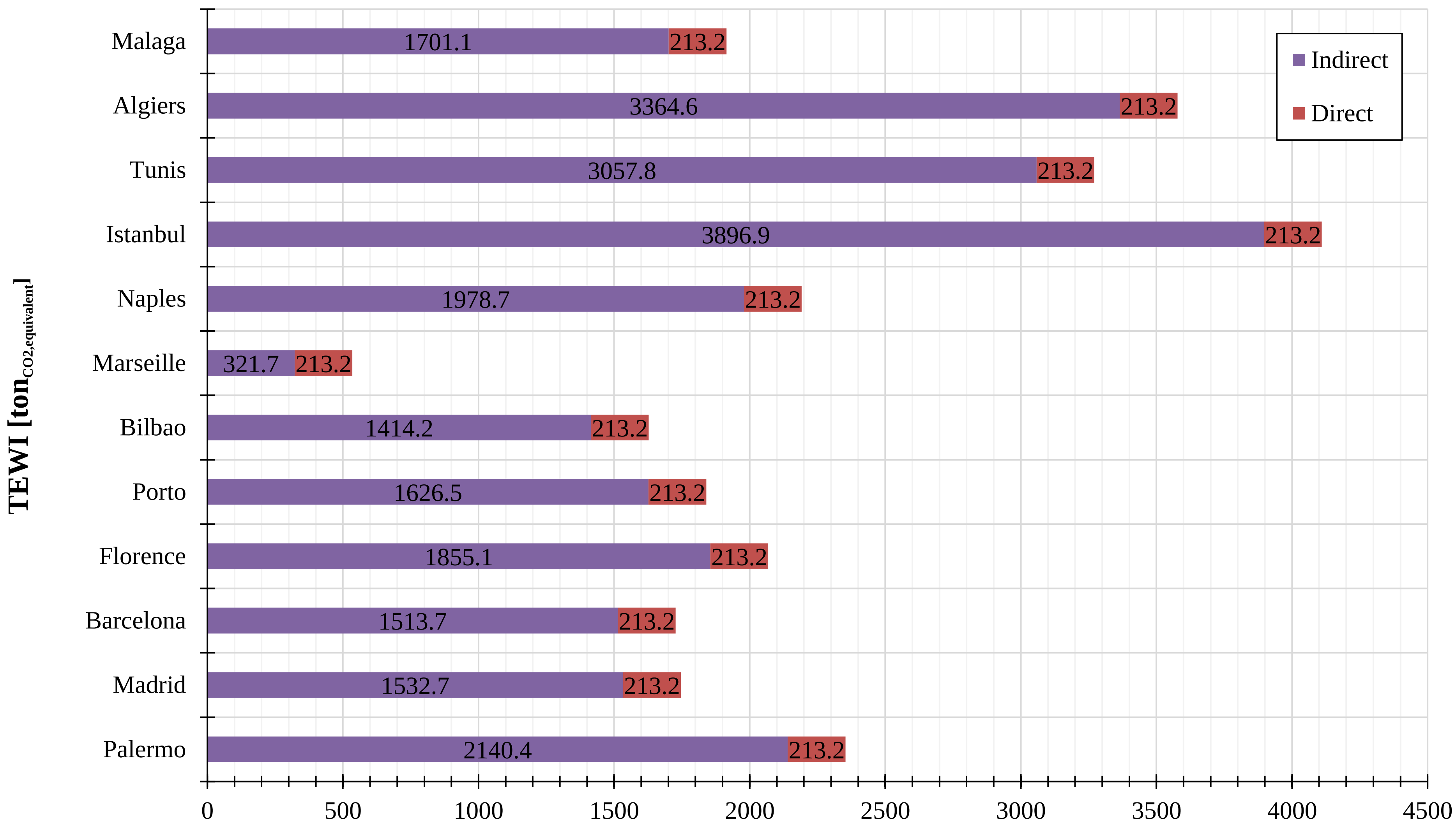


Fig. 19. Total power input of the investigated systems in heating mode at different outdoor temperatures.

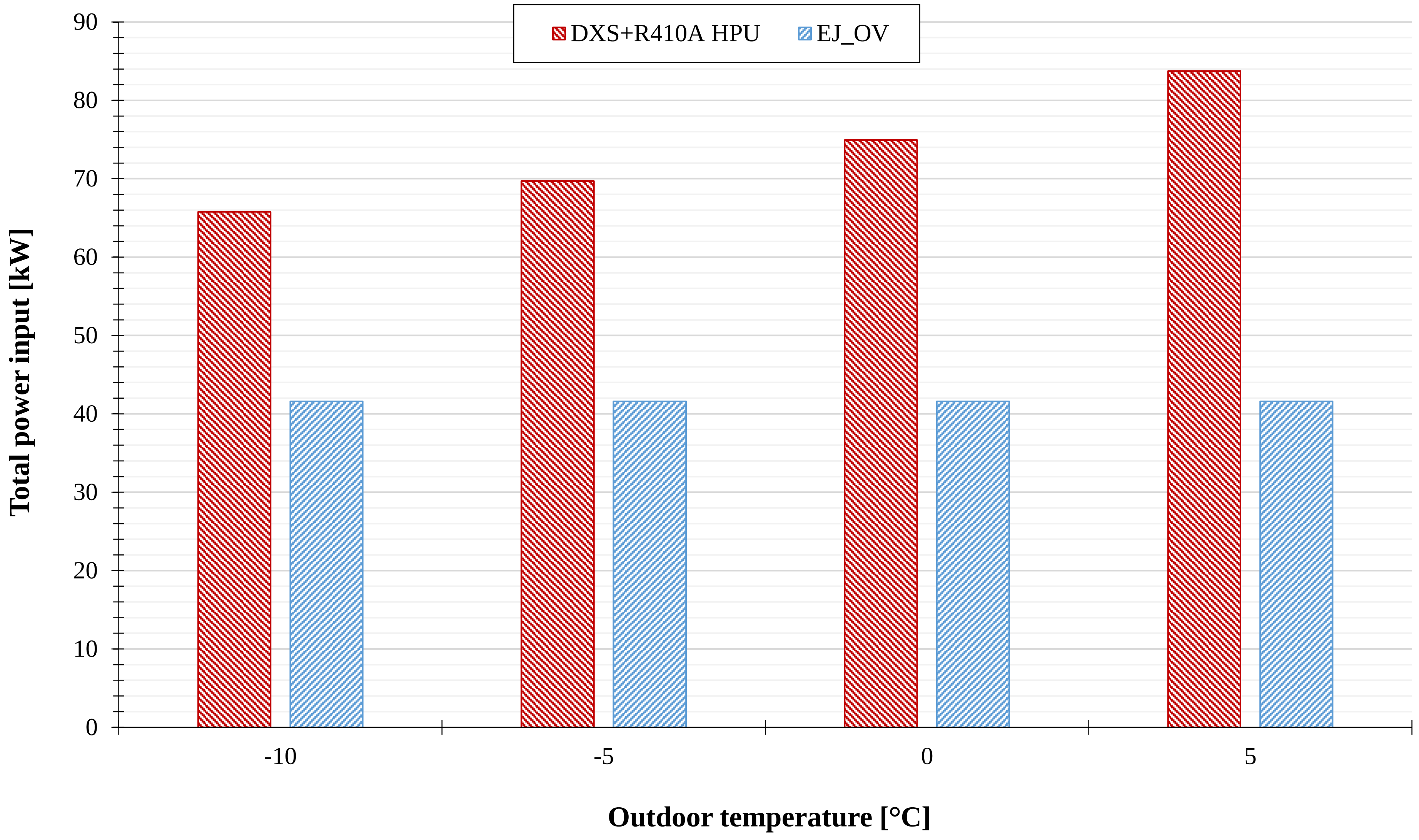


Table 1

Summary of the main findings related to the investigated systems.

Multi-ejector based systems (without integration with AC equipment)			
Reference	Evaluation	Baseline	Findings
Minetto et al. (2014b)	Theoretical	Conventional “CO_2 only” system	Energy saving of 22.5% in a supermket located in Bari (South of Italy)
Schönenberger (2016)	Assertion	“CO_2 only” system with parallel compression	Energy conservations from 15% and 25 %, depending on heat reclaim, application and climate
Hafner et al. (2014b)	Field measurements	“CO_2 only” system with parallel compression	Energy saving of 12% in a food retail store in Fribourg (Switzerland)
Haida et al. (2016)	Experimental	“CO_2 only” system with parallel compression	Improvements in COP and exergy efficiency up to 7% and 13.7%, respectively
Gullo et al. (2017b)	Theoretical	Conventional “CO_2 only” system	Energy savings from 22.2% to 26.7% in Athens (Greece)
		“CO_2 only” system with parallel compression (with and without overfed evaporators)	Energy savings from 8.6% to 22.3% in Athens
Madsen and Kriezi (2018)	Theoretical	“CO_2 only” system with parallel compression	Competitive payback periods (i.e. below 3 years) in food retail stores with a cooling capacity above 75 kW and located in cities featuring an average annual temperature above 15 °C (e.g. Athens). The payback period decreases with rise in both system size and average yearly temperature

Indirect arrangements			
Reference	Evaluation	Baseline(s)	Findings
Sawalha (2008a)	Theoretical	“CO_2-only” systems and R404A-based units	Indirect arrangements can outperform the baselines in hot climates (e.g. Phoenix, USA)
Christensen and Bertilsen (2003)	Field measurements	Average, conventional and comparable supermarket	Energy saving of 5% on the part of a R290/R744 cascade system installed in a small Danish food retail store
Karampour and Sawalha (2018)	Theoretical	Conventional “CO_2 only” system	Reduction in annual electricity use of 12.8% on the part of a R290/R744 indirect solution in Barcelona (Spain)
Gullo and Cortella (2016a)	Theoretical	HFC-based units	R1234ze(E)-based indirect systems are suitable solutions in the American climate context
Purohit et al. (2017)	Theoretical	“CO_2 only” system with dedicated mechanical subcooling “CO_2 only” system with parallel compression	R1234ze(E)-based indirect arrangements can outperform the selected baselines at outdoor temperatures above 23 °C
Sánchez et al. (2018b)	Experimental (24-hour experimental tests)	R134a direct expansion refrigerating plant	Conversion into an indirect unit using R134a in HTC leads to an average increase in total energy consumption by 45.5% in a commercial refrigeration unit plugged to a medium temperature cabinet at the investigated heat rejection levels (i.e. 43.7 °C, 32.7 °C, 23.3 °C)

Sánchez et al. (2018a)	Experimental (24-hour experimental tests)	R134a direct expansion refrigerating plant	Conversion into an indirect unit relying on either R134a or R1234ze(E) in HTC leads to average increases in total energy consumption up to 27.2% in a commercial refrigeration unit plugged to a medium temperature cabinet at the investigated heat rejection levels (i.e. 43.6 °C, 32.8 °C, 23.3 °C)
Llopis et al. (2018)	Experimental (24-hour experimental tests)	R134a/CO_2 cascade unit	At the investigated heat rejection levels (i.e. 18.8 °C, 31.6 °C, 42.2 °C): • R134a/CO_2 indirect arrangement leads to an increase in energy consumption from 3.5% to 6.2%; • R1234ze(E)/CO_2 indirect arrangement leads to a growth in energy consumption from 8.2% to 6.9%; • R290/CO_2 indirect arrangement leads to an increase in energy consumption from 10.9% to 16.1%

Cascade arrangements			
Reference	Evaluation	Baseline	Findings
Sanz-Kock et al. (2014)	Experimental	-	R134a/CO_2 cascade system offers COP values from 1.05 to 1.65 at evaporating temperatures from -40 °C to -30 °C and condensing temperatures from 30 °C to 50 °C
Souza et al. (2015) and Sanz-Kock et al. (2014)	Experimental	-	COP values of R134a/CO_2 cascade systems increase with rise in CO_2 compressor operating frequency
Sánchez et al. (2017)	Experimental (24-hour experimental tests)	R134a/CO_2 cascade unit	Growths in energy consumption up to 14% by replacing the baseline with a R134a/CO_2 indirect arrangement, depending on both the condensing temperature and the secondary fluid selected
Llopis et al. (2017)	Experimental (24-hour experimental tests)	R134a in a single-stage vapour-compression unit	The use of R513A and R450A respectively feature growths in energy consumption between -1.6% and +1.2% and from +1.3% to +6.8% compared to the baseline in medium temperature commercial refrigeration applications
Makhnatch et al. (2018)	Experimental	R134a/CO_2 cascade unit	R450A and R513A can appropriately replace R134a in cascade systems for applications featuring the middle evaporating temperature of -10 °C, the condensing temperatures between 40 °C and 60 °C and the low temperature of -35 °C
Catalán-Gil et al. (2018)	Theoretical	R513A/CO_2 cascade unit	R513A/CO_2 cascade booster systems are suitable alternatives for locations having an average annual temperature up to 13 °C (e.g. Madrid, Spain)

Solutions integrated with AC equipment			
Reference	Evaluation	Baseline	Findings
Karampour and Sawalha (2017)	Simulation models adapted from field measurements	HFC-based units	Transcritical R744 system using parallel compression is a suitable solution only for cold weathers as the heating and AC equipment is integrated into the refrigerating unit
Gullo et al. (2018b)	Theoretical	HFC-based units	Transcritical R744 system using parallel compression cannot outperform the baseline in AC mode in the Spanish climate context
Hafner et al. (2016)	Field measurements	"CO_2 only" system with parallel compression	Energy savings from 15% to 30% in the food retail store in Spiazzo (North of Italy) at outdoor temperatures between 22 °C and 35 °C, depending on both the AC reclaim and the external temperature. Furthermore, the refrigeration system can provide the whole AC reclaim of the store
Fredslund et al. (2016)	Field measurements	-	The efficiency of the vapour ejectors performing in real applications is about the same as the one evaluated in the laboratory (and above 0.25)
		"CO_2 only" system with parallel compression	Energy savings ranging from 10% to 15% at the outdoor temperature of around 30 °C

Table 2

Classification of working fluids in terms of $GWP_{100\ years}$ values (UNEP, 2015).

Range	*Classification*	*Examples*
$GWP_{100\ years} < 30$	Ultra-low or Negligible	CO_2 (R744), R290, R1234ze(E)
$30 \leq GWP_{100\ years} < 100$	Very low	-
$100 \leq GWP_{100\ years} < 300$	Low	R152a
$300 \leq GWP_{100\ years} \leq 1000$	Medium	R450A, R513A
$1000 < GWP_{100\ years} \leq 3000$	High	R134a, R410A
$3000 < GWP_{100\ years} \leq 10000$	Very high	R404A

Table 3

Physical, environmental and safety properties of the selected refrigerants (IPCC, 2013; Calm and Hourahan, 2011; Lemmon et al., 2017).

	R404A	R134a	R450A	R513A	R744	R290	R1234ze(E)	R410A
Chemical formula or composition (%wt)	44% R125 52% R143a 4% R134a	$CHF{=}CHCF_3$	42% R134a 58% R1234ze(E)	44% R134a 56% R1234ze(E)	CO_2	$CH_3CH_2CH_2$	$CHF{=}CHCF_3$	50% R32 50% R125
Molecular weight [$g \cdot mol^{-1}$]	97.60	102.03	108.70	108.40	44.01	44.10	114.04	72.58
Normal boiling point [°C]	-46.2	-26.36	-23.65	-29.87	-56.6	-42.1	-19.0	-51.4
Critical temperature [°C]	72.0	101.1	105.6	97.7	31.0	96.7	109.4	71.4
Critical pressure [bar]	37.35	40.59	39.14	38.55	73.77	42.47	36.32	49.01
Glide at 1 bar [K]	0.00	0.00	0.61	0.10		0.00	0.00	0.08
h_{fg} at -30 °C [$kJ \cdot kg^{-1}$]	189.5	219.5	203.6	195.1	303.5	412.4	201.5	253.5
h_{fg} at 40 °C [$kJ \cdot kg^{-1}$]	120.3	163.0	154.8	142.6	-	307.1	154.8	159.3
ν_v at -30 °C [$m^3 \cdot kg^{-1}$]	0.095	0.226	0.242	0.179	0.027	0.259	0.282	0.095
$GWP_{100\ years}$	3700	1300	547	573	1	3	<1	1924
ODP	0	0	0	0	0	0	0	0
Safety group	A1	A1	A1	A1	A1	A3	A2L	A1

Table 4

Summary of the investigated solutions, adopted abbreviations and assumptions necessary to implement all the simulation models related to the scenario with no AC demand.

Investigated solution	Abbreviation	Selected working fluid(s)	Running modes	Notes/References
Two R404A direct expansion units respectively serving MT and LT refrigeration loads	DXS	R404A	MT dry-expansion evaporators (i.e. t_{MT} = -10 °C)	(Gullo et al., 2017a)
			$\Delta T_{SH,MT\ evap}$ = 5 K	(Gullo et al., 2017a)
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	(Gullo et al., 2017a)
			$\Delta T_{SH,LT\ evap}$ = 5 K	(Gullo et al., 2017a)
			if $t_{ext} \leq 15$ °C: $t_{condensing} = 25$ °C if $t_{ext} > 15$ °C: $t_{condensing} = t_{ext} + 10$ °C	(Gullo et al., 2017a)

R744 multi-ejector enhanced parallel compression system with MT overfed evaporators (Fig. 1a)	EJ	R744	MT overfed evaporators (i.e. t_{MT} = -4 °C)	(Gullo et al., 2017a)
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	(Gullo et al., 2017a)
			$\Delta T_{SH,LT\ evap}$ = 5 K	(Gullo et al., 2017a)

			if $t_{ext} \leq 4$ °C:
			$t_{out,GC}$ = 7 °C
			t_{GC} = 9 °C
			$p_{GC} = p_{saturation}(t_{GC})$ bar
			
			if 4 °C < $t_{ext} \leq 17$ °C:
			$t_{out,GC} = t_{ext}$ + 3 °C
			$t_{GC} = t_{out,GC}$ - 2 °C
			$p_{GC} = p_{saturation}(t_{GC})$ bar
			
			if 17 °C < $t_{ext} \leq 27$ °C:
			$t_{out,GC} = 0.9 \cdot t_{ext}$ + 4.7 °C
			$p_{GC} = 1.6633 \cdot t_{out,GC}$ + 26.763 bar
			
			if t_{ext} > 27 °C:
			$t_{out,GC} = t_{ext}$ + 2 °C
			75 bar $\leq p_{GC} \leq$ 110 bar (Optimized)
				(Gullo et al., 2017a)

R744 multi-ejector enhanced parallel compression system with MT and LT overfed evaporators (Fig. 1b)	EJ_OV	R744	MT overfed evaporators (i.e. t_{MT} = -4 °C)	(Gullo et al., 2017a)
			LT overfed evaporators (i.e. t_{LT} = -27 °C)	(Gullo et al., 2017a)

			if $t_{ext} \leq 4$ °C:
			$t_{out,GC}$ = 7 °C
			t_{GC} = 9 °C
			$p_{GC} = p_{saturation}(t_{GC})$ bar
			
			if 4 °C < $t_{ext} \leq$ 17 °C:
			$t_{out,GC} = t_{ext}$ + 3 °C
			$t_{GC} = t_{out,GC}$ - 2 °C
			$p_{GC} = p_{saturation}(t_{GC})$ bar
			
			if 17 °C < $t_{ext} \leq$ 27 °C:
			$t_{out,GC} = 0.9 \cdot t_{ext} + 4.7$ °C
			$p_{GC} = 1.6633 \cdot t_{out,GC} + 26.763$ bar
			
			if t_{ext} > 27 °C:
			$t_{out,GC} = t_{ext}$ + 2 °C
			75 bar ≤ p_{GC} ≤ 110 bar (Optimized)
				(Gullo et al., 2017a)

R1234ze(e)/R744 indirect arrangement with MT flooded evaporators (Fig. 2a)	HFO-IND	R1234ze(e) in HTC R744 in MTC and LTC	MT flooded evaporators (i.e. t_{MT} = -4 °C)	Same energy benefits between MT overfed and MT flooded evaporators were considered to make a fair comparison
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	Same as DXS
			$\Delta T_{SH,LT\ evap}$ = 5 K	Same as DXS
			if $t_{ext} \leq 15$ °C: $t_{condensing,HTC} = 25\ °C$ if $t_{ext} > 15$ °C: $t_{condensing,HTC} = t_{ext} + 10\ °C$	(Gullo and Cortella, 2016a)
			$CR_{pump,MT\ evap} = \frac{1}{x_{R744,out\_MTevap}} = 1.5$	(Gullo and Cortella, 2016a)
			$CR_{pump,LT\ evap} = \frac{1}{x_{R744,out\_LTevap}} = 2.5$	(Gullo and Cortella, 2016a)
			$\dot{W}_{pump} = 0.01 * \dot{W}_{tot,compr}$	(Gullo and Cortella, 2016a)
			ΔT_{CC} = 2 K (i.e. $t_{evaporating,HTC} = t_{MT} - \Delta T_{CC} - \Delta T_{SH,HTC\ evap}$ with $\Delta T_{SH,HTC\ evap} = 5K$)	(Sawalha et al., 2006)

R1234ze(e)/R744 indirect arrangement with MT and LT flooded evaporators (Fig. 2b)	HFO-IND_FL	R1234ze(e) in HTC R744 in MTC and LTC	Flooded (i.e. t_{MT} = -4 °C)	Same energy benefits between MT overfed and MT flooded evaporators were considered to make a fair comparison
			Flooded (i.e. t_{LT} = -27 °C)	Same energy benefits between LT overfed and LT flooded evaporators were considered to make a fair comparison
			if $t_{ext} \leq 15$ °C: $t_{condensing,HTC} = 25$ °C if $t_{ext} > 15$ °C: $t_{condensing,HTC} = t_{ext} + 10$ °C	(Gullo and Cortella, 2016a)
			$CR_{pump,MTevap} = \frac{1}{x_{R744,out\_MTevap}} = 1.5$	(Gullo and Cortella, 2016a)
			$CR_{pump,LTevap} = \frac{1}{x_{R744,out\_LTevap}} = 2.5$	(Gullo and Cortella, 2016a)
			$\dot{W}_{pump} = 0.01 * \dot{W}_{tot,compr}$	(Gullo and Cortella, 2016a)
			$\Delta T_{CC} = 2$ K (i.e. $t_{evaporating,HTC} = t_{MT} - \Delta T_{CC} - \Delta T_{SH,HTC\ evap}$ with $\Delta T_{SH,HTC\ evap} = 5K$)	(Sawalha et al., 2006)

R290/R744 indirect arrangement with MT flooded evaporators (Fig. 2a)	R290-IND	R290 in HTC R744 in MTC and LTC	MT flooded evaporators (i.e. t_{MT} = -4 °C)	Same energy benefits between MT overfed and MT flooded evaporators were considered to make a fair comparison
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	Same as DXS
			$\Delta T_{SH,LT\ evap}$ = 5 K	Same as DXS

			if $t_{ext} \leq 15$ °C:
			$t_{condensing,HTC} = 25\ °C$
			
			if $t_{ext} > 15$ °C:
			$t_{condensing,HTC} = t_{ext} + 10\ °C$
				Same as HFO-IND

			$CR_{pump,MTevap} = \frac{1}{x_{R744,out\_MTevap}} = 1.5$	Same as HFO-IND
			$CR_{pump,LTevap} = \frac{1}{x_{R744,out\_LTevap}} = 2.5$	Same as HFO-IND
			$\dot{W}_{pump} = 0.01 * \dot{W}_{tot,compr}$	Same as HFO-IND
			ΔT_{CC} = 2 K (i.e. $t_{evaporating,HTC} = t_{MT} - \Delta T_{CC} - \Delta T_{SH,HTC\ evap}$ with $\Delta T_{SH,HTC\ evap} = 5K$)	Same as HFO-IND

R290/R744 indirect arrangement with MT and LT flooded evaporators (Fig. 2b)	R290-IND_FL	R290 in HTC R744 in MTC and LTC	Flooded (i.e. t_{MT} = -4 °C)	Same energy benefits between MT overfed and MT flooded evaporators were considered to make a fair comparison
			Flooded (i.e. t_{LT} = -27 °C)	Same energy benefits between LT overfed and LT flooded evaporators were considered to make a fair comparison

			if $t_{ext} \leq 15$ °C:
			$t_{condensing,HTC} = 25\ °C$
			
			if $t_{ext} > 15$ °C:
			$t_{condensing,HTC} = t_{ext} + 10\ °C$
				Same as HFO-IND_FL

			$CR_{pump,MTevap} = \frac{1}{x_{R744,out\_MTevap}} = 1.5$	Same as HFO-IND_FL
			$CR_{pump,LTevap} = \frac{1}{x_{R744,out\_LTevap}} = 2.5$	Same as HFO-IND_FL
			$\dot{W}_{pump} = 0.01 * \dot{W}_{tot,compr}$	Same as HFO-IND_FL
			ΔT_{CC} = 2 K (i.e. $t_{evaporating,HTC} = t_{MT} - \Delta T_{CC} - \Delta T_{SH,HTC\ evap}$ with $\Delta T_{SH,HTC\ evap} = 5K$)	Same as HFO-IND_FL

R134a/R744 cascade system (Fig. 3)	R134a-CS	R134a in HTC R744 in LTC	MT dry-expansion evaporators (i.e. t_{MT} = -10 °C)	Same as DXS and Gullo et al. (2016a)
			$\Delta T_{SH,MT\ evap}$ = 5 K	Same as DXS and Gullo et al. (2016a)
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	Same as DXS and Gullo et al. (2016a)
			$\Delta T_{SH,LT\ evap}$ = 5 K	Same as DXS and Gullo et al. (2016a)
			if $t_{ext} \leq 15$ °C: $t_{condensing,HTC} = 25$ °C if $t_{ext} > 15$ °C: $t_{condensing,HTC} = t_{ext} + 10$ °C	(Gullo et al., 2016a)
			ΔT_{CC} = 2 K (i.e. $t_{condensing,LTC} = t_{MT} + \Delta T_{SH,MT\ evap} + \Delta T_{CC}$)	Same as HFO-IND

R450A/R744 cascade system (Fig. 3)	R450A-CS	R450A in HTC R744 in LTC	MT dry-expansion evaporators (i.e. t_{MT} = -10 °C)	Same as R134a-CS
			$\Delta T_{SH,MT\ evap}$ = 5 K	Same as R134a-CS
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	Same as R134a-CS
			$\Delta T_{SH,LT\ evap}$ = 5 K	Same as R134a-CS
			if $t_{ext} \leq 15$ °C: $t_{condensing,HTC} = 25\ °C$ if $t_{ext} > 15$ °C: $t_{condensing,HTC} = t_{ext} + 10\ °C$	Same as R134a-CS
			ΔT_{CC} = 2 K (i.e. $t_{condensing,LTC} = t_{MT} + \Delta T_{SH,MT\ evap} + \Delta T_{CC}$)	Same as R134a-CS

R513A/R744 cascade system (Fig. 3)	R513A-CS	R513A in HTC R744 in LTC	MT dry-expansion evaporators (i.e. t_{MT} = -10 °C)	Same as R134a-CS
			$\Delta T_{SH,MT\ evap}$ = 5 K	Same as R134a-CS
			LT dry-expansion evaporators (i.e. t_{LT} = -35 °C)	Same as R134a-CS
			$\Delta T_{SH,LT\ evap}$ = 5 K	Same as R134a-CS
			if $t_{ext} \leq 15$ °C: $t_{condensing,HTC} = 25\ °C$ if $t_{ext} > 15$ °C: $t_{condensing,HTC} = t_{ext} + 10\ °C$	Same as R134a-CS
			ΔT_{CC} = 2 K (i.e. $t_{condensing,LTC} = t_{MT} + \Delta T_{SH,MT\ evap} + \Delta T_{CC}$)	Same as R134a-CS

Table 5

Summary of the investigated solutions, adopted abbreviations and assumptions necessary to implement all the simulation models related to the scenario including the AC demand.

Investigated solution	Abbreviation	Selected working fluid(s)	Running modes	Notes/References
Two R404A direct expansion units respectively serving MT and LT refrigeration loads (DXS) R410A chiller for AC demand (R410A CH)	DXS+R410A CH	R404A for DXS R410A for CH	$t_{evaporating,R410A\ CH} = +3\ °C$	(Karampour and Sawalha, 2015)
			$t_{condensing,R410A\ CH} = t_{ext} + 10\ °C$	(Gullo et al., 2017a)
			$\Delta T_{SH,R410A\ CH} = 5\ K$	(Gullo et al., 2017a)
R744 multi-ejector enhanced parallel compression system with MT and LT overfed evaporators and integrated with AC equipment (Fig. 4)	EJ_OV_AC	R744	$t_{HX\_AC} = +5\ °C$	(Gullo et al., 2017a)
			Quality of R744 exiting HX_AC = 1	(Gullo et al., 2017a)

R1234ze(e)/R744 indirect arrangement with MT and LT flooded evaporators and integrated with AC equipment (Fig. 5)	HFO-IND_FL_AC	R1234ze(e) in HTC R744 in MTC and LTC	t_{HX_AC} = +3 °C	Same as R410A CH
			$\Delta T_{SH,HX\_AC}$ = 5 K	Assumption (value selected to be consistent with other corresponding assumptions)
			AUX put in operation at $t_{ext} \geq 24$ °C	AUX was/were supposed to deal with AC load
R290/R744 indirect arrangement with MT and LT flooded evaporators and integrated with AC equipment (Fig. 5)	R290-IND_FL_AC	R290 in HTC R744 in MTC and LTC	t_{HX_AC} = +3 °C	Same as R410A CH
			$\Delta T_{SH,HX\_AC}$ = 5 K	Assumption (value selected to be consistent with other corresponding assumptions)
			AUX put in operation at $t_{ext} \geq 24$ °C	AUX was/were supposed to deal with AC load
R1234ze(e)/R744 indirect arrangement with MT and LT flooded evaporators for MT and LT refrigeration loads (HFO-IND_FL) R1234ze(E) chiller for AC demand (R1234ze(E) CH)	HFO-IND_FL+R1234ze(E) CH	R1234ze(e) in HTC R744 in MTC and LTC R1234ze(e) for CH	$t_{evaporating,R1234ze(E)\ CH}$ = +3 °C	Same as R410A CH
			$t_{condensing,\mathrm{R1234ze(E)CH}} = t_{ext} + 10\ ℃$	Same as R410A CH
			$\Delta T_{SH,R1234ze(E)\ CH}$ = 5 K	Same as R410A CH

R290/R744 indirect arrangement with MT and LT flooded evaporators for MT and LT refrigeration loads (R290-IND_FL) R1234ze(E) chiller for AC demand (R1234ze(E) CH)	R290-IND_FL+R1234ze(E) CH	R290 in HTC R744 in MTC and LTC R1234ze(e) for CH	$t_{evaporating,R1234ze(E)\ CH} = +3\ °C$	Same as R410A CH
			$t_{condensing,R1234ze(E)\ CH} = t_{ext} + 10\ °C$	Same as R410A CH
			$\Delta T_{SH,R1234ze(E)\ CH} = 5\ K$	Same as R410A CH
R134a/R744 cascade system for MT and LT refrigeration loads (R134a-CS) R1234ze(E) chiller for AC demand (R1234ze(E) CH)	R134a-CS+R1234ze(E) CH	R134a in HTC R744 in LTC R1234ze(e) for CH	$t_{evaporating,R1234ze(E)\ CH} = +3\ °C$	Same as R410A CH
			$t_{condensing,R1234ze(E)\ CH} = t_{ext} + 10\ °C$	Same as R410A CH
			$\Delta T_{SH,R1234ze(E)\ CH} = 5\ K$	Same as R410A CH
R450A/R744 cascade system for MT and LT refrigeration loads (R450A-CS) R1234ze(E) chiller for AC demand (R1234ze(E) CH)	R450A-CS+R1234ze(E) CH	R450A in HTC R744 in LTC R1234ze(e) for CH	$t_{evaporating,R1234ze(E)\ CH} = +3\ °C$	Same as R410A CH
			$t_{condensing,R1234ze(E)\ CH} = t_{ext} + 10\ °C$	Same as R410A CH
			$\Delta T_{SH,R1234ze(E)\ CH} = 5\ K$	Same as R410A CH

R513A/R744 cascade system for MT and LT refrigeration loads (R513A-CS) R1234ze(E) chiller for AC demand (R1234ze(E) CH)	R513A-CS+R1234ze(E) CH	R513A in HTC R744 in LTC R1234ze(e) for CH	$t_{evaporating,\mathrm{R1234ze(E)\ CH}} = +3\ °C$	Same as R410A CH
			$t_{condensing,\mathrm{R1234ze(E)\ CH}} = t_{ext} + 10\ °C$	Same as R410A CH
			$\Delta T_{SH,\mathrm{R1234ze(E)\ CH}} = 5\ K$	Same as R410A CH

Table 6

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)
EJ	$\eta_{global,HS} = -0.1004 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.4359 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.2334$ for $t_{ext} \leq 27$ °C $\eta_{global,HS} = -0.0032 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0164 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.6472$ for $t_{ext} > 27$ °C $\eta_{global,AUX} = -0.3432 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 1.3625 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) - 0.6324$ for $t_{ext} \leq 27$ °C $\eta_{global,AUX} = -0.0665 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 0.3569 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) + 0.2108$ for $t_{ext} > 27$ °C $\eta_{global,LS} = -0.0117 \cdot \left(\frac{p_{MT}}{p_{LT}}\right)^2 + 0.0439 \cdot \left(\frac{p_{MT}}{p_{LT}}\right) + 0.5496$	Gullo et al. (2017a)
EJ_OV and EJ_OV_AC	$\eta_{global,HS} = -0.1004 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.4359 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.2334$ for $t_{ext} \leq 27$ °C $\eta_{global,HS} = -0.0032 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0164 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.6472$ for $t_{ext} > 27$ °C	Gullo et al. (2017a)

	$\eta_{global,AUX} = -0.3432 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 1.3625 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) - 0.6324$ for $t_{ext} \leq 27$ °C $\eta_{global,AUX} = -0.0665 \cdot \left(\frac{p_{HP}}{p_{IP}}\right)^2 + 0.3569 \cdot \left(\frac{p_{HP}}{p_{IP}}\right) + 0.2108$ for $t_{ext} > 27$ °C $\eta_{global,LS} = -0.050 \cdot \left(\frac{p_{MT}}{p_{LT}}\right)^2 + 0.2116 \cdot \left(\frac{p_{MT}}{p_{LT}}\right) + 0.3715$	
HFO-IND, HFO-IND_FL and HFO-IND_FL_AC	$\eta_{global,R1234ze(E)} = -0.0028 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.5305$ $\eta_{global,R1234ze(E)\_AUX} = -0.0028 \cdot \left(\frac{p_{HP}}{p_{AC}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{AC}}\right) + 0.5305$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP}}{p_{LP}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP}}{p_{LP}}\right) + 0.8030$	(Gullo and Cortella, 2016a)
R290-IND, R290-IND_FL and R290-IND_FL_AC	$\eta_{global,R290} = -0.0069 \cdot \left(\frac{p_{HP}}{p_{MP}}\right)^2 + 0.0930 \cdot \left(\frac{p_{HP}}{p_{MP}}\right) + 0.4056$ $\eta_{global,R290\_AUX} = -0.0069 \cdot \left(\frac{p_{HP}}{p_{AC}}\right)^2 + 0.0930 \cdot \left(\frac{p_{HP}}{p_{AC}}\right) + 0.4056$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP}}{p_{LP}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP}}{p_{LP}}\right) + 0.8030$	(Dorin, 2018)
R134a-CS	$\eta_{global,R134a} = -0.0053 \cdot \left(\frac{p_{HP,R134a}}{p_{MP,R134a}}\right)^2 + 0.0674 \cdot \left(\frac{p_{HP,R134a}}{p_{MP,R134a}}\right) + 0.4802$	(Gullo et al., 2016a)

	$\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	
R450A-CS	$\eta_{global,R450A} = -0.0037 \cdot \left(\frac{p_{HP,R450A}}{p_{MP,R450A}}\right)^2 + 0.0482 \cdot \left(\frac{p_{HP,R450A}}{p_{MP,R450A}}\right) + 0.4978$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	(BITZER, 2018)
R513A-CS	$\eta_{global,R513A} = -0.0043 \cdot \left(\frac{p_{HP,R513A}}{p_{MP,R513A}}\right)^2 + 0.0533 \cdot \left(\frac{p_{HP,R513A}}{p_{MP,R513A}}\right) + 0.4989$ $\eta_{global,R744} = +0.0111 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MP,R744}}{p_{LP,R744}}\right) + 0.8030$	(BITZER, 2018)
R410A CH and R410A HPU	$\eta_{global,R410A} = -0.0293 \cdot \left(\frac{p_{HP}}{p_{LP}}\right)^2 + 0.2008 \cdot \left(\frac{p_{HP}}{p_{LP}}\right) + 0.3549$	Gullo et al. (2017a)
R1234ze(E) CH	$\eta_{global,R1234ze(E)} = -0.028 \cdot \left(\frac{p_{HP}}{p_{LP}}\right)^2 + 0.0419 \cdot \left(\frac{p_{HP}}{p_{LP}}\right) + 0.5305$	Frascold (2014)

Table 7

List of the correlations employed for simulating the multi-ejector block (Gullo et al., 2017a).

$\omega = a + b \cdot p_{lift} + c \cdot p_{lift}^2 + d \cdot t_{out,GC} + e \cdot t_{out,GC}^2$ for $t_{ext} \leq 17$ °C	a = 0.6610985 b = -0.2128984 c = 0.0180093 d = -0.0048173 e = 0.0002467
$\omega = a + b \cdot t_{out,GC} + c \cdot t_{out,GC}^2 + d \cdot p_{lift} + e \cdot p_{lift}^2$ for 17 °C $< t_{ext} \leq 27$ °C	a = 0.9807737 b = -0.0461714 c = 0.0015270 d = -0.1303932 e = 0.0064457
$\omega = a + b \cdot p_{lift} + c \cdot t_{out,GC}$ for 27 °C $< t_{ext} \leq 41$ °C	a = -0.0147727 b = -0.0881130 c = 0.0336677

Appendix

The most relevant parameters of VEJ (scenario including AC demand) are summarized in Table A1. It can be noticed that at the transition running modes (i.e. $t_{ext} \leq 27$ °C, $t_{out,GC} \leq 29$ °C) the pressure lift slightly varies which, as a consequence of the slightly increasing trend of the VEJ efficiency (η_{VEJ}), entails a rise in entrainment ratio (ω) values. As the transcritical operating conditions occur, the system tends to operate at high values of pressure lift to achieve significant energy savings, as experimentally verified by Haida et al. (2016). Therefore, being the VEJ efficiency slightly decreasing at these operating conditions, the entrainment ratio values diminish too (Haida et al., 2016). Also, in accordance with the experimental measurements available in the open literature, VEJ efficiencies were found to be above 0.25 (Fredslund et al., 2016) and below 0.33 (Haida et al., 2016).

Table A1. Most relevant parameters of VEJ (scenario including AC demand).

R774 pressure at primary nozzle [bar]	**R744 temperature at primary nozzle [°C]**	**R744 mass flow rate at primary nozzle [kg·s^{-1}]**	**R744 pressure at secondary nozzle [bar]**	**R744 temperature at secondary nozzle [°C]**	**R744 mass flow rate at secondary nozzle [kg·s^{-1}]**	**R744 pressure at VEJ outlet [bar]**	**R744 temperature at VEJ outlet [°C]**	**η_{VEJ} (*) [-]**	**ω [-]**
70.51	26.30	1.511	31.30	6.25	0.198	39.69	5.00	0.250	0.131
72.00	27.20	1.589	31.30	6.28	0.257	39.69	5.00	0.273	0.162
73.50	28.10	1.671	31.30	6.31	0.321	39.69	5.00	0.292	0.192
75.00	29.00	1.758	31.30	6.34	0.390	39.69	5.00	0.309	0.222
75.00	30.00	1.900	31.30	6.37	0.384	40.30	5.59	0.287	0.202
76.75	31.00	2.015	31.30	6.40	0.475	40.30	5.59	0.303	0.236
76.83	32.00	2.266	31.30	6.42	0.577	40.47	5.76	0.294	0.255
80.93	33.00	2.181	31.30	6.44	0.579	40.73	6.01	0.311	0.265
81.18	34.00	2.380	31.30	6.48	0.583	41.34	6.60	0.284	0.245
84.68	35.00	2.329	31.30	6.51	0.585	41.65	6.89	0.294	0.251
84.94	36.00	2.556	31.30	6.54	0.590	42.27	7.48	0.266	0.231
89.02	37.00	2.446	31.30	6.56	0.592	42.53	7.72	0.283	0.242
92.33	38.00	2.441	31.30	6.58	0.595	42.89	8.06	0.285	0.244
98.48	39.00	2.346	31.30	6.61	0.596	43.15	8.30	0.295	0.254
101.00	40.00	2.392	31.30	6.64	0.599	43.57	8.68	0.290	0.250
102.30	41.00	2.476	31.30	6.66	0.603	44.03	9.10	0.282	0.244
102.40	42.00	2.622	31.30	6.69	0.607	44.55	9.58	0.269	0.232
109.10	43.00	2.521	31.30	6.71	0.609	44.82	9.82	0.277	0.242
110.00	44.00	2.554	31.30	6.76	0.611	45.11	10.08	0.290	0.239

(*) defined in accordance with Elbel and Hrnjak (2008)