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Comparative assessment of low-GWP based refrigerating plants operating in hot climates

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Abstract

Carbon dioxide (CO_2 or R744) and HFO-1234ze(E) are two promising alternatives to the currently employed high-GWP working fluids for food retail applications. In this paper, two indirect refrigeration systems using respectively R1234ze(E) and CO_2 as the primary and the secondary fluid, a R744 booster configuration with parallel compression and a R744 booster solution with R290 dedicated mechanical sub-cooling are theoretically compared with a R404A multiplex direct expansion refrigerating system (baseline). The latter serves both the LT and the MT load. All the evaluated configurations are supposed to be for commercial refrigeration technologies. The results in terms of COP reveal that both the investigated "CO_2 only" configurations have the best performance at low outdoor temperatures. On the other hand, the indirect solutions can outperform all the selected refrigerating plants at the extreme climatic conditions. In comparison with the baseline, a maximum reduction in annual electricity consumption by 6.4% and 8.9% is observed for R1234ze(E) and CO_2 based systems, respectively.

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Keywords: Dedicated Mechanical Subcooling; Indirect refrigeration system; Parallel compression; R1234ze(E); R744; Transcritical CO_2 supermarket refrigeration system.

1. Introduction

The refrigeration sector consumes the major part of the total electricity required in supermarket applications [1]. In addition to this, 3 to 35% of the refrigerant charge leaks into the atmosphere depending upon the age of the

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equipment involved and its usage [2]. Consequently, there is an ever-increasing pressure towards the restriction of the indirect and direct greenhouse gases emissions worldwide in refrigerating plants. Adoption of natural or low-GWP refrigerants and the improvement of system's COP are major thrust area of research.

Nomenclature

AS	Auxiliary compressor
COP	Coefficient of Performance
CSC	Combined CO_2/R1234ze(E) secondary/cascade refrigeration system
DMS	CO_2 booster refrigeration system with R290 dedicated mechanical sub-cooling
DXS	R404A direct multiplex expansion refrigeration system
EES	Engineering equation solver
FCSC	Combined CO_2/R1234ze(E) secondary/cascade refrigeration system with also LT flooded evaporators
GWP	Global warming potential ($kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$)
h	Enthalpy per unit of mass ($kJ \cdot kg^{-1}$)
HP	High pressure
HS	High stage
HTC	High temperature circuit
LS	Low stage
LT	Low temperature
LTC	Low temperature circuit
$\dot{m}$	Mass flow rate ($kg \cdot s^{-1}$)
MT	Medium temperature
p	Pressure (bar)
PC	CO_2 booster refrigeration system with parallel compression
$\dot{Q}$	Capacity (kW)
R	Pressure ratio (-)
t	Temperature (°C)
$\dot{W}$	Power input (kW)
Subscripts	
o	Outdoor temperature (°C)
cond	Condenser
GC	Gas cooler

Although the R744 features favorable thermo-physical, safety and environmental properties, its use in "CO_2 only" configurations operating in warm weathers still needs to be deeply investigated. Its low critical temperature (31 °C, see Table 1), in fact, entails that trans-critical operating conditions can commonly occur, worsening the performance of the R744 refrigerating plants considerably. Many studies regarding this topic are available in the open literature [3-8].

Mota-Babiloni et al. [9] suggested HFO-1234ze(E) as a good alternative to the high-GWP working fluids for new refrigerators and for CO_2 cascade configurations. The authors also asserted that its lower liquid density and viscosity as compared to R134a permits the system's charge and pressure drop to be reduced. In addition to this, its negligible GWP (see Table 1) leads to insignificant harmful emissions into the atmosphere. Its performance in indirect refrigerating solutions was examined by Llopis et al. [10]. The adoption of such configurations allows reducing the refrigerant charge down to 95% [11], besides simplifying the maintenance in comparison with direct expansion technologies [12]. However, two main disadvantages associated with the secondary loop systems are the need for a pump and the presence of an additional heat exchanger. Mota-Babiloni [13,14] experimentally proved that the use of an internal heat exchanger is energy beneficial to R1234ze(E) based configurations. The data collected by Mendoza-Miranda et al. [15] from a test rig equipped with a variable speed reciprocating compressor suggested that the usage of the R1234ze(E) implies a decrease in both the cooling capacity and the COP in relation to the R134a over the investigated operating conditions.

CO_2 is also a promising secondary working fluid since it offers reduced pump's work and pipes' size and good heat transfer characteristics [16]. Furthermore, the high vapour density of the R744 allows decreasing the inner diameters, while its high saturation pressure permits reducing the compressor's size [17]. In the indirect solutions, CO_2 operates at sub-critical running modes, removing the technical and economic challenges due to very high operating pressures. Inlow and Groll [18] claimed that the secondary loop systems with CO_2 as the secondary fluid can counterbalance the drop in COP caused by the additional heat transfer level and the pump's power.

This study investigates the energetic performance of various solutions, such as a CO_2 booster system with parallel compression (PC), a CO_2 booster system with R290 dedicated mechanical sub-cooling (DMS), a combined CO_2/R1234ze(E) secondary/cascade system having MT flooded display cabinets (CSC) and a CSC with also LT flooded evaporators (FCSC). COP and annual energy consumption of the aforementioned solutions have been compared with those of a R404A direct multiplex expansion (DXS) system. Four hot climatic locations across the continents have been selected for the evaluation, i.e. Seville (Spain), Teheran (Iran), Phoenix (USA) and New Delhi (India). Table 1 summarizes the main properties of the employed refrigerants.

Table 1 Properties of the employed refrigerants [19,20]

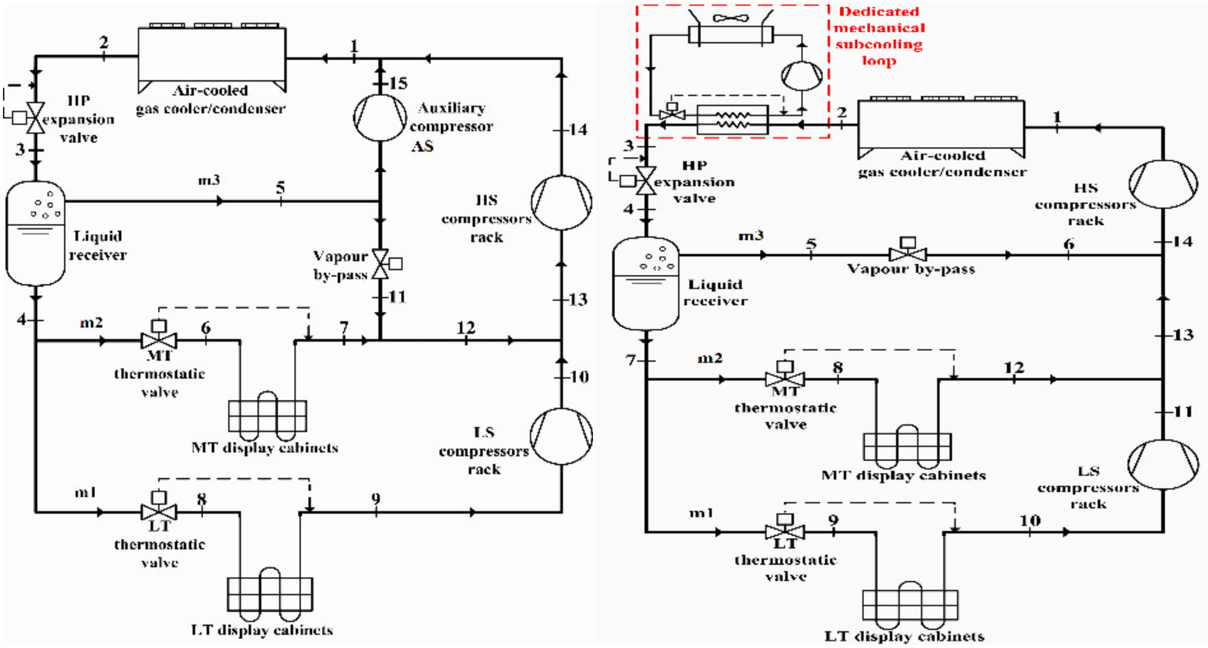
Refrigerant	*R744*	*R1234ze(E)*	*R404A*
Chemical formula or composition (% wt.)	CO_2	$CHF=CHCF_3$	44% R125 52% R143a 4% R134a
Normal boiling point, °C	-56.6	-19.0	-46.2
Critical temperature, °C	31.0	109.4	72.0
Latent heat of phase change at -30 °C, $kJ \cdot kg^{-1}$	303.5	201.5	189.5
Latent heat of phase change at 40 °C, $kJ \cdot kg^{-1}$	-	154.6	120.3
Vapor specific volume at -30 °C, $m^3 \cdot kg^{-1}$	0.027	0.282	0.095
Global warming potential ($GWP_{100\ years}$)	1	6	3700
Safety group	A1	A2L	A1
Lower flammability level, % vol.	0	7.6	None

2. Investigated systems

In the solution with parallel compression (Fig. 1 (a)), the R744 leaving the receiver is compressed from an intermediate pressure to the high one by an additional compressor. As a main consequence, the total energy consumption is reduced in comparison with a conventional booster system. The usage of a dedicated mechanical sub-cooler (Fig. 1 (b)) allows also increasing the refrigerating effect by evaporating the R290 flowing through the sub-cooling loop. In this study, the temperature of the R744 exiting the sub-cooler is set to 15 °C [5]. The combined CO_2/R1234ze(E) secondary/cascade configurations shown in Fig. 2 are selected as additional alternatives to DXS. R1234ze(E) and R744 are employed as the primary and secondary fluid, respectively. The main difference between CSC (Fig. 2 (a)) and FCSC (Fig. 2 (b)) is that the former has flooded evaporators only in the MT circuit, whereas FCSC uses them in both the MT loop and the LT one. The enhancement of the refrigerant-side heat transfer related to this type of heat exchangers allows increasing the corresponding evaporating temperature.

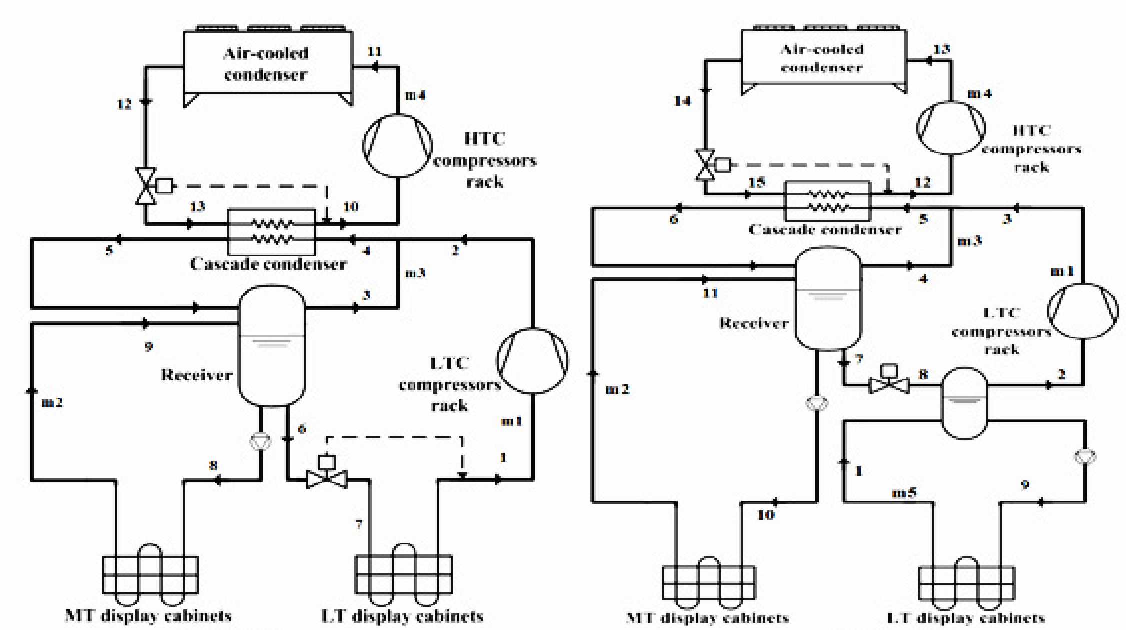
3. Thermodynamic modelling

The models are developed in Engineering Equation Solver (EES) [21] by considering all components running at steady state conditions. The operating conditions of all the investigated systems are summarized in Table 2. Furthermore, all the components are considered well-insulated and all the expansion valves are modelled assuming an isenthalpic expansion. The pressure loss across all the components is neglected and the electricity consumption related to the fans is taken as 3% of the heating capacity of the corresponding heat exchanger. To model the "CO_2 only" booster systems, the methodology suggested by Cecchinato et al. [22] is adopted. A smooth shift from subcritical running modes to transcritical ones is carried out by adopting a transition operating zone. Consequently, four zones are selected for the full range of outdoor temperatures, as already reported by Gullo et al. [5] and summarized in Table 3. The "CO_2 only" solutions are supposed to perform similarly to a conventional booster system at outdoor temperatures below 18 °C as the auxiliary compressor and dedicated mechanical sub-cooler start operating as soon as Zone III occurs.



1 (a) Parallel compression (PC) 1 (b) Dedicated mechanical sub-cooling (DMS)

Fig. 1 "CO_2 only" booster configurations



2 (a) CSC 2 (b) FCSC

Fig. 2 Combined CO_2/R1234ze(E) secondary/cascade configuration

Table 2 Operating conditions of investigated systems

Minimum condensing temperature for indirect systems	25	°C
Condenser approach temperature for indirect systems	10	°C
Minimum condensing temperature for CO_2 systems	9	°C
Condenser approach temperature for CO_2 systems	3	°C
Approach temperature for gas cooler	2	°C
Degree of sub-cooling for CO_2 systems in sub-critical conditions	2	°C
Approach temperature of the cascade condensers for indirect systems	2	°C
MT evaporating temperature (conventional/flooded)	-10/-7	°C
LT evaporating temperature (conventional/flooded)	-35/-32	°C
Internal superheating	5	°C

As regards the compressors, their global efficiencies are computed from the correlations listed in Table 4. They are derived from Frascold Software [23] for HFO compressors, BITZER Software [24] for CO_2 compressors and Dorin Software [25] for R290 compressor, respectively. The pump's circulation ratio for MT and LT flooded evaporators are taken as 1.5 and 2.5, respectively [26]. The pumps' power input is considered as 1% of the total compressors' power [27].

Table 3 Operating zones of the investigated "CO_2 only" solutions [5]

Zone	t_o (°C)	$t_{cond,outmax}$ (°C)	$t_{cond,max}$ (°C)	$t_{GC,outmax}$ (°C)	$p_{GC,max}$ (bar)
Zone I (Sub-critical, fixed t_{cond})	$t_o \leq 4$	7	9	-	-
Zone II (Sub-critical)	$4 < t_o \leq 17$	20	22	-	-
Zone III (Transition)	$17 < t_o \leq 27$	-	-	29	75
Zone IV (Trans-critical)	$27 < t_o \leq 45$	-	-	47	106

Table 4 Compressor global efficiencies

Compressor	Efficiency as a function of R
DXS, LT	$-0.0075(R^2) + 0.0652(R) + 0.5609$, R=pressure of condenser/pressure of LT evaporator
DXS, MT	$-0.0004(R^2) - 0.0021(R) + 0.6989$, R=pressure of condenser/pressure of MT evaporator
CSC and FCSC, secondary fluid circuit	$+0.0111(R^2) - 0.0793(R) + 0.8030$, R=pressure of cascade condenser/pressure of LT evaporator
CSC and FCSC, primary fluid circuit	$-0.0028(R^2) + 0.0419(R) + 0.5305$, R=pressure of condenser/pressure of cascade condenser
CO_2 sub-critical and transition, low stage	$-0.0012(R^2) - 0.0087(R) + 0.6992$, R=pressure of MT evaporator/pressure of LT evaporator
CO_2 sub-critical and transition, high stage	$-0.1155(R^2) + 0.576(R) - 0.0404$, R=pressure of condenser/pressure of LT evaporator
CO_2 transition, auxiliary compressor	$-0.172(R^2) + 0.7095(R) – 0.0373$, R=pressure of gas cooler/pressure of liquid receiver
CO_2 trans-critical, low stage	$-0.0012(R^2) – 0.0087(R) + 0.6992$, R=pressure of MT evaporator/pressure of LT evaporator
CO_2 trans-critical, high stage	$-0.0021(R^2) – 0.00155(R) + 0.7325$, R=pressure of gas cooler/pressure of MT evaporator
CO_2 trans-critical, auxiliary compressor	$-0.0788(R^2) +0.3708(R) + 0.2729$, R=pressure of gas cooler/pressure of liquid receiver
R290, sub-cooler loop	$-0.0226(R^2) + 0.1816(R) + 0.3701$, R=pressure of condenser/pressure of evaporator

Furthermore, the MT and LT cooling capacities, whose design loads are respectively assumed as 120 kW and 25 kW, are supposed to be depending on outdoor conditions [1] by employing the correlation given in equation (1).

$$\text{Load factor} = \left[1-(1-\min)\cdot\frac{(30-t_0)}{(30-5)}\right] \qquad (1)$$

in which *min* points out the minimum fraction of design load (equal to 0.66 for MT and to 0.8 for LT) and t_0 is the outdoor temperature (in °C). The temperature bins over the year of the selected locations are represented in Fig. 3. The outdoor temperature is below 15 °C for 35.6% of the time in Seville, 43.2% in Teheran, 24.8% in Phoenix and 11.9% in New Delhi. The external temperatures exceed the value of 25 °C for 22% of the time in Seville, 29.7% in Teheran, 44.6% in Phoenix and 57.5% in New Delhi. As for the intermediate running modes, they occur for 42.5% of the time in Seville, 27.1% in Teheran, 30.6% in Phoenix and 30.5% in New Delhi, respectively.

4. Results and discussion

Fig. 4 compares the resulting COPs, computed as the ratio of the sum of the total cooling capacity (MT and LT) to the sum of the total required power input (compressors, fans and, if necessary, pumps), of all investigated solutions at different outdoor temperatures. It can be noticed that the COPs of the "CO_2 only" solutions and those of the HFO based configurations remain constant at t_0 up to 4 °C and 15 °C, respectively. This is attributed to the fixed condensing temperature. Also, PC and DMS have the highest COPs at t_0 below 15 °C, whereas the indirect configurations show the worst performance. FCSC, PC and DMS are suitable alternatives to the baseline at intermediate running modes. As for HFO based solutions, FCSC exhibits the highest COPs at t_0 above 23 °C. At very high external temperatures (i.e. above 30 °C), CSC also becomes energetically competitive.

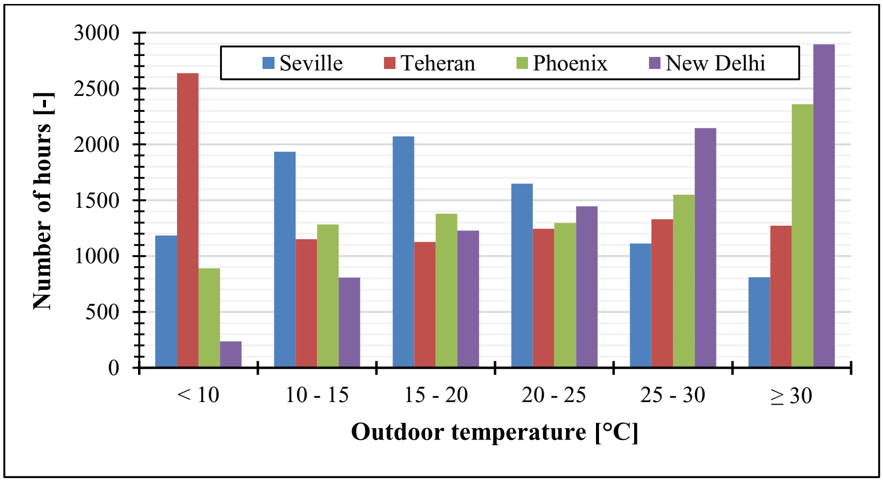


Fig. 3 Temperature bins over the year for the selected hot climatic locations

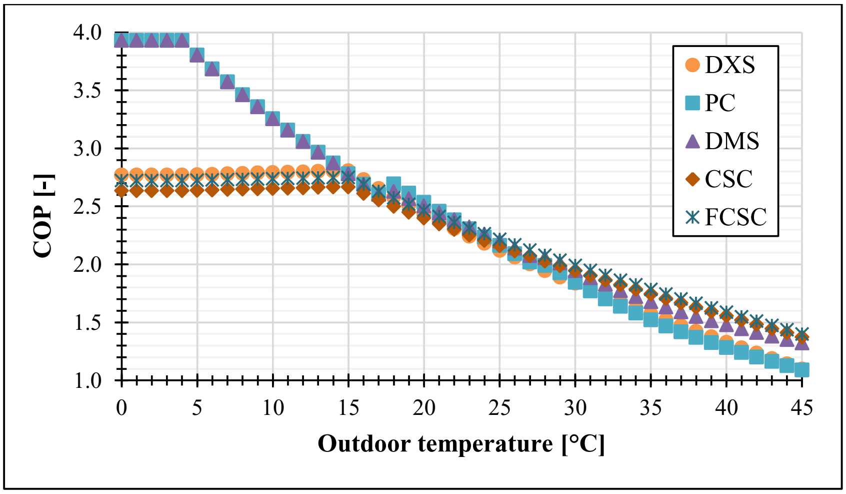


Fig. 4 Comparison of COP of the investigated solutions at different outdoor temperatures

The results related to PC and DMS are consistent with the ones presented by Gullo et al. [5]. According to the authors, the large amount of flash vapor deteriorates the performance of PC significantly at the extreme operating conditions. Furthermore, although the adoption of a dedicated mechanical sub-cooling is more beneficial at high outdoor temperatures, very high energy consumption can be attributed to DMS in these operating modes. On the other hand, PC and DMS reveal a sudden increase in COP at t_0 = 18 °C associated with the launch of the auxiliary compressor and that of the subcooling loop, respectively. The good results related to FCSC can be mainly justified by the presence of flooded evaporators at both the evaporation levels. However, FCSC and CSC show similar performance with increase in the external temperature owing to the modest difference in LT.

According to the specific climatic conditions of the different selected cities (Fig. 3), the annual energy consumption is calculated as the sum of the hourly energy used to run compressors, fans and, if necessary, pumps over the year. The results are summarized in Table 5. Being Seville characterized by the least hot climatic conditions among the evaluated locations, PC and especially DMS represent the best replacements to DXS in this place. In particular, the former achieves an energy saving by 4.3%, while the latter allows reducing the electricity consumption by 5.7%, respectively. Although Teheran reaches high outdoor temperatures over the year, the "CO_2 only" configurations can annually operate in sub-critical conditions for a large number of hours, allowing them to consume at least 6.6% less energy than the baseline. As for the hottest locations considered in this study and in comparison with DXS, DMS and FCSC drop the energy consumption by 6.3% and 5.9% in Phoenix and by 5.2% and 6.4% in New Delhi. In such locations, interesting outcomes can also be associated with CSC as it consumes 3.5% in Phoenix and 4% in New Delhi less energy than DXS.

Table 5 Comparison of annual energy consumption (MWh) of the investigated systems

System	Seville	Teheran	Phoenix	New Delhi
DXS	459.6	470.6	552.1	591.1
PC	440.0	439.5	541.1	586.3
DMS	433.2	428.6	517.3	560.6
CSC	461.6	467.7	532.9	567.2
FCSC	449.1	455.2	519.5	553.4

5. Conclusions

The COP and the annual energy consumption of four supermarket refrigeration configurations have been theoretically evaluated in this paper. The investigated solutions have involved two "CO_2 only" booster configurations and two CO_2/HFO-1234ze(E) combined secondary/cascade systems. Their performance has been compared with that of R404A direct multiplex expansion system in four hot climatic locations, i.e. Seville (Spain), Teheran (Iran), Phoenix (USA) and New Delhi (India). The most relevant conclusions drawn are:

- as for the "CO_2 only" solutions and consistently with the studies available in the open literature, the adoption of the dedicated mechanical sub-cooling has been found more energetically beneficial than the parallel compression at high outdoor temperature operations. The combination of both mechanical sub-cooling and parallel compression could be a better alternative, as suggested by Gullo et al. [5];
- both "CO_2 only" refrigeration systems and CO_2/R1234ze(E) secondary/cascade configurations perform similarly to or better than a conventional R404A direct multiplex expansion system in both hot climatic conditions (e.g. Seville and Teheran) and extremely hot weathers (e.g. Phoenix and New Delhi);
- the "CO_2 only" refrigerating plants exhibit a greater energy saving than the CO_2/R1234ze(E) secondary/cascade solutions in hot climatic conditions (e.g. Seville and Teheran);
- the increase in both the evaporating temperatures allows CO_2/R1234ze(E) secondary/cascade configurations (i.e. FCSC) to achieve better performance than that of the "CO_2 only" systems in very hot climatic conditions (e.g. Phoenix and New Delhi).

In future work, the multi-ejector concept, the integration of the refrigeration system with the air-conditioning unit and additional locations will also be considered.

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