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(Article begins on next page)

09 October 2024

Multi-ejector R744 booster refrigerating plants and air conditioning system integration – A theoretical evaluation of energy benefits for supermarket applications

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Abstract:

The multi-ejector rack is the most promising technology to push the so-called "CO_2 equator" further south and improve the global energy efficiency of R744 supermarket refrigeration systems.

This paper theoretically compares the energy consumption of a CO_2 refrigerating plant equipped with a multi-ejector unit with that of a R404A direct expansion system (DXS), of a conventional CO_2 booster configuration and of two CO_2 solutions using parallel compression. The energy benefits related to the adoption of low temperature (LT) overfed display cabinets were also assessed. Furthermore, various scenarios involving different sizes of the supermarket, integration and capacity of the air conditioning system (AC) and efficiency of the auxiliary compressors were investigated. The evaluations were carried out by considering different locations in Southern Europe. The results showed that, as a function of the selected boundary conditions, energy savings ranging from 15.6% to 27.3% could be accomplished with the multi-ejector concept over DXS.

Keywords:

Compressor Efficiency, Ejector, Overfed Evaporator, Parallel Compression, Transcritical CO_2 Refrigeration System, Warm Climates.

Nomenclature

AC	Air conditioning system
CB	R744 booster refrigeration system
COP	Coefficient of Performance [-]
DXS	R404A multiplex direct expansion refrigeration system
EES	Engineering Equation Solver
EJ	R744 booster refrigeration system using multi-ejector rack, parallel compression and medium temperature overfed evaporator
EJ_OV	R744 booster refrigeration system using multi-ejector rack, parallel compression and both low and medium temperature overfed evaporators

EJ_OV_AC	R744 booster refrigeration system using multi-ejector rack, parallel compression and both low and medium temperature overfed evaporators and integrated with the air conditioning system
GWP	Global Warming Potential [$kg_{CO_{2,equ}} \cdot kg_{refrigerant}^{-1}$]
HFC	Hydrofluorocarbon
HP	High pressure
HS	High-stage
HVAC	Heating, Ventilation and Air Conditioning
IP	Intermediate pressure
LF	Load factor [-]
LS	Low-stage
LT	Low temperature
MP	Medium pressure
MT	Medium temperature
p	Pressure [bar]
PC	R744 booster refrigeration system using parallel compression (slightly variable intermediate pressure)
PC_VAR	R744 booster refrigeration system using parallel compression (variable intermediate pressure)
t	Temperature [°C]

Greek symbols

ω	Entrainment ratio [-]

Subscripts and superscripts

AUX	Auxiliary compressor(s)
cond	Air-cooled condenser
ext	External
gc	Air-cooled gas cooler/condenser
HP	High pressure
HS	High-stage
IP	Intermediate pressure
LP	Low pressure
LS	Low-stage
LT	Low temperature
LTC	Low temperature circuit
MT	Medium temperature
MTC	Medium temperature circuit
out	Outlet
sat	Saturation

1. Introduction

The entry into force of the EU F-Gas Regulation 2014 (European Commission, 2014) and the ever growing concern to the environmental preservation pushes the European countries to find new long-term alternatives to the currently used refrigerants. Even though its high Global Warming Potential (GWP), R404A (GWP = 3922 $\text{kg}_{CO_2,equ} \cdot \text{kg}_{refrigerant}^{-1}$) is still widely employed in European supermarkets causing great direct environmental impacts due to the high annual leakage rate of the working fluids (~15-22% of the total charge). In addition to this, being commercial refrigeration applications particularly energy intensive systems, high indirect greenhouse gases emissions can also be associated with supermarkets. Thanks to its favourable environmental and thermo-physical properties, the interest in carbon dioxide as a refrigerant (R744) in food retail industry has been greatly increasing in the last few years. From the end of 2011 to October 2013, the number of the European supermarkets using "CO₂ only" refrigerating plants increased by about 117% (Shecco, 2014). On the other hand, the use of the R744-based solutions is still confined in Northern and Central Europe where these solutions can guarantee a large energy saving in comparison with the conventional solutions (Gullo et al., 2016a; Sawalha, 2008). Only 21 installations, in fact, can be counted in warm climates countries (e.g. Spain, Italy) where further efforts need to be addressed to enhance the confidence and energy efficiency of transcritical CO₂ configurations substantially. However, the R744 systems exhibit the greatest improvement potentials in terms of integration with the HVAC system, energy conservation and cost-effectiveness.

Girotto et al. (2004) and Sawalha (2008) theoretically compared the performance of a R404A direct expansion system with that of R744 refrigeration units. More recently, many researchers have focused their attention on more promising solutions employing different technologies, such as:

- parallel compression (Chesi et al., 2014; Chiarello et al., 2010; Gullo et al., 2016d, 2016b, 2016a; Hafner and Hemmingsen, 2015; Hafner et al., 2014d, 2014b; Sharma et al., 2014), based on the removal of the flash gas from the liquid receiver by employing an auxiliary compressor in order to reduce the total energy consumption. Its performance was also assessed by Gullo et al. (2016c, 2015) through the application of advanced thermodynamic tools;

- dedicated mechanical subcooling (Hafner and Hemmingsen, 2015; Hafner et al., 2014d, 2014b; Gullo et al., 2016b, 2016a; Llopis et al., 2015), which uses an additional vapour compression refrigeration system to cool down the R744 leaving the gas cooler and increase the refrigerating effect significantly. Llopis et al. (2016) demonstrated experimentally that improvements in Coefficient of Performance (COP) ranging from 6.9% to 30.3% at the evaporating temperature of -10 °C can be achieved;

- overfed evaporators (Gullo et al., 2016d), which allow rising their corresponding evaporating temperature thanks to the better usage of the heat transfer area in comparison with the traditional evaporators. The experimental campaign carried out by Minetto et al. (2014a) showed that at the air temperature of about 0 °C and at the outdoor temperature of about 16 °C, the compressor power input can be reduced approximately by 13% in relation to a conventional solution.

The energy performance of R744 refrigeration plants integrated with the HVAC system was compared with that of separated HFC-based systems by Karampour and Sawalha (2015) and Polzot et al. (2016c, 2016b, 2016a).

Another interesting approach to improve the energy efficiency of the R744 systems is based on the use of the ejectors. Thanks to the fact that they permit recovering a part of the available expansion work to pre-compress the refrigerant, this technology represents, especially in warm/hot weathers, the solution with the highest potential in energy saving, as suggested by Hafner and Hemmingsen (2015), Hafner et al. (2014d, 2014c, 2014b, 2014a) and Minetto et al. (2015, 2014b). Gullo and Cortella (2016) proved that a one-stage CO₂ system using an ejector for expansion work recovery is a more cost-effective solution than that with parallel compression. Experimental results have showed

enhancements by R744-based solutions using two-phase ejectors ranging from 7% (Elbel and Hrnjak, 2008) to 26% (Nakagawa et al., 2011). Banasiak et al. (2015) claimed that the need to handle the gas cooler pressure and, concurrently, recover some expansion work can be more successfully satisfied by means of the multi-ejector rack than by an individual fixed geometry ejector. The authors also fulfilled the design, manufacturing and the performance mapping of a multi-ejector rack operating in a CO_2 refrigeration system for commercial applications. Hafner et al. (2014c) estimated an annual energy saving on the part of an ejector supported parallel compression solution by 14% in comparison with a booster configuration employing the parallel compression in a supermarket operating in Fribourg (Switzerland). The evaluation carried out by Hafner et al. (2014a) pointed out that the adoption of the multi-ejector concept can allow achieving an increase in the efficiency of the system up to 30% in comparison with a conventional booster solution, especially during high outdoor temperature periods. According to Hafner et al. (2012), the use of a multi-ejector rack is more energetically beneficial in Southern Europe, where a decrease in compressors consumption by about 11% can be attained over a configuration without ejectors. Efficiencies of the individual ejector working in a multi-pack of 30% were experimentally measured by Banasiak et al. (2014). The use of the multi-ejector concept in a supermarket operating in Bari (Italy) leads to an energy saving by 22.5% in comparison with a conventional single-stage CO_2 refrigerating plant, as suggested by Minetto et al. (2014b). The experimental data collected by Haida et al. (2016) demonstrated that a R744 refrigeration unit using a multi-ejector pack accomplishes improvements in COP and in exergy efficiency respectively by 7% and 13.7% over a R744 system with the parallel compression.

The main target of this paper is theoretically quantifying the energy saving attainable by employing the multi-ejector rack in a R744 booster supermarket refrigeration system (EJ) in comparison with a conventional booster solution (CB), two CO_2 refrigerating plants using parallel compression with two different control strategies for the intermediate pressure (PC and PC_VAR) and a conventional HFC-based solution (DXS), used as the baseline. The latter is supposed to serve both the medium (MT) load and the LT one. The effect of the overfeeding of the LT evaporators (EJ_OV) has also been evaluated. In a configuration resembling EJ and EJ_OV, both the vapour ejectors and the auxiliary compressors play a crucial role. On the one hand, the reliability and the manufacturability of a multi-ejector rack have already been proven and even its control system has already been successfully implemented. On the other hand, one of the main challenges which still needs to be overcome is represented by demonstrating the necessity to improve the performance of the auxiliary compressors to the manufacturers. In fact, their required power input has a great contribution to the total energy consumption as they deal with large amounts of flash gas. To the best of the authors' knowledge, the energy savings related to an increase in the efficiency of the auxiliary compressors has never been studied for CO_2 refrigerating units equipped with a multi-ejector rack. This assessment is particularly interesting to configurations similar to EJ and EJ_OV and it could significantly promote the diffusion of more efficient components, a reduction in the operating costs and therefore push further towards the usage of the "CO_2 only" refrigerating plants in warm regions. For this reason, this evaluation has been carried out for various sizes of the supermarket and considering the integration of the refrigeration plant with ACs of different capacities. Furthermore, it is worth remarking that few studies about the energy performance of R744 booster supermarket refrigeration systems equipped with a multi-ejector block and integrated with AC are still available in the open literature. In addition to this, the current tendency regarding supermarket applications is having "all-in-one" CO_2 refrigeration system (Karampour and Sawalha, 2015). The simulations have been performed by considering the climatic conditions of five different warm locations, such as Rome (Italy), Lisbon (Portugal), Valencia (Spain), Athens (Greece) and Seville (Spain). The assumptions and the used methods are disclosed in Section 2, while the outcomes obtained are showed and commented in Section 3. Finally, Section 4 summarizes the main conclusions.

2. Methods

2.1. Investigated solutions

In the configuration with parallel compression (Fig. 1), an auxiliary compressor is employed to draw the vapour generated in the liquid receiver owing to the presence of the high pressure (HP) expansion valve. The refrigerant is thus compressed from an intermediate pressure (IP), which is higher than the one in the MT evaporators, to the high pressure, allowing the system to reduce the total energy consumption over CB. In CB, the flash gas share of the working fluid entering the IP receiver is throttled to the medium pressure (MP) by using the vapour by-pass valve in any operating condition. The latter is still included in "CO$_2$ only" units and stands in for the auxiliary compressor in subcritical running modes, when the amount of flash gas is below the minimum operating range of the auxiliary compressor. In these working conditions, both PC and PC_VAR operate in the same way as CB.

Fig. 1 - Schematic of a R744 booster refrigeration system with parallel compression (PC and PC_VAR).

A multi-ejector rack, whose concept was introduced by Hafner et al. (2014a, 2012), includes some vapour ejectors (typically from 4 to 6) and some liquid ejectors (usually 2), which are all fixed geometry devices of different sizes and connected in parallel among them. The former deal(s) with the refrigerant coming out of the high pressure heat exchanger to guarantee the occurrence of the optimal high-side operating pressure. This implies that a part of the vapour in the MP receiver is entrained into the vapour ejector(s) and the total amount of the refrigerant enters the IP receiver. The auxiliary compressor(s) then compress(es) a significant mass flow rate of R744 from IP (higher then MP) to the HP. The reduction in the mass flow rate of the refrigerant drawn by the HS compressors entails a substantial energy saving in comparison with conventional booster solutions. At least one of the ejectors designated for vapour removal operates permanently. Unlike them, the liquid ejector(s) start(s) operating only if the liquid indicator of the MP receiver exceeds a prefixed threshold by pumping some liquid back to the IP receiver. For this reason, the liquid ejectors were not included in this investigation. According to Minetto et al. (2014b), the resulting growth of the annually working time of the auxiliary compressors allows increasing the energy benefits associated with their use and reducing the maintenance problems.

Fig. 2a (EJ) sketches a booster system with ejectors employed for vapour pre-compression, parallel compression and overfed MT evaporators. The adoption of an internal heat exchanger located downstream of the IP receiver and upstream of the low-stage (LS) compressors permits also overfeeding the LT display cabinets (Minetto et al., 2015), as schematized in Fig. 2b (EJ_OV).

In all the evaluated cases, the auxiliary compressors were substituted by a vapour by-pass valve when they were switched off (i.e. in subcritical operating conditions) due to a limited availability of vapour inside the IP receiver.

Fig. 2 – Schematic of a R744 booster refrigeration system equipped with multi-ejector pack, parallel compression and overfed MT evaporators (EJ); (b) Schematic of a R744 booster refrigeration system equipped with multi-ejector pack, parallel compression, overfed MT evaporators and overfed LT evaporators (EJ_OV).

2.2. Simulation models
2.2.1. Description of the investigated R744 refrigeration solutions

At the beginning, the cooling capacities were chosen equal to 120 kW and to 25 kW for the MT and the LT loads (Girotto et al., 2004), respectively. This scenario intends to simulate the running modes of a typical food retail application. At a later time, the influence of the size of the supermarket was also investigated. As for the overfed evaporators, the medium evaporating temperature was increased by 6 K in both EJ and EJ_OV (Wiedenmann, 2015), while an increment by 8 K was selected for the low evaporating temperature of EJ_OV (Wiedenmann, 2015). Such a technology, in fact, allows the refrigerant to come out of the involved heat exchanger as either a saturated vapour or a fluid with a very high vapour content. In comparison with a conventional evaporator and considering the same value of pinch point temperature, the disappearance of the superheating permits obtaining a significant improvement in its overall heat transfer coefficient and, consequently, raising the working temperature. A degree of superheating for all the conventional evaporators and an increment in the refrigerant temperature by 5 K in all the suction lines were assumed. The deviation from the design running modes caused by the variations of the boundary conditions was considered by adopting the correlation suggested by Zhang (2006):

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

in which *min* denotes the minimum fraction of design load (equal to 0.66 for MT and to 0.8 for LT). Table 1 summarizes the operating conditions selected for the evaluated systems. As suggested by Cecchinato et al. (2007), a transition zone between the subcritical working modes and the transcritical ones has to be set to any CO_2 configuration in order to reduce its total energy consumption. Also, it decreases both the approach temperature of the high pressure heat exchanger and the degree of subcooling. The same method as the one proposed by Gullo et al. (2016a) was taken into account in this study. All the auxiliary compressors were supposed to start operating as soon as the transition operating conditions occurred. The consumption related to all the fans was taken as 3% of the heat capacity rejected through the corresponding heat exchanger (Karampour and Sawalha, 2015).

As showed by Bell (2004) and Minetto et al. (2005) and later thoroughly investigated by Gullo et al. (2016a), the intermediate pressure is an optimization parameter for the configuration with parallel compression. On the other hand, the present propensity is to keep it fixed (or slightly variable) so as to make the expansion process more stable and avoid high pressures inside the supermarket (Minetto et al., 2015). In this paper, the intermediate pressure was ranged from 35 bar to 40 bar for the configuration pointed out as PC and from 35 bar to 55 bar for the solution indicated as PC_VAR. The first upper limit was used to simulate the working operations of the currently state-of-the-art solutions, whereas the second one represents a technological constraint for the selected auxiliary compressor. As regards CB, its intermediate pressure was set to 35 bar.

The investigation of the multi-ejector rack was conducted by optimizing the pressure lift, i.e. the pressure difference of the two receivers (Hafner et al., 2014a). Its minimum and maximum values were taken as 4 bar (Banasiak et al., 2015) in order to guarantee an appropriate feeding of the evaporators and as 15 bar (Hafner et al., 2015), respectively. An upper limit of the efficiency of the multi-ejector rack (Elbel and Hrnjak, 2008) of 0.4 was chosen. A correlation (Table 2) derived from the experimental data collected by Haida et al. (2016) was implemented for the rack operating in transcritical conditions. As for the transition and subcritical running modes (Table 2), the experimental data suggested by Palacz et al. (2015) was employed to extrapolate a suitable correlation. In all the cases, the entrainment ratio (ω), i.e. the ratio of the mass flow rate associated with the suction flow to the mass flow rate associated with the motive flow, was computed as a function of both the optimal pressure lift and the gas cooler/condenser outlet temperature. An effectiveness of 0.5 was assumed for the internal heat exchanger belonging to EJ_OV. The

evaporators were assumed to reach the overfed conditions in any running mode. Furthermore, the quality of the refrigerant leaving the overfed heat exchangers was assumed equal to 1, which allowed neglecting the presence of the liquid ejectors.

All the implemented models were simulated by using Engineering Equation Solver (EES) (F-Chart Software, 2015) by assuming steady state working conditions and neglecting the heat losses into the surroundings and the pressure drop taking place in all the components. Furthermore, the basic thermodynamic relations were employed and all the expansion valves were simulated as isenthalpic devices. Semi-hermetic reciprocating compressors were selected for all the systems, whose global efficiencies were obtained by using BITZER Software (BITZER, 2015) for R404A-based solutions and Dorin Software (Dorin, 2015) for R744-based configurations. All the correlations obtained are listed in Table 3 as a function of the pressure ratio.

2.2.2. Description of the air conditioning system

The refrigeration system and the air conditioning unit involve 33% and 20% of the total electricity required in a supermarket (Cecchinato et al., 2010), respectively. Cecchinato et al. (2012) and Karampour and Sawalha (2015) showed that, dependent on the boundary conditions, the CO_2 refrigerating plants integrated with AC can attain lower total energy consumption than separated HFC-based configurations. The energy saving related to "all-in-one" CO_2 refrigeration system could lead to substantial reductions in their total capital investment over the conventional separated systems.

A R410A refrigeration system was selected to match the cooling demand associated with AC and separately perform along with DXS. Except for its evaporating temperature, which was taken as 3 °C (Karampour and Sawalha, 2015), the same assumptions as the ones made for DXS were adopted. Furthermore, the global efficiency of the R410A compressor was derived from BITZER Software (BITZER, 2015) and the corresponding correlation is listed in Table 3. EJ_OV was chosen as an "only-in-one" solution (and pointed out as EJ_OV_AC), which presents an additional overfed evaporator at 5 °C to satisfy the cooling reclaim of the AC. Similarly to the cooling capacities, the capacity of the AC was supposed to be linearly dependent on the outdoor temperature ranging from the design value at t_{ext} = 30 °C to a null value at t_{ext} = 4 °C. Also, it was assumed that it started operating at outdoor temperatures greater than or equal to 24 °C, otherwise free cooling was supposed to occur. Three scenarios based on the assumption of the design capacity of the AC equal to 50% (*Scenario I*), 100% (*Scenario II*) and 200% (*Scenario III*) of MT cooling capacity were investigated.

2.3. Outdoor temperatures

All the following comparisons were based on the climatic conditions in five warm climates in Southern European, i.e. Rome, Lisbon, Valencia, Athens and Seville. According to Fig. 3, the external temperature was less than or equal to 4 °C for about 5.7% of the time over the year in Rome, 0.1% in Lisbon, 1.6% in Valencia, 2% in Athens and 1.3% in Seville. It ranged from 5 °C to 17 °C for about 55.7% of the time in Rome, 57.1% in Lisbon, 48.5% in Valencia, 49.4% in Athens and 48.5% in Seville. The outdoor temperature exceeded 17 °C reaching values up to 27 °C for about 33.6% of the time in Rome, 39.9% in Lisbon, 43.9% in Valencia, 37.7% in Athens and 36.4% in Seville. Also, transcritical operations for the "CO_2-only" solutions (t_{ext} > 27 °C) occurred for about 5% of the time over the year in Rome, 2.9% in Lisbon, 5.9% in Valencia, 10.8% in Athens and 13.7% in Seville.

Fig. 3 - Number of hours per year at different outdoor temperatures in the selected locations (Remund et al., 2014).

3. Results and discussion

3.1. Comparison in terms of COP for a typical supermarket

Fig. 4 compares the COPs of all the evaluated solutions for a conventional commercial refrigeration application ($\dot{Q}_{MT,design}$ = 120 kW and $\dot{Q}_{LT,design}$ = 25 kW). It clearly shows that CB, PC and PC_VAR had better performance than DXS at outdoor temperatures up to 14 °C, whereas the baseline outperformed them starting from t_{ext} = 15 °C. According to Matthiesen et al. (2010), this value represents the current energy limit for R744 refrigerating plants defined as the "CO_2 equator". CB consumed from 1.6% to 20.5% more electricity than the baseline at external temperatures ranging from 15 °C to 40 °C. The occurrence of the transition operating conditions (17 °C < t_{ext} ≤ 27 °C) put the auxiliary compressor into operation allowing PC and PC_VAR to save up to 2.4% of energy over DXS. Consistently with the outcomes by Gullo et al. (2016a), at the extreme working modes the performance of the solution with parallel compression is significantly deteriorated on account of the large amount of flash gas generated in the receiver. At outdoor temperatures above 28 °C, the energy consumption associated with PC and PC_VAR was respectively from 4.2% to 10.2% and from 2.8% to 8.3% higher than that of DXS. In comparison with PC, a further reduction in the energy consumption up to 2.1% could be achieved by adopting PC_VAR. The reason for this result can be justified by considering that in transcritical modes:

- the optimal intermediate pressure tends to be much higher than that evaluated in transition conditions (Gullo et al., 2016a);
- at the same outdoor temperature, the auxiliary compressor running in PC reaches higher values of pressure ratio than that working in PC_VAR. However, the global efficiency of the former tends to be higher than that of the latter.

Both the configurations equipped with the multi-ejector rack could outperform all the previously mentioned systems over the investigated range of temperatures. EJ and EJ_OV respectively showed an increment in COP ranging from 77.6% to 13% and from 96.3% to 17.7% compared to DXS. In transcritical conditions, EJ and EJ_OV consumed on average 14.3% and 19.1% less energy than DXS, respectively. In fact, in relation to EJ, the overfeeding of both the evaporators entailed an additional reduction in the total power input ranging from 10.5%, computed in subcritical conditions, to 3.3%, calculated in transcritical operations. Also, in these running modes the major energy savings on the part of EJ_OV over CB were revealed which, at worst, could reach values of 41.2% and, at best, of 49.9%, respectively. As for EJ_OV, it had from 24.7% to 33.7% lower electricity consumption than PC_VAR over the selected range of temperatures.

Fig. 4 – COP of all the investigated solutions as a function of the outdoor temperature ($\dot{Q}_{MT,design}$ = 120 kW and $\dot{Q}_{LT,design}$ = 25 kW).

3.2. Annual energy consumption for a typical supermarket

The annual electricity consumption of the solutions under investigated for a typical supermarket ($\dot{Q}_{MT,design}$ = 120 kW and $\dot{Q}_{LT,design}$ = 25 kW) are listed in Table 4. In comparison with DXS, the adoption of the parallel compression was energetically justifiable only in Rome where energy saving by about 6.3% could be accomplished. With respect to the other selected cities, reductions in the energy consumption ranging from 2.3% (in Seville) to 3.6% (in Valencia) were estimated. As previously mentioned, no noteworthy difference in terms of power input was noticed between PC and PC_VAR. The annual energy consumption could be dropped by 22.4% in Rome, 19.6% in Lisbon, 20.1% in Valencia, 20.2% in Athens and 19.4% in Seville over the baseline by employing EJ. Also, the overfeeding of the LT evaporators led to consume from 24.1% (in Seville) to 27.3% (in Rome) less energy than DXS, which entailed a further energy conservation from 20.3 MWh·year^{-1} (in Rome)

to 21.3 MWh·year^{-1} (in Seville). It is important to remark that the results obtained were substantially influenced by the weather trend in the selected places. For instance, in Lisbon the "CO$_2$-only" refrigerating plants performed at outdoor temperatures ranging from 15 °C to 17 °C for most of the time. In these operating conditions and as previously described in Subsection 3.1, the performance of all the selected improved R744 systems tended to worsen as the auxiliary compressors were off. On the other hand, in Rome the aforementioned solutions mostly operated at external temperatures ranging from 8 °C to 14 °C at which even CB could outperform DXS.

In comparison with CB, a reduction in the electricity consumption ranging from 4.1% (in Lisbon) to 5.9% (in Seville) on the part of PC, from 20.5% (in Lisbon) to 22.5% (in Seville) for EJ and from 25.4% (in Lisbon) to 26.9% (in Seville) by EJ_OV were estimated. All these results are in agreement with the ones pointed out by Wiedenmann (2015).

3.3. Annual energy consumption of the integrated solution for a typical supermarket

The annual energy consumption of the HFC-based solutions (i.e. DXS for refrigeration reclaim and R410A solutions for the demand of the AC) and that of EJ_OV_AC for the conventional supermarket application ($\dot{Q}_{MT,design}$ = 120 kW and $\dot{Q}_{LT,design}$ = 25 kW) are compared in Table 5. The selected locations for this investigation were Rome, Valencia and Seville so as to consider the least warm location, the site characterized by an intermediate weather trend and the warmest city among the previously chosen places. Furthermore, only EJ_OV was assessed as it represents the most promising configuration among the ones selected in this study, as previously showed.

The outcomes obtained revealed that an annual energy saving ranging from 26% (in Rome) to 22% (in Seville) could be achieved over the separated HFC-based solutions in *Scenario I*. Furthermore, the latter consumed from 24.9% (in Rome) to 19.6% (in Seville) more electricity than the integrated CO$_2$ configuration over the year in *Scenario II*. Despite the large cooling capacity of AC in *Scenario III*, EJ_OV integrated with AC was significantly more (energetically) beneficial than the conventional solutions attaining a reduction in the annual consumption ranging from 23% (in Rome) to 16.6% (in Seville).

3.4. Influence of size of AC, global efficiency of auxiliary compressors and size of supermarket on the annual energy consumption

According to Sharma et al. (2014), the ratio of $\dot{Q}_{MT}$ to $\dot{Q}_{LT}$, defined as load factor (LF), usually ranges from 2 to 5. In this subsection, $\dot{Q}_{LT,design}$ was kept constant in all the evaluations, whereas two additional scenarios involving $\dot{Q}_{MT,design}$ equal to 65 kW (LF ~ 2.5) and to 200 kW (LF ~ 8, which represents a superstore application) were also taken into account. In addition to this, different sizes of the AC were also assessed. All the results are summarized in Table 6 in which "Improved" EJ_OV_AC stands for EJ_OV_AC having 10% higher efficiency of the auxiliary compressors than the previously assessed EJ_OV_AC. Also, the values in brackets for DXS+AC refer to the electricity consumption of the AC.

With respect to the results for LF ~ 2.5, it was noticed that energy savings ranging from 26.2% (*Scenario I*) to 23.7% (*Scenario III*) could be accomplished on the part of EJ_OV in Rome. The enhancement in the performance of the auxiliary compressors led to an additional reduction in the energy consumption from 2.4% (*Scenario I*) up to 3.4% (*Scenario III*). The latter technology is particularly promising in very warm climatic cities, such as Seville. In fact, in this location DXS+AC respectively consumed from 22.7% (*Scenario I*) to 18.2% (*Scenario III*) and from 25.1% (*Scenario I*) to 21.9% (*Scenario III*) more electricity than EJ_OV_AC and "Improved" EJ_OV_AC. The latter exhibited a further energy saving ranging from 2.6% (*Scenario I*) to 3.7% (*Scenario III*) in Rome, from 3% (*Scenario I*) to 4.2% (*Scenario III*) and from 3.6% (*Scenario I*) to 5% (*Scenario III*) in

comparison with the outcomes presented in Subsection 3.2. As regards the results with LF = 8, decrements in the annual electricity consumption over DXS ranging from 26.2% (*Scenario I*) to 22.7% (*Scenario III*) in Rome, from 23.5% (*Scenario I*) to 20% (*Scenario III*) in Valencia and from 21.5% (*Scenario I*) to 15.6% (*Scenario III*) in Seville could be obtained by EJ_OV_AC. On the other hand, "Improved" EJ_OV_AC was further able to reduce the energy consumption from 2.7% (*Scenario I*) to 3.9% (*Scenario III*) in Rome, from 3.1% (*Scenario I*) to 4.3% (*Scenario III*) in Valencia and from 3.8% (*Scenario I*) to 5.2% (*Scenario III*) in Seville compared to EJ_OV_AC.

As an example, the scenario without AC and thus based only on the performance of EJ_OV and "Improved" EJ_OV for LF ~ 5 was also investigated. The latter solution allowed dropping the annual energy consumption by 2.3% in Rome, 2.7% in Valencia and 3.3% in Seville over EJ_OV.

4. Conclusions

In this paper, the annual energy saving associated with the adoption of the multi-ejector concept has been investigated with and without overfed LT display counters and store rooms in five different warm climates European locations, i.e. Rome (Italy), Lisbon (Portugal), Valencia (Spain), Athens (Greece) and Seville (Spain). The analysis has also included two CO_2 configurations with parallel compression, a conventional R744 booster solution and a R404A multiplex direct expansion refrigeration system. The efficiencies of all the selected compressors have been derived from some manufacturers' selection software as a function of the pressure ratio. Firstly, the assessment has been carried out by assuming the running modes of a typical food retail application. At a later time, the effect of the global efficiency of the auxiliary compressors on the annual energy consumption has also been examined. Due to the fact that this parameter will play a crucial role in the near future in the diffusion of the "CO_2 only" refrigeration plants operating in warm climates, the size of the supermarket and the energy consumption associated with the refrigerating unit integrated with ACs of various capacities have also been examined. This sensitivity analysis represents the most interesting and innovative part of the present study.

The main conclusions which can be drawn from this investigation are:

- in comparison with a R404A multiplex direct expansion refrigeration solution for a typical supermarket operating in warm climates, the adoption of the multi-ejector concept on the part of a R744 booster refrigerating plant leads to an annual energy saving by about 20%. The overfeeding of the LT display cabinets permits decreasing the annual energy consumption by about 25%. Both results are consistent with the ones available in the open literature;

- the R744 solution equipped with a multi-ejector pack and both MT and LT overfed evaporators and integrated with AC leads to energy savings ranging from 16.6% to 26.2% in relation to the conventional HFC-based solutions, depending on the size of both the AC and the supermarket. This result points out that a large drop in the total capital investment can be attained by "all-in-one" CO_2 refrigeration system over typical solutions (several independent units);

- the results suggest that the manufacturability of more efficient auxiliary compressors is energetically justifiable, especially in the scenarios involving the integration with AC. In particular, this could also be economically favourable for systems running in warm/hot locations, large food retail industries and large ACs. Also, additional energy decreases ranging from 2.4% to 5.2% over the conventional system equipped with the multi-ejector rack could be achieved, according to the size of both the AC and the supermarket;

- no significant differences have been showed by the two chosen solutions with parallel compression.

The outcomes of this study point to some directions for future work:

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- the evaluation of the energy benefits related to the integration with the heat recovery system will be performed;
- a correct balance in terms of load and a suitable control strategy will be assessed for both the high-stage compressors and the auxiliary ones so as to avoid the degradation of their performance and further reduce the total energy consumption;
- an economic evaluation of all the investigated solutions will be fulfilled.

The authors do believe that "all-in-one" CO_2 refrigeration system will become extremely competitive in the near future even in terms of payback time in comparison with HFC-based solutions. Furthermore, the total capital investment associated with these solutions will be reduced by promoting their spread, while the usage of more highly efficient devices (e.g. improved ejectors, smaller heat exchangers, more efficient compressors, etc.) would permit dropping the operating costs significantly. As a result, a further move of the "CO_2 efficiency equator" further south or even its deletion could be achieved. On the other hand, a thorough assessment of the control strategies should be performed.

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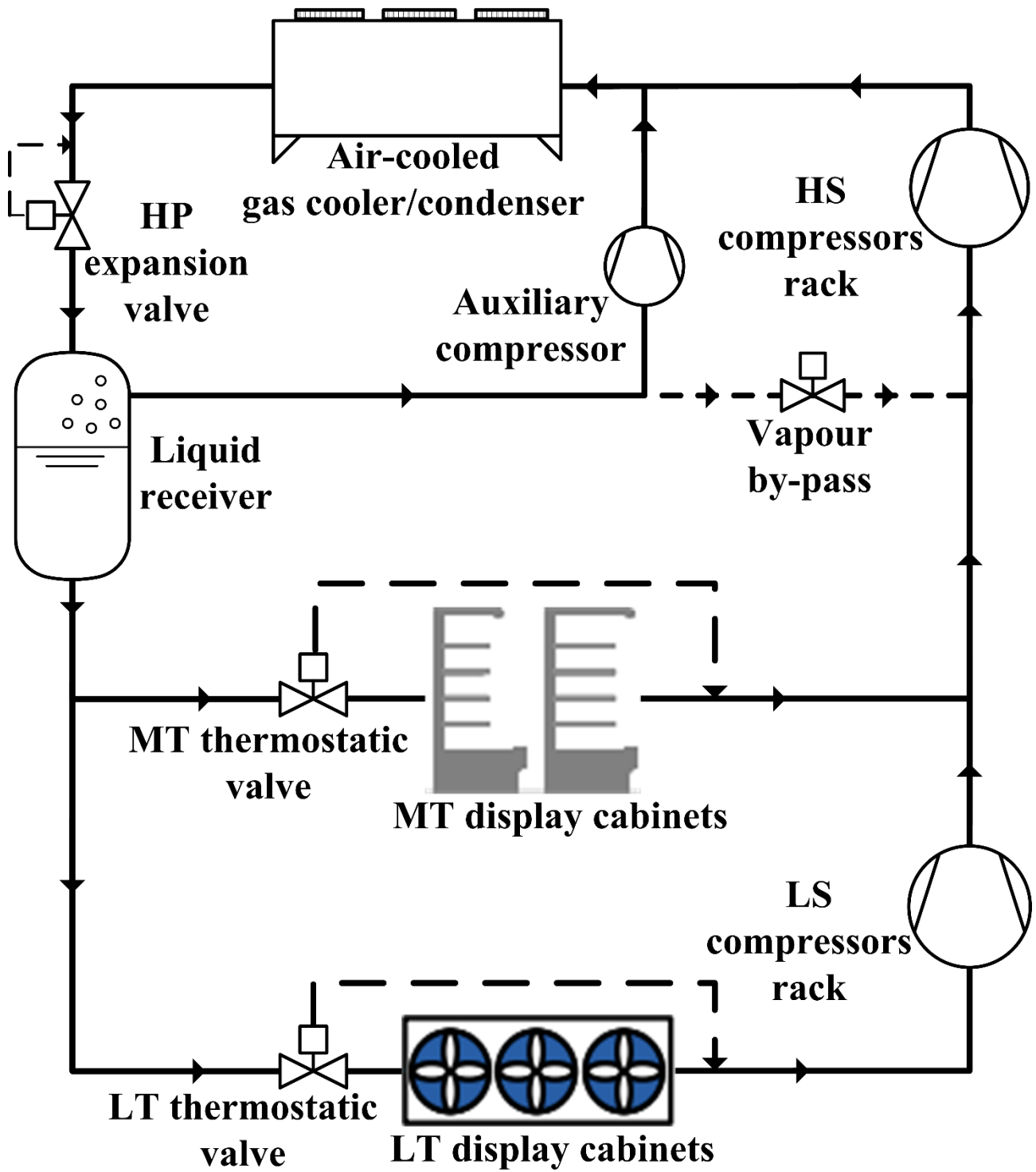benefits. In: Proceedings of the 12th IIR Gustav Lorentzen Natural Working Fluids Conference; Edinburgh, UK.

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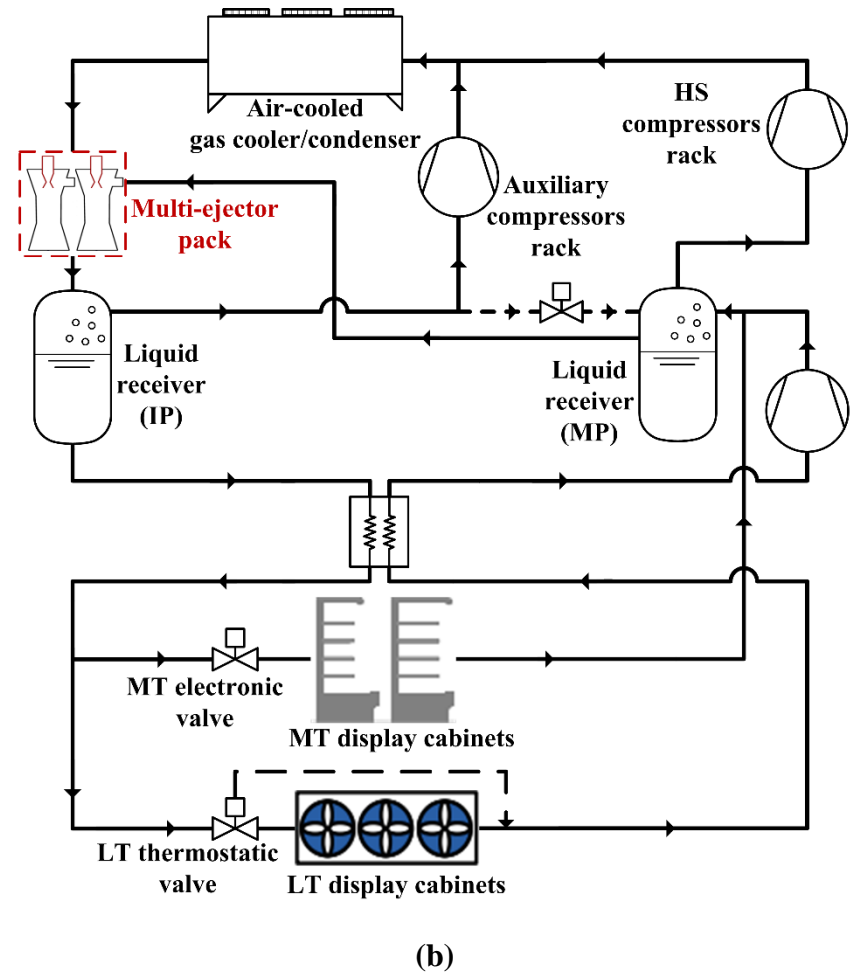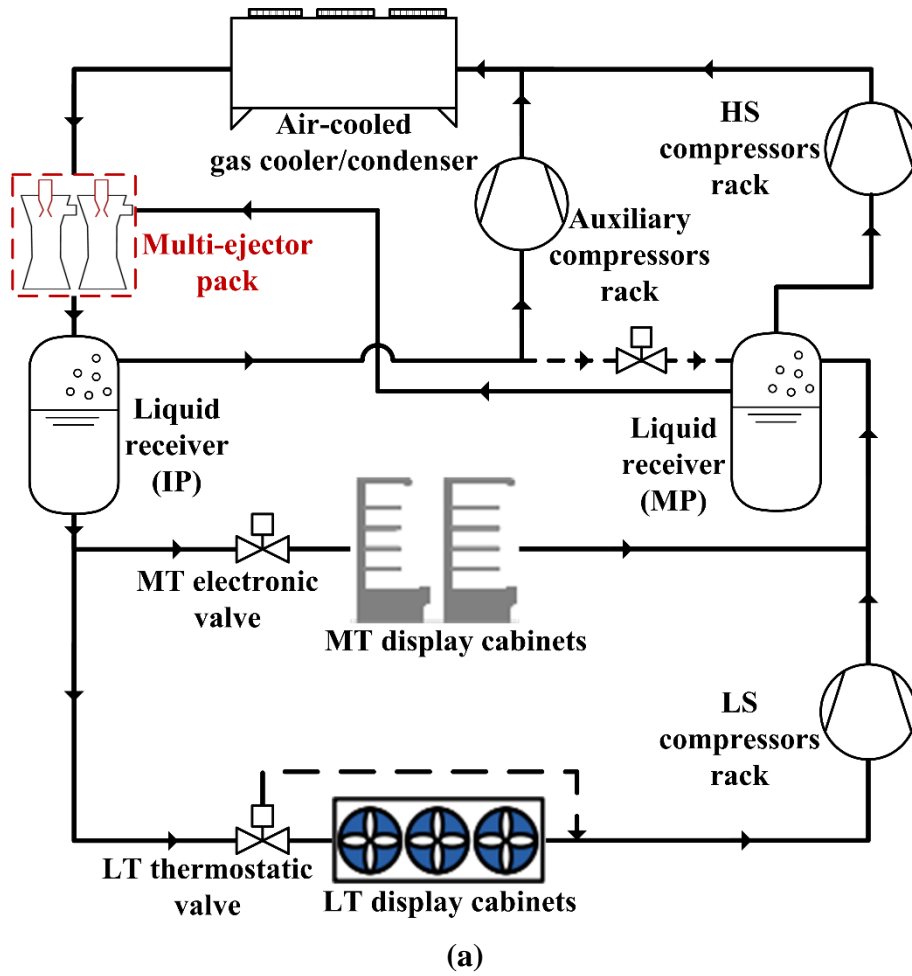
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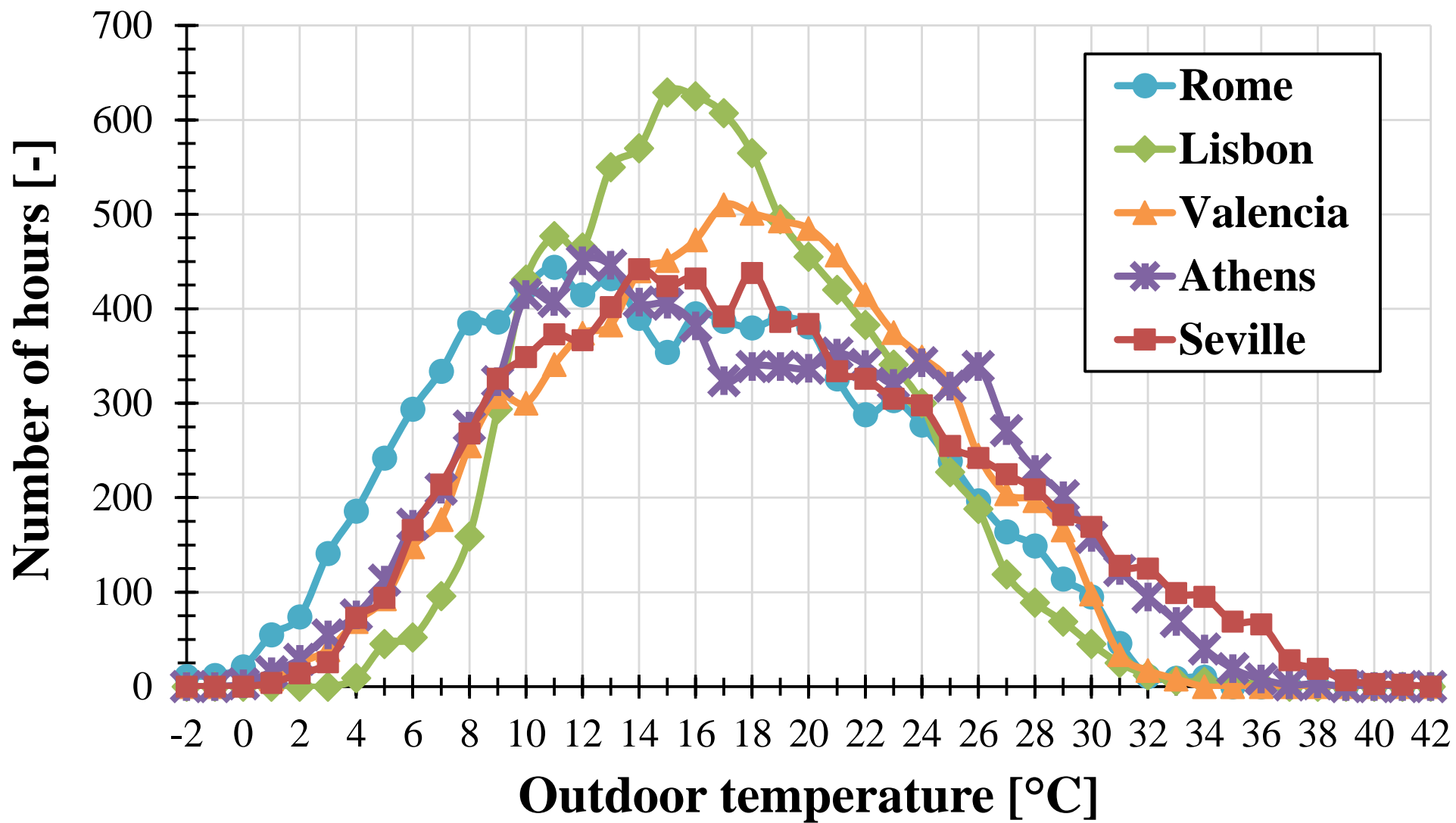
FIGURE 1

FIGURE 2

(a)

(b)

FIGURE 3

FIGURE 4