HIGHLIGHTS

- Transcritical R744 supermarket refrigeration systems are exhaustively reviewed
- Multi-ejector concept is the driving force behind their proliferation worldwide
- “All-in-one” concept will further promote their adoption worldwide
- Nowadays the adoption of these technologies can even be extended to warm locations
- Their potential of enhancement is still considerable

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Transcritical R744 refrigeration systems for supermarket applications: Current status and future perspectives

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Abstract:

Visible signs of climate change call for urgent actions on food retail industry, since such a sector is characterized by an abundant carbon footprint. Being CO_2 (or R744) recognised across the world as the most promising working fluid for supermarket applications, commercial transcritical R744 refrigeration systems have emerged as leading hydrofluorocarbon (HFC)-free technologies.
This study is intended to implement an in-depth review study covering the most important aspects related to the state-of-the-art pure R744 refrigeration plants for food retail applications, including the evolution of system architectures, some field measurements, the main available results from an energy, environmental and economic perspective as well as the indispensable future investigations.
It could be concluded that, in spite of some persisting barriers which still prevent such technologies from a wider adoption, the usage of R744 as the only refrigerant in supermarkets is no longer open to dispute, even in warm locations.

Keywords:

CO_2; Commercial refrigeration system; Field measurements, Multi-ejector; Parallel compression; System integration.

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Nomenclature

Symbols, abbreviations and subscripts/superscripts

AC	Air conditioning
CCHP	Combined cooling, heating and power
CFC	Chlorofluorocarbon
CFD	Computational fluid dynamics
CHP	Combined heat and power
CHRP	Combined heating, refrigeration and power
COP	Coefficient of Performance [-]
CTES	Cold thermal energy storage
DHW	Domestic hot water
GHG	Greenhouse gas
$GWP_{100\ years}$	Global Warming Potential over 100 years [$kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$]
HCFC	Hydrochlorofluorocarbon
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HP	High pressure [bar]
HS	High stage
HVAC	Heating, ventilation and air conditioning
IESPC	Integrated ejector supported parallel compression
IHX	Internal heat exchanger
IP	Intermediate pressure [bar]
LCCA	Life-Cycle Cost Analysis
LP	Low pressure [bar]
LS	Low stage
LT	Low temperature [°C]
MP	Medium pressure [bar]
MT	Medium temperature [°C]
ODP	Ozone Depletion Potential
p	Pressure [bar]
PCM	Phase change material
$\dot{Q}$	Cooling capacity [kW]
SEER	Seasonal Energy Efficiency Ratio [-]
t	Temperature [°C]
TEWI	Total Equivalent Warming Impact [$ton_{CO_{2,equ}}$]
tot	Total
UA	Heat exchanger conductance [$W \cdot K^{-1}$]

Greek symbols

Δ	Difference

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1. Introduction

Supermarkets are dramatically energy-consuming applications accountable for between 3% and 4% of the annual electricity consumption in industrialized countries (Reinholdt and Madsen, 2010; Tassou et al., 2011). In relation to other commercial activities, food retail sector features a very high specific energy demand (roughly between 300 kWh·m^{-2} and 600 kWh·m^{-2}). As a term of comparison, office buildings consume about between 150 kWh·m^{-2} and 200 kWh·m^{-2} (Hafner et al., 2012). Approximately between 35% and 50% of the electricity is required to run the refrigerating equipment (Lundqvist, 2000). Additionally to the refrigeration unit, lighting and heating, ventilation and air conditioning (HVAC) plants also represent energy intensive applications in food retail industry. Such an enormous need for electricity leads supermarkets to be responsible for great indirect contributions to emissions of greenhouse gases. On the other hand, this typology of commercial buildings has become one of the most vital service facilities of modern society. In fact, the total area of food retail stores in both developed and developing countries has been going from strength to strength, being galvanized by many factors, such as the rapid urbanization and the significant openness to foreign investments recently occurred (Traill, 2006). According to EY et al. (2014), the average value of food retail share over the total food market was equal to 44% in 2000 and to 62% in 2011, respectively. Furthermore, the frozen food global market is estimated to grow in sale value by 30% comparing predicted 2020's sales with 2014's values (Persistence market research, 2014).

In spite of its enormous Global Warming Potential (GWP), R404A ($GWP_{100\ years}$ = 3943 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ according to AR5) is still widely used in the European food retail sector. Therefore, due to the profound leakage rates of refrigerant into the atmosphere, an abundant direct contribution to climate change is also ascribable to supermarket applications. The estimated average annual leak rate, in fact, is around between 15% and 20% of the total charge (Hafner, 2015; Hafner et al., 2012, 2014a, 2014c, 2016; Schönenberger et al., 2014). According to Hafner (2015), Hafner et al. (2012, 2014a, 2014c, 2016) and Schönenberger et al. (2014), on a worldwide perspective R22 ($GWP_{100\ years}$ = 1760 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ and ODP = 0.055 according to AR5) is still the most employed working fluid in commercial refrigerating units, featuring a refrigerant leakage rates approximately of 30%. Also, according to SKM Enviros (2012) about 40% of greenhouse gas (GHG) refrigerant consumption in 2010 could be attributable to the food retail sector and, in particular, to large refrigeration plants operating in supermarkets (about 85%). Furthermore, it was predicted that this figure is bound to increase up to 46% in 2020 (SKM Enviros, 2012). In addition, the largest HFC market request in 2015 was owing to the commercial refrigeration sector (EPEE, 2015). However, it is worth remarking that, although HFCs are today's most massive source of greenhouse gases on global viewpoint, the applications relying on such working fluids also feature an ever-growing availability of eco-friendlier and more energy efficient replacements (Shecco, 2016a).

In 1990s the studies conducted by prof. Gustav Lorentzen (Lorentzen and Pettersen, 1993; Lorentzen, 1994, 1995) and the concomitant phase-out of ozone depleting refrigerants forced by the Montreal Protocol promoted a renewed interest in R744. Initially researchers mainly paid attention to mobile air conditioning (Lorentzen and Pettersen, 1993; Lorentzen, 1994) and heat pumping (Lorentzen, 1994, 1995; Nekså et al., 1998) units. On the other hand, first applications were mostly greeted with scepticism by the scientific community. Despite this, a significant spread of CO_2 in heat pumping water heaters, as well as a progressive shift from indirect to transcritical R744 booster refrigeration configurations in supermarket applications took hold at a later time. The initial success, the following decline and renew attention to such a refrigerant was in-depth summed up by Person (2005). The rediscovery of R744 as the only refrigerant for food retail sector also occurred as a consequence of the commencement of the EU F-Gas Regulation 517/2014 (European Commission, 2014). In fact, the adoption of such a legislative act will imply that, in order to attain the expected HFC cut (see Subsection 2.1), the average GWP of refrigerants will have to be brought from 2000 (evaluated in 2016) down to 400 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ by 2030 through the entire refrigeration sector (Shecco, 2016a). Consequently, R744 will play a crucial role in supermarkets to accomplish this goal, as these

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applications feature a dramatic direct contribution to global warming. In fact, thanks to its negligible $GWP_{100\ years}$ ($GWP_{100\ years}$ = 1 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ according to AR5) and the safety level associated with its use (i.e. non-flammability and non-toxicity), R744 is not prone to be phased out. Furthermore, this refrigerant is inexpensive and, in comparison with HFCs, shows higher latent heat, specific heat, density and thermal conductivity and lower viscosity (Kim et al., 2004). Kim et al. (2004), Cavallini and Zilio (2007) and Bansal (2012) provided a thorough overview of R744 properties, as well as of its usages in refrigeration systems. However, the adoption of "CO_2 only" (or transcritical CO_2 or pure CO_2) supermarket refrigerating units is still mainly observable in cold climate countries, whereas these technologies have been replaced by either cascade/indirect arrangements or R404A multiplex direct expansion configurations in warm areas. This has been due to the frequent occurrence of transcritical operation conditions owing to the low critical temperature of CO_2, which entails a substantial deterioration of CO_2 refrigerating plant performance operating in such climate contexts. This peculiarity of carbon dioxide implies that transcritical R744 systems can energetically compete with HFC-based systems at outdoor temperatures up to about 25 °C (Finckh et al., 2011; Sawalha et al., 2017). On the other hand, the EU F-Gas Regulation 517/2014 has also triggered a prominent innovation in "CO_2 only" solutions for high ambient temperature countries, making such HFC-free technologies mainstream for food retail sector worldwide. It is worth remarking the investment risk related to the adoption of new synthetic working with low $GWP_{100\ years}$ (e.g. R1234ze(E), R448A, R449A), as future environmental regulations could impose further restrictions on the use of such refrigerants, similarly to what occurred to chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) and what has been happening to HFCs. Also, most of these working fluids are not yet fully tested in supermarket applications. As regards ammonia (or R717) and propane (or R290), some limitations could also affect their usage in the commercial refrigeration sector due to toxicity and/or flammability hazards.

Despite the great interest drawn and to the best of the authors' knowledge, no thorough state-of-the art review on "CO_2 only" refrigeration plants for food retail applications has been noted. Therefore, the focus of this investigation lies on bridging this knowledge gap. In Section 2 the most important aspects related to the key markets are discussed, while in Section 3 the most relevant studies on transcritical R744 refrigeration systems are summed up. The peculiarities of the state-of-the-art "CO_2 only" technologies for supermarket applications are disclosed in detail in Section 4. In Section 5 the most noteworthy investigations related to the high pressure (HP) control strategy, gas cooler performance and heat recovery implementation are comprehensively described. The "all-in-one" concept, representing one of the most innovative ideas as for these solutions, and the main findings are presented in Section 6. The currently available outcomes on the energy, economic and environmental analyses are summarized in Section 7. Finally, the main conclusions and future developments are stated in Section 8 and Section 9, respectively.

2. Commercial refrigeration sector across the world

2.1. Europe

A massive increase by about 117% in the number of "CO_2 only" refrigerating systems was experienced in the European commercial refrigeration sector between the end of 2011 and October 2013 (Shecco, 2014). However, the usage of these HFC-free technologies was still widespread in Northern and Central Europe due to the aforementioned reason. In fact, only 21 installations were running in Southern Europe (i.e. Spain, Italy) over the same period of time. The coming into force of the EU F-Gas Regulation 517/2014 has been significantly fuelling the use of low-GWP working fluids in the commercial refrigeration sector. As an example, one of the major European supermarket chains recently announced the plan to convert all its food retail stores located in the UK into transcritical CO_2 refrigerating systems by the end of 2018 (r744.com, 2017a). The aforementioned legislative procedure aims at gradually decreasing HFC supply to the European market by 79% from

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2015 to 2030 compared to 2009-2012's average levels. In addition, the use of HFCs characterized by a value of $GWP_{100\ years}$ above 150 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ will be forbidden since 2022 in:

- new multipack centralised refrigerating units with a cooling capacity above 40kW, except in the primary circuit of cascade arrangements in which refrigerants with a $GWP_{100\ years}$ below 1500 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ (e.g. R134a) can be employed;
- new refrigerators and freezers (hermetically sealed) for commercial use.

This ban was confirmed on the 4th of the August 2017 due to the substantiating evidence regarding the current availability of cost-effective, technically feasible, energy-efficient and reliable alternatives to HFCs in food retail industry (European Commission, 2017).

Furthermore, the maintenance of stationary refrigerating systems employing virgin HFCs with a $GWP_{100\ years}$ above 2500 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ with will not be allowed since 2020, whereas this will be prohibited for recycled high-GWP refrigerants as of 2030.

With respect to R134a-based arrangements, it is worth remarking that R134a features an atmospheric lifetime of 13.4 years (AR5), as well as that its GWP over 20 years is 3710 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ (AR5), which is almost three times as high as the value which is usually indicated (i.e. $GWP_{100\ years}$). Also, according to Alternative Fluorocarbons Environmental Acceptability Study (2006), more than half of all R134a ever produced is still in the atmosphere. Consequently, it is possible to state that the adoption of R134a/CO_2 cascade refrigeration systems has to be strongly discouraged, especially in supermarket applications due to their substantial annual refrigerant leakages. In addition, the remarkable HFC phase-down (by 37%) which will particularly occur by 2018 is expected to cause a great rise in their price, as well as a dramatic reduction in the availability of such man-made refrigerants. Shecco (2016a) claimed that a growth in the price of R404A by 15% and in that of R407A, R410A, R407C and R134a by 10% were estimated in 2016.

The first effects of the commencement of the EU F-Gas Regulation 517/2014 in the refrigeration sector were summarized by the European Environment Agency (2016):

- the F-gas imports decreased by about 40% compared to those estimated in 2014 (both by weight and as $CO_{2,equ}$);
- the F-gas production reduced by 5% (as $CO_{2,equ}$) in 2015 in relation to 2014's levels;
- the F-gas supply fell by roughly 24% (both by weight and as $CO_{2,equ}$) as of 2014;
- the F-gas exports went down by 2% (by weight) or 1% ($CO_{2,equ}$) since 2014.

In addition to the EU F-Gas Regulation 517/2014, many countries have introduced some taxes on HFC purchase (e.g. the tax on R134a is 55.3 €$\cdot$kg^{-1} in Norway, 26 €$\cdot$kg^{-1} in Spain, 28.8 €$\cdot$kg^{-1} in Denmark, 35 €$\cdot$kg^{-1} in Sweden, 6.5 €$\cdot$kg^{-1} in Slovenia).

Thus, under the strong pressure of the EU F-Gas Regulation 517/2014 and an ever-growing concern to the environment preservation, the need for the adoption of "CO_2 only" systems as long-term eco-friendly technologies even in warm areas, such as Mediterranean Europe, has become compulsory. In fact, in spite of the remarkable debate accompanying the usage of these technologies in high ambient temperature countries, the great development triggered by the aforementioned legislative act has led to highly performing solutions suitable for any European climate context (Shecco, 2016a). This led to a noticeable growth in the number of pure CO_2 installations in Southern Europe by roughly 8 times from 2013 to 2016. Despite this, CO_2 supermarket refrigerating units still struggle to take root in warm European locations. This lower penetration into high ambient temperature country market is only due to some remaining non-technological barriers (e.g. shortage of trained installers and service technicians, little confidence in transcritical R744 supermarket refrigeration systems, social and political factors) (Minetto et al., 2018). Therefore, this has given rise to the fact that the re-positioning

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of the so-called “CO_2 efficiency equator” (Matthiesen et al., 2010) has become the most important key research area with respect to the commercial pure R744 refrigeration systems. It is worth remarking that this energy efficiency limit was presumed to pass through the northern shore of the Mediterranean in 2013 (Shecco, 2016a). Also, it is possible to notice that the number of “CO_2 only” refrigerating plants in the EU, Norway and Switzerland in 2016 was about 3 times as great as that in 2013 (Shecco, 2016a). This entails that approximately 8% of the European supermarkets are based on pure CO_2 technologies (Shecco, 2016a). In addition, it is important to highlight that there were about 11000 stores employing transcritical R744 systems worldwide and approximately 8730 of them were located in Europe (Shecco, 2016a). This means that the UE is the current leader when it comes to commercial pure R744 technologies. With putting into effect of the EU F-Gas Regulation, the move towards “CO_2 only” units is expected to intensify in the next few years. In fact, Shecco (2016b) estimated that the total number of transcritical R744 supermarket refrigeration systems installed in Europe will be equal to about 27000 in 2020 and 81000 in 2030, respectively.

To conclude, it is obvious that such a legislative act has been noticeably changing the global food retail industry, as well as affecting markets beyond Europe’s borders by inspiring regulators from various regions and countries.

2.2. North America

The enormous technological developments experienced by the European commercial refrigeration sector have significantly promoted the adoption of “CO_2 only” technologies in North America as well. This can be highlighted by taking into account that only 2 transcritical R744 supermarket systems could be counted in the USA in 2013, whereas these amounted to 68 in Canada. However, the number of these HFC-free solutions respectively increased by about 96% and 100% in 2015, being 20 installations located in California and 94 food retail stores run in Quebec. In particular, after imposing a fall on fluorinated gas emissions by 80% by 2030 in new equipment, California has become the current leader in North America with respect to regulations aimed at the environment conservation. As regards Quebec, the policy based on incentivises for companies to employ R744 as the only refrigerant has enormously encouraged the adoption of “CO_2 only” refrigerating units in supermarkets. Also, thanks to the favourable climate conditions, a negligible growth in CO_2 secondary/cascade arrangements took place in Canada between 2013 and 2015. Furthermore, a raise in such solutions by about 76% occurred in the USA over the same period of time. In addition, Shecco (2015a) claimed that the American R744 market is supposed to grow by 100 times since 2015.

However, Shecco (2015a) also highlighted the need on the part of the American government to take a more active role in providing strict regulations designed to massively propel to the usage of climate-friendlier food retail applications. This is also promoted by the fact that natural refrigerant-based technologies have been (widely) commercially available for the commercial sector and chillers in the American market since 2017 (Shecco, 2015a).

Finally, it is worth pointing out that a conventional CO_2 booster system with flash-gas removal (see Subsection 4.1) was installed for the first time ever in the USA in 2013. The unit presents three low temperature (LT) and six medium temperature (MT) compressors, an air-cooled gas cooler/condenser installed on the roof, various types of cases (i.e. LT reach-in cases, LT island cases, MT open cases, MT reach-in cases) and an array of heat exchangers for heat recovery purpose (see Subsection 5.3) connected to the system by means of HP stainless steel piping. In addition, such a solution features the implementation of the hot-gas defrost technique and the use of stepper type electronic expansion valves. In relation to a HFC-based configuration, this refrigeration unit revealed a lower environmental impact as well as comparable electricity consumption and upkeep costs (Navigant Consulting Inc., 2015).

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2.3. Japan

The Japanese market features a success story as for the adoption of transcritical R744 heat pumping units for domestic water heating purposes. As regards the commercial refrigeration sector, a great interest in R744 as the only refrigerant has emerged in the last few years with respect to the applications for convenience stores (Shecco, 2016b). On the one hand, in fact, the number of "CO_2 only" supermarket technologies increased from 190 to more than 1500 between March 2014 and 2016 (Shecco, 2016b). On the other hand, this involved almost exclusively small applications, as the High Pressure Gas Safety Act represented a dramatic barrier to the adoption of this refrigerant in large supermarkets (Shecco, 2016b). Such a restriction on the usage of R744 was removed on the 25th of July 2017 (r744.com, 2017b). Also, Japan's Ministry of Environment recently announced the implementation of natural refrigerant subsidies for food retail and food manufacturing sectors since the beginning of the 2018 financial year (r744.com, 2017c). As a consequence, it is supposed that commercial R744 refrigeration equipment will take hold in the entire Japanese food retail sector as well as that its cost will significantly decrease in the next few years.

2.4. China

The implementation of transcritical CO_2 systems in the Chinese food retail industry is currently in its infancy, as nowadays R744 is mostly employed as a secondary coolant in cascade/indirect arrangements in this sector. However, being China the world's largest HFC producer and a signatory to the Montreal Protocol, the government recently decided to take action against the enormous usage of fluorinated gases. As a consequence, it is expected that the aforementioned HFC-free technologies will gain momentum in the next few years (Shecco, 2015b). According to Shecco (2015b), this will also be strongly related to both the efficiency gains showed by the European installations and the performance of the current only one solution installed in a Chinese supermarket. Finally, it was recently announced the installation of the first Chinese transcritical R744 unit in a store located in the northern part of Beijing (r744.com, 2018a).

2.5. Other areas

Middle East's first transcritical R744 refrigeration unit was recently installed in a food retail store (2000 m^2) located in Al-Salam (Jordan). This solution, featuring the adoption of the multi-ejector concept (see Subsection 4.2.3.1.), heat recovery implementation (see Subsection 5.3), t_{MT} = -2 °C and t_{LT} = -25 °C, was defined as a test for "CO_2 only" supermarket refrigeration plants operating in high ambient temperature countries (r744.com, 2018c).

Also, the number of commercial transcritical CO_2 refrigeration systems running in South Africa (63 up to 2016) is bound to significantly increase as a consequence of their global growth (Shecco, 2016a).

3. Relevant investigations related to transcritical R744 refrigeration systems

Sienel and Finckh (2010) provided a thorough overview on the facilities and the test stands which permitted the development of commercial CO_2 refrigeration systems. The evaluation took into account the compressors, gas cooler, development of evaporators and display counters and optimization of their cost, valves, control system and the whole refrigeration unit.

Finckh and Sienel (2010) comprehensively described the methodologies for the circulation and management of the oil, the solution to the potential CO_2 relieves and the management of the possible occurrence of oil traps, as well as the control system of R744 refrigerating plants.

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Cecchinato et al. (2007) proposed a control strategy aimed at having a smooth shift from subcritical to transcritical running modes. This was based on a linear variation with respect to the outdoor temperature within an operating range circumscribed by an upper and lower limit. These constraints were defined by a couple of values of heat rejection pressure and gas cooler/condenser outlet temperature.

Two transition outdoor temperatures (i.e. 16 °C and 21 °C) for a commercial pure R744 unit for a MT application located in Glasgow (UK) were investigated by Ge and Tassou (2009). The outcomes obtained showed that an energy conservation by about 18% could be achieved by selecting 21 °C as the transition temperature rather than 16 °C.

Cecchinato et al. (2009) showed that, unlike conventional refrigerants, the energy benefits associated with the adoption of an inter-stage heat exchanger (or intercooler or de-superheater) are greater than those related to the usage of staged throttling when it comes to "CO_2 only" technologies. Additional noteworthy energy savings could be achieved with the aid of an internal heat exchanger (IHX). The researchers also estimated that, compared to a one-stage pure CO_2 refrigerating unit, such configurations are capable of attaining average enhancements in Coefficient of Performance (COP) by about 29.3% and 28.7% at evaporating temperatures respectively of -10 °C and at -30°C. On the other hand, the authors pointed out that the control of these systems could be rather challenging to be implemented.

The experimental campaign conducted by Sanchez et al. (2010) was aimed at evaluating the effect of the superheating undergone by R744 due to the electric motor cooling. The analysis considered three different evaporating temperatures (0 °C, -10 °C, -17 °C), four compressor speed values (1150 rpm, 1300 rpm, 1450 rpm, 1600 rpm) and discharge pressures between 74.2 bar and 104.9 bar. The outcomes brought to light a maximum reduction in cooling capacity and COP respectively equal to 20% and 23%, as well as an increase in power input and discharge temperature respectively up to 5% and 28% were also measured.

Torrella et al. (2011) experimentally evaluated the energy benefits associated with the usage of an IHX on the part of a "CO_2 only" refrigerating unit at three different evaporating temperatures (i.e. -5 °C, -10 °C, -15 °C), two different gas cooler outlet temperatures (i.e. 31 °C, 34 °C) and heat rejection pressures between 74.5 bar and 105.9 bar. In comparison with the same system with no IHX, the results showed that a negligible increase in power input is obtained, whereas COP and cooling capacity can be increased up to 12%. Also, the compressor discharge temperature was raised up to 10 °C over the investigated running modes, as well as the thermal effectiveness of IHX was found to be mainly depending on both the evaporating temperature and the gas cooler pressure.

Hafner et al. (2011) suggested the use of oil-free compressors as a means of enhancing both the energy efficiency and cost-effectiveness of transcritical R744 systems. However, the authors highlighted that new maximum discharge temperatures need to be identified, as well as the compressor design has to be adapted with respect to the selected application.

Chesi et al. (2012) developed an experimental apparatus enabling the investigation various thermodynamic cycles as well as the performance evaluation of specific components (i.e. compressors, heat exchangers and expansion valves) related to transcritical R744 systems. In particular, the authors studied the effect of the usage of IHX on both the COP and the cooling capacity at two different suction pressures (i.e. 26 bar and 33 bar) and three different gas cooler exit temperatures (i.e. 20 °C, 30 °C and 40 °C). The results revealed that, although such a heat exchanger is particularly advantageous with rise in temperature lift (i.e. increases up to 30% and 20%, respectively), its use can lead to undesirable discharge temperatures (up to 180 °C).

Cabello et al. (2012) experimentally compared the advantages of the vapour injection technique into three different points (i.e. before IHX, after IHX and before the suction line of the compressor) over a basic single-stage unit. The measurements revealed that similar enhancements are achieved by these configurations with a maximum increase in cooling capacity and COP equal to 9.8% and 7%,

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respectively. The researchers also assessed a maximum drop in the discharge temperature by 14.7 °C, as well as they suggested the adoption of such solutions in warm/hot climates.

The experimental campaign implemented by Sanchez et al. (2014a) showed that at the heat rejection temperature of about 15 °C, a one-stage CO_2 system performs better in subcritical than in transcritical running mode. On the other hand, at the temperatures of about 20 °C and 25 °C, the opposite result was assessed. The authors claimed that these outcomes were due to the poor effectiveness of the selected condenser. The consequent installation of an inverter in the compressor led to noteworthy enhancements in COP and cooling capacity in both subcritical and transcritical operating conditions for low compressor speeds.

Sanchez et al. (2014b) collected some experimental data in order to assess the advantageous related to the use of IHX in two different positions. The results obtained disclosed that the adoption of the configuration with IHX at the gas cooler exit and that with dual IHX are beneficial over all the investigated conditions. These expedients respectively lead to a maximum increment in COP by 10.6% and 13% compared to the basic solution with no IHX. Furthermore, the presence of such a heat exchanger at the liquid receiver exit is not always favourable. Also, the usage of IHXs permits decrementing the optimal heat rejection pressure, whereas increase in discharge temperature up to 20 °C can be reached.

According to Llopis et al. (2015b), high-GWP refrigerating systems for LT applications can be more successfully replaced with cascade/indirect arrangements using climate-friendly working fluids rather than with "CO_2 only" configurations.

4. Evolution of transcritical R744 refrigeration systems for supermarket applications

Many architectures as early transcritical R744 refrigeration plants for food retail applications have been suggested (Dispenza et al., 2005; Cecchinato et al., 2012b; Kaiser and Fröschle, 2010; Sawalha, S., 2008b, 2013; Girotto et al., 2004; Tassou et al., 2011; Matthiesen et al., 2010; Sienel and Finckh, 2010; Finckh and Sienel, 2010).

The R744 refrigerating layout proposed by Girotto et al. (2004) was found to consume 10% more energy than a R404A multiplex system in the North of Italy. In order to enhance the system performance, the authors suggested the usage of a double-stage compression for the MT unit, suction of the vapour in the liquid receiver and reduction of the gas cooler approach temperature (or $\Delta T_{approach}$, i.e. difference in temperature between the outgoing refrigerant and the ingoing cooling medium).

Dispenza et al. (2005) proved the feasibility as well as the suitability of a three-stage R744 refrigeration technology for a hypermarket located in Sicily (Italy). The researchers recommended the adoption of such a configuration for warm climate applications needing a cooling tower.

The centralized R744 systems theoretically investigated by Sawalha (2008b) consume from 4% to 12% less energy than a R404A direct expansion configuration in Stockholm (Sweden). However, the author also showed that these solutions are not suitable replacements for HFC-based systems in high ambient temperature countries.

All-CO_2 cascade refrigeration systems became popular substitutes for subcritical CO_2-based solutions (i.e. cascade/indirect arrangements) in an effort to phase out the chemical refrigerants employed in MT and LT circuits. Also, in comparison with cascade arrangements using man-made working fluids, the oil management is easier to be implemented (Finckh and Sienel, 2010; Matthiesen et al., 2010; Tassou et al., 2011), as well as these technologies allow overcoming some limitations, such as service and installation complexity (Tassou et al., 2011). On the other hand, an all-CO_2 cascade configuration is characterized by an elaborate control system, besides featuring a decrease by about 3% in COP for each increase by 1 K in the condensing temperature (Kaiser and Fröschle, 2010). Consequently, as regards "CO_2 only" systems for large commercial refrigeration installations the spotlight has been on

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booster-based layouts (See Section 4.1). Sienel and Finckh (2010) claimed that the booster configuration can reduce the LT circuit components by 75% compared to all-CO_2 cascade solutions leading to a significant cost reduction. Also, the reliability of CO_2 booster technologies was highlighted by Finckh and Sienel (2010). Tsamos et al. (2017b) estimated an energy saving by about 2% on the part of a CO_2 booster system compared to an all-CO_2 cascade arrangement in both moderate and warm climate contexts.

Finally, it is worth evaluating the current availability of transcritical R744 supermarket refrigerating system components, today's CO_2 compressors can cover all the capacity, from the small (rotating compressors) to the very large capacities (turbo-compressors) (Nekså et al., 2016). As for the CO_2 gas coolers, Nekså et al. (2016) highlighted that counter-flow solutions are usually more desirable. Plate-heat exchangers are also available nowadays for both gas cooler and evaporators. However, Girotto (2017) claimed that although the usage of shell and tube heat exchangers avoid the issues related to thermal stress, plate gas coolers are currently the first choice with respect to "CO_2 only" supermarket applications. The reason for this lies in the fact that these components are more efficient, compact and cost-effective. Nekså et al. (2016) stated that the CO_2 evaporators can also be based on plate-in-shell heat exchangers for bit larger capacities. Tube-in-fin heat exchangers with small diameter pipes for applications using air as the external fluid are also obtainable at the present time. Javerschek et al. (2017a) affirmed that the new R744 compressor generation can lead to an annual energy saving by 13% in Helsinki (Finland), Strasbourg (France) and Athens (Greece) in relation to standard R744 compressor range.

4.1. Basic booster layout (1st generation)

Many researchers have studied the performance of the basic booster refrigeration system with flash gas removal (Ge and Tassou, 2011a, 2011c; Sharma et al., 2015; Shilliday, 2012). The first prototype was developed in the framework of the EU Project "Life" at Danish Technological Institute in June 2006 and installed in a small Danish store in March 2007. This solution featured an energy saving and a drop in the carbon footprint respectively by about 4% and 52% in comparison with a parallel R404A system (European Commission, 2008). The system has been operating since then with no noteworthy problems, leading in a short time to 200 installations in Northern Europe (Matthiesen et al., 2010).

As sketched in Fig. 1, this solution presents two or more booster compressor(s) (i.e. COMP_LO) (hence the name booster configuration) to serve the LT evaporators, which lift the pressure from LT to MT level. Also, first proposed solutions did not permit removing the flash gas (via the vapour by-pass valve indicated as BPV_1 in Fig. 1) generated in the liquid receiver at intermediate pressure (IP) (Sawalha, 2013; Sawalha et al., 2015). However, the energy benefits associated with the flash gas removal (Sawalha et al., 2015) have led such a technique to become best practice in "CO_2 only" supermarket refrigeration systems (Ge and Tassou, 2011a, 2011c; Shilliday, 2012).

Fig. 1. Transcritical R744 booster supermarket refrigeration system and corresponding p-h diagram (Ge and Tassou, 2011a).

Also, conventional booster units with flash gas removal (Fig. 1) are defined as the 1st generation and thus the "old" benchmark of transcritical R744 supermarket refrigeration systems (Nekså et al., 2016; Hafner et al., 2016).

Shi et al. (2010) developed a dynamic model of a CO_2 booster supermarket refrigerating system, which was efficaciously validated by employing some field data. The system performance could be

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accurately predicted at moderate speeds of the gas cooler fans with the aid of a two-dimensional heat exchanger model. Also, the authors showed that the natural convection due to the gas cooler frame height significantly affects the system performance as the fans are off.

Shilliday (2012) implemented and validated against some laboratory data a simulation model of a R744 booster system. The evaluation, which considered the energy data related to a small food retail store located in in Northern Ireland, demonstrated that such a technology performs similarly to a HFC-based unit.

The COPs measured by Sharma et al. (2015) in a laboratory-scale R744 booster refrigeration unit ranged from 3.3 to 1.4 at external temperatures from 10 °C to 35 °C. At a later time, the performance of the aforementioned unit was compared by Fricke et al. (2016) with that of a similar sized laboratory-scale R404A direct expansion system. According to the results obtained, the transcritical CO_2 configuration had on average 15% greater COPs at outdoor temperature ranging from 15.6 °C to 31.1 °C, revealing values between 4.1 and about 2.

Sawalha et al. (2015) collected some field data for 4-18 months in five Swedish supermarkets employing three different "CO_2 only" layouts. These measurements disclosed that increments in energy efficiency up to 16% can be attained by removing the flash gas from the IP receiver, whereas an increase in the evaporating temperature by between 1 and 3 K and an improvement in the efficiency of compressors enhance COP_{tot} up to 14%. In addition, according to Sawalha et al. (2017), "new" installations (i.e. booster-based architectures) are characterized by higher $COPs_{tot}$ at outdoor temperatures below 24 °C, as well as by an energy saving by 20% compared to HFC systems in an average-size supermarket in Stockholm. This outcome was based on the filed measurements collected for 7-9 months in three different Swedish food retail stores using conventional HFC solutions. In like manner Finckh et al. (2011) also showed that a CO_2 booster arrangement performs similarly to or better than a conventional HFC system at external temperatures up to 24 °C. In particular, the researchers estimated that in Frankfurt (Germany) a CO_2 booster solution leads to an energy conservation by about 18% at temperatures up to 10 °C and by 13.8% from an annual basis over the aforementioned system.

Ommen and Elmegaard (2012) successfully validated a numerical model applying a thermodynamic diagnosis based on the characteristic curves method to a CO_2 booster supermarket refrigeration system. The assessment was implemented in both subcritical and transcritical running modes in steady state. The results indicated that the cost of the LT cooling product is about twice as high as that of the MT one.

Heerup and Fredslund (2016) estimated that measured energy consumption related to a small Danish supermarket is about 23% higher than that computed mainly due to the periodic fluctuating load. Also, in comparison with 8 other similar installations, this solution is found to be very energetically efficient.

Gullo et al. (2017) theoretically assessed energy savings between 7.5% and 17% in cold and mild climates (i.e. Oslo, London, Frankfurt, Milan) on the part of a conventional booster system over a R404A unit.

Pure R744 systems are characterized by substantial differences between the heat rejection and the heat absorption pressure due to the occurrence of transcritical operating conditions. This leads to enormous exergy destruction rates related to the expansion valve and thus to highly depreciated performance with rise in cooling medium temperature (Fazelpour and Morosuk, 2014; Cavallini and Zilio, 2007). This marked fall in energy efficiency takes place in both warm locations and heating mode. The adoption of some technological expedients permits partially overcome this drawback. For this reason, researchers' attention has turned to the enhancement of both energy efficiency and cost-effectiveness of more promising solutions than the conventional booster configuration.

In the next subsections, the most relevant characteristics and some practical aspects, as well as some field measurements related to the most state-of-the-art R744 supermarket refrigeration systems were presented.

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4.2. State-of-the-art transcritical R744 refrigeration systems for supermarket applications: 2nd generation and 3rd generation

4.2.1. Parallel compression (2nd generation)

In a conventional R744 booster system, the amount of flash gas removed from the liquid receiver and thus compressed by the high stage (HS) compressors goes up significantly with rise in outdoor temperature. Gullo et al. (2016a) estimated that in transcritical running modes the flash gas mass flow rate is on average equal to 45% of the total mass flow rate. As a consequence, extremely poor performance can be ascribable to such a technology in high ambient temperature countries. A method, which leads to modest enhancements in COP, is that based on the compression of a part of or the total amount of the flash gas from IP to HP with the aid of one or more parallel (or auxiliary) compressor(s). Consequently, the HS compressors are unload in favour of the auxiliary compressors with rise in outdoor temperature. It is important to highlight that such a solution is energy beneficial up to moderate external temperatures. As reported by Gullo et al. (2017b), in fact, compared to a R404A direct expansion unit, the energy efficiency limit commonly experienced by the 1st generation technology at outdoor temperatures above about 14 °C can be pushed up to 27 °C by adopting the 2nd generation solution. At more extreme operating conditions, its performance is significantly compromised due to the massive amount of flash gas generated in the liquid receiver (Gullo et al., 2016a). As highlighted by Javerschek et al. (2015), the advantages related to the adoption of such a solution are strongly depending on the bin hours per year for which parallel compressor can be employed.

Nowadays the configuration employing this technology (Fig. 2) and implementing space and tap water heating recovery (see Subsection 5.3) is pointed out as the 2nd generation as well as the "current" benchmark of transcritical CO_2 supermarket refrigeration systems (Nekså et al., 2016; Hafner et al., 2016).

Fig. 2. Transcritical R744 booster supermarket refrigeration system outfitted with parallel compression and implementing heat recovery (Schönenberger, 2016).

According to Javerschek et al. (2016), at severe operation conditions the flash gas by-pass valve closes and the vapour in the liquid receiver is thus drawn by the parallel compressor. This means that this additional component is put into operation as soon as both the high pressure and the opening degree of the aforementioned valve exceed prefixed thresholds concurrently. The parallel compressor can be alternatively connected to the suction line of the HS compressors via a 3-way valve. This allows taking advantage of employing an additional variable-speed compressor at moderate outdoor temperatures and relatively large cooling loads.

The importance of optimizing the intermediate pressure in a CO_2 refrigeration cycle using parallel compression was demonstrated by many researchers (Bell, 2004; Minetto et al., 2005). The existence of this can be justified by taking into account that both the cooling capacity and auxiliary compressor power input go up with drop in intermediate pressure. According to Minetto et al. (2005), the optimum value of this variable is affected by the displacement ratio of the parallel to the main compressor. The authors also proved the feasibility and the reliability of such a technology with the aid of a test rig, as well as they suggested a solution to the possible issue associated with the oil recapture. However, the refrigerating units which are using such a technology are being run at a fixed (or slightly variable up to 40 bar) intermediate pressure. This allows both a more stable feeding of the expansion valves and avoiding high pressures in the sale area, as reported by Minetto et al. (2015). Also, Gullo et al. (2017a) showed that the energy conservation related to the optimization of the

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intermediate pressure is negligible compared to the same system with slightly variable IP in Mediterranean European climate context. The reason for this lies in the fact that the same auxiliary compressor was selected for these two units. It is worth remarking that at the present time the upper limit regarding the highest pressure which can be reached by the refrigeration plants during shut down periods is set to between 40 and 45 bar. On the other hand, according to Hafner et al. (2014b, 2014d) and Hafner and Hemmingsen (2015) the value of this constraint will be pushed up to 60 bar in the near future.

Da Ros (2005) showed the optimum values of COPs, intermediate and heat rejection pressures at gas cooler exit temperatures above 25 °C.

The investigation by Sarkar and Agrawal (2010) revealed that the optimal intermediate pressure is more affected by the evaporating temperature rather than by the gas cooler pressure. Furthermore, the authors also claimed that the adoption of an auxiliary compressor is a more effective technology in relation to both the usage of parallel compression with subcooler and that of double-stage compression with flash gas by-pass.

Chiarello et al. (2010) carried out an experimental study on a system with parallel compressor mainly operating in subcritical conditions. The data demonstrated the reliability, as well as the good energy efficiency of this technology. The researchers also recommended paying close attention to the design temperature, as this parameter remarkably affects the operating period of the auxiliary compressor and thus the efficiency of the whole system.

Wiedenmann et al. (2014) suggested focussing on the design of the auxiliary compressors in order to suitably select the minimum suction volume rate of the smallest parallel compressor. This would permit a substantial increase in their operating time which would also lead to the enhancement of the overall performance as well as to an increase in the compressor lifetime. Furthermore, the researchers also pointed out that the oil management can be implemented in the same way as for the HS compressors. Up to now, the large number of installations in operation based on this technology have allowed decreasing the costs and obtaining efficient control systems.

Chesi et al. (2014) indicated that, depending on the boundary conditions, the usage of parallel compression theoretically enhances the COP by more than 30% in relation to a basic one-stage CO_2 system. Also, the authors proved both theoretically and experimentally that the performance of such a solution is strongly influenced by the liquid separator efficiency. Displeasing superheating and pressure drop were found to be significant sources of performance deterioration.

Sharma et al. (2014a) carried out a theoretical performance assessment of eight commercial refrigeration systems working in different American locations. The results indicated that the configuration with parallel compression can be energetically competitive with a R404A direct expansion solution at annual average temperatures up to about 14 °C.

Javerschek et al. (2015) estimated that parallel compression technology involves an increase in COP between 8.4% and 13.6% at outdoor temperatures ranging from 25 °C to 42.5 °C. Also, the authors assessed enhancements in Seasonal Energy Efficiency Ratio (SEER) by between 2.9% and 6.4% in Tokyo (Japan) with respect to the bin hours per year for which auxiliary compressor can be run.

Gullo et al. (2016b, 2016d) and Gullo and Hafner (2017b) highlighted that the advanced exergy analysis is a great design tool even for "CO_2 only" supermarket refrigeration systems with parallel compression. Also, such a technology presents on average 18.7% higher both COP and exergy efficiency than the basic transcritical CO_2 unit at cooling medium temperatures ranging from 30 °C to 50 °C, as computed by Gullo et al. (2015).

Fritschi et al. (2016) delved into the operating regimes of a one-stage CO_2 unit with parallel compression at which a growth in its efficiency by at least 10% in comparison with a CO_2 system with flash-gas injection is attained. These conditions are reached at gas cooler outlet temperatures above 27 °C, evaporating temperatures below -7 °C and intermediate pressures up to 45 bar. The authors assumed that the energy savings related to their occurrence lead to the compensation of the

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cost associated with the additional compressor. The evaluation was carried out with the aid of a numerical model validated against some experimental data.

The control strategies and their influence on the annual performance of this configuration were evaluated by Javerschek et al. (2016) from both the analytical and experimental point of view. The outcomes obtained suggested that the optimal intermediate pressure depends on both the load ratio and the condensing/gas cooler outlet temperature.

The vapour injection technique is considered as an alternative to the usage of an auxiliary compressor. On the other hand, Bella and Kaemmer (2011) experimentally showed that, despite the good efficiencies of the former, the technology employing parallel compression is more reliable as it has no issues in terms of both vibrations and the intermediate pressure control.

4.2.2. Overfed evaporators

The superheating of the refrigerant at the evaporator outlet is the most conventional strategy to control the mass flow rate flowing thorough any refrigerating unit. Such an expedient brings about remarkable irreversibilities as the reachable maximum evaporating temperature is restricted due to the pinch point temperature (i.e. difference between air inlet temperature and refrigerant temperature at evaporator outlet) across this heat exchanger. In particular, the superheating implementation is especially detrimental to "CO_2 only" refrigeration technologies owing to the substantial $\Delta p/\Delta T$ ratio and the enormously advantageous heat transfer performance. According to Finckh et al. (2011), an annual energy saving by 3% can be accomplished for each increase by 1 K in MT. This dramatic energy penalization can be reduced by overfeeding the evaporators with the aid of the liquid ejectors (Hafner et al., 2012, 2014a, 2014c; Haida et al., 2016b; Schönenberger et al., 2014) all year round (Hafner and Banasiak, 2016; Hafner et al., 2014c, 2016; Schönenberger et al., 2014) and, less attractively, by employing a pump (Girotto, 2012; Gullo et al, 2016c, 2017b). In fact, on the one hand, these technologies reveal similar results from the energy perspective, as the main benefits related to their usage is associated with the increase in the evaporator operating temperature and the pump power input is negligible (Girotto, 2012; Minetto et al., 2014a). On the other hand, many well-known drawbacks can be associated with the adoption of pumps in supermarket applications (e.g. cavitation issues, cost), meaning that the usage of liquid ejectors should be preferred. Also, overfed evaporators can be combined with parallel compressors with great results (Girotto, 2012; Gullo et al, 2016c, 2017b), even in warm regions.

Minetto et al. (2014a) suggested an innovative method based on the usage of a liquid ejector to overfeed various evaporators arranged in parallel in a commercial pure CO_2 system, as sketched in Fig. 3. Besides proving the reliability and the control stability of this solution, the experimental results disclosed that the compressor power input can be dropped by roughly 13% compared to a conventional configuration. The evaluation was implemented by considering the external temperature of about 16 °C and the air temperature of approximately 0 °C.

Fig. 3. Transcritical R744 refrigeration system with overfed evaporators by employing a liquid ejector (Minetto et al., 2014a).

Gullo et al. (2016c) theoretically estimated that a "CO_2 only" solution equipped with MT overfed evaporators and parallel compression allows accomplishing annual energy savings from 8.2% (in Seville, Spain) to 12.3% (in Rome, Italy) compared to a R404A system in Southern Europe. Also, further energy savings between 2.3% and 2.6% could be attained by adopting a de-superheater.

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A refrigeration plant similar to the one mentioned above was found to consume from 12.4% in a warm climate (i.e. Athens) to 22.7% in a cold weather (i.e. in Oslo, Norway) less electricity than a R404A unit (Gullo et al., 2017b).

Karampour and Sawalha (2018) claimed that the simultaneous implementation of parallel compression and MT and LT overfed evaporators leads to an energetically and economically promising HFC-free solution for warm climates. Such a system, in fact, was found to perform similarly to R404A- and R449A-based direct expansion systems as well as to a R290/R744 indirect solution in Barcelona (Spain), featuring an energy conservation of about 15% compared to a conventional CO_2 booster unit. However, higher energy savings could be ascribable to a R717/R744 indirect arrangement at the same boundary conditions.

Haida et al. (2016b) numerically studied two liquid ejectors of different size hosted in a multi-ejector pack (see Subsection 4.2.3.1). The researchers claimed that short lengths of the pre-mixer and mixer allow improving the performance of both devices over the investigated range of operating conditions. Also, wide diffuser angles enhance the mass entrainment ratio (i.e. ratio of the mass flow rate associated with the suction flow to the mass flow rate associated with the motive flow) and decrease the diffuser length.

4.2.3. Ejectors

The kinetic energy content associated with any refrigerant and owing to the pressure drop taking place in a conventional expansion valve is lost as friction heat. The higher the cooling medium temperature, the more remarkable this drawback is for any vapour-compression refrigeration system. Due to the low critical temperature of CO_2 and the consequent occurrence of transcritical running modes, prominent irreversibilities can thus be attributed to the isenthalpic expansion process in pure CO_2 systems. On the one hand, this causes a more dramatic deterioration of the overall performance than in HFC-based solutions with rise in hot sink temperature. On the other hand, this means that higher potential enhancements in performance can be achieved by replacing a conventional HP expansion valve with a work recovery device (i.e. ejector or expander) in a "CO_2 only" unit. As an example, the published literature suggests that CO_2 and R134a (and R410A) ejectors allow usually recovering respectively about between 20% and 30% and below 20% of the expansion work (Lawrence and Elbel, 2015). This work recovery would lead to a pre-compression of a part of the refrigerant and thus to a higher compressor suction pressure. As two-phase ejectors are much simpler in manufacturing, operation and controlling than expanders, being also easily damageable due to the presence of a large amount of liquid, the researchers' attention has been considerably focusing on such devices. Ejectors also feature cheapness, absence of moving parts and ability to handle two-phase flows with no damage. Furthermore, this technology represents, especially in warm/hot weathers, the solution with the highest potential in energy saving for food retail applications (Hafner and Hemmingsen, 2015; Hafner et al., 2016, 2014d, 2014c, 2014b, 2014a, 2012; Minetto et al., 2015, 2014b; Schönenberger, 2016). Bilir and Ersoy (2009) also claimed that two-phase ejectors are particularly promising for refrigerating plants running in tropical countries and desert area. According to Hafner (2015), the market share of the ejector-based R744 refrigerating systems in food retail industry is likely to be between 50% to 80% for new installations in 2020. Experimental campaigns have showed that transcritical R744 refrigerating systems employing an ejector for work recovery can accomplish improvements in COP between 7% (Elbel and Hrnjak, 2008) and 26% (Nakagawa et al., 2011).

Banasiak and Hafner (2013) highlighted the noteworthy influence of the wall roughness on the ejector performance, especially when it comes to long mixing sections.

The effect of the irreversibilities on the efficiency of a CO_2 ejector (defined according to Elbel and Hrnjak, 2008) was assessed by Banasiak et al. (2014b). The authors concluded that the closest attention has to be paid to the mixing section. On the other hand, the analysis brought to light that the

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components of an ejector need to be simultaneously evaluated to suitably optimize the device geometry.

It is worth mentioning that the conventional individual constant-geometry ejector is not able to ensure an accurate control of the discharge pressure and, simultaneously, implement expansion work recovery effectively (Banasiak et al., 2015). Therefore, two control strategies have drawn interest: the usage of various fixed-geometry ejectors consisting of the so-called "multi-ejector rack" (Hafner et al., 2012, 2014a) and the adoption of an individual device possessing a movable conical needle in the motive nozzle, closing/opening the nozzle throat (Liu et al., 2012). However, Nekså at al. (2016) suggested that the latter involves very limited ejector efficiencies and operation range, especially in off-design conditions. Smolka et al. (2016) numerically compared the performance of fixed- and controllable-geometry ejectors using convergent and convergent-divergent nozzles operating in a CO_2 refrigerating unit. The former exhibited high and easily predictable efficiencies over the investigated operating range. As for the controllable-geometry ejectors, these devices disclosed even better performance as the motive nozzle throat decreases up to about 35% and much lower efficiencies beyond this value. On the other hand, the authors claimed that it is extremely difficult to predict the needle positions (in terms of the needle insertion distance) bringing about satisfactory performance for controllable-geometry ejectors over the studied running modes. Girotto (2017) highlighted that spindle-based ejectors are suitable devices only for a specific mass flow rate/capacity. It is worth remarking a few experimental data related to this solution are currently available, meaning that its real performance and reliability with respect to the control implementation are still unknown. However, nowadays about 50 stores employing multi-ejector arrangement can be counted (Girotto, 2017). Furthermore, experimental study by Banasiak et al. (2015) also revealed that the heat rejection pressure can be satisfactorily controlled by a multi-ejector pack in supermarket applications.

Smolka et al. (2013) developed the mathematical model which, at a later time, was fully validated by Palacz et al. (2015) and employed by Palacz et al. (2017a, 2017b, 2017c), Smolka et al. (2016), Haida et al. (2016b, 2018a, 2018b) and Bodys et al. (2016, 2017) to numerically look into the multi-ejector concept.

The range of running modes in which the homogenous equilibrium model can be applied with a reasonable accuracy to a CO_2 stream flowing through a two-phase ejector was examined by Palacz et al. (2015). The results suggested that, at operating regimes near or above the critical point, the motive mass flow rate can be predicted with errors below 10% over the experimental data. However, the accuracy deteriorates together with the lowering values of pressure at which the expansion process in the motive nozzle invokes evaporation/cavitation (evaporation onset point).

Bodys et al. (2016) numerically contrasted the performance of fixed ejectors with and without swirl flow at the inlet of the motive and suction nozzles. These devices were accommodated in a multi-ejector block of a transcritical CO_2 refrigeration system for supermarket applications. The outcomes showed that positive effects on ejector performance can be obtained by means of a suitable rotational speed of the motive stream. However, the mass flow entrainment ratio goes up less than 4%, implying that this expedient cannot be considered as an appropriate solution to enhance the COP of the investigated technology.

Palacz et al. (2017c) optimized the shape of four CO_2 ejectors improving their efficiency by 6%. In comparison with the base ejector shape, all the devices had the tendency to need for a more extended mixing chamber, a greater value of the motive nozzle diverging angle and a higher motive nozzle outlet diameter. On the other hand, the optimized mixer diameter resulted similar to that of the base ejector shape. Consequently, the enhanced ejectors feature less marked turbulences in the mixing chamber, a smoother expansion process within the motive nozzle and a more uniform velocity field in the mixing section.

Palacz et al. (2017a) optimized the geometry of two CO_2 vapour ejectors by applying two different algorithms, i.e. genetic and evolutionary algorithm. The overall ejector efficiency maximization was chosen as the objective function. The results revealed a strong relation between mixer diameter and ejector performance. As a result of the application of both the aforementioned algorithms, the mixing

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section length and the mixer diameter in the optimal design were respectively found to be greater than and about the same as the baseline design. Also, the optimization procedure led to an increase in the overall ejector efficiency by 2%. Finally, the authors claimed that CO_2 ejector geometry can be effectively optimized over a limited range of running modes with the aid of the presented methodology.

Palacz et al. (2017b) compared the accuracy of the homogeneous equilibrium and homogeneous relaxation models as applied to evaluate the performance of CO_2 ejectors designed for supermarket applications. The researchers concluded that the accuracy enhancement of the homogeneous relaxation approach is unacceptable possibly due to the relaxation time formulation. The study was based on heat rejection pressures between 47 bar and 94 bar and temperatures at motive nozzle inlet between 6 °C and 36 °C, respectively.

Haida et al. (2018a) developed and validated a reduced-order model to accurately evaluate the nozzle mass flow rate of two-phase ejectors in dynamic simulations of supermarket refrigeration systems with negligible time effort.

The performance of CO_2 ejectors for refrigeration and air conditioning supermarket applications was mapped by Haida et al. (2018b). Also, the researchers developed the approximation functions of the ejector pressure lift as a function of the external temperature at air conditioning operating conditions to attain the best efficiency of each ejector.

4.2.3.1. Multi-ejector concept (3rd generation)

The multi-ejector concept was developed by Hafner et al. (2014a, 2012) and involves the simultaneous adoption of highly performing technologies for transcritical R744 supermarket applications (i.e. two-phase ejectors, overfed evaporators, parallel compression). Such a combination of expedients permits maximizing the achievable energy advantages, especially in warm climates. The multi-ejector rack available in the market relies on a block accommodating 4-6 vapour ejectors and 2 liquid ejectors, being all fixed geometry devices of various size and connected in parallel. Besides keeping the optimal high side operating conditions in any running mode, the ejectors aimed at vapour removal pre-compress a part of the refrigerant from the medium pressure (MP) to IP. As a consequence, the suction pressure of parallel compressors is 3÷10 bar higher than that of HS compressors belonging to a conventional booster solution (Hafner and Hemmingsen, 2015; Hafner et al., 2014b, 2014d). This in turn implies a copious unloading of the latter to the detriment of the former (Hafner and Banasiak, 2016; Hafner et al., 2016; Fredslund et al., 2016). The significant usage of the auxiliary compressors entails the achievement of high overall energy efficiencies and the reduction in their maintenance issues (Minetto et al., 2014b) all year round. An additional energy benefit related to the multi-ejector concept is represented by the increase in the operating temperature of the MT evaporators with the aid of the liquid ejectors, as explained in the Subsection 4.2.2. At least one of the vapour ejectors is permanently employed, whereas the devices designed for liquid pumping are put into operation as the liquid indicator exceeds a prefixed threshold in the MP receiver. The required capacity is constantly satisfied by varying the combination of vapour ejectors. The control logic for each ejector is based on a solenoid shut-off valve at the inlet of the motive nozzle and a check valve at the inlet of the suction nozzle. A schematic of a transcritical R744 booster supermarket refrigeration system equipped tin multi-ejector block is displayed in Fig. 4.

Fig. 4. Transcritical R744 booster supermarket refrigeration system outfitted with multi-ejector module (Hafner et al., 2012; Hafner et al. 2014a).

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The solution relying on the multi-ejector concept and performing heat recovery (see Subsection 5.3) represents the 3rd generation of "CO_2 only" supermarket refrigeration systems (Nekså et al., 2016; Hafner et al., 2016).

Girotto (2017) claimed that, depending on the external temperature profile, a liquid ejector with an efficiency of 8% can enhance the overall annual performance by 15%, whereas a vapour ejector with a peak efficiency of 30% by 5%, respectively.

Multi-ejector concept was applied for the first time ever in a Swiss large supermarket ($\dot{Q}_{MT}$ = 120 kW, $\dot{Q}_{LT}$ = 55 kW) in the region of Fribourg in summer 2013 (average annual temperature = +9.6 °C) (Schönenberger et al., 2014; Hafner et al., 2014c). As regards this installation, Schönenberger et al. (2014) and Hafner et al. (2014c) discussed many practical aspects, such as the suitable number of ejectors and the switching from superheated to overfed mode (and vice versa) on the part of the MT evaporators, with the aid of some field measurements. In Fig. 5 the energy consumption of the aforementioned plant is contrasted with the average energy consumption of three CO_2 refrigeration systems with parallel compression operating in just as many similar stores. It is possible to notice that the former consumes about 14% less electricity over the investigated period of time.

Fig. 5. Comparison of the energy consumption of transcritical R744 booster refrigeration systems with and without ejector support in Switzerland (Schönenberger et al., 2014).

Banasiak et al. (2014a) experimentally proved that the efficiency of an individual ejector belonging to a multi-ejector pack can achieve values of 0.3 depending on the discharge pressure and temperature, the pressure lift (i.e. difference in pressure between the two receivers) and the evaporating pressure.

Banasiak et al. (2015) designed, manufactured and implemented a performance mapping of such a prototype for commercial refrigeration applications. The 1D computational model developed by Banasiak and Hafner (2011) was originally employed for designing the vapour ejector with the smallest capacity, whereas the other three devices were simply scaled up. The results related to Banasiak et al. (2015)'s investigation revealed that the ejector efficiency is above 0.3 over a wide range of the investigated operating conditions. According to the authors, the oil management can be suitably performed both within the liquid separator and upstream of the gas cooler. This was also highlighted by Nekså et al. (2016) and Hafner et al. (2016). Also, its control system was already effectively implemented (Banasiak et al., 2015, 2014a; Hafner et al., 2016; Hafner et al., 2014c; Schönenberger et al., 2014; Schönenberger, 2016).

According to Kriezi et al. (2015, 2016), the range of operation conditions at which an ejector performs in the best possible way is significantly influenced by its geometrical parameters. By scaling the ejector, in fact, no significant changes in terms of efficiency are observed. On the other hand, the ejector performance is substantially modified by altering any of its characteristic dimension. For this reason, as the operating conditions are significantly different over the year, Kriezi et al. (2016) suggested employing a liquid ejector designed for summer running modes and another for winter operation regimes. A suitable liquid ejector for summer conditions allows the vapour ejectors to recover more work. Ejectors designated for vapour removal can be suitably used for pumping some liquid in summertime. Kriezi et al. (2015, 2016) also highlighted that it is possible to design ejectors featuring acceptable performance over a wide range of running modes for various applications. In order to achieve this target, close attention has to be paid to many factors, such as ejector geometry, typology of application, system control, boundary conditions.

Minetto et al. (2015) carried out a review regarding the technological aspects, attained knowledge and some experimental results on the subject of auxiliary compressors combined with ejectors.

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Haida et al. (2016a) implemented an experimental comparison between a R744 refrigeration unit using parallel compression with and without multi-ejector pack. Enhancements in COP and exergy efficiency up to 7% and 13.7% could be respectively obtained. Multi-ejector module efficiencies up to 0.33 were measured as a function of the pressure lift and the motive and suction conditions. Also, the authors claimed that the parallel compressor rack needs to be adapted to the usage of the multi-ejector block.

The experimental studies disclosed that the energy efficiency can be increased by 10% between the first and the second generation, whereas at more severe operating conditions additional enhancements by 20% can be accomplished between the second and the 3rd generation, as reported by Nekså et al. (2016). Also, the theoretical assessment conducted by Gullo et al. (2017b) suggested that the adoption of the multi-ejector concept leads to higher COPs in relation to a R404A direct expansion unit at outdoor temperatures up to 40 °C.

Fredslund et al. (2016) presented field data from installations located in various places. The results revealed that the efficiencies of the vapour ejectors measured in the laboratory are very similar to those evaluated in such supermarkets (and above 0.25). The authors also recommended paying close attention to the compressor sizes (so as to meet the required load suitably and have the auxiliary compressors as long as in operation), pressure lift (which should be low, especially for dry expansion evaporators) and the oil return design.

Also, Hafner et al. (2016) reported that:

- abundant decreases in the mass flow rate related to the HS compressors to the disadvantage of the parallel compressors occur, whether or not AC demand is requested (see Section 6);
- the AC demand can lead to the complete switching off of the HS compressors which, in turn, implies that the parallel compressors deal with the total vapour mass flow rate;
- the number of the vapour ejectors in operation strongly depends on the outdoor temperature (and thus on the AC need).

The analysis implemented by Bodys et al. (2017) proved that a multi-ejector block features high and stable performance over the whole range of the evaluated running modes for food retail applications.

The field measurement study by Schönenberger (2016) revealed that the number of the compressor on-off cycles in parallel vapour compression systems supported by ejectors are reduced in relation to conventional solutions. Furthermore, the author estimated an energy saving on the part of aforementioned enhanced technology from 15% to 25 % over CO_2 refrigerating plants employing an auxiliary compressor depending on the heat reclaim, application and weather conditions.

As mentioned above, the adoption of the multi-ejector concept prompts the increase in MT driven by the recovery of the expansion work between HP and MP. Furthermore, as suggested by Minetto et al. (2015), an IHX can be integrated into a R744 ejector supported parallel vapour compression system as a means of overfeeding the LT evaporators too, as schematized in Fig. 6. This leads to the need for a low pressure (LP) accumulator to trap the liquid which cannot possibly evaporate in the heat exchanger pointed out as IHX D in Fig. 6 (Schönenberger, 2016). Also, IHX C (Fig. 6) is employed for heating up R744 before the refrigerant is drawn by the LS compressors in any running mode.

Fig. 6. Transcritical R744 booster supermarket refrigeration system outfitted with multi-ejector module and having both MT and LT overfed evaporators (Schönenberger, 2016).

The available literature suggests that the use of the multi-ejector concept permits increasing MT and LT respectively by 6 K and by 8 K in any operating condition (i.e. even in subcritical modes) and,

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consequently, reducing the frost formation and number of defrost cycles over a conventional solution (Hafner et al., 2014c; Hafner and Banasiak, 2016; Schönenberger et al., 2014; Schönenberger, 2016).

As summarized in Fig. 7, Gullo et al. (2017b) theoretically estimated that the 2nd generation offers energy saving between 1% in Oslo and 5.7% in Athens, whereas the 3rd generation consumes from 17.8% in Oslo and 26.7% in Athens in relation to the 1st generation. On the other hand, the multi-ejector concept leads to energy savings between 16.9% in London (UK) and 23.4% in Oslo over the 2nd generation. It is worth remarking that the heat recapture implementation was not considered in this investigation.

Fig. 7. Annual energy consumption of the 1st, 2nd and 3rd generation of "CO_2 only" supermarket refrigeration systems compared to that of a R404A direct expansion unit in the European climate context (Modified from Gullo et al., 2017b).

Further improvements in the overall system performance can be achieved by promoting the interchangeability of HS and parallel compressors (i.e. principle of pivoting) according to the running modes (Hafner, 2017; Banasiak et al., 2015; Hafner et al., 2014a, 2015; Minetto et al., 2015; Pardiñas et al., 2017a, 2017b). As suggested by Hafner (2017), the compressors can be linked to either the HS or the parallel suction group with the aid of on/off valves located upstream of them. This would also lead to a "gap-free" control of the refrigeration load, besides lowering the installation cost as well as enhancing the compactness of the unit (Hafner, 2017). The need for a more flexible usage of HS and parallel compressors was highlighted by Hafner and Banasiak (2016). Also, Javerschek et al. (2017b) claimed that multi-ejector concept can successfully reduce the required nominal displacement. Furthermore, Hafner et al. (2015) suggested connecting the low stage (LS) compressors to either the IP accumulator or the MP separator, depending on the capacity ratios of the different levels of temperature. At a later time, Pardiñas et al. (2017a) theoretically showed that it is advisable to adopt the first technique as the growth in the power input related to LS compressors is balanced out by the better operating conditions of parallel compressors.

Girotto (2017) summed up the main advantages associated with R744 multi-ejector enhanced parallel compression system as follows:

- substantial reduction of the compressor discharge temperature thanks to both the MP liquid receiver and the overfed evaporators, being advantageous in terms of lifetime of the oil, components on the discharge line and de-superheater employed for heat recovery (see Subsection 5.3);
- increased protection against liquid in the compressor suction manifolds thanks to both the MP liquid receiver and the implemented active methods aimed at limiting the liquid level;
- a decrease in the total installed swept volume compared to a single compression system, as showed by Chiarello et al. (2010);
- a simplified control of evaporators;
- gaining energy competitiveness even at outdoor temperatures up to between 40 °C and 42 °C.

A multi-ejector module was recently installed in Georgia (USA), representing the first food retail ever relying on this technology in North America. In such a climate context and compared to a conventional booster unit, it was claimed that such a solution is capable of raising peak energy savings by 11.3% and between 15% and 23% in non- and optimized operating conditions, respectively (r744.com, 2017d).

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4.3. Other state-of-the-art transcritical R744 refrigeration systems for supermarket applications

4.3.1. Technologies aimed at cooling down R744 exiting the gas cooler/condenser

As mentioned above, the high outdoor temperatures give rise to transcritical running modes causing a considerable difference between rejection and absorption pressures in “CO_2 only” refrigeration plants. As consequences, the refrigerant entering the evaporators features very high quality (i.e. reduction in refrigerating effect) as well as the total power input increases significantly. The growth in cooling effect combined with a decrease in discharge pressure can be attained by cooling down R744 coming out of the gas cooler/condenser with the aid of:

- a dedicated mechanical subcooling, representing the most promising technology for this purpose in supermarket applications;
- an absorption chiller, which can be driven by recovering waste heat (e.g. from the gas cooler/condenser);
- thermoelectric subcooler, whose working principle is based on the Peltier effect (Schoenfield et al., 2012).

4.3.1.1. Dedicated mechanical subcooling

The solution with dedicated mechanical subcooling involves the usage of a further vapour-compression refrigeration unit (i.e. a self-contained unit), as sketched in Fig. 8. This is employed for cooling down R744 coming out of the gas cooler/condenser by promoting the vaporization of the refrigerant circulating in the subcooler loop, which is typically a plate heat exchanger. According to Gullo et al. (2016a), this expedient leads to an average reduction by about 64% in the quality of R744 entering the HP expansion valve in transcritical operation conditions. Consequently, a substantial increase in the refrigerating effect and thus in the overall energy efficiency are attained (Gullo et al., 2016a; Llopis et al., 2015a, 2016). Also, Gullo et al. (2016a) and Llopis et al. (2015a, 2016) highlighted that significant drops in the optimal heat rejection pressure can also be attributed to such a solution.

Fig. 8. Transcritical R744 refrigeration system with dedicated mechanical subcooling (Llopis et al. 2015a).

The theoretical evaluation carried out by Llopis et al. (2015a) suggested that the use of a R134a mechanical subcooling cycle improves COP by 13.7% at -5 °C for a single-stage CO_2 configuration and by 13.1% at -30 °C for a CO_2 cycle with double-stage compression and intercooling over the basic cycle. This evaluation was implemented at the outdoor temperature of 30 °C and with a degree of subcooling of 5 °C.

The experimental data gathered by Llopis et al. (2016) revealed that enhancements in COP ranging from 6.9% to 30.3% at the evaporating temperature of -10 °C can be achieved.

Mazzola et al. (2016) compared the advantages of three different subcooling techniques with the aid of some field data. The investigated technologies involved a dedicated chiller employed as a mechanical subcooling, a water/CO_2 subcooler used for integrating the CO_2 refrigeration system with the air conditioning (AC) unit and the usage of a groundwater source. In relation to a conventional CO_2 booster system and at outdoor temperatures ranging from 40 °C to 48 °C, the authors estimated

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energy savings by roughly 25%, 36% and 30% on the part of the aforementioned solutions, respectively.

Sanchez et al. (2016) experimentally proved that the use of a R290 dedicated mechanical subcooling permits attaining enhancements in cooling capacity as well as in COP between 27.2% and 42.8% and between 5.1% and 19.3%, respectively. The evaluation was performed at the evaporating temperature of -10 ºC and at two different temperatures of the water going into the gas cooler (i.e. 30 ºC, 35 ºC). At the same boundary conditions, the researchers found that the use of IHX increases the cooling capacity and the COP from 1.9% to 11.7% and from 0.8% to 12.2%, respectively.

Beshr et al. (2016) carried out three laboratory tests at steady state conditions in order to implement an experimentally validated simulation model of a CO_2 booster refrigerating plant with a R134a dedicated mechanical subcooling.

The advantages of the application of the advanced exergy analysis to transcritical CO_2 solutions using the aforementioned solution were demonstrated by Gullo et al. (2016d) and Gullo and Cortella (2016b).

A system possessing this expedient combined with a parallel compressor and including domestic hot water (DHW) production was installed in a hypermarket located in Alzira (Spain) in 2013 (Frigo-Consulting LTD, 2014). The gas cooler outlet temperature was kept below 26 °C all over the year, leading to energy savings up to 10% over a HFC-based solution. The configuration investigated by Eikevik et al. (2016) is also capable of performing in subcritical running modes all the year round. This is due to the fact that R290 flowing through the subcooling loop is employed as the cooling medium for the R744 gas cooler at high outdoor temperatures. According to the implemented field tests, the R290 unit should start operating at the external temperature of 23.5 °C, whereas the developed simulation models suggested that it should be put into operation at 19.4 °C. In addition, the authors estimated:

- a seasonal COP respectively equal to 3.22 in Oslo, 2.99 in Paris (France) and 2.85 in Madrid;
- a growth in COP up to 15% with rise in the velocity of the air circulating through the CO_2 condenser;
- an enhancement in COP from 8.84% at 18.7 °C to 6.91% at 40 °C by adopting a new design of the CO_2-R290 condenser.

The results obtained by Nebot-Andrés et al. (2017) suggested that the system equipped with the dedicated mechanical subcooling outperforms the corresponding cascade arrangement in MT commercial refrigeration applications.

Dai et al. (2017) recommended the adoption of such a solution for low evaporating temperatures and hot climate contexts.

The energy performance of a "CO_2 only" refrigeration plant outfitted with a dedicated mechanical subcooling using zeotropic mixture was investigated by Dai et al. (2018). Compared to pure R32, the outcomes suggested that the overall COP can be enhanced by 4.91% at the evaporating temperature of -5 °C and external temperature of 35 °C by adopting R32/R1234ze(Z) (55/45).

It is important to highlight that, although such a technology is particularly efficient at very high hot sink temperatures, it also features a high investment cost, global complexity and complicated control strategy, as suggested by Gullo et al. (2016a) and Girotto (2017). In particular, Girotto (2017) remarked that many drawbacks can be related to this solution, such as the risk/admissibility (i.e. flammability, toxicity) associated with the working fluid employed in the subcooler cycle and the need to utilize a double loop (i.e. increase in the complexity and cost as well as reduction in the overall system efficiency). Also, heat recovery (see Subsection 5.3) for space heating and DHW production could also be difficult to be performed.

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4.3.1.2. Subcooling by means of an absorption chiller

Arora et al. (2011) compared the energy and exergy performance of a transcritical CO_2 refrigeration unit coupled with a single effect H_2O-LiBr absorption system with that of a conventional pure R744 refrigeration plant. The combined solution employs the evaporator of the absorption chiller to cool down CO_2 exiting the gas cooler/condenser, a turbine as the replacement for the conventional expansion valve as well as the heat recovered from its HP heat exchanger to feed the absorption unit. The authors estimated an increase in refrigerating capacity, in COP and in exergy efficiency by 14.05%, 14.20% and 3.67% compared to the conventional solution at the evaporating temperature of –25 °C and CO_2 subcooler exit temperature of 40°C.

The results obtained by Salajeghe and Ameri (2016) reveal that their proposed solution is a suitable substitute for a basic "CO_2 only" unit with and without IHX operating for LT applications. The investigated system involved a single effect H_2O-LiBr absorption chiller to reduce the temperature of R744 exiting the gas cooler/condenser, an IHX located downstream of the HP heat exchanger and a solid oxide fuel cell to drive both the compressor of the transcritical R744 vapour-compression unit and the generator of the absorption unit.

4.3.1.3. Thermoelectric subcooler

Schoenfield et al. (2012) experimentally proved the feasibility associated with the coupling of a transcritical R744 refrigeration plant with a thermoelectric subcooler employed for cooling down R744 coming out of the gas cooler/condenser. In relation to a basic "CO_2 only" refrigeration unit, the combined system presented an enhancement by 3.3% in COP with a corresponding increase by 7.9% in cooling capacity.

The results obtained by Sarkar (2013) suggest that the adoption of a thermoelectric subcooler on the part of a transcritical R744 refrigeration plant permits decreasing its heat rejection pressure, the compressor pressure ratio and the compressor discharge temperature as well as improving the volumetric cooling capacity compared to a conventional "CO_2 only" solution. In addition, the researcher estimated a maximum enhancement in COP by 25.6% and a maximum decrease in high pressure by 15.4% over the investigated operation conditions.

4.3.2. Thermal energy storages

Supermarket applications feature highly fluctuating refrigeration loads between opening and closing time owing to many factors, such as variable indoor thermo-hygrometric conditions, blind removal and disturbance of customers. Therefore, significant performance enhancements could be achieved by shifting a part of the refrigeration reclaim from more adverse (i.e. daytime) to more favourable (i.e. night-time) running modes with the aid of cold thermal energy storages (CTESs). Besides leading to peak load cuts and thus energy consumption reductions, these technologies also potentially permit cost savings and equipment size decrements. According to Fidorra et al. (2015b), CTESs can be classified from the capacity viewpoint as follows:

- small capacity storages, which use a phase change material (PCM) directly employed in the display counters to attenuate the product temperature variation (Alzuwaid et al., 2015; Manescu et al., 2017);
- medium capacity storages, aimed at shifting the loads from daytime to nigh-time operating conditions by using a large ice-bank storage or an ice slurry tank (Ferrandi and Orlandi, 2013; Heerup and Green, 2014; Polzot et al., 2016a, 2016b; Fidorra et al., 2015b, 2016; Nöding et al., 2016);
- large capacity storages, meaning soil and aquifers (Ohannessian and Sawalha, 2014; Leiper et al., 2014; Rehault and Kalz, 2012).

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Another distinction is generally made in terms of working principle, i.e. these technologies can be latent and sensible thermal storages. Fidorra et al. (2015b) also highlighted the difference between auxiliary and integrated storages. The auxiliary reservoirs, which are connected to the refrigerating unit, undergo a charge process as the outdoor temperature falls and/or in night-time periods. In fact, at these operation modes the system is characterized by high energy efficiency and, possibly, by low cost of the electricity too. However, the discharge process involves the cooling down of the refrigerant leaving the gas cooler/condenser so as to enhance the overall performance at high external temperatures. Similar technologies were investigated by Ferrandi and Orlandi (2013), Heerup and Green (2014) and Ohannessian and Sawalha (2014). The integrated storages are installed between the evaporator and the space needing to be cooled down. Their target is to reduce the short-term fluctuations of the refrigeration load by employing either PCMs in the display cabinets or directly the secondary loop of indirect supermarket arrangements. Unlike the first case, this typology of storages can even operate as the refrigeration system is switched off to keep the cooled space temperature.

4.3.2.1. Small capacity storages

Manescu et al. (2017) proposed an innovative concept concerning the use of a cold energy storage located on the top of the display cabinet (Fig. 9). Also, the heat exchange between the evaporator and the cold storage is based on the thermosiphon effect. In addition to the conventional advantages ascribable to CTESs, such a solution involves the usage of unused space within the display counter. To the best of the authors' knowledge, this is the only available investigation which relates transcritical R744 supermarket refrigerating systems to CTESs.

Fig. 9. a) Charging mode and normal operation of the cabinet, b) normal operation, c) charging mode of the CTES, d) discharging by thermosiphon circulation (Manescu et al., 2017).

4.3.2.2. Medium capacity storages

Ferrandi and Orlandi (2013) suggested the use of a sensible cold storage (from 12 m^3 to 27 m^3) to cool down R744 leaving the HP heat exchanger in the daytime. An additional evaporator was employed for recharging the cold storage during the night. The adoption of the cold storage allows the transcritical running modes to take place at higher outdoor temperatures. Furthermore, the authors claimed that this expedient is more energetically beneficial in summertime.

The theoretical study by Heerup and Green (2014) assessed the energy and economic benefits related to an ice storage integrated into a CO_2 booster configuration with flash gas by-pass valve. The evaluation considered two climate contexts (i.e. a cold and a warm location), five night-time tariffs and two case studies. The results obtained disclosed that daytime peak power by about 50% can be achieved at summer conditions, as well as reductions in energy consumption between 5% and 15% can be accomplished on a yearly basis. On the other hand, the amortization period was supposed to be longer than 5 years, being unacceptable for supermarket applications.

Polzot et al. (2016a) proved that, in order to equal the energy consumption of a R134a/R744 cascade system operating in the North of Italy, a conventional CO_2 booster system coupled with a 950 m^3 fire prevention tank (i.e. sensible CTES) needs to adopt parallel compression. Also, Polzot et al. (2016b) estimated energy savings between 3% and 5% compared to a separated solution by thermally integrating a heat pumping unit with the aforementioned system.

Fidorra et al. (2015b) showed that, for the selected case studies, the use of a CTES at 15 °C in a CO_2 booster system with parallel compression (Fig. 10) leads to total energy savings and total cost reductions up to 5.6% an 5.9%, respectively. On the other hand, the adoption of a CTES at 0 °C (i.e.

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ice bank storage) (Fig. 10) brings the total energy consumption and the total cost down to 3.5% and 3.9%, respectively. The better results obtained in the first case were due to the location of the charge storage, whose additional load could be satisfied via the auxiliary compressor.

Fig. 10. Layout of storage integration: a) reference layout, b) improved layout, c) possible integration with air conditioning system (Fidorra et al., 2015b).

Fidorra et al. (2016) examined four various booster refrigeration layouts integrated with both latent and sensible CTESs in terms of power input drop and efficiency gain. The evaluation was based on outdoor temperatures between 10 °C and 40 °C and steady state operations. The study extensively described many aspects, such as the different advantageous and disadvantageous of the aforementioned solutions and the required equipment.

The energy advantageous related to the decoupling of a CO_2 refrigeration system from the heating unit through the adoption of heat storage were evaluated by Nöding et al. (2016). The study involved many different control strategies of heat rejection pressure, leading to a daily energy saving by 8.5% over a conventional R744 unit in a typical day in January in Braunschweig (Germany) in the most realistic investigated scenario.

4.3.2.3. Large capacity storages

Leiper et al. (2014) gathered some field measurements from the 1st of September 2013 to the 1st of January 2014 in two CO_2 booster systems, i.e. with and without geothermal loop. Both stores were located in the UK and featured similar cooling capacity, whereas the gross floor area of the ground coupled supermarket was 20% greater. The results revealed that the geothermal loop impedes the occurrence of transcritical running modes all year round, as well as it permits a reduction in the annual energy consumption by 24.6%.

In the review on the research regarding CO_2 supermarket refrigeration systems and heat recovery carried out at KTH Royal Institute of Technology, Karampour et al. (2015) reported that a long term storage working as a subcooler can reduce the annual electricity consumption by 6%. This value was also estimated by Ohannessian and Sawalha (2014) by adding a thermal energy storage to a CO_2 booster refrigeration plant (with no flash gas removal) using a ground source heat pumping unit for heating purposes and operating in Stockholm. On the other hand, the researchers also reported that a CO_2 booster solution performing heat recovery (see Subsection 5.3) consumes approximately 8% less electricity than that employing a ground source heat pump unit.

The state-of-the-art supermarket located in Germany and investigated by Rehault and Kalz (2012) (Fig. 11) can reduce the total energy consumption by about 20% over a standard food retail store. This target was accomplished with the aid of innovative technologies for lighting, heating/ventilation and refrigeration. The refrigerating plant, in fact, is a CO_2 booster unit equipped with borehole heat exchangers employed as a subcooler in summertime and as a further heat source in the cold months. In addition, the heating mode occurs only in subcritical running modes and relies on a dedicated compressor. The authors also claimed that up to 50 % of the heat rejected through de-superheater can be recovered in wintertime.

Fig. 11. Schematic of a R744 booster supermarket refrigeration system with borehole heat exchanger (Rehault and Kalz, 2012).

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4.3.3. Water-cooled condensers/gas coolers

Interesting improvements in the overall energy efficiency can be achieved by adopting water-based techniques aimed at enhancing the gas cooler performance, such as:

- the adiabatic cooling of the air occurring upstream of the heat exchanger so as to raise the air relative humidity and thus the temperature difference between R744 and the air. This leads to extended periods of time over the year when the refrigerating system is capable of performing in subcritical running modes. In such a solution, mains water can be employed;
- spraying water directly on the gas cooler heat transfer area. As a consequence of the water evaporation, some heat is removed from the gas cooler walls and therefore from R744. This expedient relies on the usage of treated water to avoid deposits and corrosive effects.

A simplified schematic of evaporative condenser/gas cooler is represented in Fig. 12.

Fig. 12. Simplified schematic of a R744 booster supermarket refrigeration system equipped with parallel compression and an evaporative condenser/gas cooler (Visser, 2017).

Wiedenmann et al. (2014) pointed out that a high potential can be associated with the adiabatic cooling. The advantages related to the adoption of this technique were theoretically assessed by Bellstedt et al. (2010) in some Australian cities. The authors concluded that a dry-boosted transcritical R744 system is a suitable alternative for R404A units in such a climate context.

Lozza et al. (2007) estimated energy savings between 3.2% and 5% over a conventional air-cooled "CO_2 only" configuration by spraying water droplets during severe summer temperatures. The results, which were based on a location in the North of Italy, were found to be strongly depending on the water-spray activation temperature. Also, the lower this temperature, the higher the energy savings (and water consumption as well as the enhancement of the maintenance expenditures) were. Despite the limited energy benefits, the authors claimed that this solution is able to reduce the investment cost (i.e. undersized compressors, heat exchangers and electric equipment) thanks to the substantial lowering in the energy consumption (i.e. about 25%) at peak operating conditions.

Fornasieri et al. (2008) claimed that the evaporating cooling is an especially beneficial technology in dry climate contexts (e.g. Tripoli). On the other hand, such a solution can also be suitably employed in humid climates (e.g. Bangkok) particularly as the external temperature is above 35 °C. Thanks to the peculiar properties of R744, the researchers suggested applying this expedient to a limited fraction of the air stream flowing through the gas cooler. This entails insignificant penalizations in COP and enormous water savings. Furthermore, the field measurements gathered in a CO_2 supermarket system located in Adelaide (Australia) proved that the water consumption in a real application is more substantial than the predicted value at high outdoor temperatures. This was due to a short time of experimental data collection, climatic anomaly and water dissipation.

On the other hand, Girotto (2017) highlighted that water-based techniques are mainly suitable for industrial applications, as water is a limited resource. This is particularly true when it comes to warm countries (e.g. Spain, Portugal, Greece, Southern Italy and Balkan Peninsula) where its usage is often subject to stringent limitations, especially in summertime. Also, other issues, such as need for water treatment systems, corrosion, little flexibility, etc., can be related to these solutions. Similar assertions were also made by Visser (2014, 2015), although this author claimed that water-based expedients permit reducing the design pressures, energy consumption in various climate contexts as well as running and operating costs in relation to both synthetic and other natural refrigerants.

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The most southern R744 supermarket refrigeration solution in North America was installed in Dunwoody (USA) in July 2014. As claimed by Shecco (2015a), this solution is outfitted with an adiabatic gas cooler, leading to an energy conservation by 6.3% over a R407A-based technology.

4.3.4. Thermally-driven technologies

A commercial building features remarkable electricity, refrigeration and cooling/heating demands, which are generally satisfied by the electricity delivery from the national grid, vapour compression refrigeration systems and heating units, respectively. On the other hand, the overall efficiency of these processes is modest due to both the poor performance of the power plants and the distribution losses in the grid.

A significant energy saving is then offered by adopting a tri-generation (or CCHP or CHRP) system. Such a solution consists of a co-generation (or CHP) unit coupled with a thermally-driven refrigerating unit for cooling or even refrigeration. This technology permits also overcoming the disadvantage related to the enormous mismatch between the refrigeration and heating needs. A few installations involving supermarket applications are currently available in the USA, the UK and Japan (Tassou et al., 2011). According to Tassou et al. (2011), the main barriers which impede the adoption of this technology are the strong difficulty in correctly evaluating the energy conservation related to its adoption, as well as little experience and availability of field measurements from food retail sector. In addition, nowadays they are uniquely employed in installations having operating temperatures above 0 °C. The authors also claimed that its diffusion could be promoted by a substantial increase in the accessibility to biofuels. In addition, Tassou et al. (2010) highlighted the need to develop further investigations involving the enhancement of their efficiency and cost-effectiveness, the implementation of units for LT applications as well as of a suitable control logic to optimize the integration of these technologies with thermal and power systems. Suamir et al. (2012) showed that the integration of a CO_2 supermarket refrigeration system with a tri-generation plant in Manchester area (UK) allows reducing the energy consumption by 30% and the environmental impact by 43%, as well as it offers a payback time of 3.2 years in comparison with a R404A refrigerating unit. The experimental and numerical results related to the study by Suamir and Tassou (2013) recommended employing the cooling released by a tri-generation plant to cool down R744 flowing through an all-CO_2 cascade arrangement. This permits the system to operate in subcritical operating modes all year round, as well an energy saving by 30% and a reduction in carbon footprint above 40% in relation to a conventional solution in a food retail store located near Manchester (UK). The study by Ge et al. (2013) suggested that a microturbine tri-generation system can generate above 90% of the electricity required by an all-CO_2 cascade arrangement operating in a supermarket located in the UK.

Wiedenmann et al. (2014) provided an interesting overview about various promising technologies for R744 supermarket refrigerating applications. In particular, the author suggested that in summertime the large amount of heat rejected through the HP heat exchanger can be employed for running an adsorption chiller aimed at cooling down CO_2 coming out of the gas cooler. According to the results obtained for a refrigerating plant operating in Switzerland ($\dot{Q}_{nominal}$ = 86 kW at $t_{evaporation}$ = -9 °C), the adsorption chiller should be small in order to be able to work for long periods over the year. Furthermore, the high investment costs and the resulting high payback periods do not promote the diffusion of such a solution.

4.3.5. Expanders

An expander is another work producing device which had gained a modest interest in the last few years as regards food retail industry. The main benefits related to such a technology are the increase in COP and the reduction in the heat rejection pressure in transcritical running modes (Nickl et al., 2005).

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Nickl et al. (2005) developed a three-stage work extracting expander for a transcritical CO_2 refrigerating system. To guarantee an optimal integration into this system, a liquid/vapour separator was adopted. The results revealed that the expander featured an isentropic between 0.65 and 0.70, which led to enhancements in COP above 40% compared to a conventional system. At a later time, this solution was installed in a Swiss supermarket and investigated by Riha et al. (2006a). In particular, two expander stages were used for substituting the conventional HP expansion valve, whereas the remaining stage was employed as the alternative to the conventional vapour by-pass valve. In subcritical running modes, the compressor of the expander performed similarly to a parallel compressor. The authors estimated that, compared to both a R404A unit and a conventional CO_2 system, energy conservations up to 25% in cold regions and up to 30% in warm countries can be achieved by opting for such a technology.

Riha et al. (2006b) proposed incorporating an expander into a CO_2 refrigeration unit equipped with double-stage compression (i.e. main compressor and auxiliary compressor of the expander/compressor), a recooler and an intercooler to enhance the performance of a basic system in moderate climate contexts. The expander/compressor was estimated to be around 33% less expensive than the main compressor.

Yang et al. (2007) theoretically proved that the optimal high pressure is mainly affected by the gas cooler exit and evaporating temperatures, whereas the compressor and expander efficiencies have a scarce influence.

Cho et al. (2009) estimated an improvement in COP by 28.3% over the basic CO_2 unit by employing a work-producing expander with an efficiency of 0.3. With respect to the same baseline, the researchers also showed that a two-stage solution with intercooler and a double-stage with vapour injection can respectively achieve 13.1% and 18.3% greater COPs.

The theoretical study carried out by Pérez-Garcia et al. (2013) indicated that a one-stage CO_2 unit with an expander can outperform various basic configurations with and without IHX. In addition, the authors suggested the adoption of such a heat exchanger on the part of the former only at gas cooler temperature below 40 °C.

The field tests implemented by Wenzel and Ullrich (2012) highlighted an increase in the COP by 9.9%, as well as a drop in the discharge temperature and heat rejection pressure respectively by 6 K and 4.7 bar. The same refrigeration plant was previously exhaustively studied by Gerber and Wenzel (2011).

Wenzel and Ullrich (2012) also indicated that the test and demonstration refrigerating unit (i.e. a CO_2 booster unit) connected to a Swiss demonstration supermarket features an enhancement in the COP by 9.3%, and a decrease in the discharge temperature and high pressure respectively by 14 K and 6 bar.

Dieckmann (2013) designed, manufactured and tested a scroll CO_2 expander for commercial refrigeration applications. The investigation demonstrated that isentropic efficiencies between 0.60 and 0.65 can be reached. The internal leakages between the scrolls were found to be responsible for the detected inefficiency, as well as suboptimal thrust bearing performance caused incrementally lower efficiencies. On the other hand, the mechanical design was found to be robust.

A commercial R22 direct expansion unit leads to substantial energy conservations compared to a CO_2 refrigerating system using an expander at outdoor temperatures between 34 °C and 50 °C, as theoretically pointed out by Purohit et al. (2016).

The results by Shet et al. (2016) revealed that at the evaporating temperature of roughly -10 °C and at the gas cooler exit temperature of about 35 °C, a single-stage CO_2 solution with an expansion turbine increases its COP and its 2nd law efficiency respectively by 25.72% and 30.19% over a basic solution.

Pérez-Garcia et al. (2016) theoretically showed that, at the evaporating temperature of -10 °C and at the gas cooler outlet temperature of 35 °C, the usage of an expander raises the COP by 27.2% over a conventional CO_2 system.

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The theoretical investigation by Joneydi Shariatzadeh et al. (2016) suggested that a transcritical CO_2 refrigeration system with expander and without IHX is a suitable alternative for the basic CO_2 solution from both the energy and exergy viewpoint.

Singh et al. (2016) theoretically proved that a "CO_2 only" system with expander has comparable performance with that of a transcritical CO_2 solution with flash gas inter-cooler in typical Indian climate conditions.

Purohit et al. (2017b) suggested the use of a work extracting expander for supermarkets located in warm countries. In comparison with a conventional booster unit, the authors theoretically estimated that this technology leads to energy savings between 53.7 $MWh \cdot y^{-1}$ (in Seville) and 118.1 $MWh \cdot y^{-1}$ (in New Delhi, India), as well as to an additional investment recovery time between 3 years (in Phoenix, USA) and 4.7 years (in Teheran, Iran).

The theoretical evaluation by Hafner et al. (2014b) showed that the energy benefits related to the usage of an expander in commercial CO_2 refrigeration solutions is substantially depending on its efficiency. For small applications this technology shows promising results, although it cannot outperform the ejector-based system. On the other hand, dramatically poor performance can be ascribable to this solution in large food retails. In addition to this, Nekså et al. (2010) suggested that even parallel compression is a more energy beneficial expedient than an expander for MT commercial refrigeration applications at outdoor temperature between 30 °C and 40 °C.

It could be concluded that the results by Riha et al. (2006a) were enormously overestimated. Also, expanders are economically unattractive for supermarket applications and still far from being competitive with the ejectors. In addition to this, they can get easily damaged by multi-phase flows. The field data collected from such a solution installed in Switzerland in 2009 ($\dot{Q}_{nominal}$ = 85 kW at $t_{evaporation}$ = -9 °C) brought Wiedenmann et al. (2014) to conclude that the reaching of the optimal heat rejection pressure was negatively influenced by the recurrent on/off of the compressors due to the variations of the cooling need. Therefore, the researchers claimed that, to this point in time, the need to overcome such technological barriers, the low safety in operating conditions and the high investment costs have confined the usage of such a technology in a very thin market niche.

4.3.6. Water loop heat pump unit

Polzot et al. (2017) recommended the adoption of a water loop heat pump unit as a means of reducing the energy consumption related to the commercial "CO_2 only" refrigerating systems. The authors theoretically proved that a R744 supermarket refrigeration plant coupled with such a technology is a suitable solution for cold climates, providing an energy saving by 9.4% over a HFC system. To the best of the authors' knowledge, this investigation is the only currently available study combining transcritical CO_2 refrigerating technologies for food retail applications and water loop heat pump units.

5. Heat rejection pressure control strategy, gas cooler performance and heat recovery implementation

5.1. Optimal heat rejection pressure

It is a well-known fact that an optimal gas cooler pressure has to be identified as a function of the gas cooler outlet temperature as soon as the transcritical operations take place in a pure CO_2 refrigerating unit (Cavallini and Zilio, 2007; Cecchinato et al., 2009; Kim et al., 2004; Lorentzen, 1994).

Casson et al. (2003) suggested the usage of a two-stage throttling based on an appropriate fixed value of the differential pressure as a means of concurrently (and suitably) controlling the evaporator and gas cooler performance in transcritical running modes. This approach was implemented in a supermarket for the first time ever by Girotto et al. (2004).

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In order to evaluate the optimum heat rejection pressure, many correlations have been proposed depending on the selected application (Chen and Gu, 2005; Brown et al., 2002; Liao et al., 2000; Kauf, 1999; Sawalha, 2008a; Ge and Tassou, 2011a). As for commercial refrigeration systems, Sawalha (2008a) suggested one as a function of both the outdoor temperature and $\Delta T_{approach}$ of the gas cooler. A more accurate correlation was introduced by Ge and Tassou (2011a) by relating the optimum high pressure to the isentropic efficiency of the HS compressor rack, the external temperature and the effectiveness of IHX located downstream of the gas cooler. According to Yang et al. (2015), the application of the typical discharge pressure optimization procedure causes deviations from COP_{max} up to 71%. Therefore, the authors proposed a new method to substantially reduce these losses down to 0.7%. Cabello et al. (2008) compared the experimental measurements of the optimum high pressure related to a single-stage CO_2 system to the outcomes obtaining by applying four correlations from the literature. The authors concluded that significant reduction in COP can be associated with small errors in optimal pressure value. Some researchers proposed the application of real-time control strategies of the optimal discharge pressure (Peñarrocha et al., 2014; Zhang and Zhang, 2011; Cecchinato et al., 2010a, 2012a). This approach is a more efficient and robust solution than the direct use of the correlations from the published literature, as pointed out by Cecchinato et al. (2010a). However, the authors also showed that the application of the correlations from the open literature does not deteriorate the results remarkably with respect to supermarket applications.

Cecchinato et al. (2010b) suggested a novel architecture for a transcritical CO_2 air-cooled chiller relying on an innovative control strategy so as to properly regulate the optimum high pressure. The unit assures dependable operations and a simple management, besides cooling propylene glycol down to -8 °C for commercial refrigeration installations. COP values ranging from 3.1 to 2 were experimentally measured for external temperatures between 18 °C and 35 °C.

Finally, it is worth remarking that the adoption of some of the state-of-the-art R744 technologies lead to significant reductions in the optimum high pressure. For instance, in comparison with a conventional booster unit, Gullo et al. (2016a) computed average decreases related to the usage of a dedicated mechanical subcooling between 5.3% and 6.3% over the investigated range of outdoor temperatures. At the same boundary conditions, the researchers also evaluated that parallel compression brings the optimal heat rejection pressure on average down to 4.2%. Also, the experimental evaluation by Sanchez et al. (2016) proved that the use of such a technology allows reducing the heat rejection pressure especially with rise in cooling medium temperature, enhancing the stability of the refrigeration unit as well as minimizing the sharp reduction in COP as the system works at discharge pressures below the optimum value.

5.2. Gas cooler performance

As previously explained, the performance of a transcritical CO_2 refrigeration system is strongly depending on the heat rejection pressure and, in turn, on both the implemented control technique and the gas cooler/condenser behaviour. This implies that the maximization in the system energy efficiency can be achieved only by well designing, i.e. appropriate evaluation of number of circuits and their arrangement. As a consequence, the development of accurate simulation models plays a crucial role in the appropriate design of gas coolers and thus in the enhancement in the overall energy efficiency.

A model for a R744 cross-flow gas cooler based on a finite element method and employing published correlations was implemented by Yin et al. (2001) to assess the CO_2- and air-side heat transfer and pressure drop. After its experimental validation, it was found that the model was capable of predicting the gas cooler capacity within ±2% as well as the pressure drop associated with CO_2 well within the range of experimental error.

Ge and Cropper (2009) developed a mathematical model of finned-tube CO_2 gas coolers, which was validated against the test results available in the open literature. This was used for investigating different pipe circuit arrangements so as to increase the heat capacity or minimize $\Delta T_{approach}$ of the

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gas coolers. It could be concluded that both $\Delta T_{approach}$ (drops up to 12.1 K) and the heat capacity (i.e. growths up to 51.5%) can be enhanced by raising the number of HP heat exchanger circuits.

The water-cooled gas cooler model built by Sanchez et al. (2012), which was developed by relying on the finite-volume methodology and experimentally validated, involved a coaxial heat exchanger operating under steady state conditions. The results obtained demonstrated that the heat transfer rates can be computed with an uncertainty lower than ±10% as well as the outlet temperatures of both fluids can be calculated with an uncertainty below ±3 °C. At last, the authors highlighted that the application of the ε-NTU methodology overestimates the thermal effectiveness of the gas cooler with a value of 100% in 79.77% of the tests.

Ge et al. (2015) implemented an experimentally validated model of a CO_2 finned-tube gas cooler to predict the effect of various controls and sizes of such a component on the overall system performance.

Heerup and Bramsen (2016) mapped the performance of a gas cooler/condenser installed in a Danish supermarket by employing some field data gathered from the 15th of June to the 13th September 2013. In transcritical running modes, discrepancies up to three times were estimated between the mean values of $\Delta T_{approach}$ in the actual installation and the values predictable from the manufacturer data. Based on the collected measurements, a correlation was developed so as to predict the mean UA-values as a function of the cooling capacity, outdoor temperature and gas cooler exit temperature. Its accuracy was found to be high and significantly related to both the load cyclic variations and heat exchanger geometry.

Tsamos et al. (2017a) experimentally studied the performance of two CO_2 air cooled finned-tube gas coolers/condensers (i.e. 3-row and 2-row heat exchangers), as well as their influence on the overall system efficiency. The evaluation took into account various air temperature inlet, main fan speeds and air volumetric flow rates in both subcritical and transcritical running modes. The results pointed out that the larger heat exchanger presents higher COPs thanks to lower values of $\Delta T_{approach}$ and thus larger cooling capacities.

Santosa et al. (2017) looked into the local refrigerant and air heat transfer coefficients in plain fin-and-tube gas cooler coils by implementing a computational fluid dynamics (CFD) simulation model. This was experimentally validated with respect to heat rejection rate and air outlet temperature. The authors claimed that a horizontal slit on the fin between the first and second row of tubes allows enhancing the heat rejection rate of the gas cooler by between 6% and 8%. This means that smaller heat exchangers for a given heat rejection capacity can be employed or lower high pressures can be reached. Furthermore, some correlations aimed at evaluating the overall refrigerant and air heat transfer coefficients were also suggested.

Tsamos et al. (2017c) developed an experimentally validated simulation model of two different gas cooler designs. The model is capable of predicting the performance of the gas cooler (i.e. refrigerant temperature profiles, pressure drop across the gas cooler, air and refrigerant heat transfer coefficients and heat rejection) in transcritical operating conditions, as well as calculating COP of the investigated system.

Rossetti et al. (2017) implemented an innovative 3D CFD model to simulate fin and tube CO_2 gas cooler performance with reduced computational efforts compared to detailed models based on the description of each single fin. The researchers claimed a satisfactory accuracy can be obtained, meaning that the total heating capacity can be predicted with an accuracy within 2.5%, as well as the maximum deviation in the local temperature on the copper curves is 3.8 as CO_2 temperature is equal to 108 °C.

5.3. Heat recovery implementation

Heat recapture from "CO_2 only" refrigeration systems is one of the most appealing techniques as regards commercial sector. For this reason, this has become standard in transcritical R744

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refrigerating unit for food retail applications all over the world, as it offers a noteworthy chance to further reduce the consumption (Ge and Tassou, 2014; Sawalha, 2013) and their carbon footprint (Ge and Tassou, 2014), as well as leads to satisfactory payback times (Tambovtsev et al., 2011; Reinholdt and Madsen, 2010). In fact, it is a well-known fact that the gas cooling is a peculiar process taking place by sensible cooling. This implies that R744 can more suitably fit the water temperature profile than conventional refrigerants and thus more effectively produce hot and warm water at various temperatures. This expedient in transcritical R744 refrigeration systems is also fostered by the high discharge temperatures obtained on the part of the HS compressors. Also, it is worth remarking that remarkable consumptions (i.e. about 20% of the total energy needed) are ascribable to the heating reclaim in a supermarket (Hafner et al., 2014a), being necessary for various processes at different levels of temperature, such as DHW production, space heating, fresh air pre-heating or snow melting and hygienic cleaning purposes.

Ge and Tassou (2011b) developed a supermarket simulation model integrating the refrigeration system, the HVAC unit and the building envelope. After its validation against some experimental data, this was used for demonstrating that 40% of the space heating demand can be satisfied by recovering heat from a CO_2 booster refrigerating plant (Ge and Tassou, 2011c). In addition, Ge and Tassou (2011c) estimated similar performance between this technology and a conventional R404A system. The evaluation was based on a climate context in the North of England. Ge and Tassou (2014) employed the aforementioned model for investigating four different control strategies of the heat rejection pressure related to an all-CO_2 cascade unit operating in a Northern city in the UK. In comparison with the same refrigerating system utilizing a gas boiler for heating purposes, a suitable control logic leads to cost savings up to 5.6% besides providing up to about 47% of the heating demand.

The comparison carried out by Zsebik et al. (2014) recommended that, in comparison with a R717 refrigerating unit, the more utilized the recovered heat, the more energetic and economic competitive a CO_2 refrigeration plant is. Furthermore, an amount of heat ranging from 0.55 kW to 0.92 kW can be recaptured every kW of required cooling capacity in winter season.

Denecke and Hafner (2011) theoretically compared three different heat capture strategies relying on hot gas heat recovery in a CO_2 booster system located in Tromsø (Norway). The authors concluded that the combination of series-connected heat pumping units with the adoption of the optimum COP technique is the most energetically beneficial strategy. However, wide discrepancies between the theoretical results and the outcomes related to the field measurements were found.

As suggested by Sawalha (2013) and Tambovtsev et al. (2011), the heat recovery in transcritical R744 supermarket refrigerating systems can be appropriately performed with the aid of two additional heat exchangers (i.e. de-superheaters) located upstream of the conventional gas cooler/condenser (Fig. 13). The three heat exchangers, which are assembled in series, can be completely by-passed by using the corresponding 3-way valve arranged upstream of each heat exchanger. The first de-superheater is located downstream of the HS compressors so as to produce DHW, followed by a similar heat exchanger aimed at satisfying the space heating. The gas cooler/condenser is possibly employed for cooling down CO_2 leaving either the first or second de-superheater in order to enhance the overall energy efficiency.

Fig. 13. Transcritical R744 booster supermarket refrigeration system implementing heat recovery (Polzot et al., 2017).

Sawalha (2013) highlighted that the higher the gas cooler pressure, the larger the amount of recoverable heat is. However, this also leads to both a growth in the total energy consumption and a simultaneous drop in the refrigerating effect. Also, the author pointed out the need to strike a balance

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between the amount of heat to be recaptured and the overall system energy efficiency as the gas cooler/condenser is put into operation. In fact, on the one hand, the usage of such a heat exchanger allows the refrigerating unit to substantially enhance its performance. On the other hand, it also reduces the recoverable heating capacity, since the refrigerant mass flow rate decreases. As already thoroughly proved by many researchers (Ge and Tassou, 2014; Sawalha, 2013; Tambovtsev et al., 2011), the amount of recaptured heat and global energy efficiency can be concurrently maximized for commercial R744 refrigeration systems by implementing a suitable control strategy. Sawalha (2013) extensively investigated the controlling trend to appropriately perform heat recovery to provide an average-size Swedish food retail store with the whole heating need. This was based on:

- first of all, the gas cooler/condenser had to be run at fully capacity to reach as large degree of subcooling as possible;
- secondly, the high pressure is fixed to a maxim optimal value and the gas cooler/condenser capacity is dropped by firstly slowing fan speed down, later switching fans off (i.e. heat removed from the gas cooler by natural convection) and finally by-passing the gas cooler/condenser by using its 3-way valve.

Karampour and Sawalha (2017) found that this strategy is consistent to real applications.

Reinholdt and Madsen (2010) studied two different operating strategies to implement heat recovery for various purposes. The first control logic, which relied on the maximization of $COP_{cooling}$, permitted satisfying from 18% to 30% at low outdoor temperatures and 100% of the heating demand at high external temperatures. On the contrary, the second control strategy, which was aimed at maximizing the amount of recovered heat, allowed recovering at least 35% of the total heating reclaim all over the year.

Tambovtsev et al. (2011) showed the first field measurements related to a "CO_2 only" refrigerating solution implementing heat recovery in a supermarket, besides exhaustively discussing its control aspects. The results revealed that total COPs between 5 and 9 can be achieved at outdoor temperatures ranging from -15 °C and 5 °C. At the same boundary conditions, a heat pumping unit would have let to COPs between 1.5 and 5.

Hafner et al. (2012) theoretically estimated that a CO_2 refrigerating plant can satisfy 68% of the heating demand related to a supermarket running in Northern Europe. The remaining load can be covered by means of either a heat pump system with a variable speed compressor or, in order to reduce the investment cost, an electrical auxiliary heat.

The study by Sawalha (2013) also suggested that a CO_2 booster solution with no flash gas removal and performing heat recapture consumes 4% less electricity than separated HFC-based systems in an average-size food retail store located in Stockholm.

Ge and Tassou (2014) also asserted that the optimal amount of heat reclaim to be recovered is substantially depending on the relative cost of both the gas and the electricity.

Abdi et al. (2014) compared the control logic implemented by Sawalha (2013) with that of a real supermarket operating in the North of Sweden. The assessment showed that an amount of heat of 130% of the required cooling capacity can be theoretically recovered from the investigated CO_2 booster system. On the other hand, the implemented experimental campaign proved that only between 30% and 60% of the total recoverably heat is actually employed. Also, the authors claimed that such a technology is rather competitive with a geothermal heat pump.

Karampour and Sawalha (2014a) developed a simulation model of a CO_2 booster system with flash gas by-pass valve installed in a Swedish supermarket and based on both some field measurements and the control technique suggested by Sawalha (2013). The results indicated that such a configuration can achieve a seasonal performance factor of 4 in heating mode. This leads to the conclusion that in cold regions this technology can satisfactorily cover the whole refrigeration and heating demands by applying a suitable control strategy.

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The usage of IHXs in different positions on the part of a CO_2 booster system with flash gas by-pass does not involve large enhancements in the $COP_{cooling}$ in comparison with the same configuration without IHXs, as theoretically studied by Karampour and Sawalha (2014b). On the contrary, this expedient is particularly beneficial in subcritical running modes as heat recapture is performed, leading to increase in COP_{total} up to 11% over the investigated operating conditions.

Sharma et al. (2014b) theoretically delved into the potential related to the heat recovery from a CO_2 booster solution in favour of a desiccant regeneration, water heater and space heating in 16 American cities. It was found that the warm the wheatear, the more water and space heating demand can be covered. Also, the refrigeration plant was capable of satisfying on average 91% of water heating and 37% of space heating needs with a humidity of 55% within the selected food retail store, whereas these values amounted respectively to 85% of water heating and 29% of space heating reclaims with a humidity of 35%. Finally, the authors stated that a transcritical CO_2 unit and a R404A direct expansion system can meet on average 65% and 54% of the total heating demand, respectively.

Fidorra et al. (2015a) suggested an innovative approach aimed at successfully comparing the total energy costs related to different heat recovery strategies as the heating demand and the energy price change.

Colombo et al. (2010, 2014, 2015, 2016) developed a simulation model of a conventional CO_2 booster system which was validated against some experimental results. In this system, the heat rejected through the HP heat exchangers was employed for feeding the hot water services, the floor heating unit and either an absorption chiller aimed at providing the AC need or a district heating unit. In particular, Colombo et al. (2016) recently showed that total energy consumption of a supermarket located in Wales (UK) can be reduced by 3% by employing this solution in place of R404A systems.

Shi et al. (2017) implemented an experimentally validated simulation model of a R744 booster solution operating in a Dutch food retail store. The outcomes recommended that such a refrigerating unit performing heat recovery allows bringing the primary energy consumption down to 13% compared to the same refrigerating system having a floating condensation temperature and relying on a boiler for the heating demand.

Karampour and Sawalha (2017) made some simulation models adapted from field measurements based on a conventional booster system (with flash gas by-pass) implementing heat recovery and employing a R410A unit for AC purposes. The authors estimated an increase in the annual energy consumption respectively by 15% in Paris (France) and by 20% in Barcelona compared to separated HFC-based units.

The need for parallel compression on the part of a transcritical CO_2 booster system performing heat recapture and running in the Italian climatic context was demonstrated by Polzot et al. (2016c). Additionally, the researchers pointed out that the usage of a further evaporator as a heat source (i.e. load evaporator) (Fig. 13) permits the investigated solution to provide the entire space and tap water heating reclaim. It worth remarking that a load evaporator is an additional heat exchanger employed in heating mode and aimed at covering the whole heat required by the refrigeration system as the refrigeration load is not sufficient. According to Sheehan et al. (2016), this purpose can be achieved by utilizing either an air-cooled evaporator located outdoors or a self-contained solution. The latter (Fig. 14a) is a more cost-effective solution having no frost formation issues, whereas the former is noisy and voluminous and it could require a special coil arrangement and frequent defrost cycles. As an alternative, geothermal storages can be adopted for such a purpose (Hafner et al., 2014e), as well as to perform some subcooling and provide free cooling in summertime (Fig. 14b).

Fig. 14. (a) Self-contained unit (Sheehan et al., 2016) and (b) geothermal storages (Hafner et al., 2014e) to promote heat recapture on the part of R744 booster supermarket refrigeration systems at low refrigeration loads.

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Sheehan et al. (2016) gathered some field measurements from an English convenience store with a sale are of 250 m^2 ($\dot{Q}_{MT}$ = 26 kW at t_{MT} = -10 °C, $\dot{Q}_{LT}$ = 7 kW at t_{LT} = -35 °C). These were integrated with some data obtained in a laboratory. The investigated CO_2 refrigeration system involved 3 HP heat exchangers for the purpose of: DHW production, space heating and an additional heat source to provide a suitable amount of heating in case of low cooling load. The outcomes suggested that the proposed heat source increases the recoverable heating capacity from 8% at 75 bar to 18% at 90 bar. The investigated system featured overall values of COP respectively equal to 5.45 and 4.79 at night-time and day-time operations, whereas its standard COP added up to 4.62.

Karampour and Sawalha (2018) estimated a seasonal performance factor of 3.9 for a conventional CO_2 supermarket booster solution in heating mode and of 3.6 for a R407C air source heat pumping unit, respectively. Also, the authors claimed that the total energy cost associated with the heat recovery implementation is approximately 50% lower than that for district heating and 20% lower than that related to a R407C air source heat pump unit, respectively.

The multi-ejector concept was adopted in a hypermarket located in Timisoara (Romania) in 2015. A decrease in the electricity consumption up to 13% over a configuration with parallel compression was predicted thanks to the use of this technology along with both the DHW production and the heat recovery implementation (Frigo-Consulting LTD, 2015).

A R744 multi-ejector enhanced parallel compression system applying heat recovery was adopted in a football stadium (capacity for 68000 people) in Madrid (Spain). This was designed for an outdoor temperature of 38 °C as well as for cooling capacities of 95 kW at the evaporating temperature of -10 °C and 45 kW at -28 °C/-30 °C. It was theoretically predicted that such a solution will lead to an energy conservation of 15% compared to a conventional R404A system (r744.com, 2018b).

6. "All-in-one" concept

Most of the electricity required in a supermarket (i.e. about 33%) is employed for running the food preservation system (Cecchinato et al., 2010c). Additional significant contributions, i.e. approximately 20% of the total energy intake, are due to heating and AC reclaims (Cecchinato et al., 2010c). Also, although all these demands can be simultaneously request, their behaviour with respect the time is significantly different. In fact, the space heating/AC and refrigeration needs are dramatically time varying over the year (Karampour and Sawalha, 2017), whereas the tap water heating load is generally constant all year round (Karampour and Sawalha, 2017) and negligible compared to the other reclaims (less than 10%÷15% of the total heating need according to Karampour and Sawalha, 2017). As a means of copiously dropping the total investment, maintenance and running costs (Hafner et al., 2015, 2014e), the "all-in-one" (or "fully integrated") technologies have become the current tendency as regards transcritical CO_2 supermarket refrigeration solutions (Karampour and Sawalha, 2015). The energy and economic benefits related to these highly integrated units can only be attained by employing R744 as the only refrigerant as a result of its unique properties. Karampour and Sawalha (2017) also included other additional advantages concerning such technologies. Firstly, the reduction in the intricacy in terms of communications between various operation and maintenance entities in charge for putting all the different units into operation. Furthermore, such solutions are very compact and imply a few extra components compared to a conventional "CO_2 only" system. On the other hand, many units are required as well as extra space is necessary when it comes to separated HFC-based systems. "All-in-one" configurations are tailor-made systems which successfully satisfy the entire refrigeration, AC and DHW demands, alongside matching most of or even the whole heating reclaim of the selected food retail store (Hafner et al., 2016; Hafner and Banasiak, 2016; Hafner et al., 2015; Karampour and Sawalha, 2015, 2017). However, it is worth remarking that R410A ($GWP_{100\ years}$ = 1924 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$ according to AR5) is today's most widely used working fluid in chillers and air conditioning units, including supermarket applications. On the other hand, due to the aforementioned reasons, this refrigerant is bound to be replaced with a climate-

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friendly alternative, despite the current shortage of its appropriate replacement. Consequently, the interest in "fully integrated" R744 configurations has been greatly intensifying. Approximately 10÷15 "all-in-one" commercial "CO_2 only" units could be counted in Sweden in August 2016 (Karampour and Sawalha, 2017) as well as many others located in various other countries (Danfoss, 2016; Hafner et al., 2016).

The theoretical study conducted by Cecchinato et al. (2012b) demonstrated that a transcritical CO_2 unit integrated with a R717 chiller and a boiler performs similarly to a separated solution, i.e. a R410A heat pumping system employed for supplying cold/hot water and a R404A system for providing the refrigeration load. The study was carried out by selecting a medium-size supermarket and three different cities (i.e. Treviso, Stockholm, Singapore).

Karampour and Sawalha (2015) suggested adopting the configuration schematized in Fig. 15 for temperate/cold climates. The authors also claimed that the parallel compressor should put into operation in summertime with and without AC reclaim, whereas this system should operate in the same way as a conventional booster in wintertime. At outdoor temperatures above 25 °C (i.e. warm places), this solution should include a separated HFC or different AC unit. At a later time, these results were experimentally confirmed by means of some field measurements collected in a small-medium size supermarket located in the middle of Sweden in January and in July 2014 (Karampour and Sawalha, 2017). The system was found to be capable of properly satisfying the whole AC need and a large part of the heat reclaim for space heating and DHW. Also, COPs in heat recovery modes similar to or better than those of conventional heat pump units were estimated.

Fig. 15. "All-in-one" transcritical R744 booster supermarket refrigeration system equipped with parallel compression, corresponding P-h diagram and implemented heat recovery control strategy (Karampour and Sawalha, 2015).

The evaluation carried out by Karampour and Sawalha (2017) leads to the conclusion that an "all-in-one" R744 refrigerating solution with parallel compression is an appropriate technology only for cold climates. In fact, as showed in Fig. 16 such equipment (S1) offers an energy saving of 11% in Stockholm, whereas it consumes respectively 9% in Paris and 12% in Barcelona more electricity compared to separated HFC-based units (S3). Also, Karampour and Sawalha (2017) assessed energy savings on the part of this HFC-free unit of 19% in Stockholm, 17% in Paris and 1% in Barcelona over R410A-based units in AC mode. Furthermore, this technology consumes 5% less electricity in Stockholm, whereas it features 14% in Paris and 11% in Barcelona higher energy consumptions than HFC heat pumping units. At a later time, the authors estimated that the adoption of parallel compression allows "CO_2 only" supermarket refrigeration plants to achieve comparable values of SEER to that of a R410A chiller in Stockholm (about 4.5) and in Barcelona (about 4).

Fig. 16. Annual electricity use comparison of an "all-in-one" transcritical R744 booster supermarket refrigeration system in Barcelona, Paris and Stockholm ($\dot{Q}_{MT}$ = 200 kW at t_{MT} = -8 °C, $\dot{Q}_{LT}$ = 35 kW at t_{LT} = -32 °C, $\dot{Q}_{AC}$ = 35 kW at $t_{outdoor}$ = 32 °C, $\dot{Q}_{heating}$ = 190 kW at $t_{outdoor}$ = -20 °C) (Karampour and Sawalha, 2017).

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The design evaluation implemented by Visser (2017) suggested that an "all-in-one" CO_2 booster unit outfitted with both parallel compression and an evaporative condenser represents a suitable alternative to a CO_2/NH_3 cascade arrangement in Kuala Lumpur (Malaysia). The sketch of such a technology is depicted in Fig. 12.

The "fully integrated" R744 refrigeration system with parallel compression and subcooling via a ground thermal storage located in Evanston (USA) enables achieving an energy saving by 60% over traditional systems (Cyclone Energy Group, 2013). This also features several other eco-friendly technologies.

In march 2016, an "all-in-one" R744 refrigerating plant using parallel compression was put into operation in Conegliano (Italy). The unit is able to provide a cooling capacity of 45 kW at -35 °C, an AC load of 71 kW and the whole heat reclaim with the aid of a load evaporator for a 600 m^2 food retail store. The system showed energy saving by 10% (r744.com, 2016).

As mentioned above, in some running modes the required heating demand can be only satisfied by running the refrigeration system in transcritical conditions (Sawalha, 2013). This turns further in favour of the ejectors-based configurations thanks to its large potential of energy saving (Schönenberger et al., 2014; Hafner et al., 2014a; Hafner et al., 2014c). As theoretically showed by Gullo et al. (2018), in fact, such technologies reduce the required total power input by from 36.8% to 50.3% over separated HFC-based units at outdoor temperatures ranging from -10 °C up to 5 °C.

Hafner et al. (2016) collected some field data from the integrated ejector supported parallel compression (IESPC) unit installed in Spiazzo (Italy) (Fig. 17) between the 1^{st} of May and the 30^{th} of October 2015 at outdoor temperatures ranging from 22 °C to 35 °C. Values of pressure lift from 5 bar to 10 bar were evaluated. The results showed that, besides providing the whole AC demand, such a configuration consumes from 15% to 30% less energy than the system with parallel compression depending on the AC needs and external temperature.

Fig. 17. "All-in-one" transcritical R744 booster supermarket refrigeration system equipped with multi-ejector rack (IESPC unit) (Hafner et al., 2016).

A "fully integrated" CO_2 system equipped with multi-ejector module was installed in Italy's largest hypermarket (10000 m^2) in Milan (Danfoss, 2016). The use of this technology along with an integrated control of the HVAC unit, light and refrigeration load should bring the energy consumption down to around 50% in comparison with more conventional installations.

An "all-in-one" multi-ejector based solution ($\dot{Q}_{MT}$ = 360 kW at t_{MT} = -2 °C, $\dot{Q}_{LT}$ = 72 kW at t_{LT} = -26 °C) was implemented in a supermarket located close to Turin (Italy) (r744.com, 2018d).

As mentioned above, the usage of the auxiliary compressors, especially for the refrigerating systems integrated with the AC unit, is substantially extended as a multi-ejector rack is adopted (Hafner et al., 2016; Fredslund et al., 2016). Therefore, Gullo et al. (2017a) suggested the manufacturability of 10% more efficient parallel compressors. The researchers, in fact, estimated increases in energy saving between 2.4% and 5.2% compared to a conventional R744 ejector supported parallel system, depending on both the selected climate context and the AC unit and store size. Also, according to Gullo et al. (2017a), this expedient is particularly (energetically and economically) reasonable for large supermarkets, remarkable AC demands and installations in warm/hot regions.

Fredslund et al. (2016) experimentally showed that energy savings from 10% to 15% for a configuration integrated with the AC system at about 30 °C can be attained. Despite some issues caused by the oil return, a reduction in the power input by around 4% for a solution with no integration

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with the AC unit ($\dot{Q}_{MT}$ = 120 kW, no overfed evaporators) at roughly 27 °C was estimated. Promising results are also expected in heat recovery mode.

Hafner et al. (2015) defined as "next generation of CO_2 supermarket refrigeration systems" an all-in-one unit outfitted with two multi-ejector blocks (i.e. one dedicated to AC need and the other to refrigeration load) being able to reach a pressure lift of 15 bar. Therefore, the performance of this configuration was studied for different AC loads. A full size laboratory supermarket refrigeration system resembling this technology was extensively described by Hafner and Banasiak (2016).

Gullo et al. (2018) theoretically evaluated that R744 multi-ejector enhanced parallel compression systems required from 3.1% up to 18% less power input in AC mode than separated HFC-based units at outdoor temperatures between 25 °C and 40 °C, as depicted in Fig. 18.

Fig. 18. Total power input of the investigated systems in AC mode at different outdoor temperatures (Gullo et al., 2018).

Kvalsvik et al. (2017) theoretically predicted that a CO_2 refrigerating plant involving two multi-ejectors modules and coupled with the AC unit ($\dot{Q}_{MT}$ = 10 kW at t_{MT} = -3.2 °C, $\dot{Q}_{LT}$ = 3 kW at t_{LT} = -28.6 °C, $\dot{Q}_{AC}$ = 20 kW at t_{AC} = +12.6 °C) consumes 53% more electricity than separated R410A-based solutions at the design outdoor temperature of 45 °C (i.e. Indian climate context). Also, the researchers highlighted that an energy performance enhancement by 30% experienced by the "CO_2 only" technology would lead to 7% greater power input compared to the aforementioned solutions. On the other hand, no overfed evaporators were considered in this assessment, as well as the analysis should have been based on annual energy consumption basis.

The theoretical assessment by Pardiñas et al. (2017a) related to a typical Norwegian supermarket revealed that the adoption of parallel compression is beneficial at outdoor temperatures above 10 °C, leading to energy savings from 7% to 19% in relation to a conventional booster configuration (between 15 °C and 30 °C). Also, it was found that the usage of a multi-ejector module is advantageous at external temperatures higher than 25 °C. Further energy benefits can be accomplished with the aid of an AC multi-ejector rack (8.3% at 30 °C and 8.6% at 25 °C). The analysis took into account the integration with an AC unit having the design load of 35 kW at the external temperatures of 30 °C.

Pardiñas et al. (2017b) compared two different IESPC configurations (i.e. with and without AC multi-ejector module) for a typical Norwegian medium-sized food retail store ($\dot{Q}_{MT}$ = 60 kW, $\dot{Q}_{LT}$ = 10 kW, $\dot{Q}_{AC}$ = 45 kW, design outdoor temperature of 30 °C) in steady state operations. The results showed that these two solutions perform similarly in subcritical running modes. On the other hand, the heat rejection pressure was found to be strongly related to the external temperature, MT, selected system and AC evaporating pressure. The researchers also highlighted that the performance improvement associated with the AC multi-ejector block has to compensate the additional cost as well as the complicated system implementing the heat rejection pressure control. Also, the solution without AC multi-ejector module and relying on the compressor interchangeability strategy was pointed out as a more reasonable technology as AC evaporation pressures could be kept at about 35 bar. Finally, the authors recommended the compressor manufacturers to enlarge the operational envelope of their equipment to further enhance the overall system performance.

As recently highlighted by Girotto (2016) and Hafner (2107), the usage of water as an energy carrier between a refrigeration system and AC/heating equipment implies many drawbacks, such as the involvement of many heat exchangers with the resulting increase in energy efficiency deterioration, corrosiveness and high investment costs. To overcome these drawbacks, the researchers suggested taking advantage of the favourable properties of R744 by employing this refrigerant in direct cooling

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and heating fan coils and air curtains installed inside the building (Fig. 19). This would lead to a reduction in the cost for the heating and cooling unit, a fall in the energy need as well as in the time required by the HVAC unit to be installed (Girotto, 2016; Hafner, 2017).

Fig. 19. Transcritical R744 supermarket booster refrigeration system equipped with parallel compression relying on space heating and cooling with water as the energy carrier (a), direct space heating with R744 (b) and direct space cooling with R744 (c) (Girotto, 2016).

7. Summary of findings

7.1. Practical aspects and energy evaluations

The findings associated with the 2nd and the 3rd generation of transcritical R744 supermarket refrigeration systems are summarized in Table 1 and Table 2, respectively. Also, the main results of today's accessible investigations on the "all-in-one" concept are presented in Table 3a (transcritical R744 supermarket booster refrigeration systems as baselines) and Table 3b (supermarket refrigeration systems using synthetic working fluids as baselines). Finally, the most relevant outcomes related to currently available energy assessments for the state-of-the-art R744 supermarket technologies are summed up in Table 4a (transcritical R744 supermarket booster refrigeration systems as the baselines) and Table 4b (supermarket refrigeration systems using synthetic working fluids as baselines). In particular, the values having a positive sign refer to the fact that the investigated technology outperforms the suggested baseline for all these Tables. On the contrary, negative values point out the opposite result.

Table 1. Summary of the findings associated with the main investigations on parallel compression.

Table 2. Summary of the findings associated with the main investigations on the multi-ejector concept.

Table 3a. Summary of the energy benefit assessments associated with the main investigations on the "all-in-one" concept (transcritical R744 supermarket refrigeration systems as baselines).

Table 3b. Summary of the energy benefit assessments associated with the main investigations on the "all-in-one" concept (supermarket refrigeration systems using synthetic working fluids as baselines).

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Table 4a. Summary of the energy benefit assessments associated with the main investigations on the state-of-the-art R744 technologies for supermarket applications (transcritical R744 supermarket refrigeration systems as baselines).

Table 4b. Summary of the energy benefit assessments associated with the main investigations on the state-of-the-art R744 technologies for supermarket applications (supermarket refrigeration systems using synthetic working fluids as baselines).

The findings related to Table 1, Table 2, Table 3a, Table 3b, Table 4a and Table 4b brought to light that:

- the technology involving the dedicated mechanical subcooling reveals that energy savings by between 3% and 23% can be attained over a R404A unit, depending on the selected weather conditions;
- the energy benefits associated with parallel compression do not justify the adoption of R744 in mild/warm locations, especially when it comes to "all-in-one" configurations;
- as highlighted in Fig. 20, R744 multi-ejector enhanced parallel compression systems can outperform both R404A direct expansion solutions and R1234ze(E)/R744 indirect units even in warm locations (Gullo and Hafner, 2017a). As examples, compared to such HFO-based equipment the aforementioned HFC-free technologies lead to energy savings at least up to 20.9%, 16.7% and 9.3% in Chicago (USA), Atlanta (USA) and Miami (USA), respectively. It is worth emphasizing that no heat recovery was performed in this study;
- the usage of the multi-ejector rack permits attaining between 20% and 30% higher $COPs_{heating}$ than a conventional booster system;
- a R744 multi-ejector enhanced parallel compression system integrated with the AC unit features energy savings between 15.6% and 26.2% in relation to separated HFC-based units in Mediterranean Europe, depending on the size of both the AC equipment and the selected food retail store;
- the filed data related to the supermarket located in Dunwoody (USA) suggest that an adiabatic condenser is capable of modestly decreasing the annual electricity consumption by 6.3% over a R407A system;
- promising results can be ascribable to transcritical CO_2 supermarket refrigeration systems with geothermal loop;
- a dramatic shortage of investigations on the "all-in-one" R744 multi-ejector enhanced parallel compression technologies, with particular reference to experimental studies.

Fig. 20. Comparison of annual energy consumption related to R744 multi-ejector enhanced parallel compression systems and R1234ze(E)/R744 indirect arrangements in the American climate context (Gullo and Hafner, 2017a).

By considering the previously summarized results, it is possible to claim that at the present time the so-called "CO_2 equator" in Europe has disappeared, meaning that transcritical R744 refrigerating

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solutions can be adopted with great energy efficiencies in any European climate context by adapting the system layout to the peculiar characteristics of R744.

7.2. Economic evaluations

CO_2 refrigeration technologies show reduced installation costs with respect to distribution pipes and the refrigerant charge (Matthiesen et al., 2010; Shilliday, 2012). Matthiesen et al. (2010) estimated a cost saving by about 73% by installing copper pipes in a CO_2 solution over a R404A system. The evaluation of the running costs and the moderate refrigerant leakages are additional cost-effective aspects when it comes to CO_2 refrigerating solutions.

According to the report by EMERSON Climate Technologies (2010), a CO_2 booster unit features an investment cost 48% greater than that of R404A technologies.

Tassou et al. (2011) claimed that transcritical CO_2 units are from 10% to 30% more expensive than R404A systems due to both the unique controls and components needed and the high working pressures.

The economic assessment for a small supermarket in the North of Ireland carried out by Shilliday (2012) suggested that the capital cost of a CO_2 booster system is 63% higher than that of a R404A unit. The reason for this lies in the fact that HS compressors are particularly costly (24% of the additional cost), as well as owing to the price and the number of components (e.g. valves, receiver, gas cooler, control systems). The cost of R744 was reported to be about 4 times lower than that of R404A.

The adoption of parallel compressor implies reductions in total cost of the final product ranging from 2.6% to 10.5% compared to a conventional single-stage CO_2 system at cooling medium temperatures between 30 °C to 50 °C, as computed by Gullo et al. (2015). At heat sink temperatures from 30 °C to 42 °C, Gullo and Cortella (2016a) evaluated that the total cost investment for the solution with a two-phase ejector is on average about 12% higher than that for a one-stage CO_2 unit. On the other hand, a drop in total cost of the final product on average by up to 25.2% was also computed at the same boundary conditions.

According to Girotto (2015), a return on investment of about 2.5 years can be obtained on the part of a multi-ejector based system (with intercooler/air-cooled gas cooler-condenser) compared to R404A units, as showed in Fig. 21. The evaluation took into account the investment and running costs, design point of 38 °C and cooling capacities of 250 kW at MT and 50 kW at LT, respectively. At the same boundary conditions, a transcritical CO_2 refrigeration plant with dedicated mechanical subcooling present a return on investment above 10 years.

Fig. 21. Return on investment (based on investment and running costs) related to a conventional R744 supermarket booster refrigeration system, a R744 supermarket booster refrigeration system equipped with multi-ejector block, R404A direct expansion units and a R744 supermarket booster refrigeration system outfitted with dedicated mechanical subcooling ($\dot{Q}_{MT}$ = 250 kW, $\dot{Q}_{LT}$ = 50 kW, design point = 38 °C) in a warm climate (Girotto, 2015).

Tsamos et al. (2017b) claimed that the solution with parallel compression has a payback period of about 3 years in London and 2 years in Athens.

Between 2008 and 2016 it was noticed that the efficiency of transcritical CO_2 supermarket refrigerating units went up to 25% and, simultaneously, a drop in the equipment cost by 30% occurred (Shecco, 2016a). This led to the fact that "CO_2 only" supermarket systems currently feature a similar

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to or between 5% and 10% higher total installation cost than conventional HFC units (Shecco, 2016a). In particular, it was found that the price of a R744 multi-ejector enhanced parallel compression system is nowadays, at worst, 10% higher than that of typical refrigerating units (Shecco, 2016a). On the other hand, Shecco (2016a) highlighted that the overall lifecycle cost of the solutions relying on natural refrigerants is lower than that associated with HFC-based technologies. Finally, further economic benefits are expected to be derived from the ever-growing economy of scale related to "all-in-one" HFC-free equipment.

7.3. Environmental impact evaluations

EMERSON Climate Technologies (2010) suggested that a R744 booster solution in place of a R404A system brings the carbon footprint of a supermarket down to 42%.
Shilliday (2012) pointed out that the Total Equivalent Warming Impact (TEWI) associated with a booster configuration running in Northern Ireland is about 50% lower than that of a HFC-based unit.
Beshr et al. (2014) compared the environmental impact of various supermarket refrigeration systems in six different American cities. The results revealed that, although a R744 booster solution performs worse in some of the selected locations, such a technology is more eco-friendly than both N-40-based arrangements and a R404A multiplex unit in mild and cold climates with annual leak rate above 2%.
Llopis et al. (2015b) indicated that cascade/indirect arrangements using low-GWP working fluids for LT applications are more environmentally beneficial than the investigated "CO_2 only" systems.
Compared to a R744/R134a cascade system, Gullo et al. (2016a) estimated reductions in TEWI from 9.7% in Athens to 24.9% in Valencia (Spain) by using parallel compression and from 9.6% in Athens to 24.4% in Valencia by adopting a dedicated mechanical subcooling.
At a later time, Gullo et al. (2016e) computed drop in TEWI between about 31% and 56% for the configuration with both mechanical subcooling and parallel compression in five Mediterranean European locations in relation to a R404A direct expansion unit.
The solution suggested by Colombo et al. (2016) was capable of cutting the carbon footprint related to a store located in Wales by 34% over R404A units.
The study carried out by Tsamos et al. (2017b) highlighted that a reduction in TEWI by respectively 3.6% in London and 5% in Athens can be accomplished with the aid of parallel compression over a conventional booster unit.
Karampour and Sawalha (2018) estimated that TEWI associated with supermarket refrigeration solutions relying on man-made working fluids (i.e. R404A, R449A) is 2-7 times as high as that of natural refrigerant-based systems (i.e. R744, R717, R290) in Stockholm and Barcelona.

8. Conclusions

The ever-growing commitment to the protection of the environment, as well as the cheapness, acceptable safety level related to its usage and its favourable thermo-physical properties opened the door to wider approval of R744 as the only refrigerant in food retail industry. Therefore, being HFCs potent greenhouse gases, transcritical R744 systems have become standard for the commercial refrigeration sector all over the world as a result of their enormous technological advancements.

In this investigation the state-of-the-art "CO_2 only" solutions for supermarket applications have been critically reviewed. These are being continuously cost and efficiency optimized, yielding promising results even in warm countries.

The most relevant conclusions which can be drawn from this research work are:

- an additional diffusion of these HFC-free technologies in some countries (e.g. USA) should be further stimulated with the aid of some legislative acts;
- the dedicated mechanical subcooling is not bound to significantly contribute to the spread of transcritical R744 refrigerating solutions in food retail stores;

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- the multi-ejector concept is the main driving force behind the proliferation of transcritical CO_2 supermarket refrigeration systems worldwide and especially in warm climates. Also, additional profuse cost savings are expected to be achieved by adopting the "all-in-one" concept;
- thermal energy storages are supposed to lead to further promising energy efficiency enhancements;
- a noteworthy attention needs to be devoted to the stores located in warm climates where an abundant lack of both confidence in such technologies and trained technicians is evident (Minetto et al., 2018). This can be considered the current main brake on the broad diffusion of these solutions in high ambient temperature countries;
- transcritical R744 supermarket technologies present similar to or between 5% and 10% higher total installation cost than conventional HFC units (Shecco et al., 2016a). This gap is expected to be filled following on from a larger scale mass production of such systems, especially the ones relying on "all-in-one" concept;
- the adoption of CO_2 as the only refrigerant in food retail applications implies great environmental benefits, even in warm places.

Finally, it is important to highlight that, although ejector-based "CO_2 only" systems represent the most promising solutions for supermarket applications in any climate context, various drawbacks still need to be overcome, such as their high cost (especially related to the control devices) as well as their considerable complexity (leading to possible complicated maintenance operations).

9. Future developments

By considering the findings summarized in the previous Sections, it can be concluded that new available technologies will permit the proliferation of transcritical CO_2 supermarket solutions across the world. This target will be successfully achieved provided that the remaining non-technological barriers are overcome. However, in order to accomplish this goal, many investigations on various perspective are still needed.

With respect to CFD assessments, the implementation of the following work should be taken into account:

- additional more thorough CFD simulation models need to be developed in order to improve the accuracy of ejector performance prediction as well as its design. These have to be validated against some field measurements, which implies that massive experimental campaigns have to be carried out. In particular, as suggested by Kriezi et al. (2015, 2016), additional experimental evaluations regarding the effect of the geometrical characteristics of ejectors on their functionality, the system performance and the capacity control are indispensable. Also, the possible benefits related to the switching from winter to summer ejector design as the heat recapture is implemented need for in-depth studies (Kriezi et al., 2015, 2016). As a consequence, the performance of the ejectors would significantly enhance, as well as the confidence in such a technology would also be boosted;
- the implementation of more sophisticated CFD simulation models are also necessary for gas cooler/condensers. In addition, the investigation by Heerup and Bramsen (2016) suggests mapping the gas cooler/condenser performance in both design and off-design operation conditions for many geometries and flow regimes should also be realized. This would allow developing a suitable and generalized methodology aimed at their cost-effective selection.

From the thermodynamic viewpoint, significant efforts should be devoted to:

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- the application of advanced thermodynamic tools, such as thermoeconomic diagnosis and advanced exergy analysis, to transcritical R744 solutions based on field measurements in order to permit the selected system to permanently operate at the most cost-effective and thermodynamically efficient running modes (Ommen and Elmegaard, 2012; Gullo et al., 2016b; Gullo and Hafner, 2017b);
- the assessment of the energy and economic benefits related to the usage of two different multi-ejector blocks (i.e. one dedicated to AC need and the other to refrigeration demand), to direct heating and cooling fan coils and air curtains and to the interchangeability of main and auxiliary compressors (i.e. reduction in compressor number) (Hafner, 2017; Pardiñas et al., 2017a, 2017b). This has to be supported by means of field measurements;
- appropriately evaluate the performance of “all-in-one" R744 multi-ejector enhanced parallel compression systems in severe climate contexts (e.g. India), as suggested by the results derived from Kvalsvik et al. (2017). In fact, it has already been widely recognized that these technologies are the most highly efficient and climate friendly solutions for the European commercial refrigeration sector.

Although thermal energy storages have been gaining great ground, additional research work is still essential:

- the solution proposed by Manescu et al. (2017) seems to be a promising expedient. However, additional investigations are needed, such as the heat flow evaluation in the cold storage unit, experimental validation, assessment of the heat transfer within the cold storage and the thermosiphon system;
- substantial efforts should be addressed to geothermal storages as, despite the growing interest in these solutions, their real benefits still need to be thoroughly studied. Future work is necessary when it comes to field measurements collection, implementation of a suitable heat recovery strategy, appropriate design of the heating system, economic evaluations, assessment of geothermal storage function, control and sizing, etc.;
- despite being promising technologies, Fidorra et al. (2016, 2015b) pointed out the need for additional and more in-depth evaluations involving economic aspects and the part load operations of the compressors should be addressed to the CO_2 systems employing CTESs. Additionally, the storage heat transfer process and suitable control strategies for CTESs should also be assessed (Fidorra et al., 2016, 2015b), as well as these results need to be validated against some experimental data (Fidorra et al., 2016, 2015b);
- the decoupling the heating reclaim and refrigeration demand with the aid of thermal energy storages is also significantly worth being investigated, as suggested by the outcomes obtained by Nöding et al. (2016).

Finally, the future work should also involve:

- the enhancement of the efficiency related to R744 compressors as well as the enlargement of their operational envelope;
- the implementation of life-cycle cost analyses (LCCA) related to “all-in-one" R744 multi-ejector enhanced parallel compression systems;
- as the state-of-the-art technologies for supermarket applications has matured, a suitable transfer process of these solutions into small applications (e.g. CO_2 condensing units) and heat pumping systems needs to be considered.

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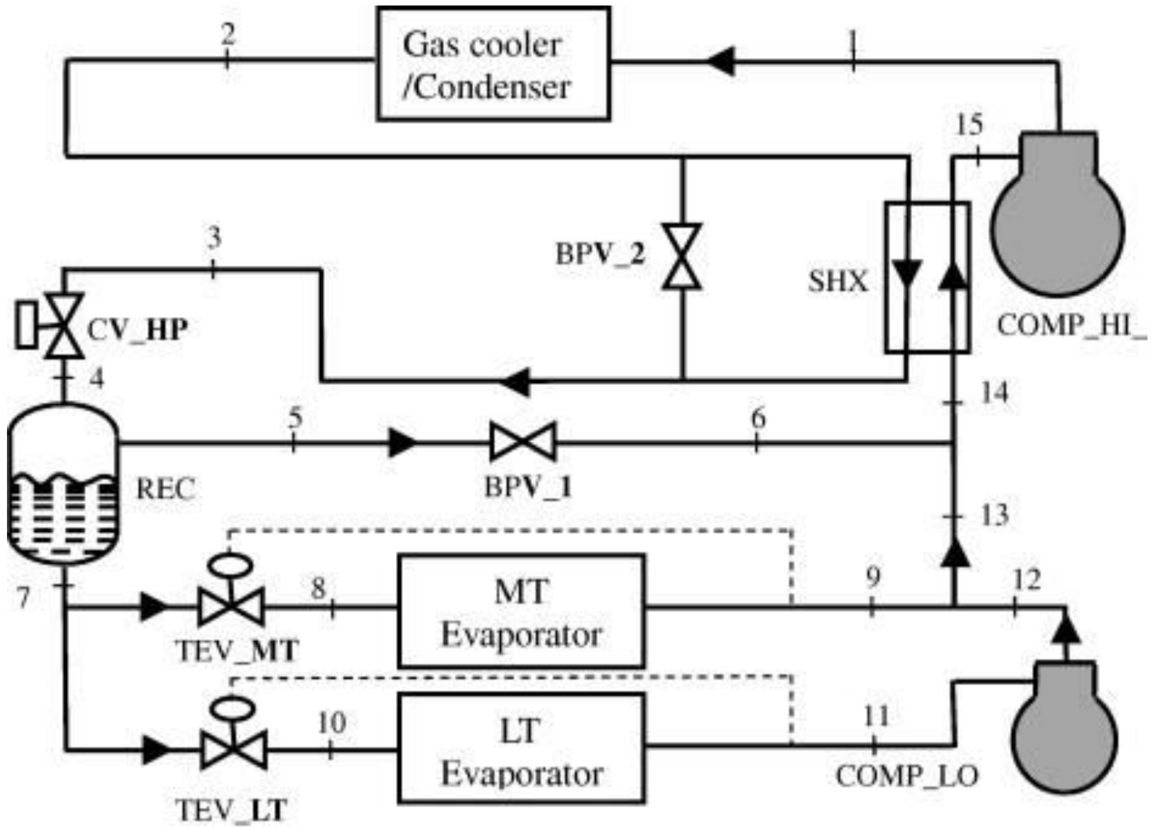
Figure 1



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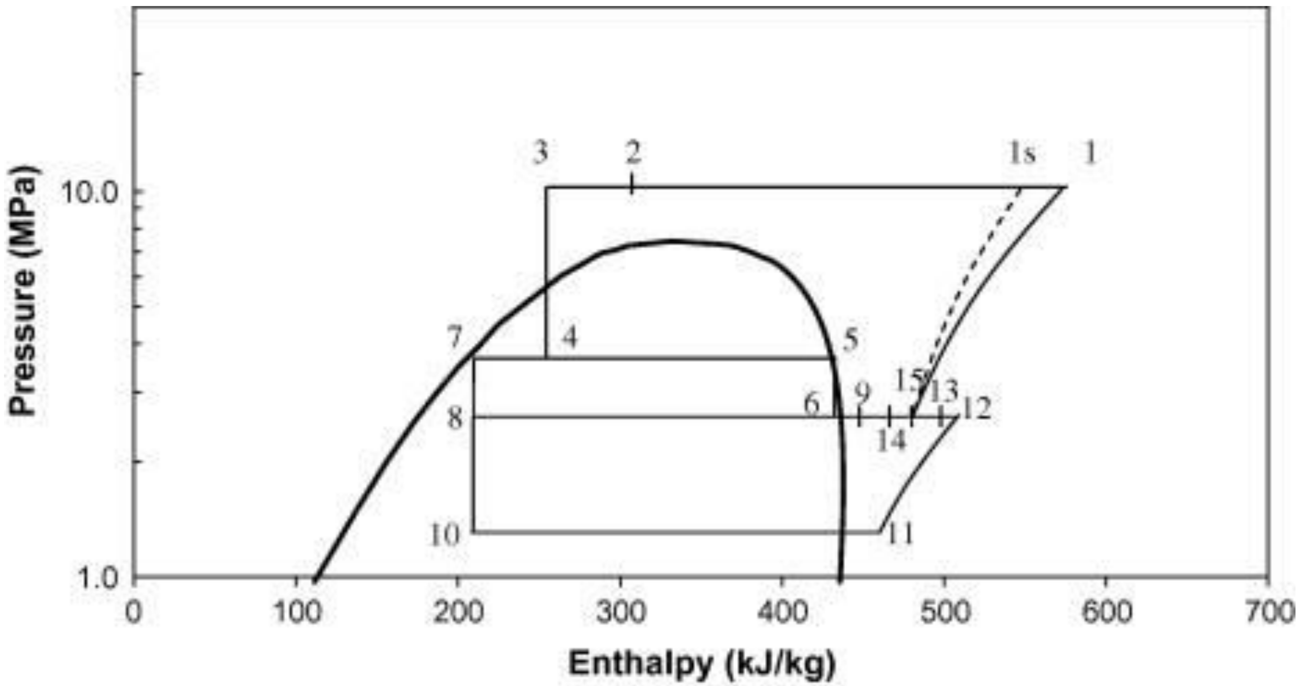

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Figure 2

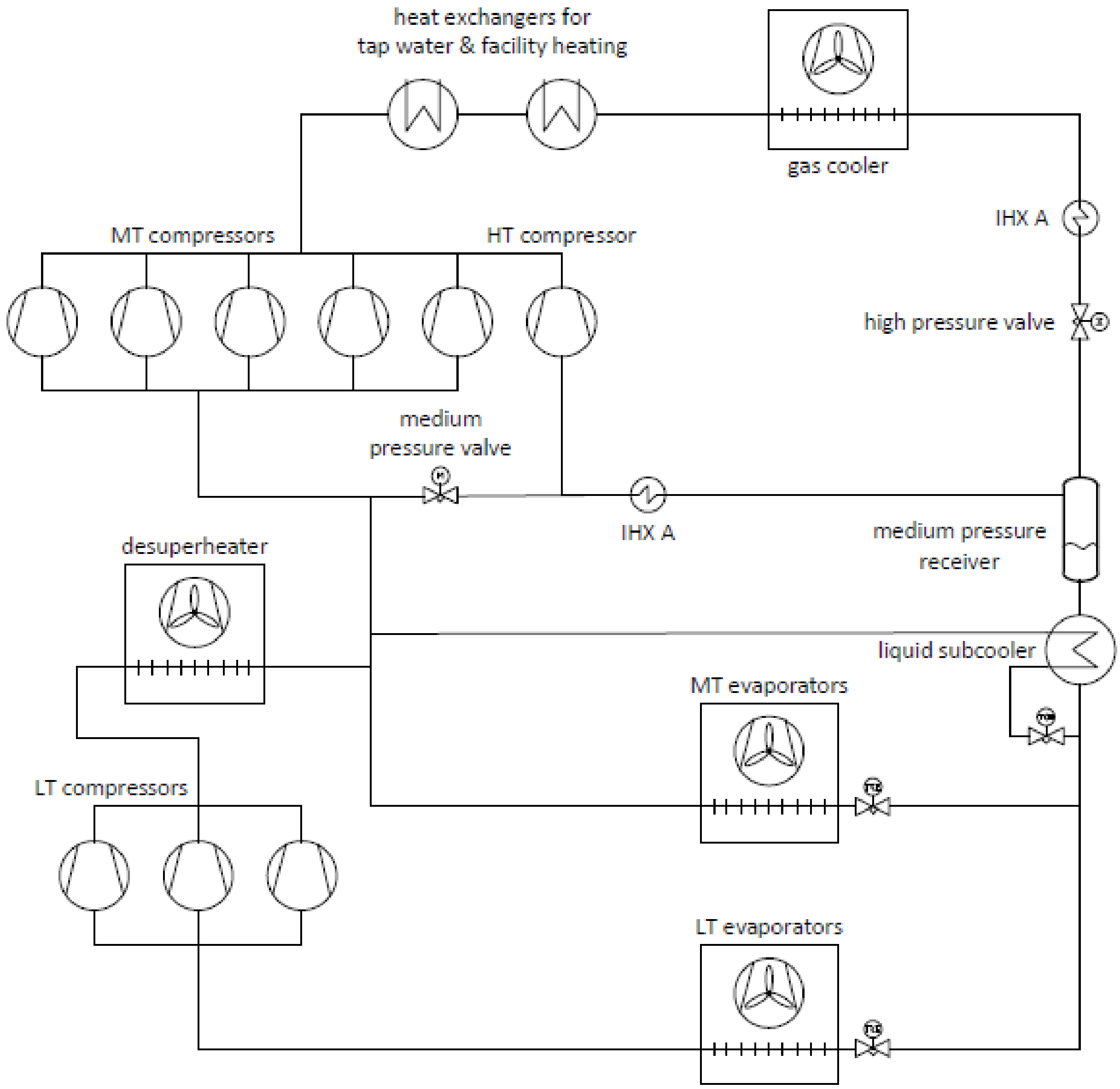

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Figure 3

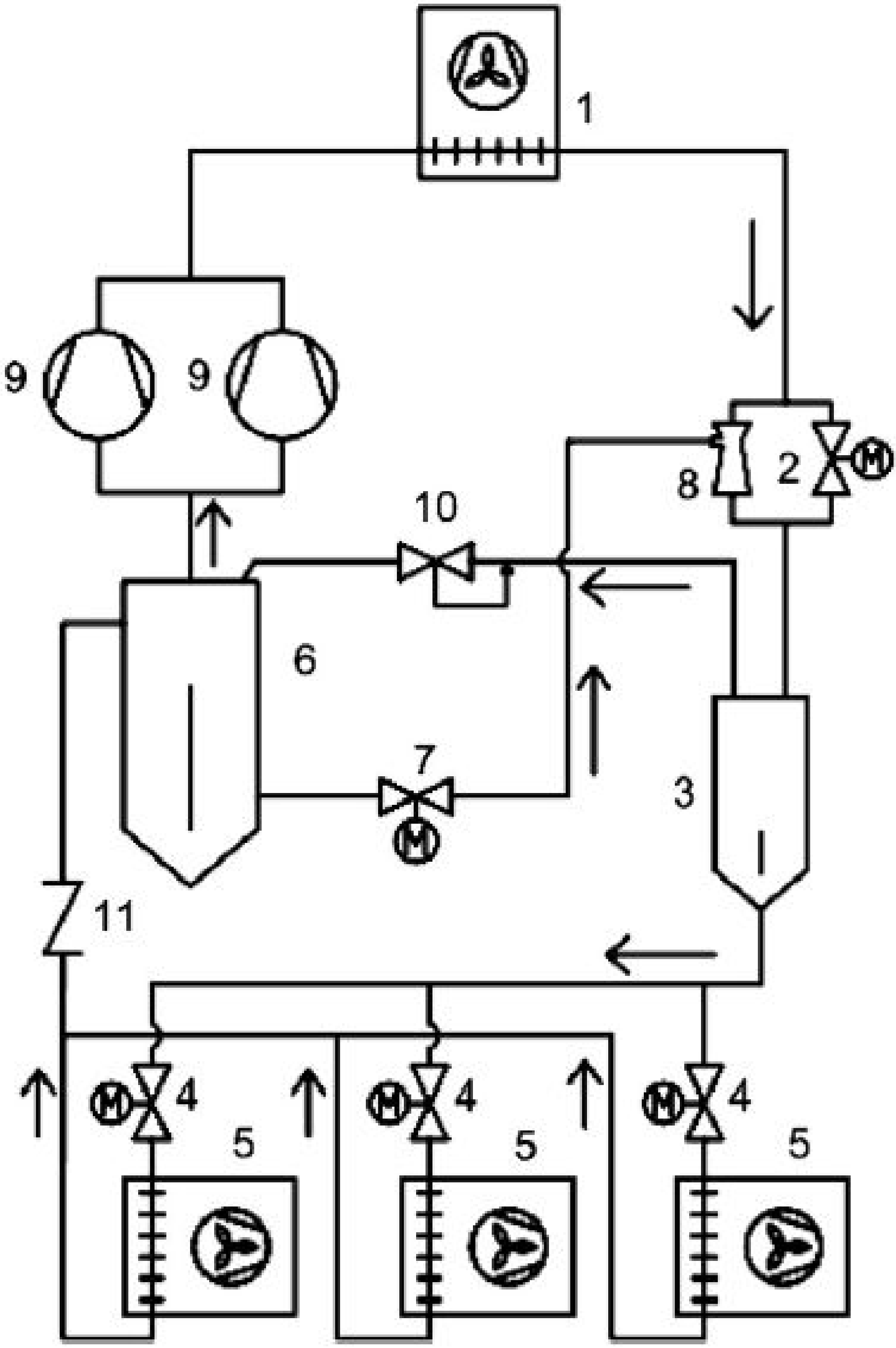

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Figure 4

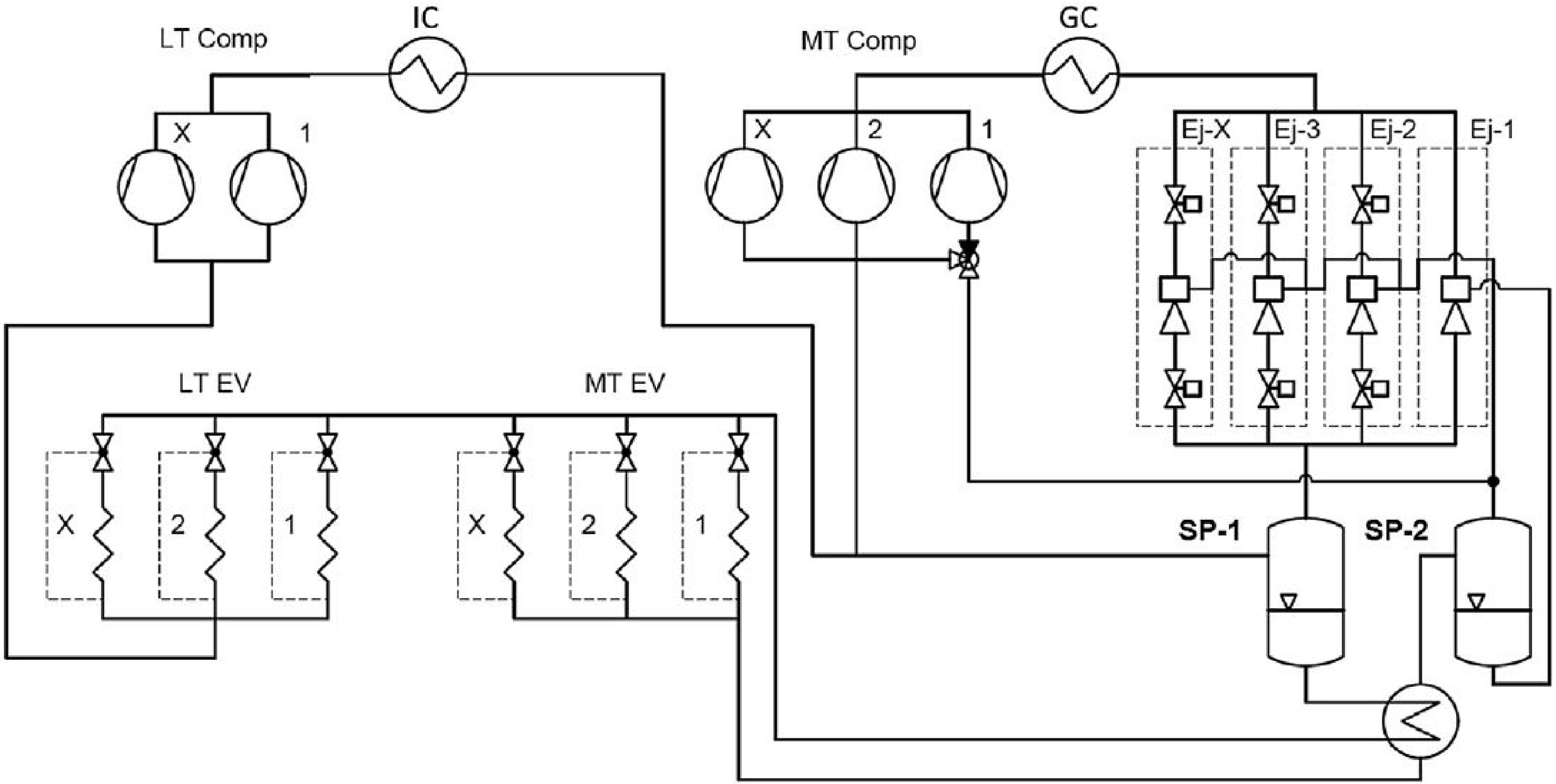

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Figure 5

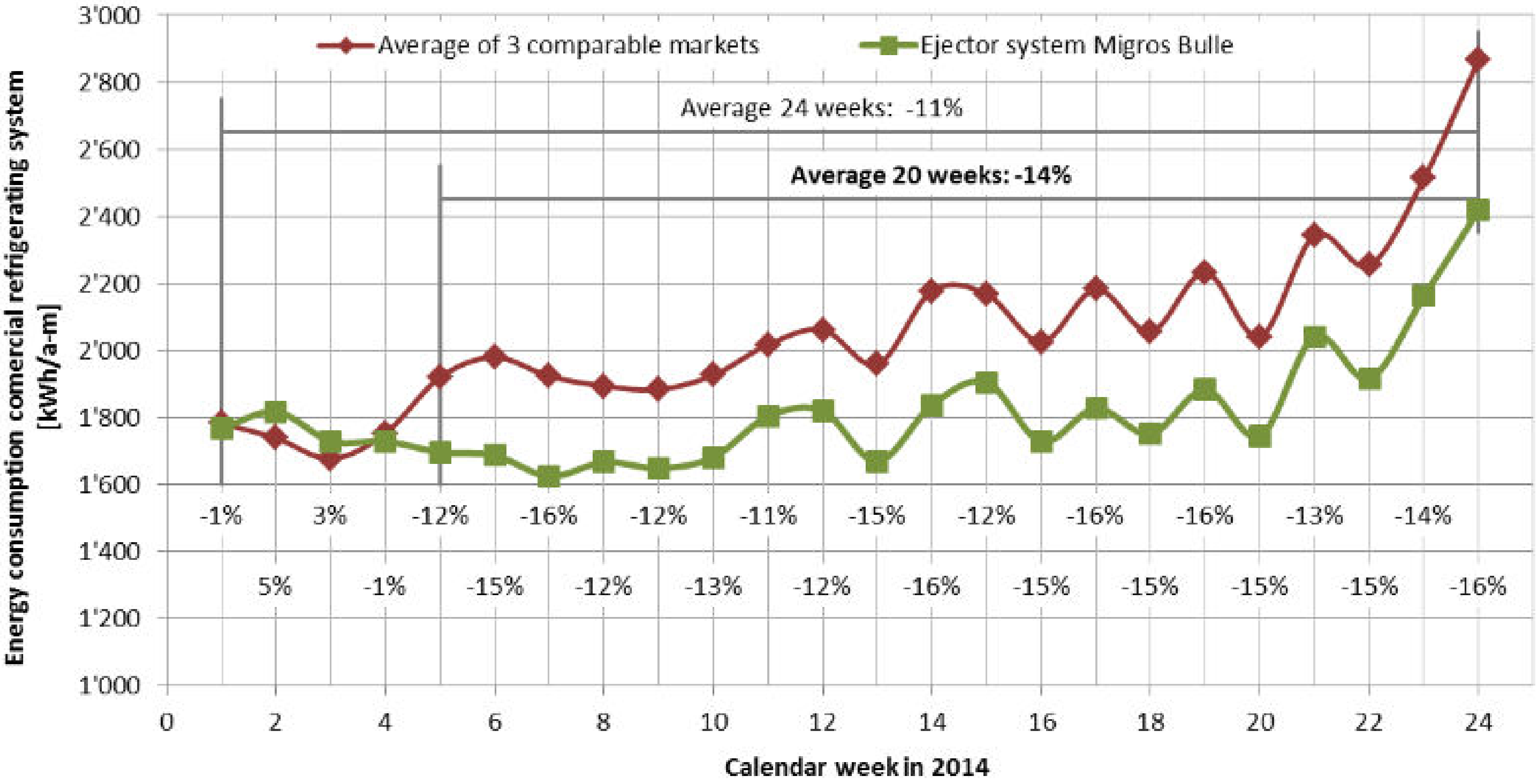

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Figure 6

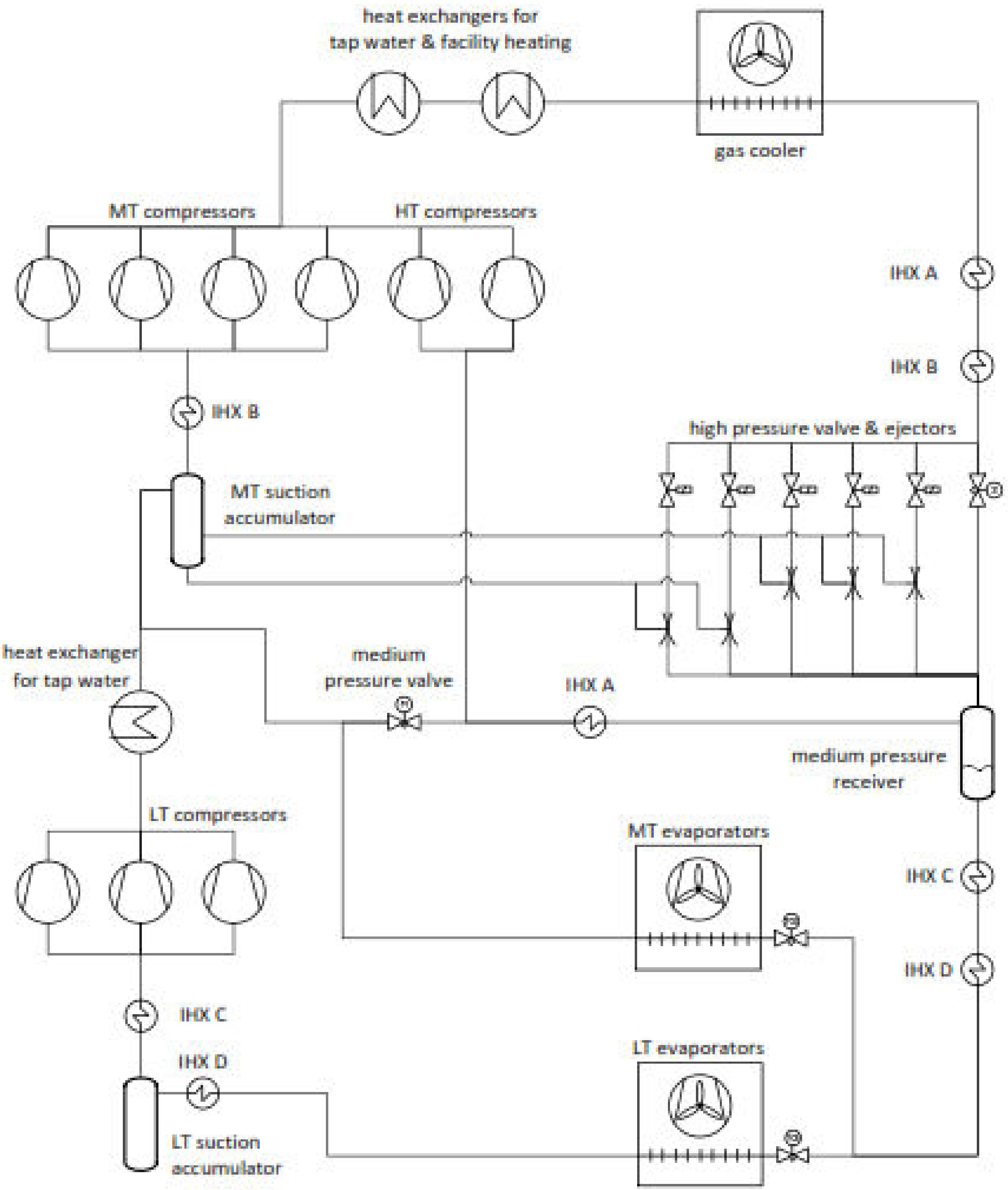

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Figure 7

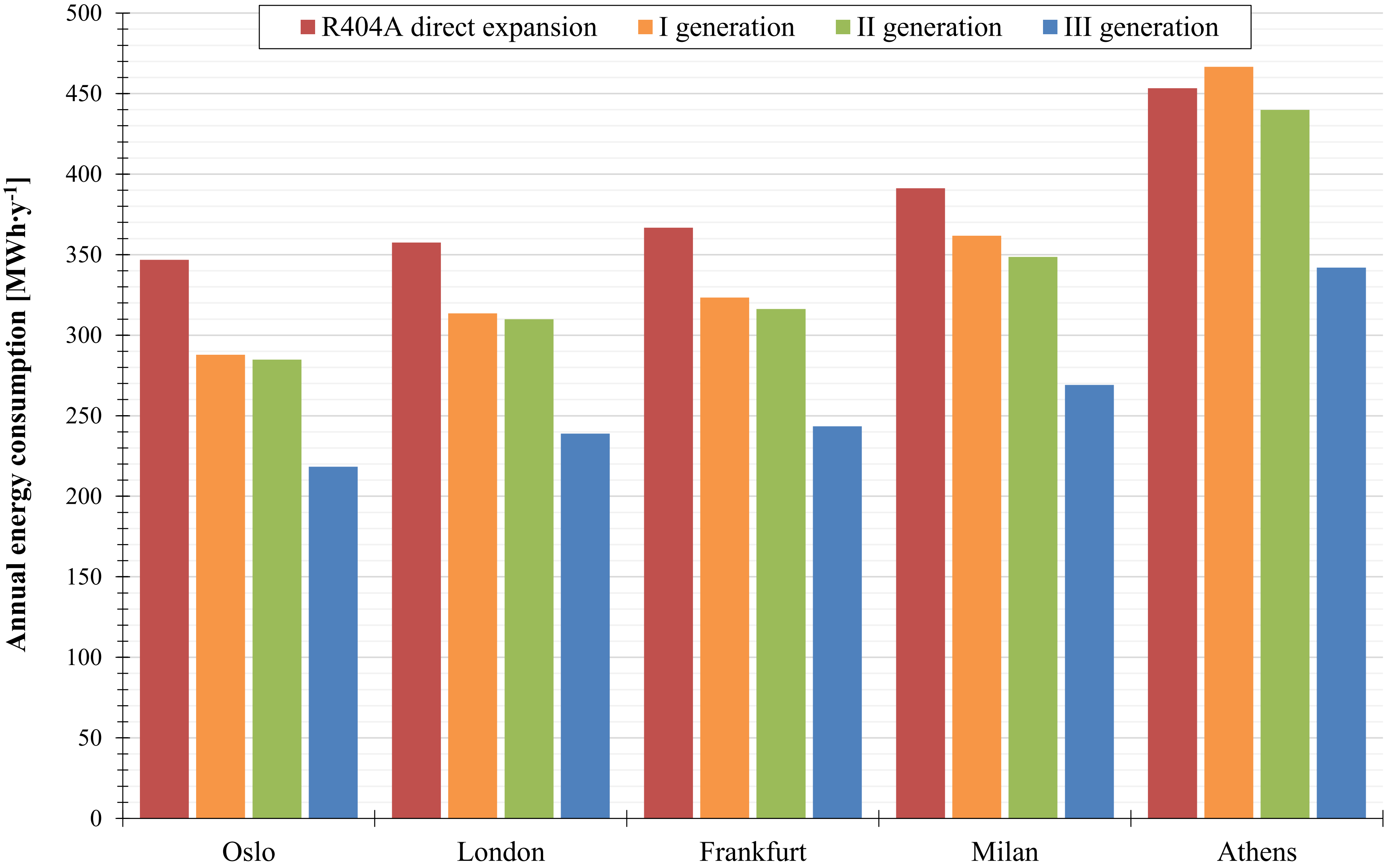

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Figure 8

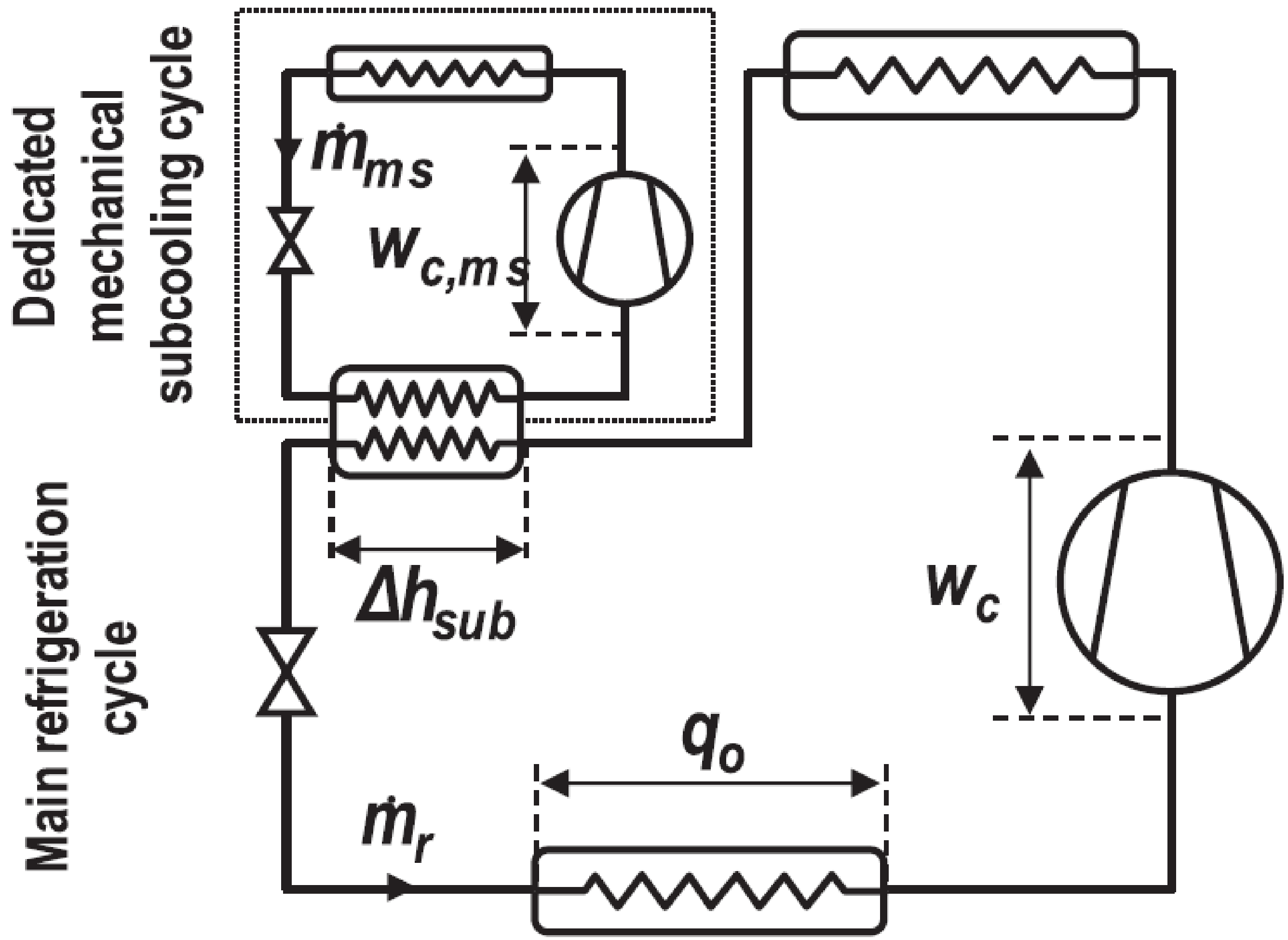

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Figure 9

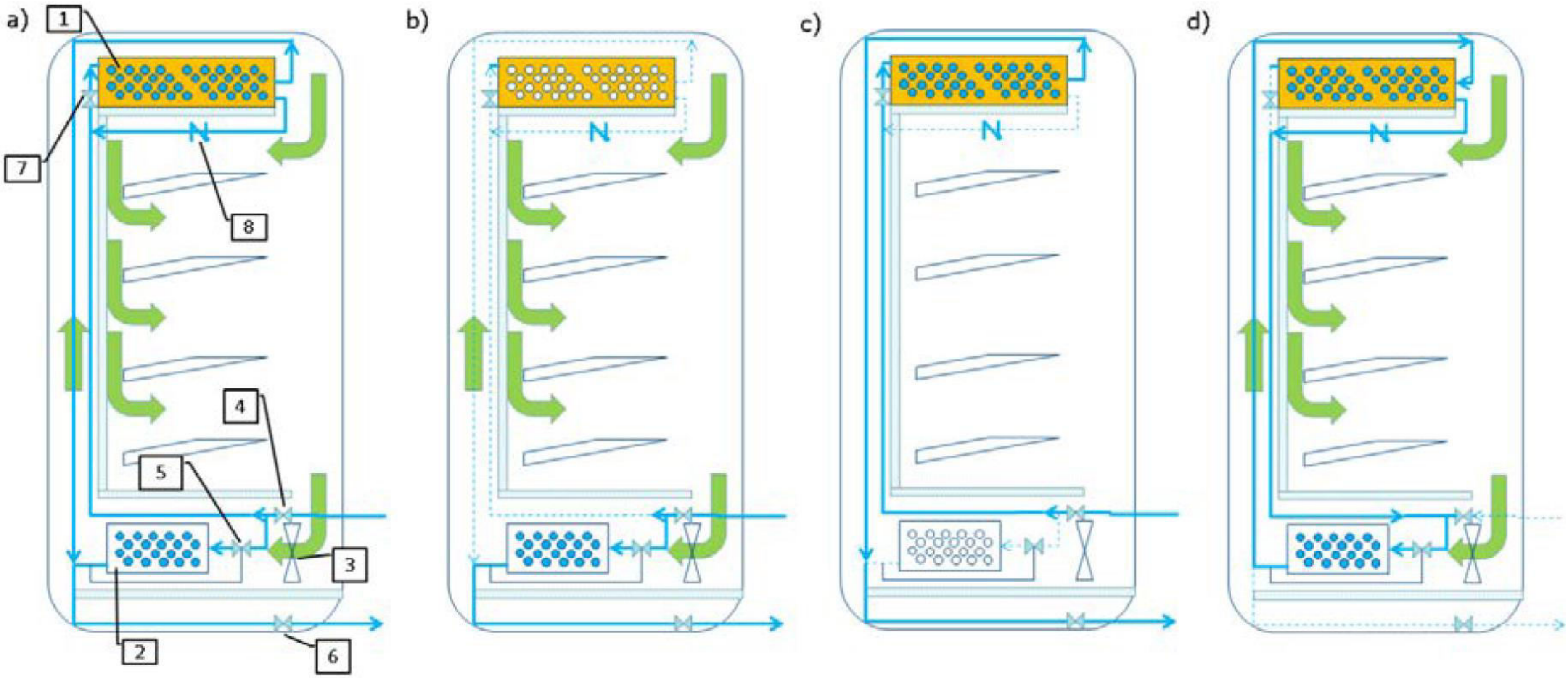

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Figure 10

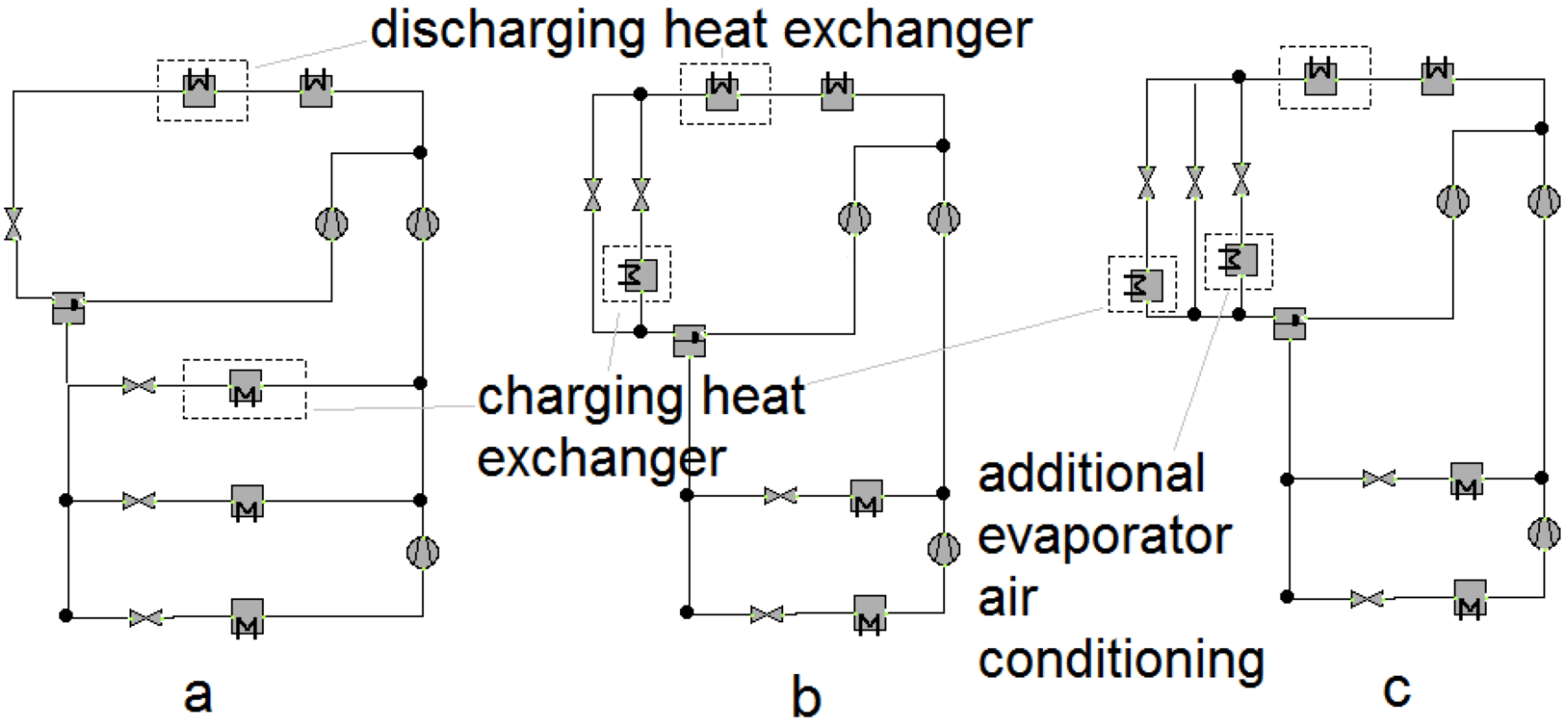

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Figure 11

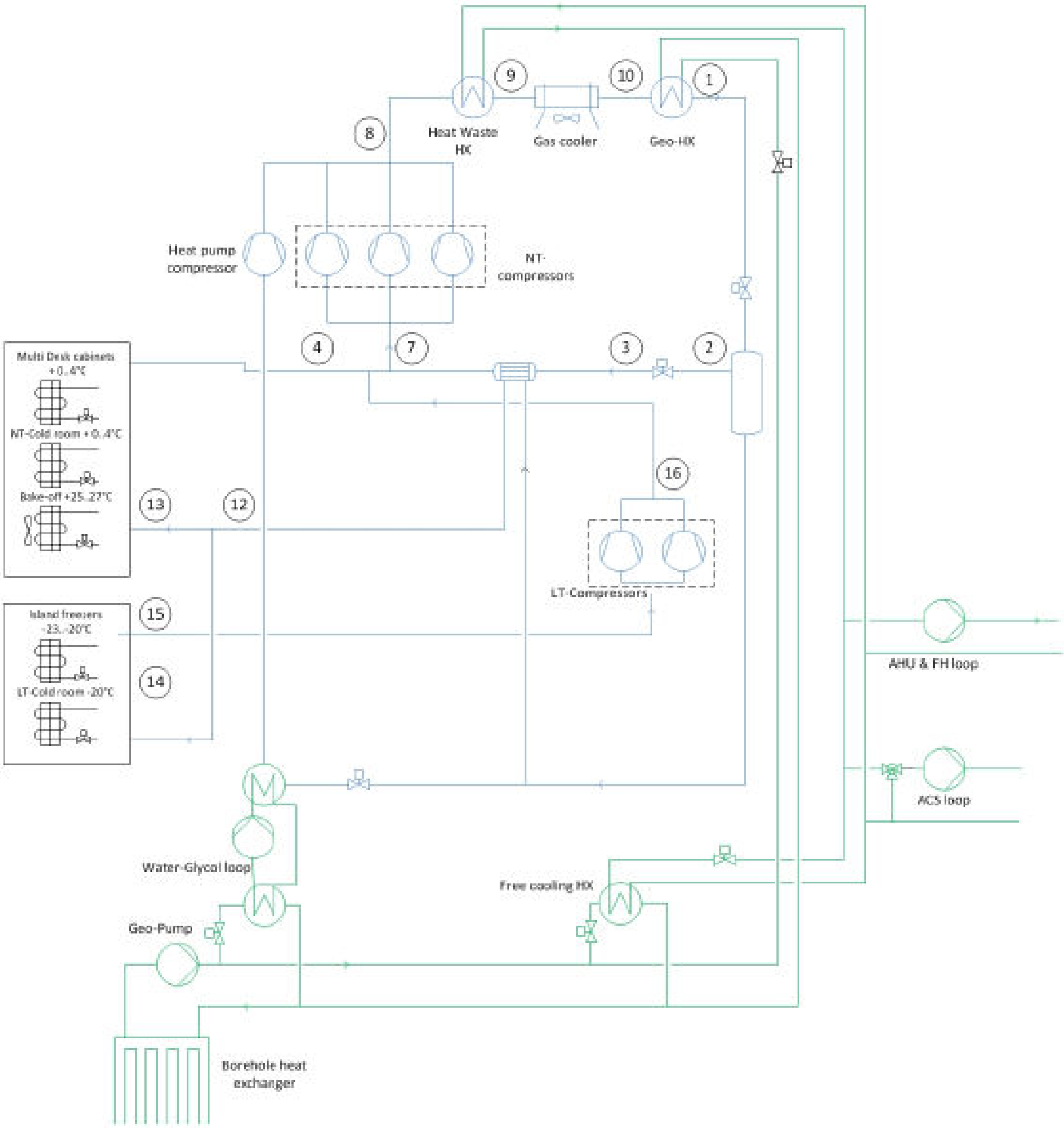

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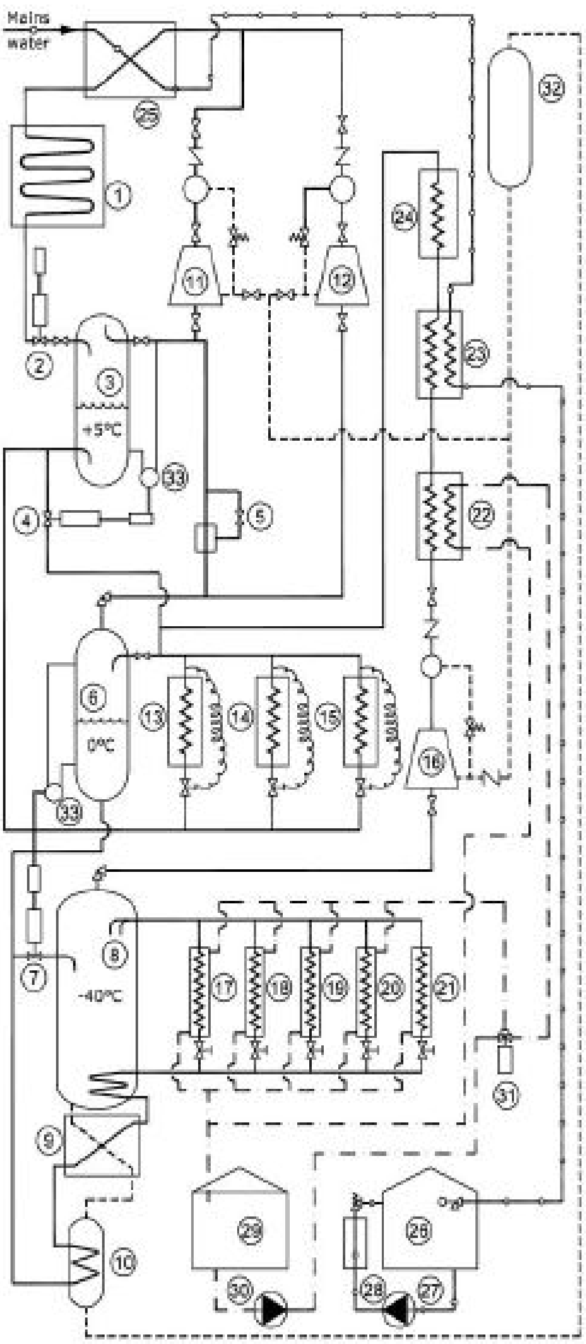
Figure 12



Legend

1	Hybrid CO_2 gas cooler/evaporative condenser
2	Compressor discharge pressure regulator
3	+5°C 1st stage expansion vessel
4	2nd stage expansion to interstage
5	Back pressure regulator
6	0°C intercooler/Hi stage suction trap
7	3rd stage expansion from 0 to −40°C
8	−40°C suction trap
9	Oil still heat exchanger
10	Oil drain vessel
11	Parallel compressors
12	High stage compressors
13	Work Area evaporators
14	Fresh make-up air coolers
15	Process Area and AC evaporators
16	Booster compressors
17-19	Cold Store evaporators
20	Blast freezer evaporator
21	Transfer Area evaporator
22	Defrost glycol heater in booster discharge
23	2nd stage water heater in booster discharge
24	Transfer Area reheat in booster discharge
25	1st stage water heater
26	Potable hot water tank
27	Hot water circulating pump
28	Hot water consumers
29	Defrost fluid tank
30	Defrost fluid circulating pump
31	Glycol diverting valve to heating or evaporator defrost
32	Compressor oil reservoir

—— CO2 lines
–◦–◦– Potable water from main to storage tank
– – – Temper −55 defrost fluid
------- Compressor oil recovery & re-use

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Figure 13

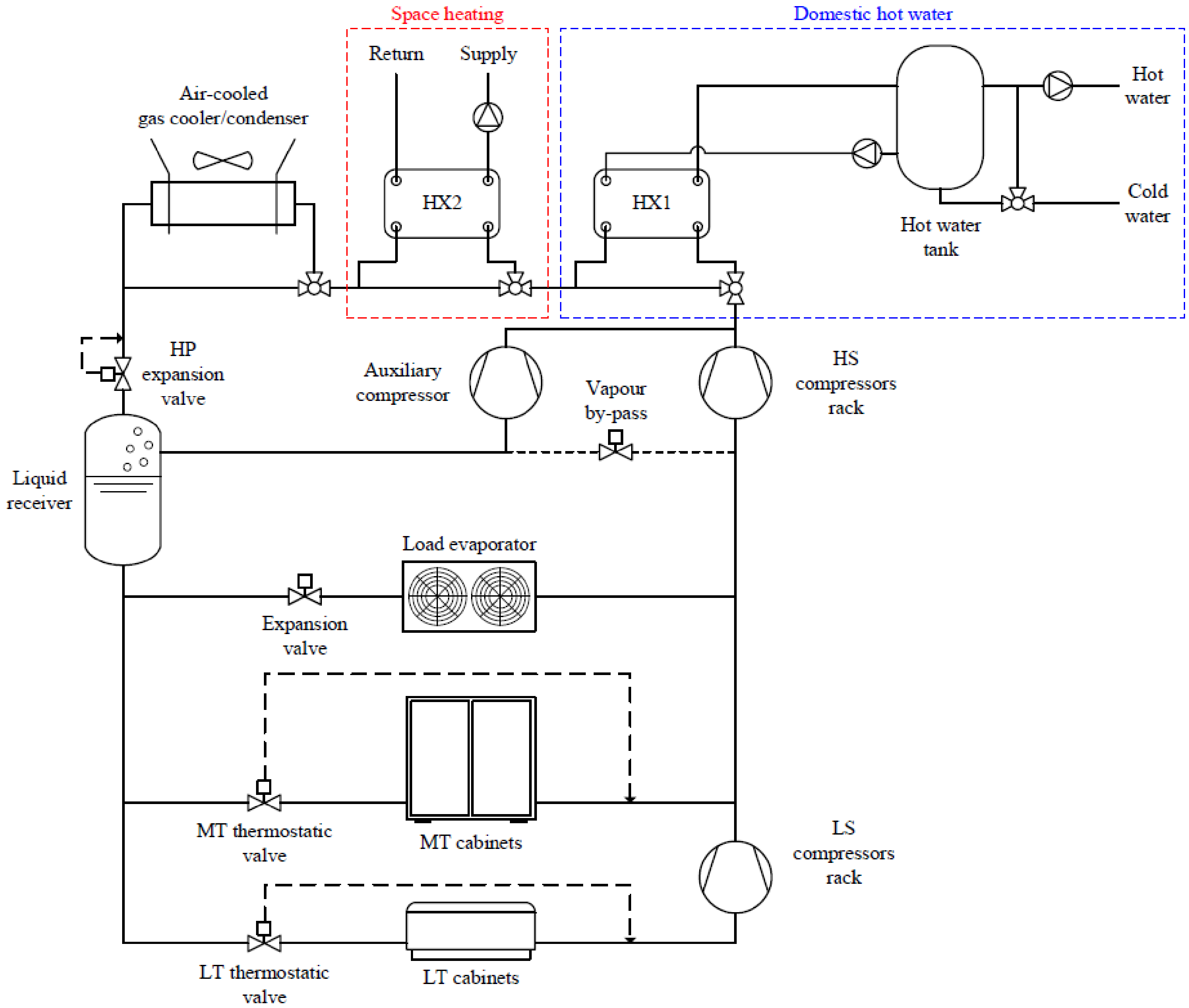

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Figure 14a

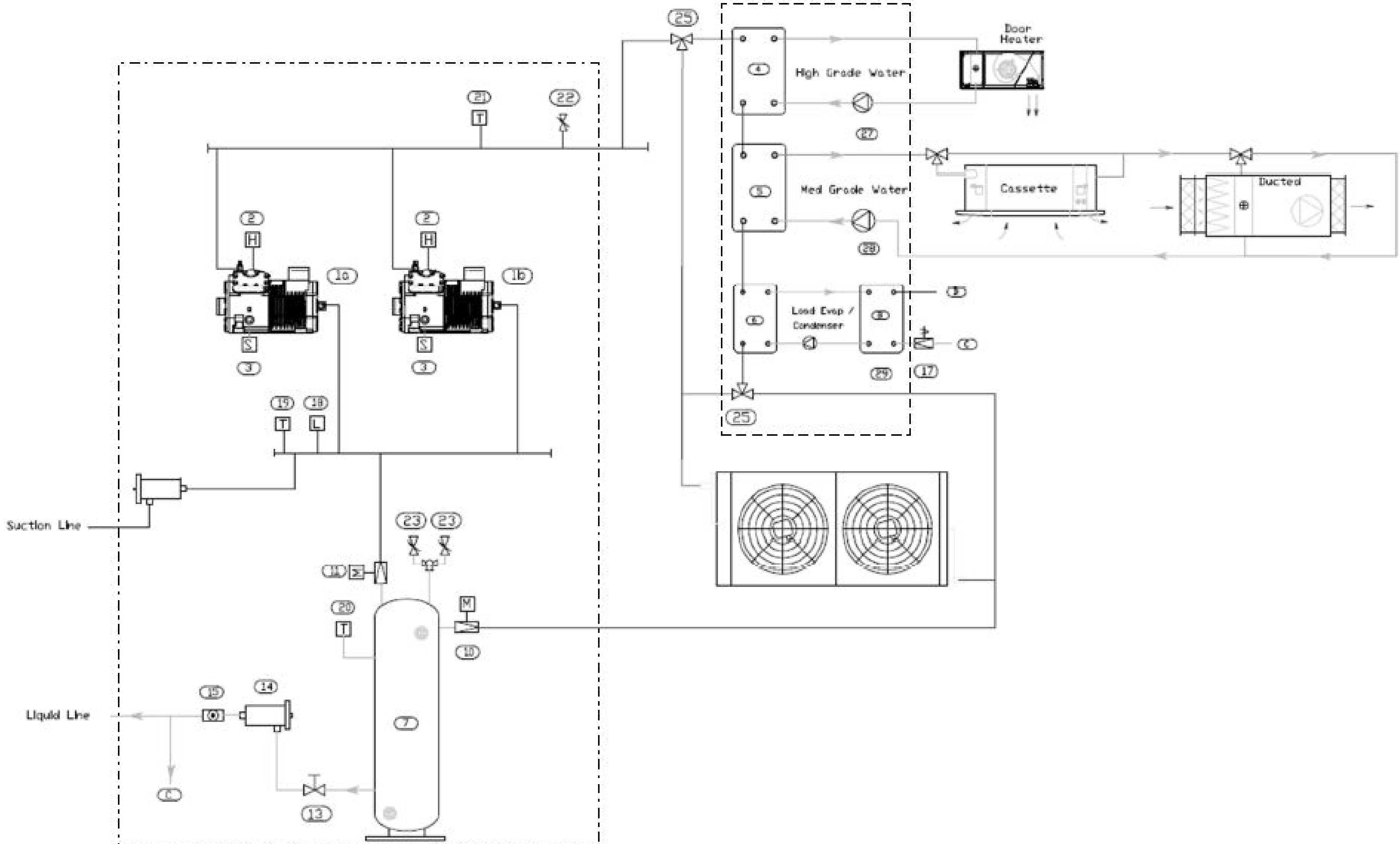

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Figure 14b

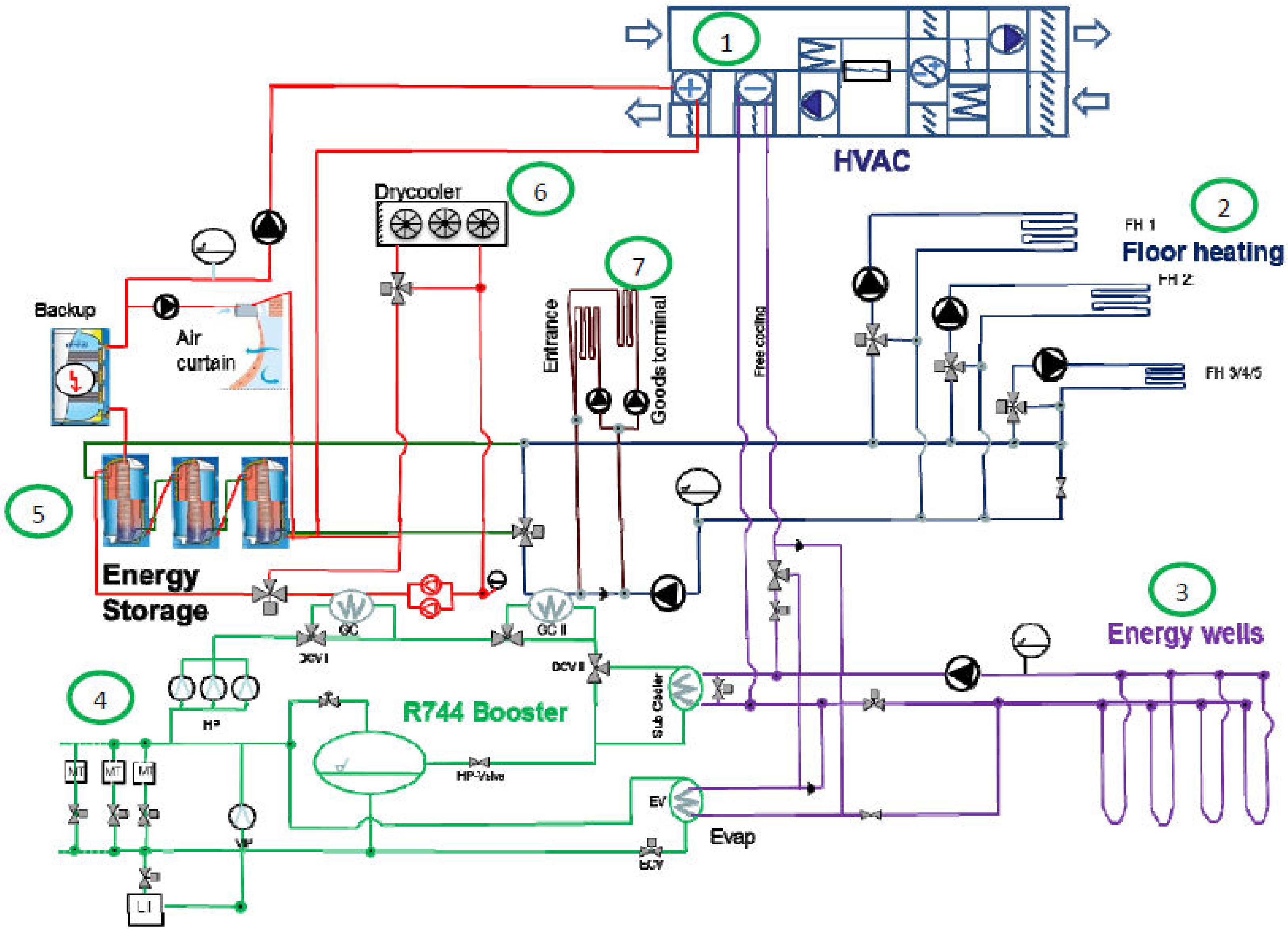

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Figure 15

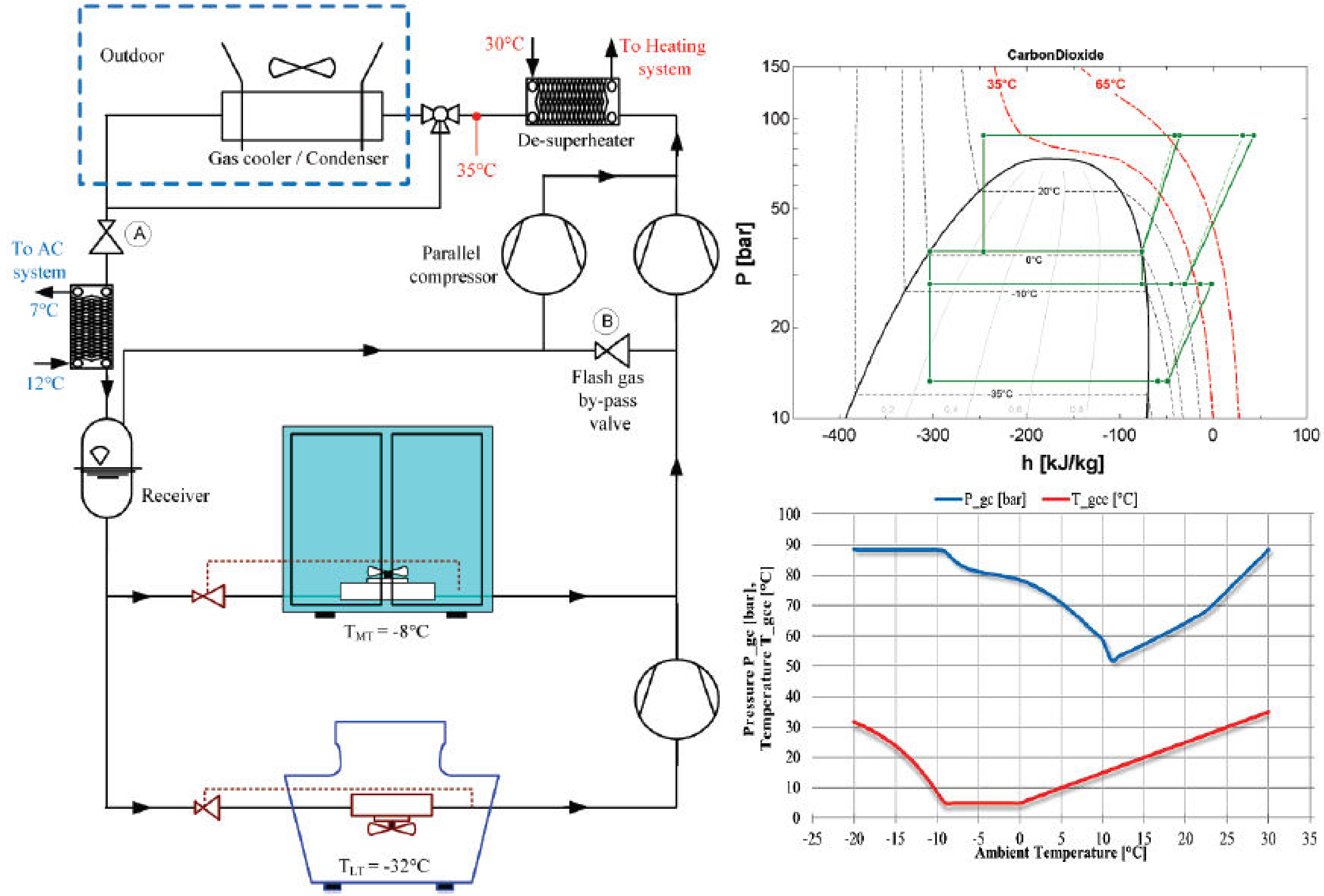

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Figure 16

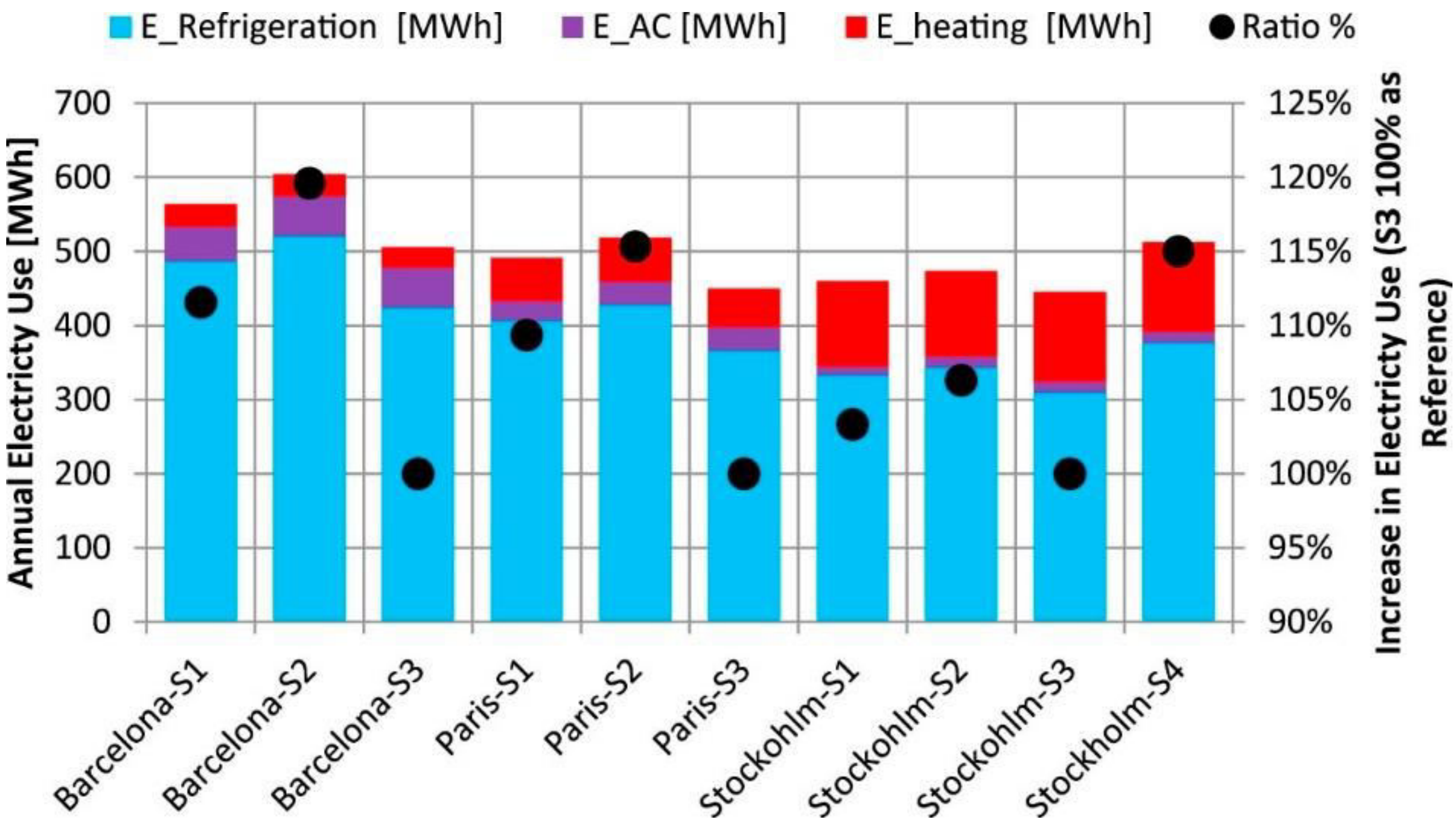

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Figure 17

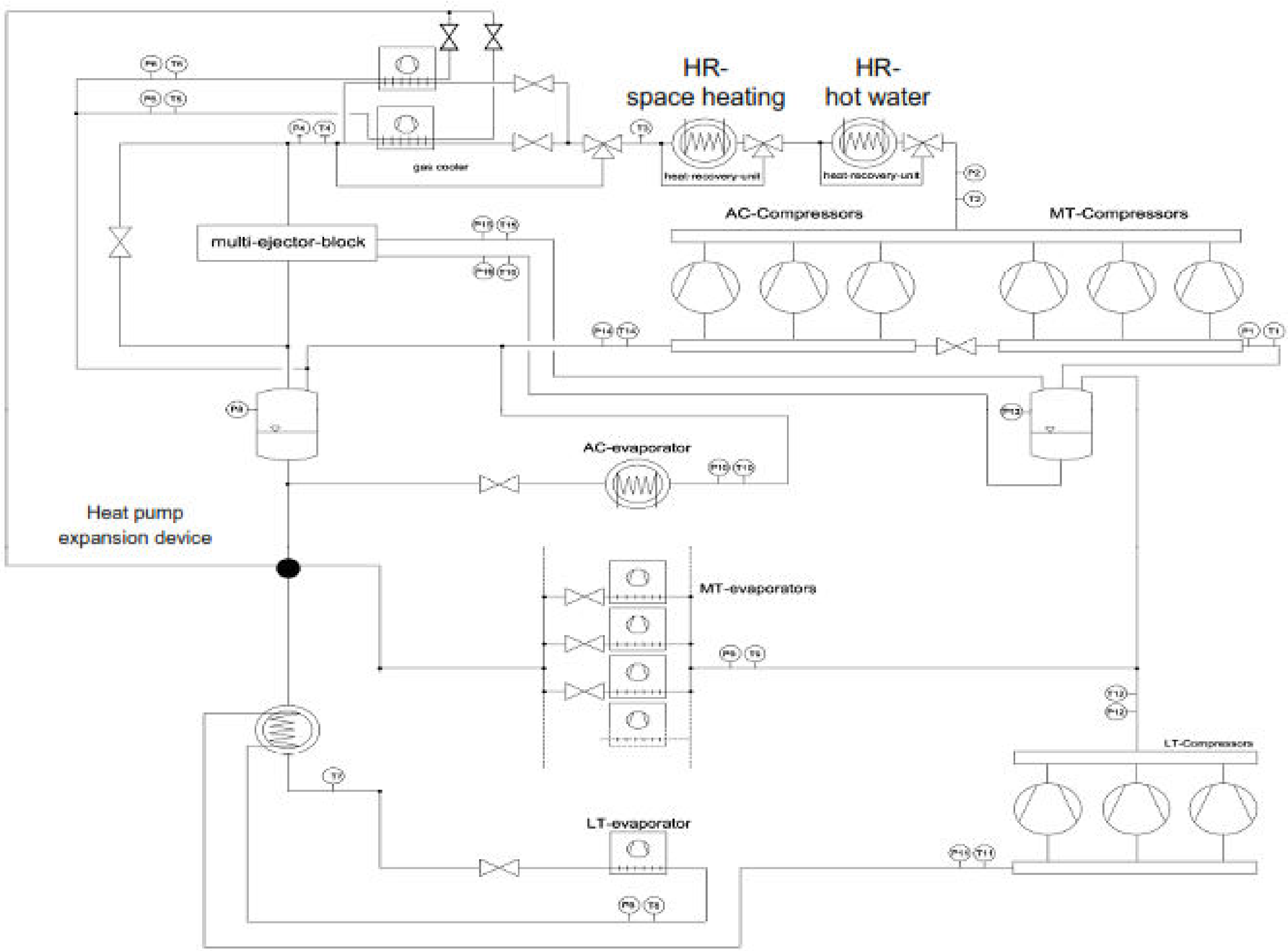

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Figure 18

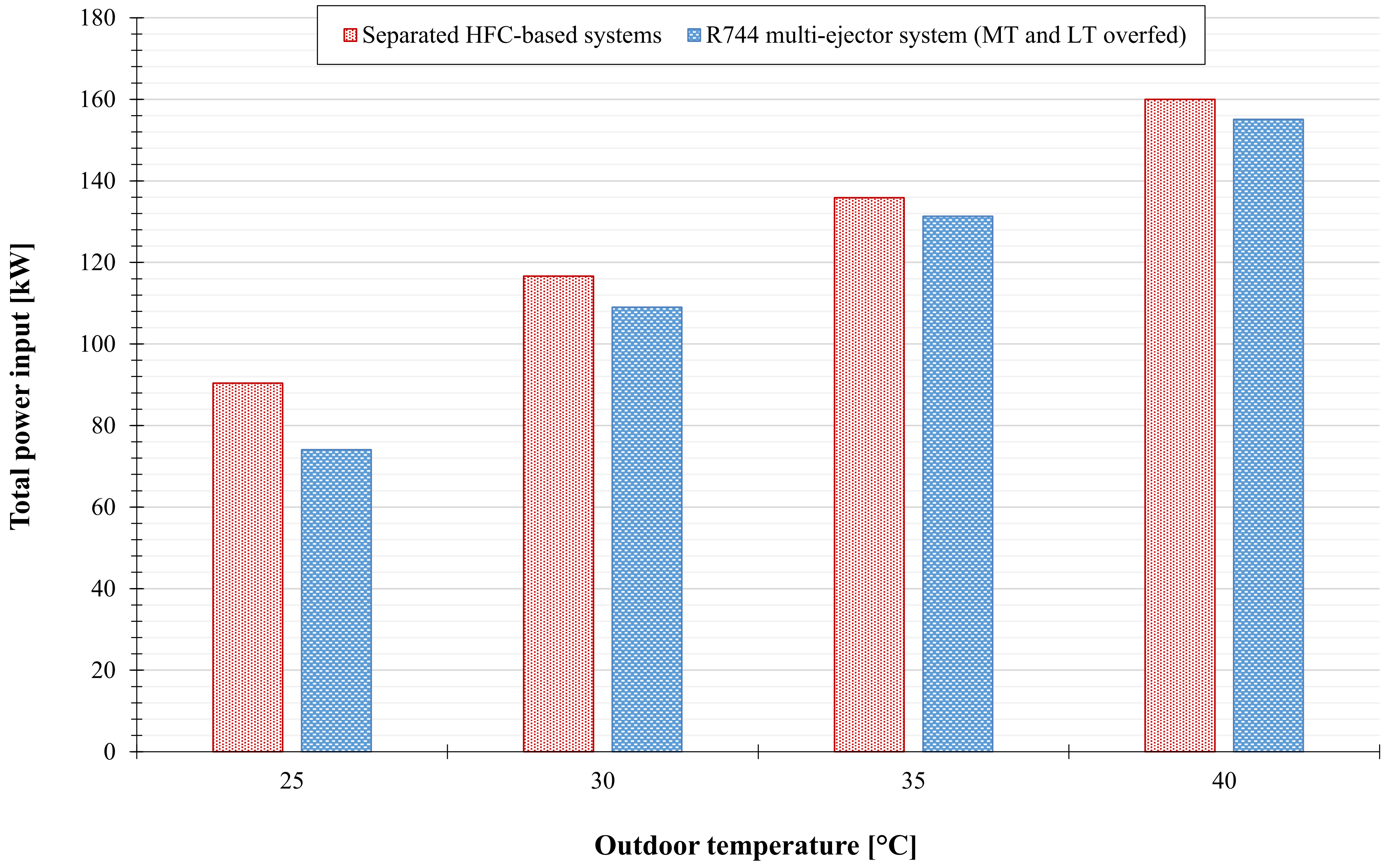

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Figure 19a

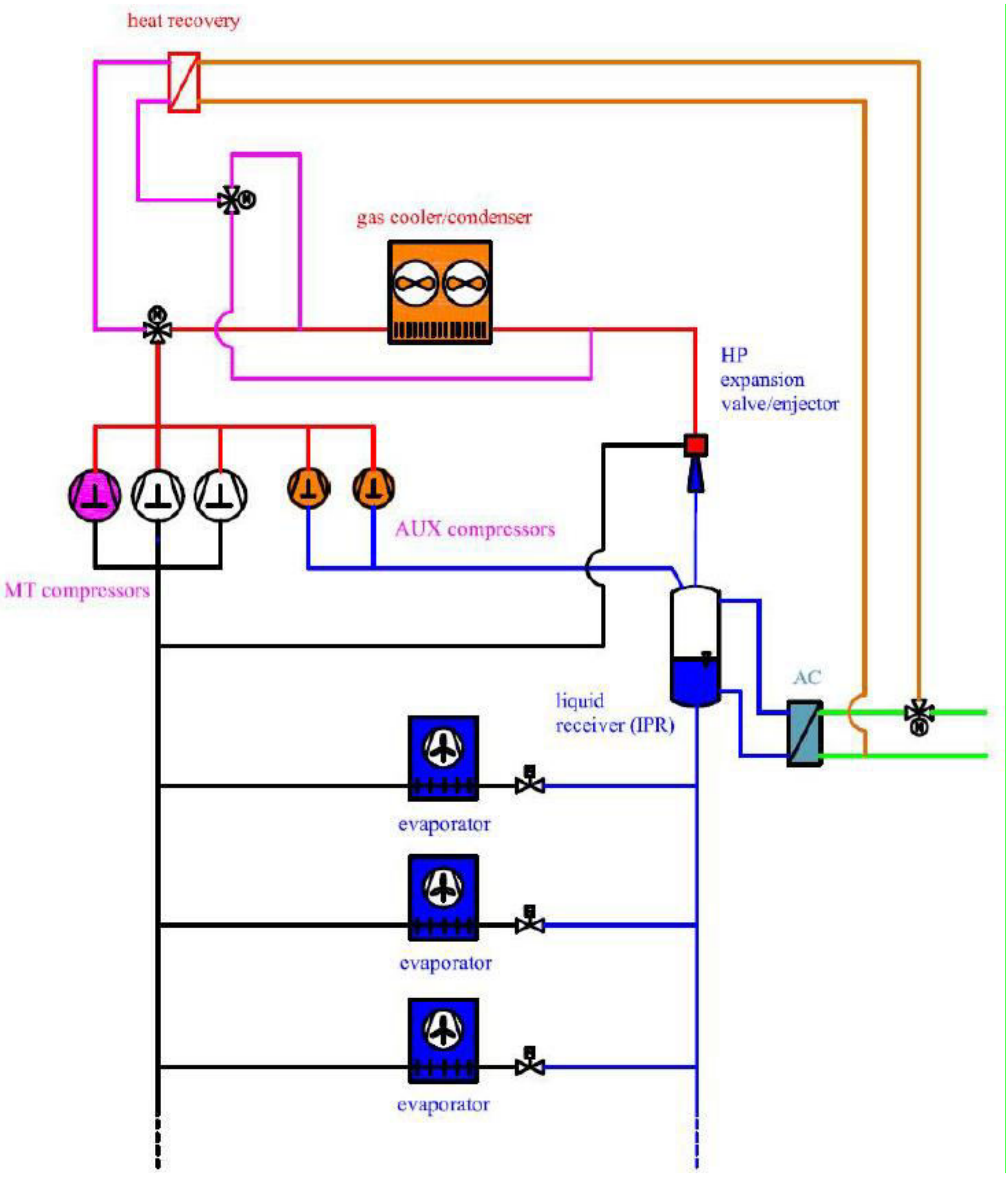

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Figure 19b

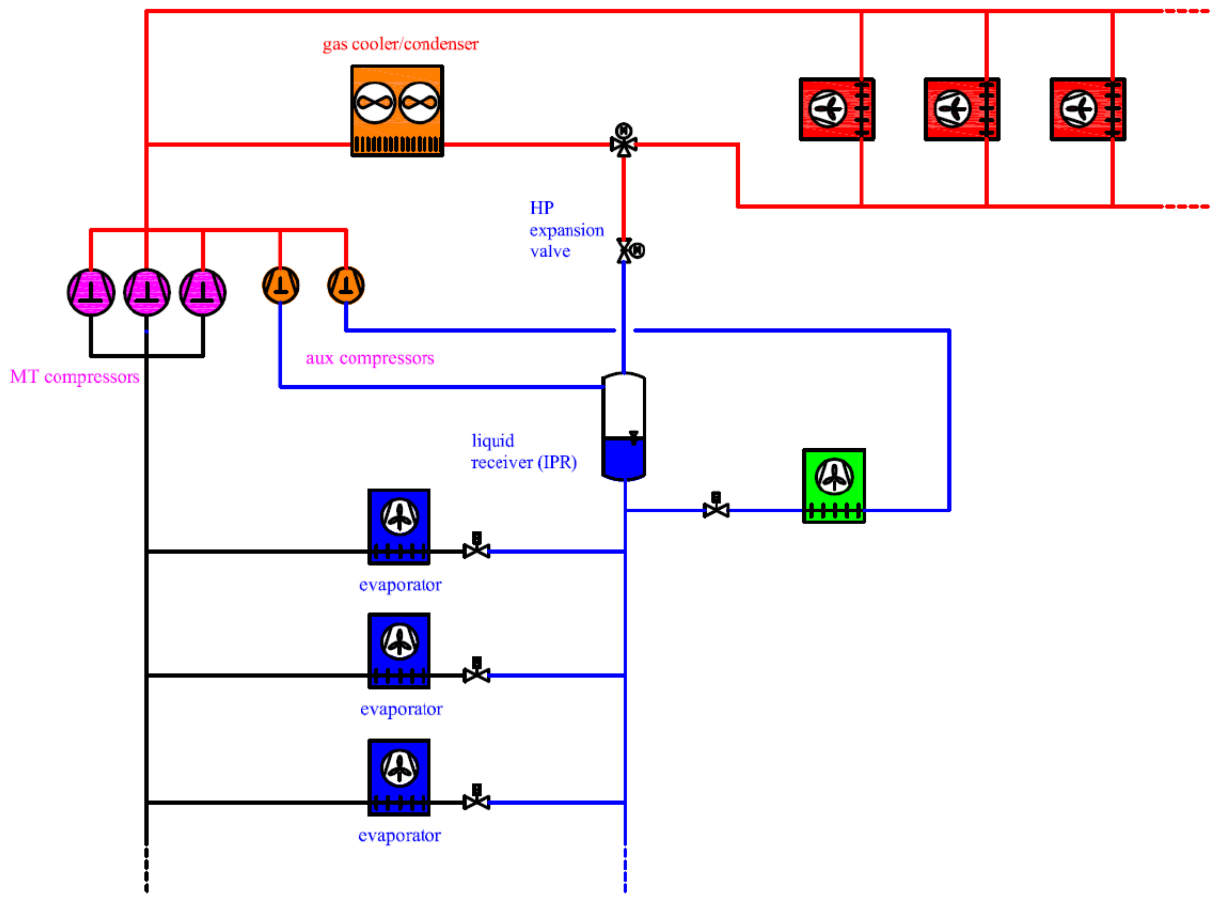

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Figure 19c

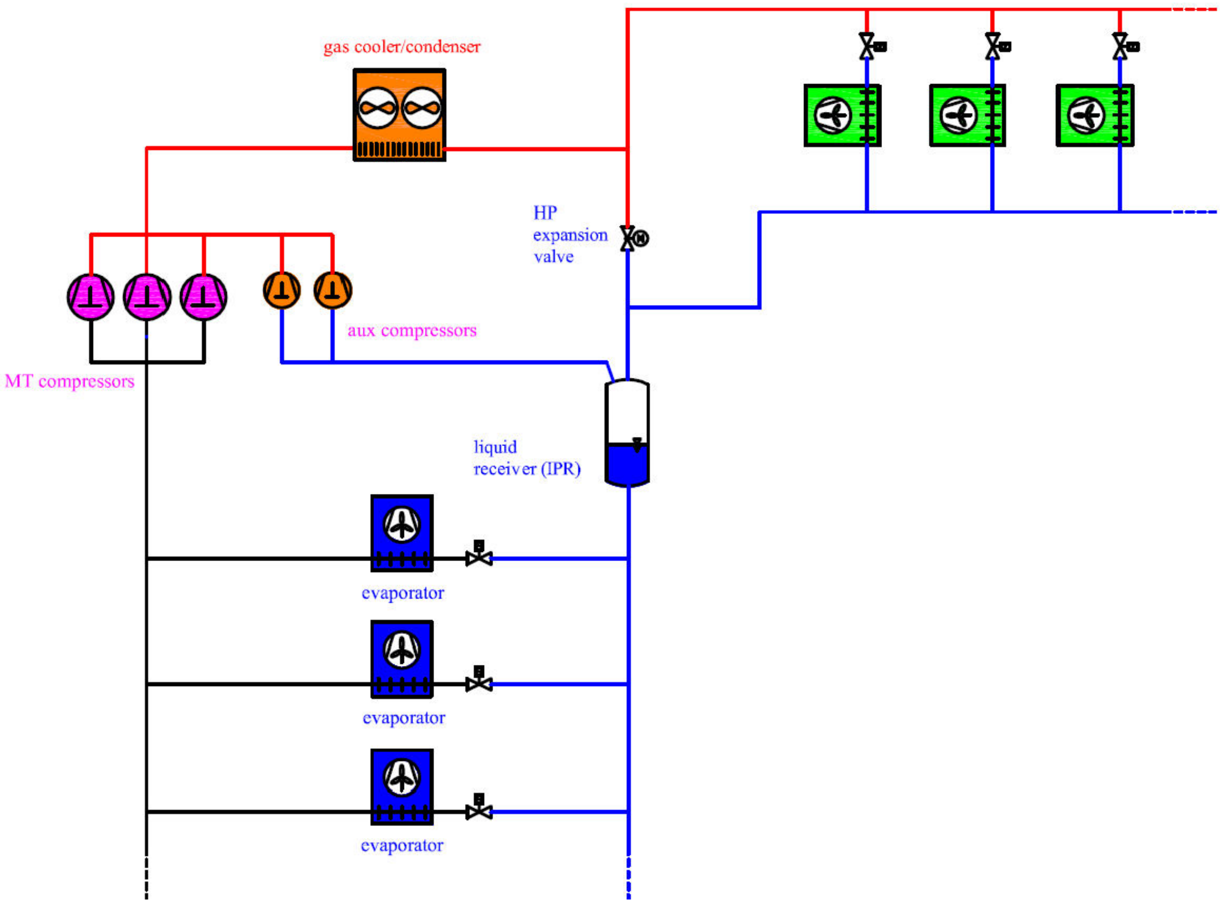

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Figure 20

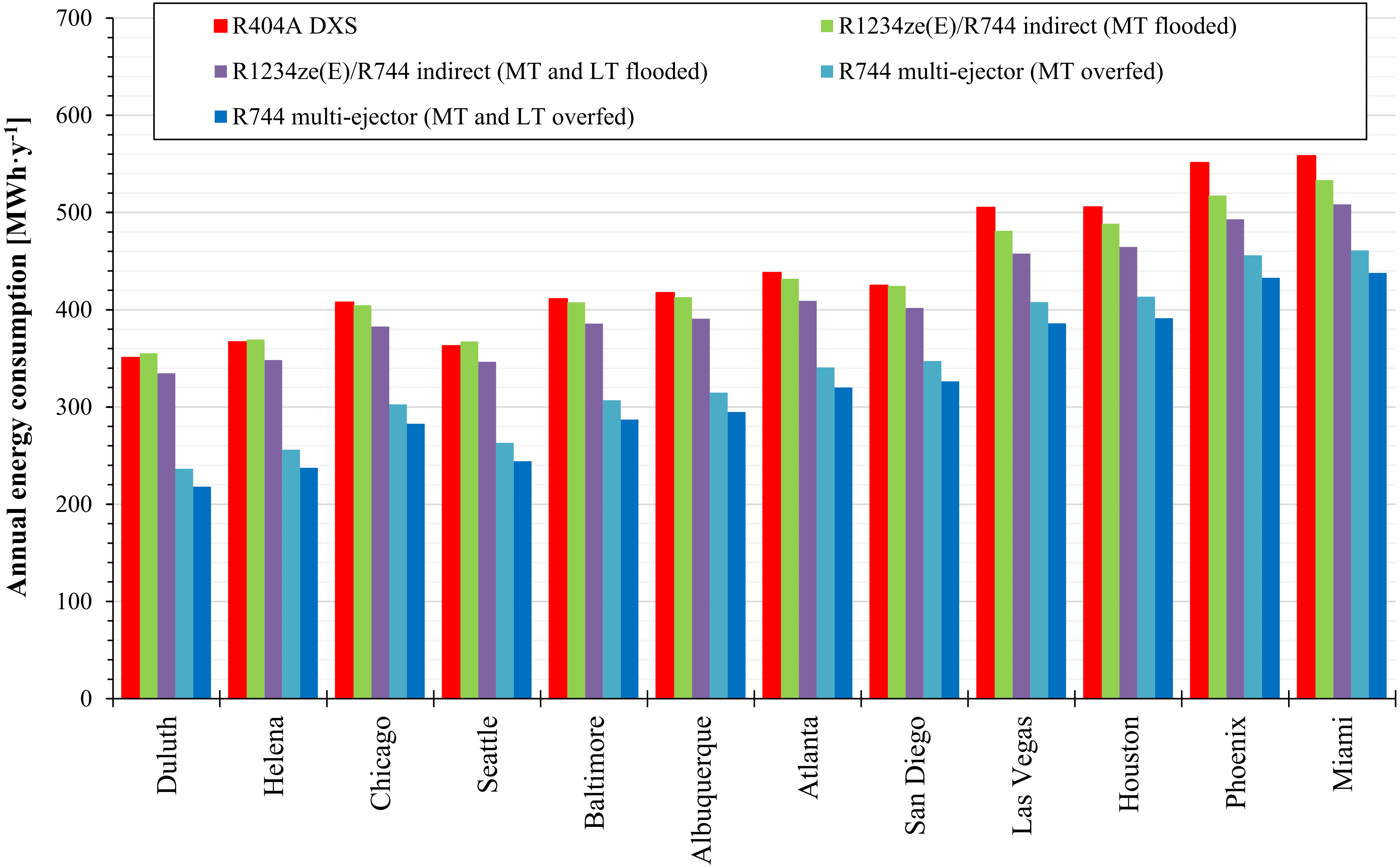

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Figure 21

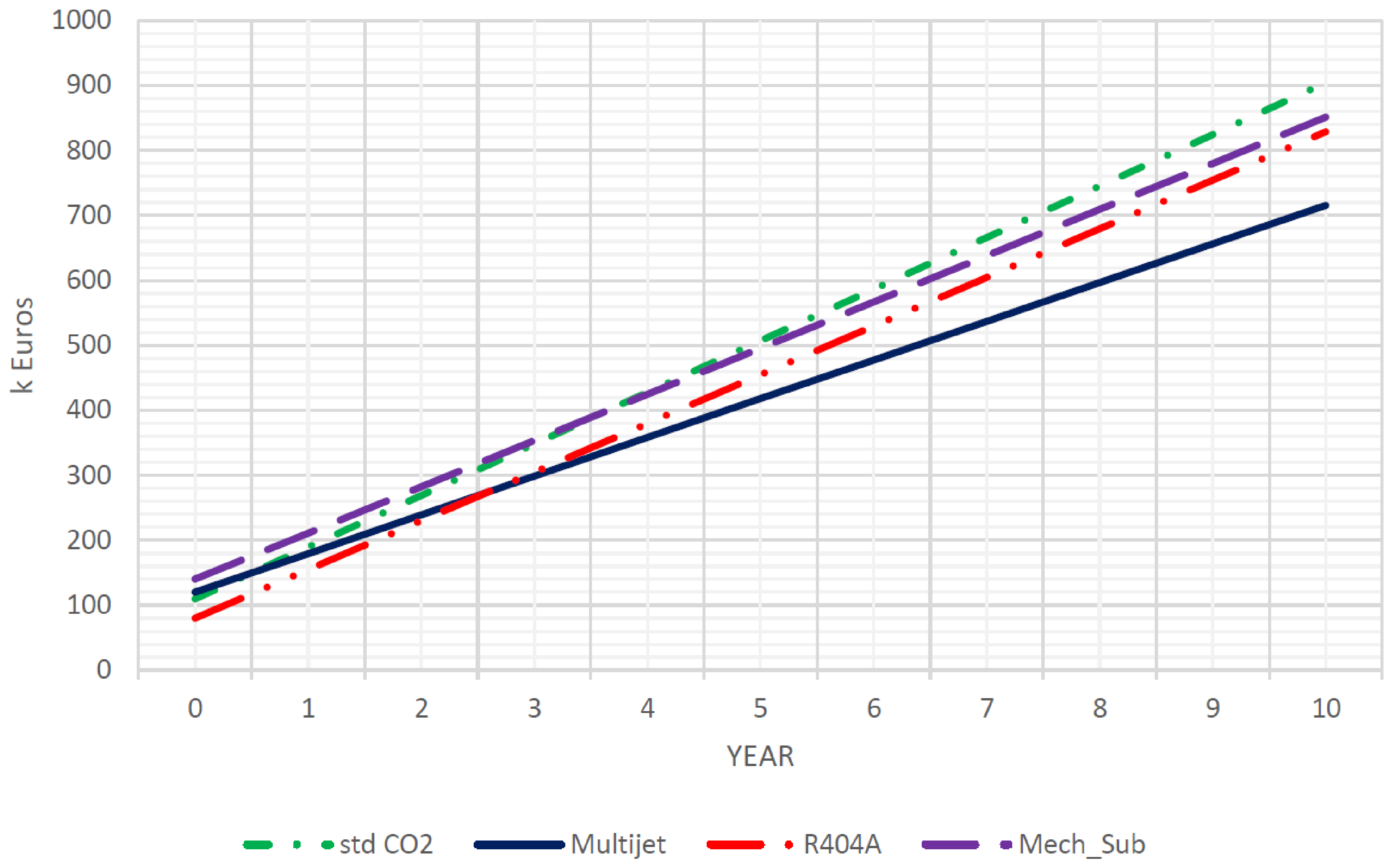

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Table 1

Summary of the findings associated with the main investigations on parallel compression.

Reference	Evaluation	Baseline	Findings
Bell (2004)	Theoretical	-	Existence of optimal intermediate pressure
Minetto et al. (2005)	Experimental	-	Existence of optimal intermediate pressure Proof of feasibility and reliability Solution to oil recapture
Bella and Kaemmer (2011)	Experimental	Vapour injection technique	More reliable solution
Chesi et al. (2014)	Experimental	-	Promising energy performance Energy efficiency strongly depends on liquid separator efficiency Displeasing superheating and pressure drop represent prominent sources of performance deterioration
Chiarello et al. (2010)	Experimental	-	Proof of reliability Design temperature plays a crucial role
Da Ros (2005)	Theoretical	-	Optimal values of COPs, intermediate and heat rejection pressures at gas cooler exit temperatures above 25 °C

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Fritschi et al. (2016)	Numerical model validated against some experimental data	One-stage CO_2 system	Achievement of operating regimes at which a growth in efficiency by at least 10%
Gullo et al. (2017a)	Theoretical	Parallel compression with fixed intermediate pressure	Parallel compression with optimized intermediate pressure performs similarly in Southern European
Gullo et al. (2017b)	Theoretical	R404A system	Energetically advantageous at outdoor temperature up to 27 °C
Gullo et al. (2016a)	Theoretical	Booster system	Reduction in optimal heat rejection pressure on average by 4.2%
Gullo et al. (2015)	Theoretical	One-stage CO_2 system	Average increase in both COP and exergy efficiency by 18.7% at cooling medium temperatures of 30 °C÷50 °C
Javerschek et al. (2016)	Theoretical	-	Optimal intermediate pressure depends on both the load ratio and the condensing/gas cooler outlet temperature
Javerschek et al. (2015)	Theoretical	-	Energy advantages depend on the bin hours per year for which parallel compressor can be employed
		Booster system	Increase in COP by 8.4%÷13.6% at outdoor temperatures 25 °C÷42.5 °C
Minetto et al. (2015)	-	-	Fixed or slightly variable intermediate pressure in real applications

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Nekså et al. (2010)	Theoretical	Expander	More energy beneficial for MT applications at outdoor temperatures 30 °C÷40 °C
Sarkar and Agrawal (2010)	Theoretical	-	Optimal intermediate pressure mainly depends on evaporating temperature More effective technology compared to other “CO_2 only” systems
Sharma et al. (2014a)	Theoretical	R404A system	Energetically competitive at annual average temperatures up to about 14 °C
Visser (2017)	Assertion	R744/NH_3 cascade system	Suitable alternative if combined with an evaporative condenser/gas cooler in Kuala Lumpur
Wiedenmann et al. (2014)	Practical experience	-	Parallel compression design plays a crucial role Ever-decreasing costs and available efficient control systems

Table 2

Summary of the findings associated with the main investigations on the multi-ejector concept.

Reference	Evaluation	Baseline	Findings
Hafner et al. (2016) and Fredslund et al. (2016)	Field measurements	Parallel compression	Enormous unloading of HS compressors
Minetto et al. (2014b)	Assertion	Parallel compression	Reduction in maintenance issues
Gullo et al. (2017b)	Theoretical	R404A system	Higher COPs at outdoor temperatures 0 °C÷40 °C
Hafner, (2017), Banasiak et al., (2015), Hafner et al., (2015), Minetto et al., (2015) and Pardiñas et al., (2017a, 2017b)	Assertion/Theoretical	-	Interchangeability of HS and parallel compressors could significantly enhance the overall energy efficiency
Banasiak et al. (2015), Nekså et al. (2016) and Hafner et al. (2016).	Assertion/Experimental	-	Oil management can be efficiently implemented
Kriezi et al. (2015, 2016)	Experimental	-	Geometrical parameters enormously affect the ejector performance
Haida et al. (2016a)	Experimental	Parallel compression	Enhancements in COP and exergy efficiency up to 7% and 13.7% Parallel compressor rack needs to be adapted to the usage of multi-ejector block
Nekså et al. (2016)	Assertion	Parallel compression	Enhancements by 20% at severe operating conditions

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Fredslund et al. (2016)	Field measurements	-	Close attention has to be paid to the compressor sizes, pressure lift and the oil return design
Bodys et al. (2017)	Experimental	-	High and stable performance for food retail applications
Schönenberger (2016)	Assertion	Parallel compression	Energy saving by 15%÷25 % depending on heat reclaim, application and weather conditions
Hafner et al. (2014c), Hafner and Banasiak (2016), Schönenberger et al. (2014) and Schönenberger (2016)	Field measurements	Booster system	Increase in MT and LT by 6 K and by 8 K in any operating condition Reduction in frost formation and number of defrost cycles

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Table 3a

Summary of the energy benefit assessments associated with the main investigations on the "all-in-one" concept (transcritical R744 supermarket refrigeration systems as baselines).

Reference	Evaluation	Technology	Baseline	Location	Quantification of energy benefit	Parameter related to energy benefit
Javerschek et al. (2016)	Theoretical	Parallel compression with optimized IP	Parallel compression with fixed IP (= 40 bar)	Athens Strasbourg Edinburgh	+4.2% +2.1% +0.6%	SEER
Karampour and Sawalha (2015)	Theoretical	Parallel compression	Booster system	Sweden	Modest enhancement Up to +14%	COP_{tot} in wintertime COP_{tot} in summertime
Hafner et al. (2015) and Hafner (2017)	Assertion	Multi-ejector (equipped with two modules)	-	-	-	Next generation of "CO_2 only" supermarket refrigeration systems will feature two multi-ejector blocks and direct cooling and heating fan coils and air curtains

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Table 3b

Summary of the energy benefit assessments associated with the main investigations on the "all-in-one" concept (supermarket refrigeration systems using synthetic working fluids as baselines).

Reference	Evaluation	Technology	Baseline	Location	Quantification of energy benefit	Parameter related to energy benefit
Karampour and Sawalha (2017)	Simulation models adapted from field measurements	Parallel compression	Separated HFC-based units	Stockholm Paris Barcelona	+11% -9% -12%	Annual energy saving
Karampour and Sawalha (2015)/Karampour and Sawalha (2017)	Theoretical/Simulation models adapted from field measurements	Parallel compression	Separated HFC-based units	-	-	Suitable technology only for cold climates
Danfoss (2016)	Assertion	Multi-ejector	Conventional system	Milan	+50%	Annual energy saving
Cyclone Energy Group (2013)	Field measurements	Parallel compression and ground thermal storage	Conventional system	Evanston	+60%	Annual energy saving
r744.com (2016)	Field measurements	Parallel compression	Conventional system	Conegliano	+10%	Annual energy saving

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Table 4a

Summary of the energy benefit assessments associated with the main investigations on the state-of-the-art R744 technologies for supermarket applications (transcritical R744 supermarket refrigeration systems as baselines).

Reference	Evaluation	Technology	Integration system	Baseline	Location	Quantification of energy benefit	Parameter related to energy benefit
Minetto et al. (2014b)	Theoretical	Multi-ejector	-	Conventional R744 system	Bari	+22.5%	Annual energy saving
Ferrandi and Orlandi (2013)	Theoretical	Cold storage	-	Conventional R744 system	North of Italy	+5% +28% in peak power	Daily energy saving Peak power
Fidorra et al. (2015b)	Theoretical	Parallel compression+CTES at 15 °C	-	Booster system	-	+5.6% +5.9%	Energy saving Total cost reduction
Fidorra et al. (2015b)	Theoretical	Parallel compression+CTES at 0 °C (i.e. ice bank storage)	-	Booster system	-	+3.5% +3.9%	Energy saving Total cost reduction
Tsamos et al. (2017)	Theoretical (validated against laboratory measurements)	Parallel compression	-	Booster system	London Athens	+3.6% +5.0%	Annual energy saving
Hafner et al. (2012)	Theoretical	Multi-ejector	Heat recovery	Parallel compression	Athens	+11.4%	Annual energy saving

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Hafner et al. (2014a)	Theoretical	Multi-ejector	Heat recovery	Booster system	 Athens Frankfurt Trondheim	+20%÷+30% +17% +16% +5%	$COP_{heating}$ in wintertime COP COP COP
Frigo-Consulting LTD (2015)	Assertion	Multi-ejector	Heat recovery	Parallel compression	Timisoara	+13%	Annual energy saving
Schönenberger et al. (2014); Hafner et al. (2014c)	Field measurements	Multi-ejector	-	Parallel compression	Fribourg	+10%	Energy saving in wintertime
Schönenberger et al. (2014); Hafner et al. (2014c)	Field measurements	Multi-ejector	-	Booster system	Fribourg	+18%	Energy saving in wintertime
Leiper et al. (2014)	Field measurements	Geothermal loop	-	Booster system	Locations in the UK	+24.6%	Annual energy saving
Fredslund et al. (2016)	Field measurements	Multi-ejector	AC integration	Parallel compression	Various locations	+10%÷+15% (depending on AC needs)	Energy saving
Hafner et al. (2016)	Field measurements	Multi-ejector	AC integration	Parallel compression	Spiazzo	+15%÷+30% (depending on AC needs)	Energy saving at outdoor temperatures +22 °C÷+35 °C
Pardiñas et al. (2017a)	Theoretical	Multi-ejector (equipped with two modules)	AC integration	Booster system	Outdoor temperature above 15 °C	+7%÷+19% +8.3%	Energy saving

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				Multi-ejector (equipped with a module)	Outdoor temperature: 30 °C		
Pardiñas et al. (2017b)	Theoretical	Multi-ejector (equipped with two modules)	AC integration	Multi-ejector (equipped with a module)	Subcritical running modes Transcritical running modes	Similar performance Optimal heat rejection depends on external temperature, MT, selected system and AC evaporating pressure	-
r744.com (2017d)	Field measurements	Multi-ejector	-	Booster system	Georgia	+11.3% and +15%÷2+3% in non- and optimized operating conditions	Peak energy saving

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Table 4b

Summary of the energy benefit assessments associated with the main investigations on the state-of-the-art R744 technologies for supermarket applications (supermarket refrigeration systems using synthetic working fluids as baselines).

Reference	Evaluation	Technology	Integration system	Baseline	Location	Quantification of energy benefit	Parameter related to energy benefit
Hafner and Hemmingsen (2015)	Theoretical	Mechanical subcooling	-	R404A system	28 cities worldwide	+3%÷+23%	Annual energy saving
Hafner and Hemmingsen (2015)	Theoretical	Multi-ejector	-	R404A system	28 cities worldwide	+11%÷+28%	Annual energy saving
Gullo et al. (2016a)	Theoretical	Parallel compression, Mechanical subcooling, Parallel compression+Mechanical subcooling	-	R744/R134a cascade system	Valencia Athens	Negligible difference	Annual energy consumption
Gullo et al. (2016e)	Theoretical	Mechanical subcooling	-	R404A system	Valencia Athens Seville Rome Lisbon	+3.5% +4.2% +3.9% +5.8% +2.8%	Annual energy saving
Gullo et al. (2017b)	Theoretical	Parallel compression	-	R404A system	Oslo London Frankfurt Milan Athens	+17.9% +13.3% +13.8% +10.9% +3.0%	Annual energy saving
Gullo et al. (2017b)	Theoretical	Multi-ejector	-	R404A system	Oslo London Frankfurt Milan Athens	+31.8%÷+37.1% +27.9%÷+33.2% +28.5%÷+33.6% +26.2%÷+31.2% +19.9%÷+24.6%	Annual energy saving

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Purohit et al. (2017a)	Theoretical	Parallel compression	-	R404A system	Seville Teheran Phoenix New Delhi	+4.3% +6.6% +2.0% +0.8%	Annual energy saving
Purohit et al. (2017a)	Theoretical	Parallel compression	-	R1234ze(E)/R744 indirect arrangements	Seville Teheran Phoenix New Delhi	+2.0%÷+4.7% +3.4%÷+6.0% -4.2%÷-1.5% -6.0%÷-3.4%	Annual energy saving
Purohit et al. (2017a)	Theoretical	Mechanical subcooling	-	R404A system	Seville Teheran Phoenix New Delhi	+5.7% +8.9% +6.3% +5.2%	Annual energy saving
Purohit et al. (2017a)	Theoretical	Mechanical subcooling	-	R1234ze(E)/R744 indirect arrangements	Seville Teheran Phoenix New Delhi	+3.5%÷+6.2% +5.8%÷+8.3% +0.4%÷+2.9% -1.3%÷+1.2%	Annual energy saving
Gullo and Hafner (2017a)	Theoretical	Multi-ejector	-	R404A system	American climate context	+17.3%÷+37.8%	Annual energy saving
Gullo and Hafner (2017a)	Theoretical	Multi-ejector	-	R1234ze(E)/R744 indirect arrangements	American climate context	+7.5%÷+38.6%	Annual energy saving
Polzot et al. (2016a)	Theoretical	Parallel compression+Cold storage	-	R134a/R744 cascade arrangement	North of Italy	Negligible difference	Annual energy consumption
Rehault and Kalz (2012)	Field measurements	Geothermal loop	Heat recovery	Standard food retail store	Location in Germany	+50%	Annual energy saving
Gullo et al. (2017a)	Theoretical	Multi-ejector	AC integration	Separated HFC-based units	Rome Valencia Seville	+22.7%÷+26.2% +20.0%÷+24.0% +15.6%÷+22.7%	Annual energy saving
Kvalsvik et al. (2017)	Theoretical	Multi-ejector (equipped with two modules)	AC integration	Separated R410A-based solutions	Outdoor temperature of 45 °C	-53%	Power input

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Shecco (2015)	Field measurements	Adiabatic condenser	-	R407A system	Dunwoody	+6.3%	Annual energy saving
r744.com (2018b)	Theoretical	Multi-ejector	-	R404A system	Madrid	+15%	Annual energy saving

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