Energy and environmental performance assessment of R744 booster supermarket refrigeration systems operating in warm climates

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Energy and Environmental Performance Assessment of R744 Booster Supermarket Refrigeration Systems operating in Warm Climates

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Abstract:

This paper presents a comparison among different solutions for multi-stage evaporator refrigeration systems for supermarkets in terms of annual energy consumption and environmental impact (Total Equivalent Warming Impact). Eight configurations were studied: a R744/R134a cascade refrigeration system (baseline), a conventional and an improved R744 booster system, two R744 booster solutions with dedicated mechanical subcooling, a R744 booster with parallel compression and two solutions which combined the parallel compression and the mechanical subcooling. The comparison was based on the weather data in Valencia (Spain) and in Athens (Greece).

The results showed that all the enhanced configurations may achieve a comparable energy saving to the one of the baseline in both the selected weather conditions. Furthermore, they entailed benefit in environmental terms reducing the Total Equivalent Warming Impact (TEWI) by at least 9.6% beside the cascade solution.

Keywords:

Annual Energy Consumption, CO_2, Compressor Efficiency, Dedicated Mechanical Subcooling, Parallel Compression, TEWI.

Nomenclature

BRA	British Refrigeration Association
CB	Conventional Booster Refrigeration System
COP	Coefficient of Performance
CS	Cascade Refrigeration System
E	Electrical energy consumption, $kWh \cdot year^{-1}$
EES	Engineering Equation Solver

GWP	Global Warming Potential, $kg_{CO_2} \cdot kg^{-1}_{refrigerant}$
HFC	Hydrofluorocarbons
HP	High Pressure
HS	High Stage
HT	High Temperature
IB	Improved Booster Refrigeration System
IEA	International Energy Agency
L	Annual leakage rate, $kg \cdot year^{-1}$
LS	Low stage
LT	Low Temperature
MS7	Booster Refrigeration System with dedicated Mechanical Subcooling down to 7 °C
MS15	Booster Refrigeration System with dedicated Mechanical Subcooling fixed to 15 °C
MT	Medium Temperature
m	Refrigerant charge, kg
n	System operating life time, year
p	Pressure, bar
PC	Booster Refrigeration System with Parallel Compression
PCMS	Booster Refrigeration System with Parallel Compression and Mechanical Subcooling
t	Temperature, °C
TEWI	Total Equivalent Warming Impact, kg_{CO_2}

Greek symbols

α	Recycling factor, %
β	Indirect emission factor, $kg_{CO_2} \cdot kWh^{-1}$
Δ	Difference
η	Efficiency
χ	Amount of saturated vapour sucked by the auxiliary compressor, $kg \cdot s^{-1}$

Subscripts and superscripts

appr	Approach
AUX	Auxiliary compressor
cc	Cooling Capacity
cond	Condenser
el	Electrical
ext	External
gc	Gas cooler
global	Global
int	Intermediate pressure
MAX	Maximum
out	Outlet
SUB	Subcooler

1. Introduction

R744 is a natural refrigerant which is capable of providing comparable or even better performance than synthetic refrigerants when employed in subcritical conditions. From the thermo-physical point of view, it shows lower viscosity and higher latent heat, thermal conductivity, density, volumetric cooling capacity and specific heat than HFCs (Ge and Tassou, 2011). The interest in carbon dioxide as refrigerant has been increasing rapidly in the last few years thanks to its advantageous thermo-physical properties, non-flammability, non-toxicity and negligible Global Warming Potential (GWP). Due to its low critical temperature (30.98 °C), transcritical operations take place as soon as the outdoor temperature exceeds a threshold level which can be as low as 15 °C (Girotto et al., 2004). As a main consequence, high exergy destruction rates can be associated with the expansion valve of a CO_2 transcritical machine, which cause a large decline in its performance. As proposed by Fazelpour and Morosuk (2014), a solution to reduce such irreversibilities is to utilize an economizer downstream of the gas cooler. This concept is, in some extent, similar to the one of using a mechanical subcooling loop presented in this paper. Furthermore, in such operating conditions an optimal value of high pressure, which maximizes the Coefficient of Performance (COP), has to be identified as a function of the gas cooler outlet temperature (Kim et al., 2004).

As regards the employment of the carbon dioxide in warm climates, on the one hand R744 is being widely utilized in the low temperature circuit of cascade systems in which it can perform in the most favourable conditions. On the other hand, its use as main refrigerant at high external temperatures still needs to be further studied, evaluating all the aspects which involve thermodynamic, environmental and economic analyses.

In spite of these drawbacks, CO_2 is a promising working fluid in different applications such as commercial refrigeration systems, which exhibit both indirect and direct environmental impacts. The indirect ones are mainly attributed to the generation of the electricity used to run the system under consideration. Since CO_2 performs much worse than synthetic refrigerants in warm weathers, it is important to identify any improvements to accomplish at least similar energy consumption. The solutions to this issue include both the aspects of energy saving (e.g. closed display cabinets) and of energy efficiency (improvement of booster systems, condenser fans with variable speed motors, etc.). As far as the direct emissions are concerned, carbon dioxide can reduce them drastically since its GWP is negligible.

Cascade refrigeration systems are efficient solutions even in warm climates. In such configurations, R744 can operate in subcritical conditions when it is used in the low stage circuit and it allows decreasing energy consumption of pumps and pipes size, as well as it features good heat transfer properties (Ge and Tassou, 2011). The refrigerant flowing in the high stage can be chosen among different fluids, both natural (R717, R290, R1270) and synthetic ones (R134a, R404A). Due to the flammability and/or the toxicity of the former and to the high GWP of R400 series, R134a is a good candidate to be used as primary refrigerant of cascade systems with cooling capacity over 40 kW (European Commission, 2014). Nevertheless, it is worth remarking that European Union pushes towards the use of natural refrigerants and that many countries have levied taxies on HFCs purchase (e.g. Spain) and/or promoted the spread of eco-friendly solutions (e.g. Belgium).

1.1. Literature review

Ge and Tassou (2011) carried out a sensitivity analysis of the main parameters which affect the performance of a R744 booster cycle over the range of outdoor temperatures from 25 to 40 °C. The results showed that the optimal gas cooler pressure can be determined as a function of the isentropic efficiency of the high stage compressors rack, the outdoor temperature and the effectiveness of the internal heat exchanger located downstream of the gas cooler. In particular, in order to improve the COP, both a high isentropic efficiency and a high effectiveness of the previously mentioned heat exchanger are required, while the intermediate pressure should be kept as low as possible.

A comparison in terms of both annual energy consumption and costs between commercial refrigeration systems using carbon dioxide and direct expansion solutions using R404A was made by Girotto et al. (2004). They concluded that the solution using the natural refrigerant has 10% higher energy consumption than the other one when both of them are run in the North Italy. The higher consumption can be attributed to the cooling load at medium temperature (MT) since the system at low temperature (LT) points out similar consumption to the one of an equivalent R404A system. The authors also proposed some possible enhancements, such as decreasing the approach temperature of the gas cooler, using two-stage compression for the MT system and sucking the vapour from the liquid receiver.

As for the possible improvements of the conventional refrigeration system, several solutions have been suggested. One of the most interesting systems is the adoption of a dedicated mechanical subcooling. Thornton et al. (1994) proved the existence of an optimal subcooling temperature which is mainly affected by the temperature at which heat of condensation is rejected into the external heat sink and the one of the heat source from which vaporization heat is extracted.

The experimental investigation by Qureshi and Zubair (2013) showed experimentally that a subcooling loop can improve both the cooling capacity and the exergetic efficiency of a basic system. Llopis et al. (2015a) evaluated the potential enhancements which could be fulfilled by utilizing a subcooling cycle on the part of both a conventional CO_2 system and a CO_2 unit with two-stage compression with intercooling. The former operated at evaporating temperatures of 5 °C and -5 °C, whereas a low temperature equal to -30 °C was chosen for the latter. In both cases the outdoor temperature ranged from 20 °C to 35 °C. The authors studied the use of R290, R1270, R1234yf, R161, R152a and R134a as refrigerant for the subcooler loop concluding that the achievable improvements are independent of the selected working fluid. At the outdoor temperature of 30 °C and with a degree of subcooling of 5 °C, the use of R134a allows increasing COP by 9.5% at 5 °C, by 13.7% at -5 °C and by 13.1% at -30 °C.

Hafner and Hemmingsen (2015) estimated an energy saving varying from 11% to 28% related to a R744 unit with a R290 dedicated mechanical subcooling loop beside a R404A direct expansion system. The evaluation was attained taking into account weather conditions in several cities located all over the world.

Parallel compression is another propitious solution which allows enhancing the performance of a conventional CO_2 refrigeration system by compressing a part or the entire amount of vapour generated in the liquid receiver from the intermediate pressure to the high one.

Sarkar and Agrawal (2010) claimed that the parallel compression is a more effective solution to enhance the efficiency of a single-stage CO_2 cycle than both the configuration with two-stage compression and flash gas by-pass and the one with parallel compression and subcooler. The assessment was fulfilled considering gas cooler outlet temperatures from 30 to 60 °C and the evaporating temperatures of -45, -20 and 5 °C. They also proved the existence of an optimal intermediate pressure which affects the performance of the overall system markedly. Furthermore, the outcomes highlighted that such variable is mainly influenced by the evaporator temperature, concluding that the use of the auxiliary compressor is more beneficial at very low values of the latter.

The importance of the intermediate pressure as a key parameter of the optimization procedure was also underlined by Chiarello et al. (2010) and by Minetto et al. (2005). The former also performed an experimental campaign and implemented a theoretical model in order to demonstrate the obtainable enhancements. The experimental apparatus, even though it showed interesting results, was mainly run in subcritical conditions.

A test rig was built by Minetto et al. (2005) to prove the feasibility of this solution and the possibility to overtake the technological issues associated with it, such as the lubricant oil return. The authors also showed theoretically that a CO_2 booster system with parallel compression can accomplish good results in terms of both cooling capacities and COPs in comparison with a conventional one. They claimed that the intermediate pressure can be controlled by a variable-frequency drive as a function of the swept volume ratio of the compressors.

Da Ros (2005) compared a conventional single stage R744 system operating at the evaporating temperature of -10 °C with a solution with parallel compression and with the one with two-stage compressor. The author took into account both constant and freely variable pressure difference between the gas cooler and the liquid receiver. In all the evaluated cases, the system with the two-stage compressor performs better than the other ones.

An alternative to the system with auxiliary compressor is the one which uses a single compressor and vapour injection at intermediate pressure. This solution was compared experimentally to that with dedicated auxiliary compressor by Bella and Kaemmer (2011). The assessment was conducted ranging the intermediate and the high pressures and keeping the evaporating temperature equal to -10 °C. They found out that even though this solution is able to achieve good efficiencies, the system with dedicated compressor is preferable in terms of both evaluated COP and of practical constraints, such as vibrations and ease to control the intermediate pressure.

The annual energy consumption evaluation for a supermarket located in Bari (Italy) carried out by Minetto et al. (2014) points out that the solution with parallel compression can attain a large energy saving in comparison with a conventional system.

A theoretical model was implemented by Chesi et al. (2014) to find out the best operating conditions as well as the effect of liquid separator efficiency, intermediate pressure and compressors volumetric flow ratio on the performance of the overall system. The authors asserted that the largest enhancements in cooling capacity arise at gas cooler pressures different from those at which maximum COPs occur. An experimental investigation was also implemented considering different high pressures, gas cooler outlet temperatures and evaporating temperatures in order to evaluate the main parameters which influenced the system under investigation. Both the analyses confirmed that the use of an auxiliary compressor can improve the performance of a conventional CO_2 refrigeration system. Pressure drop in the suction line, unintentional superheating and a faulty separation between vapour and liquid within the receiver can strongly worsen the performance of the real cycle.

Gullo et al. (2015) compared from the energetic, exergetic and exergoeconomic point of view a system with parallel compression with a basic solution. The analyses were based on a system with cooling capacity of 100 kW. The cooling medium temperatures were varied from 30 to 50 °C, whereas the evaporating temperature was kept equal to -10 °C. The outcomes showed that the improved system can reduce the energy consumption on average by 18.7% leading to an increase in total purchased equipment cost by 23.4% beside the single-stage refrigeration system. Furthermore, a decrement in the irreversibilities of the throttling valve of about 50% was also reached.

Hafner et al. (2014a) made a comparison among different R744 refrigeration configurations operating in warm climates. The evaluated running modes were: evaporating temperature of – 6 °C at the cooling capacity of 25 kW, -8 °C at 50 kW and -10 °C at 100 kW. As regards the solution with dedicated mechanical loop, two different approaches were considered. In the first approach, the R744 subcooler outlet temperature was fixed to 20 °C, whereas in the second one the subcooler was capable of providing a cooling capacity of 30 kW. In the case with evaporator cooling capacity equal to 100 kW, the configuration with fixed subcooler outlet temperature along with the solution which combines the parallel compression and the ejector show the highest COPs over the outdoor temperatures range varying from 30 to 42 °C. The solutions with liquid receiver and auxiliary compressor presents slightly lower COPs than those of the solution with flash gas and parallel compression in all the evaluated cases. Only transcritical operations and single-stage evaporation systems were taken into account.

A comparison in terms of annual energy consumption for four different locations in China was presented by Hafner et al. (2014b). It can be concluded that a system with dedicated mechanical subcooling can perform better than a R404A system even in warm weathers achieving an energy saving by 16%.

Polzot et al. (2015) compared the performance of three different booster refrigeration solutions with a R744/R134a cascade refrigeration system. The authors studied a conventional booster solution and a solution using a water tank as a cold storage with and without parallel compression. The simulation was carried out considering the weather trend in Genoa (Italy), which is characterized by a mild climate. The results obtained suggested that comparable energy consumption with that of the cascade refrigeration system can be accomplished by employing an auxiliary compressor.

An experimental campaign was conducted by Sawalha et al. (2015) in five Swedish supermarkets. The outcomes underlined that large improvements can be realized by using a subcooler and by removing the flash gas generated in the intermediate pressure liquid receiver. Further enhancements can be reached by reducing external superheating and by increasing both the evaporating temperatures and the compressors overall efficiency.

Sanz-Kock et al. (2014) collected experimental measurements regarding a R134a/CO_2 cascade refrigeration system varying the evaporating temperature from -40 to -30 °C and the condensing temperature from 30 to 50 °C, respectively. The experimental setup, which had an air-cooled heat exchanger downstream of the low stage (LS) compressor to cool CO_2 down when outdoor conditions allowed it, exhibited cooling capacities from 4.5 to 7.5 kW. Several parameters were taken into account, such as performance of the compressors, temperature difference in the cascade condenser, cooling capacity and COP. At constant LS evaporating temperature of -30 °C, an average decrement in COP by 18% is obtained for each increase in the condensing temperature of the high stage (HS) circuit by 10 °C. On the other hand, keeping HS condensing temperature at 40 °C, the COP brings down on average by 12% for each decrease in the LS evaporating temperature by 5 °C. Furthermore, the authors found out that the LS condensing temperature does not influence COP significantly. Cooling capacity was observed to be more dependent on LS evaporating temperature than on HS condensing temperature. The experimental results by Souza et al. (2015) confirmed that COP of such solution goes up with decrease in the CO_2 compressor operating frequency.

The main target of this paper is to follow up the energy improvements that a R744 booster refrigeration system may fulfil by adopting a dedicated mechanical subcooling and/or an auxiliary compressor. A conventional CO_2 booster system, an improved one with reduced gas cooler approach temperature, two CO_2 booster system with dedicated mechanical subcooling, a booster with parallel compression and two CO_2 booster solutions which combine the parallel compression and a subcooling loop have been investigated. The annual energy consumption and the environmental impact in terms of TEWI of all the previously mentioned configurations have been compared with those of a CO_2/R134a cascade system (baseline), considering the weather trends in Valencia (Spain) and in Athens (Greece). In section 2 the methods including modelling approach and assumptions are presented, whereas the results are shown in the following section. Finally, the computed outcomes are compared with the other works and the main conclusions are stated in section 4.

1.2. Investigated solutions

A cascade system consists of two circuits (LS and HS) which interact with each other by using the so-called cascade condenser. The vaporization of the refrigerant in the high temperature stage takes place as a consequence of the heat rejection into the refrigerant flowing in the LS loop. This means that the cascade condenser operates as a condenser for the LS circuit and as an evaporator for the HS one, with an appropriate temperature difference between the two selected working fluids. In the case of commercial refrigeration plants, the HS loop has some additional evaporators which are used to cool down chilled food display cabinets and cold rooms.

In this paper, a cascade refrigeration system (Fig. 1) in which R134a and R744 flow respectively through HS and LS circuit was considered as baseline. In such circumstances, carbon dioxide always operates in subcritical conditions.

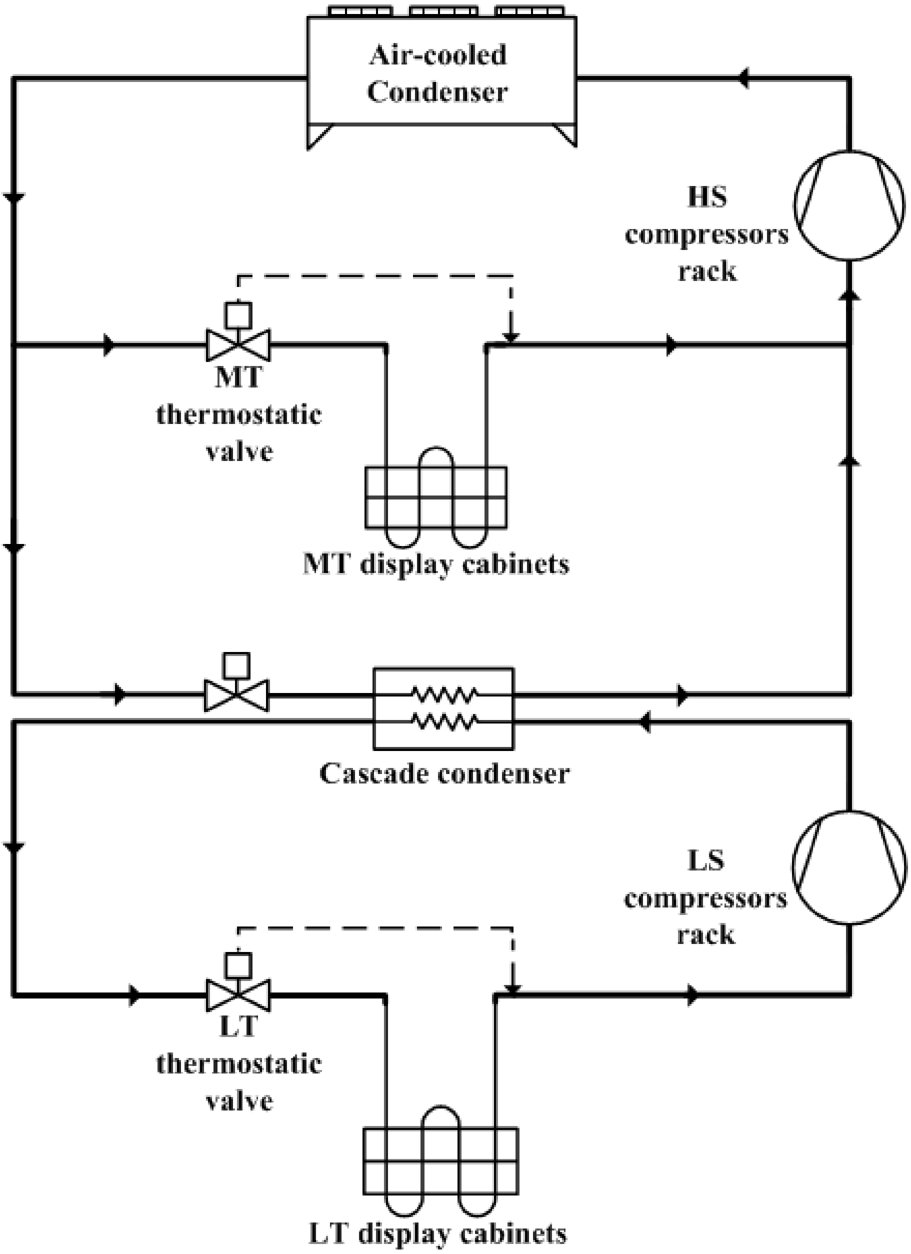

Fig. 1 - Schematic of a cascade refrigeration system.

Unlike the cascade refrigeration system, R744 booster system uses carbon dioxide in both MT and LT display cabinets. A low stage compressors rack behaves as a booster in order to rise the refrigerant pressure from the low pressure to the medium one. The heat exchanger at high pressure acts as condenser at low outdoor temperatures, otherwise it operates as gas cooler in transcritical conditions. In the latter case, ingoing CO_2 undergoes a reduction in temperature without involving no phase change. The outgoing fluid from the condenser/gas cooler is throttled and split into vapour and liquid within the liquid receiver. Vapour is thus throttled to the medium evaporating pressure and compressed along with the mass flow rates exiting LS compressors and MT display cabinets to the high pressure.

Fig. 2 refers to a typical layout of a conventional booster system for commercial refrigeration applications.

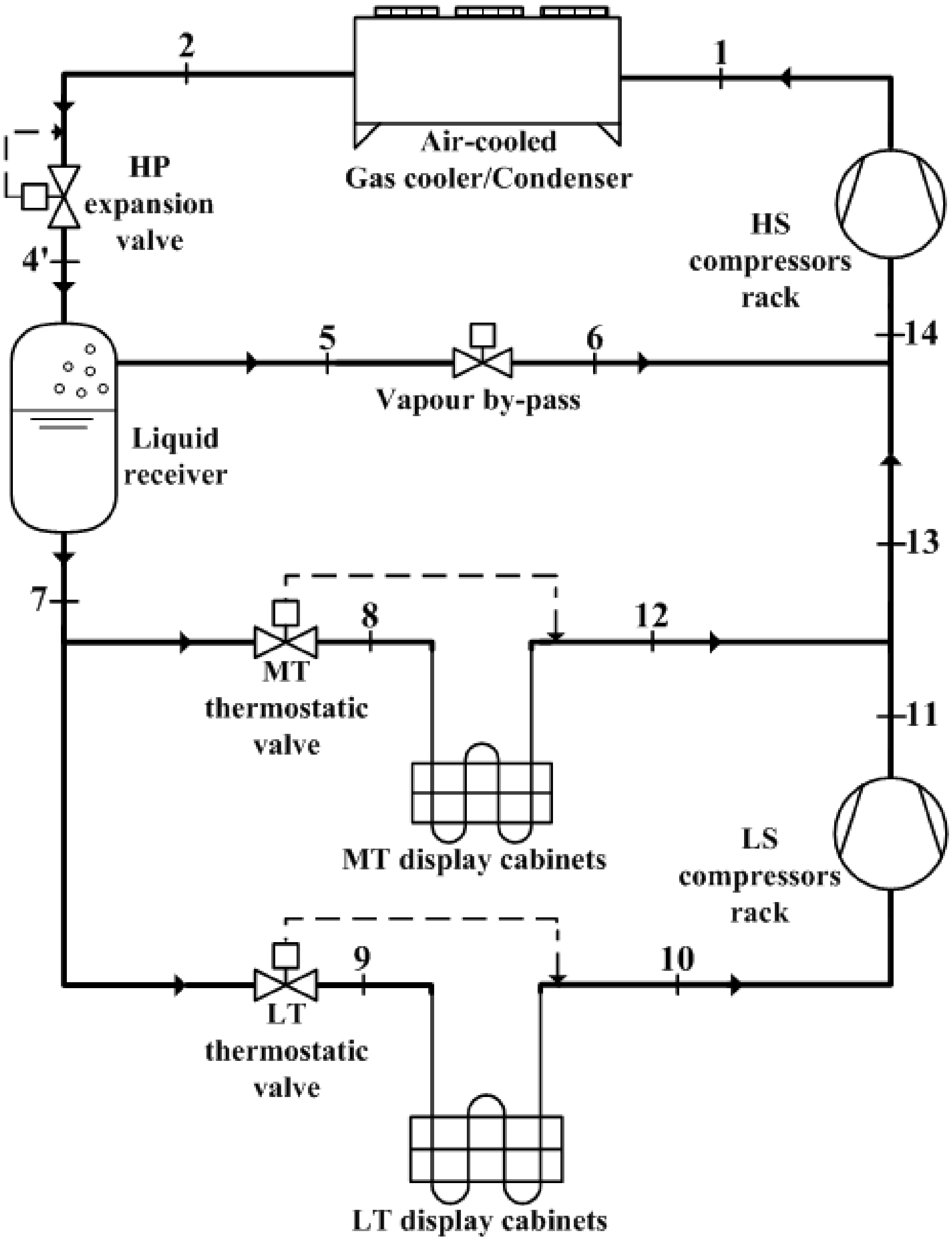

Fig. 2 - Schematic of a conventional R744 booster refrigeration system.

The configuration with dedicated mechanical subcooling represents an interesting solution in order to drop the energy consumption of a R744 refrigeration systems which are run in warm climates (Hafner et al., 2014a; Hafner et al., 2014b; Llopis et al., 2015a). Such solution, which is schematised in Fig. 3, includes a vapour-compression refrigeration unit cooling CO_2 down after the gas cooler/condenser. This leads to a decrease in the refrigerant quality entering the liquid receiver and thus to an enhancement in performance. Furthermore, the optimal gas cooler pressure is also reduced in comparison with the conventional configuration allowing to bring down the energy consumption of the overall system. In the present paper, a mechanical subcooling loop using R290 as refrigerant was taken into account, which began operating as soon as transition conditions took place.

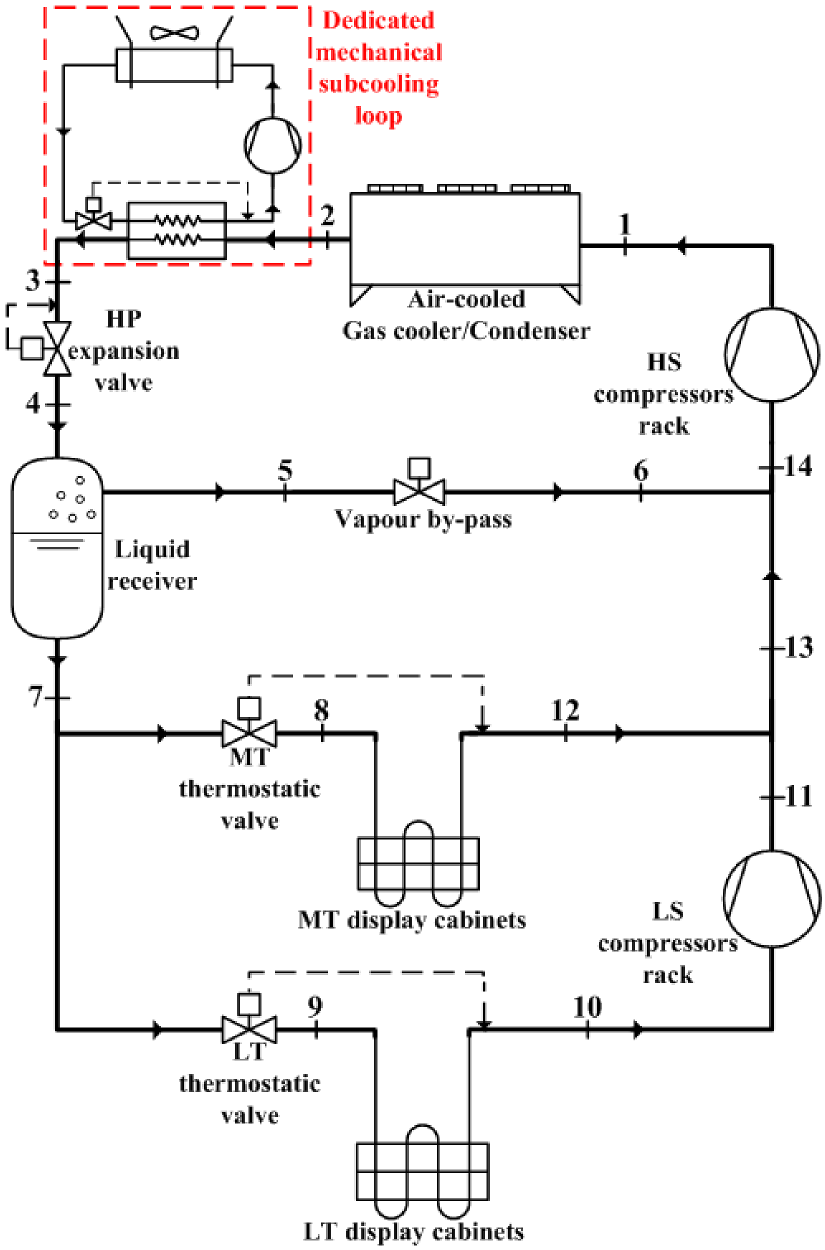

Fig. 3 - Schematic of R744a refrigeration system with dedicated mechanical subcooling.

Fig. 4 compares a conventional booster system (dotted line) with a booster solution with dedicated mechanical subcooling (solid line) in a *log(p)-h* diagram. The difference is represented by the thermodynamic transformation indicated as 2 – 3, which stands for the subcooling process before CO_2 enters the high pressure (HP) expansion valve (3 – 4). On the contrary, the refrigerant leaving the gas cooler/condenser is directly throttled (2 – 4') in a conventional booster solution.

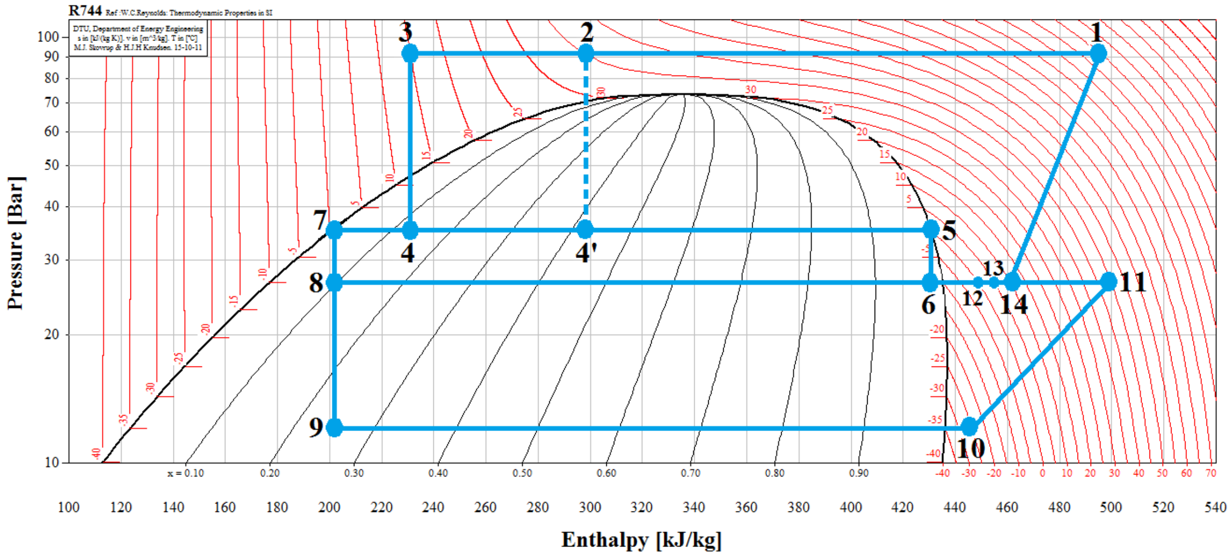

Fig. 4 - Log(p)-h diagram of R744 booster refrigeration systems with and without dedicated mechanical subcooling.

The amount of vapour which is produced due to the HP expansion valve increases as the external temperature rises. This leads to an increment in the amount of refrigerant which needs to be compressed from the medium pressure to the high one, implying a growth in the energy consumption associated with the high stage compressor (especially during summer time). Fig. 5 shows this phenomenon taking into account the case of the "improved booster" (IB) underlining as, in transcritical operations, the mass flow rate of the saturated vapour was on average equal to 45% of the total mass flow rate. It also pointed out that both the total and the vapour mass flow rate increased suddenly at temperatures over 38 °C due to the high gas cooler outlet temperature and thus to the high quality of the refrigerant entering the liquid receiver. The amount of vapour became equal to 50% of the refrigerant total mass flow rate at the external temperature of 40 °C.

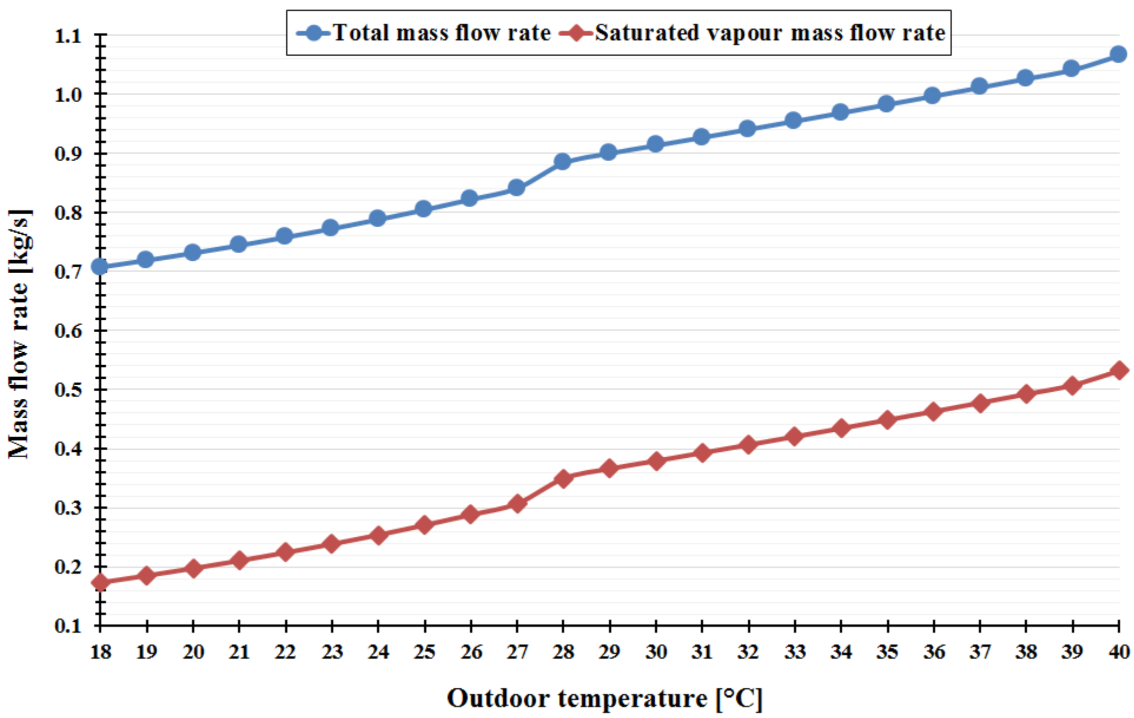

Fig. 5 - Total mass flow rate and vapour saturated mass flow rate of R744 improved booster refrigeration system (IB) operating in transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

A solution to this problem is to adopt an additional compressor (Fig. 6) on the purpose of sucking either a part or the entire amount of vapour from the intermediate pressure liquid receiver to the gas cooler pressure. As suggested by Chiarello et al. (2010), Minetto et al. (2005) and Sarkar and Agrawal (2010), in this case the overall system has to be optimized in terms of both the gas cooler pressure and the intermediate one.

In this paper, in order to guarantee a suitable minimum mass flow rate for the auxiliary compressor, the latter was run as soon as transition conditions were fulfilled. The thermodynamic cycle of a booster system with parallel compression is described in l*og(p)-h* diagram in Fig. 7.

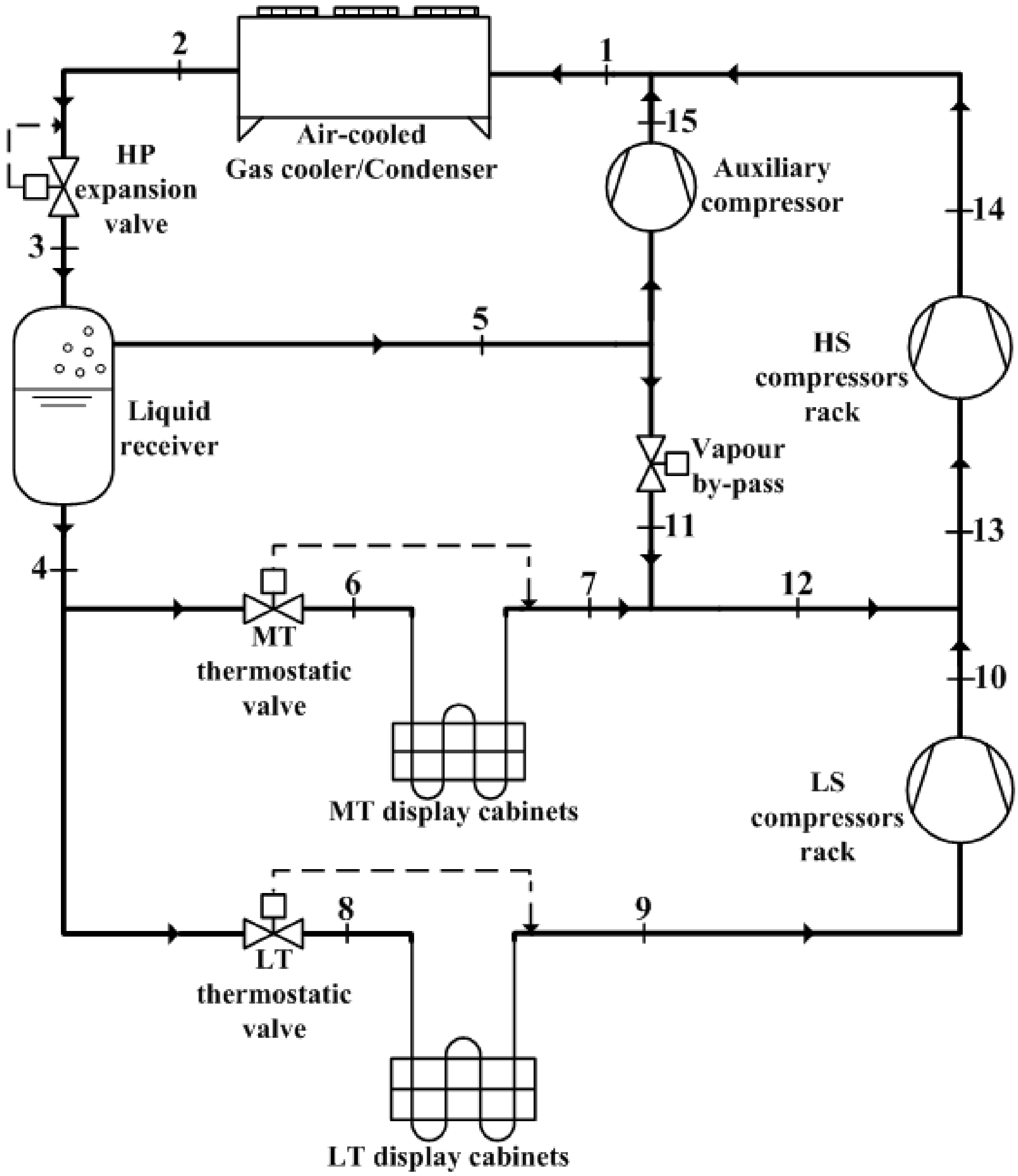

Fig. 6 - Schematic of a R744 booster refrigeration system with parallel compression.

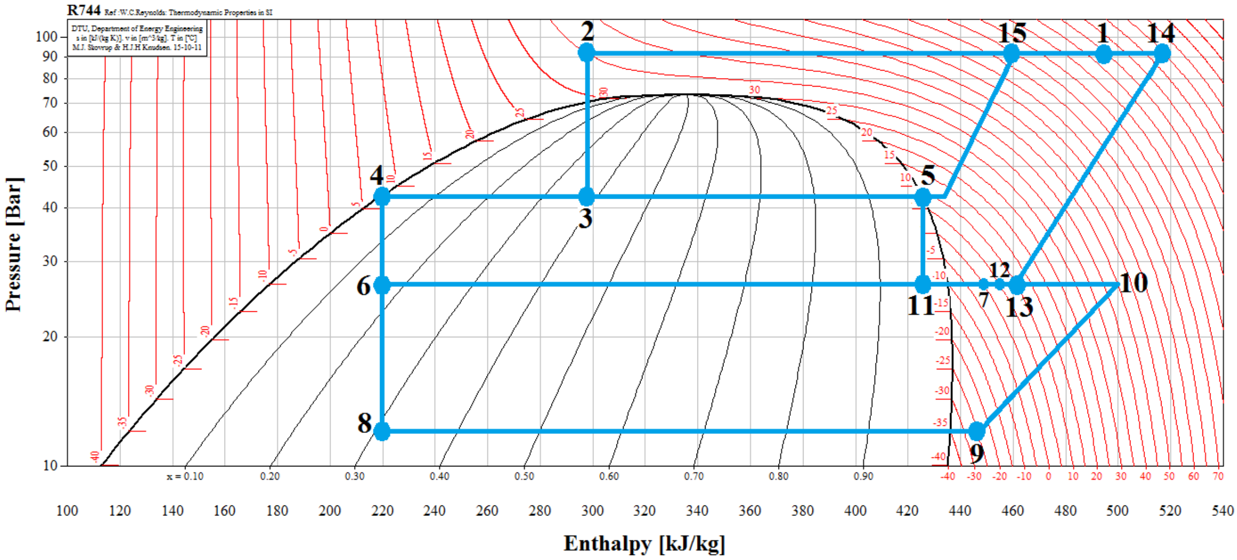

Fig. 7 - Log(p)-h diagram of a R744 booster refrigeration system with parallel compression.

The use of a mechanical subcooler loop allows lessening the energy consumption of a conventional CO_2 refrigeration machine. This solution can be combined with the parallel compression in order to derive the benefits associated with both systems starting from the transition conditions (Fig. 8). It is worth underling that such system acted in the same way as a conventional booster configuration in subcritical operations.

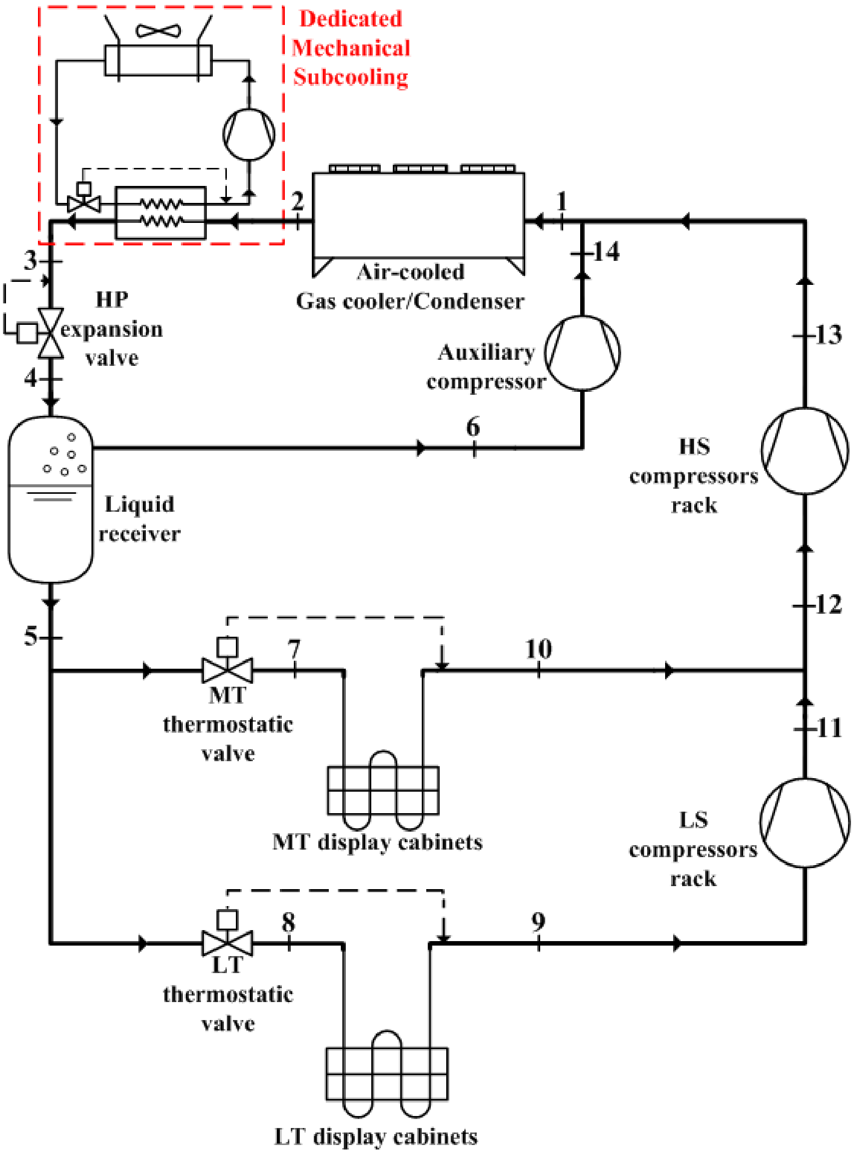

Fig. 8 - Schematic of a R744 booster refrigeration system which combines the parallel compression and the mechanical subcooling.

Log(p)-h diagram of the previously mentioned solution is represented in Fig. 9.

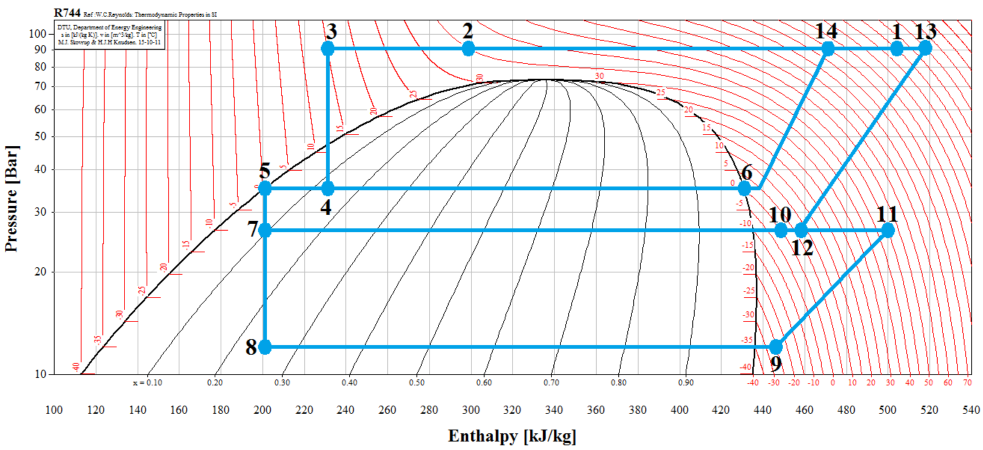

Fig. 9 - Log(p)-h diagram of a R744 booster refrigeration system which combines parallel compression and mechanical subcooling.

2. Methods

2.1. Case studies

The calculations were based on the conditions of a typical European supermarket, as presented by EMERSON Climate Technologies (2010). The common running modes for all of the evaluated solutions are summarized in Table 1. In EMERSON Climate Technologies (2010), LT was set to -35 °C for the R404A centralized direct expansion solution (baseline) and to -32 °C for the booster system. The reason for this difference lies in the better CO_2 performance during the vaporization process than that of R404A (Girotto et al., 2004). Since the estimation of these benefits is quite difficult, the authors preferred considering operating conditions more general, which were represented by R404A centralized direct expansion solution for LT and by the cascade system for MT. Furthermore, the superheating was split into two parts: useful superheating, which took place within the evaporators, and the external one, which arose in the suction line.

Table 1 – Operating conditions in common with all the evaluated systems.

MT load	97	kW
LT load	18	kW
MT	-10	°C
LT	-35	°C
Useful superheating	5	K
External superheating	5	K
Condenser/Gas cooler fan power	4.5	kW_{el}
MT evaporator fans, lights, defrost	10	kW_{el}
LT evaporator fans, lights, defrost	4	kW_{el}

Table 2 shows the specific running modes for the different systems under investigation and the acronym used from now on to indicate the different selected solutions. The approach temperature in a heat exchanger can be defined as the difference between the temperature of the outgoing hot fluid and the one of the ingoing cold fluid. An “improved” booster (IB) solution was proposed in addition to the conventional configuration to investigate the effect of the reduction in the approach temperature of the gas cooler/condenser.

Table 2 – Specific operating conditions of the evaluated systems.

	Cascade refrigeration system (CS)	Conventional R744 booster refrigeration system (CB)	Improved R744 booster refrigeration system (IB)	R744 booster with subcooler outlet temperature varying down to 7 °C (MS7)	R744 booster with subcooler outlet temperature equal to 15 °C (MS15)	R744 booster with parallel compression (PC)	R744 booster with parallel compression and dedicated mechanical subcooling using R290 (PCMS290)	R744 booster with parallel compression and dedicated mechanical subcooling using R1270 (PCMS1270)
Minimum condensing temperature [°C]	25	10	9	9	9	9	9	9
Approach temperature of the condenser [K]	10	10	3	3	3	3	3	3
Approach temperature of the gas cooler [K]	-	5	2	2	2	2	2	2
Temperature difference of the cascade condenser [K]	5	-	-	-	-	-	-	-
Intermediate pressure [bar]	-	35	35	35	35	Optimized	Optimized	Optimized

The following additional assumptions were made:

1. pressure drop was neglected;
2. refrigerant of the subcooling loop underwent an internal superheating of 5 K and condenser fan power of the mechanical subcooling loop was set to 1 kW;
3. selected working fluids for the subcooling loop were R290 for MS7, MS15 and PCMS290 and R1270 for PCMS1270;
4. approach temperature of the condenser of the mechanical subcooling loop added up to 8 K;
5. approach temperature of the evaporator of the mechanical subcooling loop for MS7 amounted to 5 K;
6. with respect to MS15, PCMS290 and PCMS1270, it was assumed that the compressors of the respective subcooling loop operated at as high evaporating temperature as possible in accordance with their corresponding operating envelope;
7. no heat transfer into the surroundings was considered.

2.2. Simulation model

Models of the cycles were implemented in Engineering Equation Solver (EES) (F-Chart Software, 2015a), which were based on the mass and energy balance, as well as the components relations.

As previously explained, a CO_2 booster refrigeration system operates in subcritical conditions when outdoor temperature is low enough, whereas transcritical operations occurs at high external temperatures. In order to improve its performance, it is necessary to define a transition zone which arises at intermediate outdoor temperatures and which is dependent on the ability of the condenser to reject heat into the surroundings. A procedure similar to the one adopted by Cecchinato et al. (2007) was used to determine the conditions in which transcritical operations took place. As sketched in Fig. 10, four operational zones were defined. Zone I referred to the subcritical conditions in which, independently of the external temperature, the condensing temperature was kept equal to its minimum value, in accordance with Table 2. As a result of this, the energy consumption was constant.

The shift from Zone I to Zone II depended on the approach temperature of the condenser and on the outdoor temperature. Since in both Zone I and Zone II a degree of subcooling of 2 K was selected, it occurred at values of external temperatures over 4 °C for IB, MS7, MS15, PC, PCMS290 and PCMS1270, whereas it took place over -2 °C for CB. Starting from such conditions, the system entered the subcritical zone in which the condensing temperature was ranged according to the external temperature (Zone II). The R744 condenser outlet temperature could be evaluated as follows:

$$t_{out,cond} = t_{ext} + \Delta T_{appr,cond} \qquad (1)$$

The transition zone (Zone III) was the one where the system moved gradually from the subcritical conditions to the transcritical ones. In accordance with the chosen condenser approach temperatures (Table 2), these conditions arose at external temperatures higher than 17 °C for IB, MS7, MS15, PC, PCMS290 and PCMS1270 and higher than 10 °C for CB. According to Cecchinato et al. (2007), Zone III could be defined by identifying an upper and a lower limit in terms of high pressure and gas cooler/condenser outlet temperature. Thus, the high pressure heat exchanger varied its working conditions linearly (segment A - B in Fig. 10) in accordance with the previously mentioned limits and with the external temperature. The maximum condenser outlet temperature and the maximum condensing one were set to 20 and 22 °C (point A in Fig. 10), respectively. The minimum gas cooler outlet temperature and the upper limit of high pressure, indicated as point B in Fig. 10, were chosen equal to 29 °C and 75 bar (Table 3 and Table 4), respectively. In these running modes, the approach temperature of the gas cooler/condenser decreased from 3 K (point A in Fig. 10) for IB, MS7, MS15, PC, PCMS290 and PCMS1270 to 2 K (point B in Fig. 10) and from 10 K (point A in Fig. 10) to 5 K (point B in Fig. 10) for CB. The system operated along the conditions represented by the segment A - B in Fig. 10, following for IB, MS7 and MS15 the below equations:

$$\mathrm{t}_{\mathrm{out,gc}} = 0.9 \cdot \mathrm{t}_{\mathrm{ext}} + 4.7 \qquad (2)$$

$$p_{gc} = 1.6633 \cdot t_{out,gc} + 26.763 \qquad (3)$$

and

$$t_{out,gc} = 0.6429 \cdot t_{ext} + 13.571 \qquad (4)$$

$$p_{gc} = 1.6633 \cdot t_{out,gc} + 26.763 \qquad (5)$$

for CB.

Furthermore, this control strategy allowed reducing the degree of subcooling of the condenser/gas cooler from 2 K to a null value.

Transcritical conditions (Zone IV) took place at outdoor temperatures over 27 °C for IB, MS7, MS15, PC, PCMS290 and PCMS1270 and over 24 °C for CB and an optimal gas cooler pressure had to be evaluated as a function of the external temperature (Kim et al., 2004).

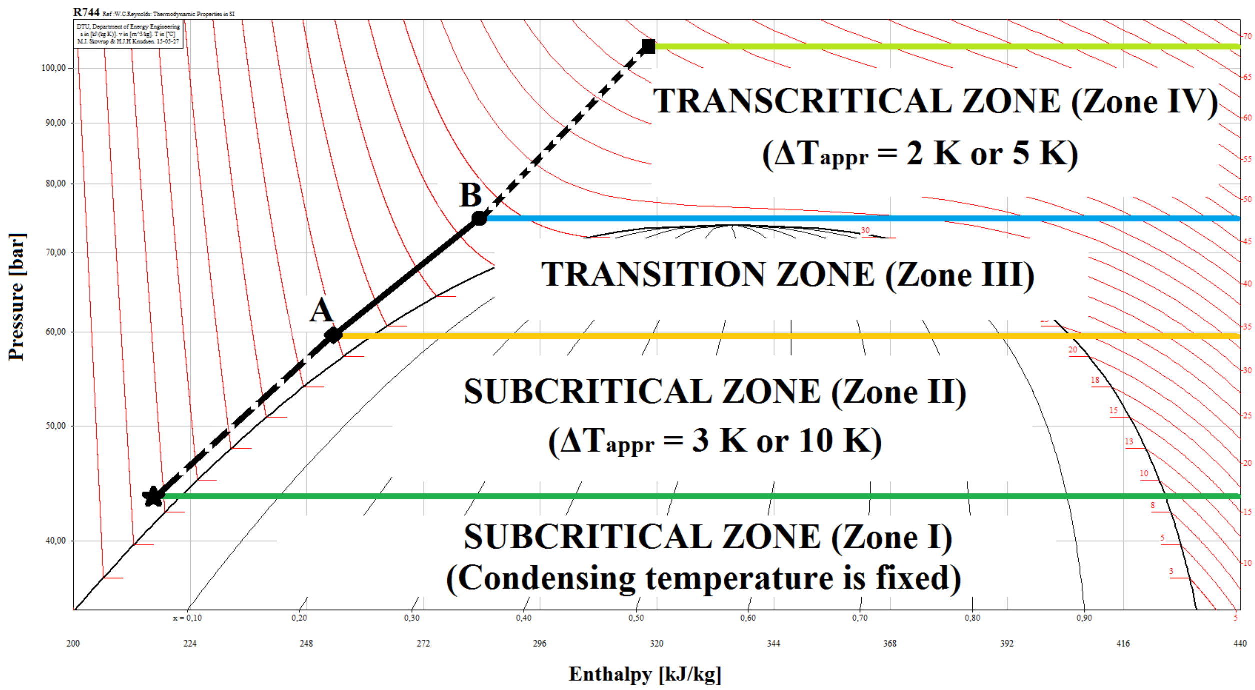

Fig. 10 - Definition of the operating zones for the R744 refrigeration booster systems under investigation.

Table 3 for CB and Table 4 for IB, MS7, MS15, PC, PCMS290 and PCMS1270 summarize the operating zones previously explained.

Table 3 - Operating zone for CB.

Zone	t_{ext} [°C]	$t_{out,condMAX}$ [°C]	$t_{condMAX}$ [°C]	$t_{out,gcMAX}$ [°C]	$p_{gc,MAX}$ [bar]
I	$t_{ext} \leq -2$	8	10	-	-
II	$-2 < t_{ext} \leq 10$	20	22	-	-
III	$10 < t_{ext} \leq 24$	-	-	29	75
IV	$24 < t_{ext} \leq 40$	-	-	45	106

Table 4 - Operating zone for IB, MS7, MS15, PC, PCMS290 and PCMS1270.

Zone	t_{ext} [°C]	$t_{out,condMAX}$ [°C]	$t_{condMAX}$ [°C]	$t_{out,gcMAX}$ [°C]	$p_{gc,MAX}$ [bar]
I	$t_{ext} \leq 4$	7	9	-	-
II	$4 < t_{ext} \leq 17$	20	22	-	-
III	$17 < t_{ext} \leq 27$	-	-	29	75
IV	$27 < t_{ext} \leq 40$	-	-	42	106

Two different approaches were taken into account for MS7 and MS15:

a) R744 subcooler outlet temperature could be reduced down to 7 °C, which is a typical value of minimum condenser outlet temperature for a CO_2 booster system;
b) R744 subcooler outlet temperature was set equal to 15 °C.

As for PCMS290 and PCMS1270, a constant subcooler outlet temperature equal to 15 °C was chosen. Furthermore, the chosen compressor using R1270 presented a maximum evaporation temperature of 5 °C, which was twice as low as that of that employing R290.

Table 5 sums up the chosen optimization variables of the R744 booster refrigeration systems under investigation which were adopted during the optimization procedures. The DIRECT algorithm Method was implemented in case of two or more independent variables, otherwise the Golden Section search Method was applied (F-Chart Software, 2015b). In all cases, the minimization of the total energy consumption was chosen as objective function.

Table 5 – Independent variables for the optimization procedures of the R744 booster refrigeration systems under investigation.

	Transition conditions	Transcritical conditions
CB and IB	-	p_{gc}
MS7	$t_{out,SUB}$	p_{gc}, $t_{out,SUB}$
MS15	-	p_{gc}
PC	p_{int}, χ	p_{gc}, p_{int}, χ
PCMS290	p_{int}	p_{gc}, p_{int}

PCMS1270	p_{int}	p_{gc}, p_{int}

The global efficiencies of the compressors using R134a and R744 were derived from BITZER Software (BITZER, 2015) as a function of the pressure ratio (Table 6 and Table 7), whereas Dorin Software (Dorin, 2015) was utilized to obtain R290 and R1270 compressors performance. Particularly, their computation was based on the suction and the discharge conditions exhibited by the manufacturer as the boundary conditions varied. All the chosen compressors were semi-hermetic reciprocating ones and all their suggested technological constraints were respected.

Table 6 - Compressors global efficiencies in subcritical and transition conditions.

Configuration	Compressor global efficiency
CS	$\eta_{global,R134a} = -0.0053 \cdot \left(\frac{p_{HP,R134a}}{p_{MT,R134a}}\right)^2 + 0.0674 \cdot \left(\frac{p_{HP,R134a}}{p_{MT,R134a}}\right) + 0.4802$ $\eta_{global,R744} = 0.0111 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right) + 0.803$
CB, IB, MS7, MS15, PC, PCMS290 and PCMS1270	$\eta_{globalHS,R744} = -0.1155 \cdot \left(\frac{p_{HP,R744}}{p_{MT,R744}}\right)^2 + 0.5762 \cdot \left(\frac{p_{HP,R744}}{p_{MT,R744}}\right) - 0.0404$ $\eta_{globalLS,R744} = -0.0012 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right)^2 - 0.0087 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right) + 0.6992$
Auxiliary compressor for PC	$\eta_{globalAUX,R744} = -0.172 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right)^2 + 0.7095 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right) - 0.0373$
Auxiliary compressor for PCMS290 and PCMS1270	$\eta_{globalAUX,R744} = -0.0507 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right)^2 + 0.2073 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right) + 0.4635$
R290 Subcooler loop	$\eta_{global,R290} = -0.0939 \cdot \left(\frac{p_{HT,R290}}{p_{LT,R290}}\right)^2 + 0.4966 \cdot \left(\frac{p_{HT,R290}}{p_{LT,R290}}\right) - 0.0449$
R1270 Subcooler loop	$\eta_{global,R1270} = -0.047 \cdot \left(\frac{p_{HT,R1270}}{p_{LT,R1270}}\right)^2 + 0.3442 \cdot \left(\frac{p_{HT,R1270}}{p_{LT,R1270}}\right) + 0.0299$

Table 7 - Compressors global efficiencies in transcritical conditions.

Configuration	Compressor global efficiency
CS	$\eta_{global,R134a} = -0.0053 \cdot \left(\frac{p_{HP,R134a}}{p_{MT,R134a}}\right)^2 + 0.0674 \cdot \left(\frac{p_{HP,R134a}}{p_{MT,R134a}}\right) + 0.4802$ $\eta_{global,R744} = 0.0111 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right)^2 - 0.0793 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right) + 0.803$
CB, IB, MS7, MS15, PC, PCMS290 and	$\eta_{globalHS,R744} = -0.0021 \cdot \left(\frac{p_{HP,R744}}{p_{MT,R744}}\right)^2 - 0.0155 \cdot \left(\frac{p_{HP,R744}}{p_{MT,R744}}\right) + 0.7325$

PCMS1270	$\eta_{globalLS,R744} = -0.0012 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right)^2 - 0.0087 \cdot \left(\frac{p_{MT,R744}}{p_{LT,R744}}\right) + 0.6992$
Auxiliary compressor for PC	$\eta_{globalAUX,R744} = -0.0788 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right)^2 + 0.3708 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right) + 0.2729$
Auxiliary compressor for PCMS290 and PCMS1270	$\eta_{globalAUX,R744} = -0.0272 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right)^2 + 0.2117 \cdot \left(\frac{p_{HP,R744}}{p_{int,R744}}\right) + 0.2476$
R290 Subcooler loop	$\eta_{global,R290} = -0.0226 \cdot \left(\frac{p_{HT,R290}}{p_{LT,R290}}\right)^2 + 0.1816 \cdot \left(\frac{p_{HT,R290}}{p_{LT,R290}}\right) + 0.3701$
R1270 Subcooler loop	$\eta_{global,R1270} = -0.0761 \cdot \left(\frac{p_{HT,R1270}}{p_{LT,R1270}}\right)^2 + 0.5103 \cdot \left(\frac{p_{HT,R1270}}{p_{LT,R1270}}\right) - 0.1814$

In this paper the results were compared in three different ways. The first comparison referred to all the systems using R744 as main refrigerant. The other ones compared all the evaluated solutions in terms of annual energy consumption and TEWI assuming CO_2/R134a cascade refrigeration system as baseline.

2.3. Outdoor temperatures

Valencia (Spain) and Athens (Greece) were identified as locations in order to be able to make a comparison in two warm climates with different external temperatures distribution.

The outdoor temperatures were obtained by using Meteonorm (Remund et al., 2014) and their distribution is shown in Fig. 11.

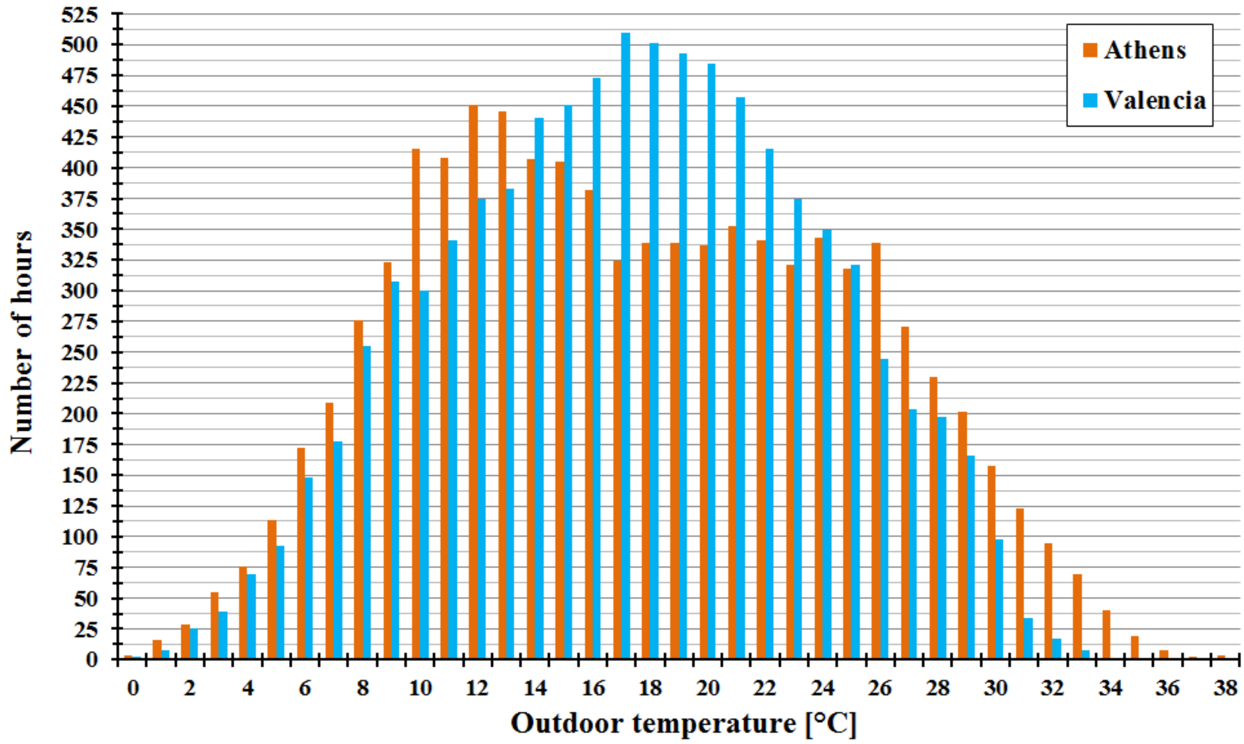

Fig. 11 - Number of hours per year at different outdoor temperatures in Athens (Greece) and Valencia (Spain).

As can be seen in Fig. 11, outdoor temperature was higher than 27 °C for about 11% of the time in Athens and 6% in Valencia. In these conditions, transcritical operations occurred for CB, IB, MS7, MS15, PC, PCMS290 and PCMS1270. On the other hand, transcritical operations for the conventional booster took place at external temperatures higher than 24 °C, which arose for more than 21% of the time in Athens and 14% in Valencia.

Transition conditions took place for CB, IB, MS7, MS15, PC, PCMS290 and PCMS1270 at outdoor temperatures higher than 17 °C, which occurred for about 49% of the time in Athens and 50% in Valencia. As regards CB, it was run in transcritical conditions for about 57% of the time in Athens and 61% in Valencia.

This outcome points out the importance associated with the choice of a suitable optimisation strategy for the transition zone.

2.4. Total Equivalent Warming Impact

The Total Equivalent Warming Impact (TEWI) is a parameter which assesses both the direct and the indirect emissions of greenhouse gases. The former are due to refrigerant leakages, whereas the latter are due to CO_2 emissions associated with the process of electricity generation. It is defined as (BRA, 2006):

$$TEWI = TEWI_{direct} + TEWI_{indirect} \qquad (6)$$

$$TEWI_{direct} = GWP \cdot L \cdot n + GWP \cdot m \cdot (1 - \alpha) \qquad (7)$$

$$TEWI_{indirect} = E \cdot \beta \cdot n \qquad (8)$$

As far as the evaluated solutions are concerned, the below assumptions were made:

- the GWP of R134a, R744, R1270 and R290 were selected equal to 1430, 1, 1.8 and 3, respectively (EMERSON Climate Technologies, 2010; The Australian Institute of Refrigeration, 2012);
- an annual leakage rate for the systems using CO_2 as refrigerant of 15% was chosen (EMERSON Climate Technologies, 2010; Shilliday, 2012), whereas it added up to 5% for the mechanical subcooler cycle since it was supposed to be completely confined in the machinery room (Llopis et al., 2015b);
- the operating life of all selected systems was evaluated equal to 10 years (EMERSON Climate Technologies, 2010; Shilliday, 2012);
- the R134a charge in HS circuit of the cascade refrigeration system was chosen equal to 2 $kg_{R134a} \cdot kW_{cc}^{-1}$ (EMERSON Climate Technologies, 2010);
- the CO_2 charge in LS circuit of the cascade refrigeration system was chosen equal to 1 $kg_{R744} \cdot kW_{cc}^{-1}$;
- the CO_2 charge in all booster refrigeration systems was chosen equal to 1.2 $kg_{R744} \cdot kW_{cc}^{-1}$ (Shilliday, 2012);
- R290 and R1270 charge were selected equal to 70 kg;
- 95% of the refrigerant was assumed to be recycled (EMERSON Climate Technologies, 2010);
- the CO_2 emission due to the electricity generation added up to 0.241 $kg_{CO_2} \cdot kWh^{-1}$ (Llopis et al., 2015b) in Valencia and to 0.720 $kg_{CO_2} \cdot kWh^{-1}$ (IEA, 2011) in Athens.

3. Results

In this section a comparison in terms of performance evaluation and optimal gas cooler pressure among the R744 refrigeration systems was firstly reported. Then, the evaluation of the energy consumption and TEWI of all the solutions under consideration was also accomplished.

3.1. Performance comparison among the evaluated R744 refrigeration systems

Reducing the approach temperature of the gas cooler/condenser enhances the performance of the R744 booster system thanks to a lower optimal gas cooler pressure in comparison with the conventional solution at the same external temperature. The additional adoption of a mechanical subcooling allows attaining a further drop in the optimal high pressure and a growth in the specific refrigerating effect promoted by the decrease in the quality of the refrigerant entering the liquid receiver. According to Fig. 12, MS15 showed a reduction in quality by about 64.9% in comparison with IB in transcritical conditions. It could also be noticed that the former was characterized by a decreasing trend, whereas MS7 had a growing one. The reason of this outcome lies in the fact that the state point representing the subcooler outlet conditions for MS15 shifted along an isothermal transformation, process which drives to a reduction in the quality of the refrigerant as the high pressure (and thus the outdoor temperature) goes up. Vice versa, MS7 underwent the opposite phenomenon since the temperature was not fixed. In transition operations, the refrigerant quality for IB exhibited on average 38.6% higher values than those showed by MS15. CO_2 quality of MS15 presented an average value of 0.14 at external temperatures ranging from 18 °C to 40 °C. MS7 pointed out quality values on average 63.6% lower than those shown by IB. CB, IB and MS7 also had a sudden increase in quality as transcritical conditions occurred, which became even more marked at very high external temperatures. This outcome underlines further the need for the adoption of a subcooler loop for refrigeration systems operating in warm weathers. It is worth noticing that PCMS290 and PCMS1270 had a similar behaviour in relation to that showed by MS15.

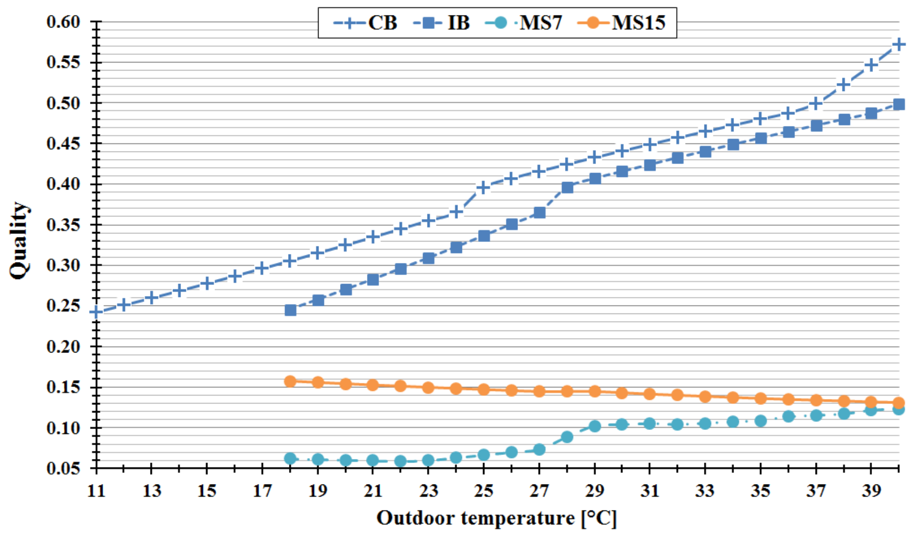

Fig. 12 - Quality of R744 entering the liquid receiver as a function of the outdoor temperature in transition and transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

In Fig. 12, the results associated with CB were presented starting from t_{ext} = 11 °C as the transition operations occurred at lower outdoor temperatures than IB, MS7 and MS15, according to Table 3.

The trend of the temperature at the subcooler outlet regarding MS7, as pointed out in Fig. 13, was approximately constant for external temperatures up to 27 °C and then it began increasing as transcritical conditions approached. As soon as these operations arose, this trend became more significant. This result can be justified taking into account that the adoption of the mechanical subcooling leads to two benefits: the former is associated with the reduction in refrigerant temperature entering the liquid receiver, which implies a growth in the refrigerating effect, whereas the second benefit is associated with the reduction in high pressure, which allows dropping the energy consumption. Since no optimal high pressure could be accomplished in the transition region, the system needed to achieve the highest degree of subcooling in order to be capable of enhancing its performance in such conditions. On the contrary, the system could be provided with both the benefit associated with the subcooling and the one associated with the reduction in gas cooler pressure as soon as transcritical conditions took place. As a result of this, the system was able to achieve the optimal conditions at higher subcooler outlet temperatures than in the previous case.

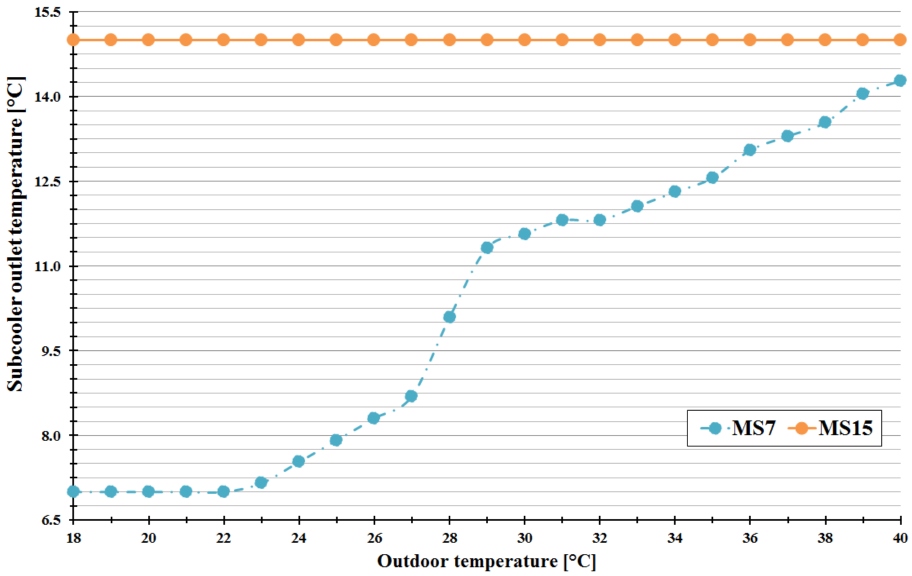

Fig. 13 - Subcooler outlet temperature as a function of the outdoor temperatures in transition and transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

Fig. 14 highlights the high reduction in the mass flow rate of the saturated vapour exiting the liquid receiver related to the use of the mechanical subcooling. At very high outdoor temperature, the saturated vapour mass flow rate of the system with parallel compression was about five times higher than the ones of the solutions which combined parallel compression and mechanical subcooler loop.

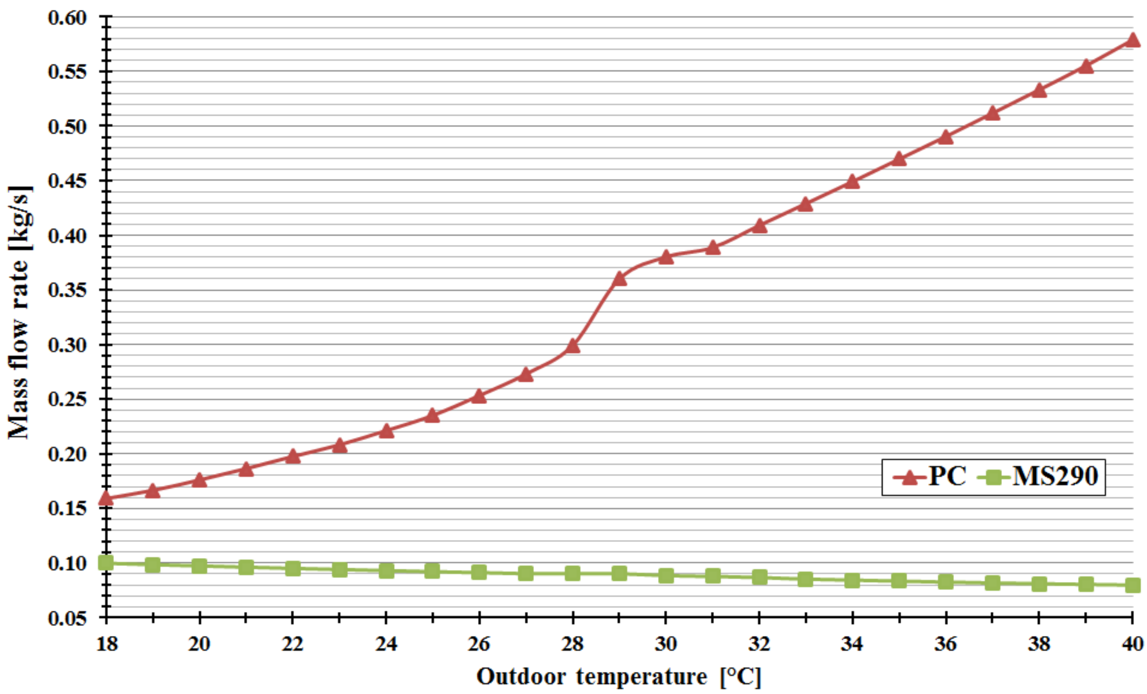

Fig. 14 - Saturated vapour mass flow rate of PC and PCMS290 as a function of the outdoor temperature in transition and transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

As previously mentioned, the intermediate pressure becomes a key parameter for the optimization procedure in case of adoption of an auxiliary compressor. The results obtained indicated that, when both the parallel compression and the mechanical subcooler loop were run, the influence of the intermediate pressure was almost negligible and the performance of the system was mainly affected by the high pressure since the subcooler outlet temperature was kept constant. The reason of this lies in the low quality of the refrigerant flowing into the liquid receiver thanks to the presence of the subcooler. Fig. 15 shows distinctly this outcome underling as the intermediate pressure of PCMS290 (and that of PCMS1270) tended to be as low as possible in accordance with the technological constraints of the selected additional compressor. Furthermore, as for PC, it was possible to notice a sudden increase in the optimum intermediate pressure as soon as the transcritical operations took place. This result could be associated with the increment in the amount of vapour which had to be sucked at high external temperatures. In these conditions, in fact, the intermediate pressure had to be risen in order to be capable of decreasing the electrical consumption of the additional compressor. The average value of the optimal intermediate pressure in the range of outdoor temperatures from 18 °C to 27 °C was equal to 40 bar. The liquid receiver pressure was as low as 44.51 bar at 28 °C and it reached an average value equal to 48.13 bar at external temperatures from 29 °C to 40 °C.

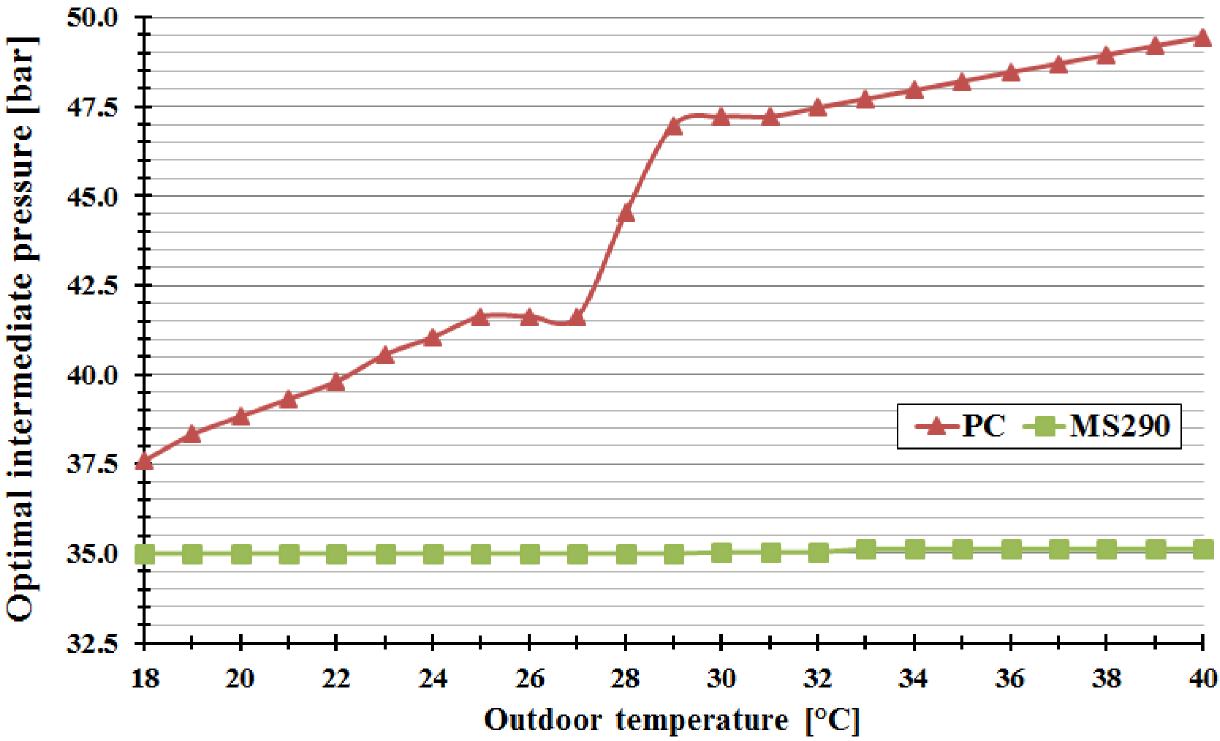

Fig. 15 - Optimal intermediate pressure as a function of the outdoor temperature in transition and transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

Fig. 16 compares the optimal gas cooler pressure values which were achieved by all the evaluated CO_2 booster solutions in transcritical conditions.

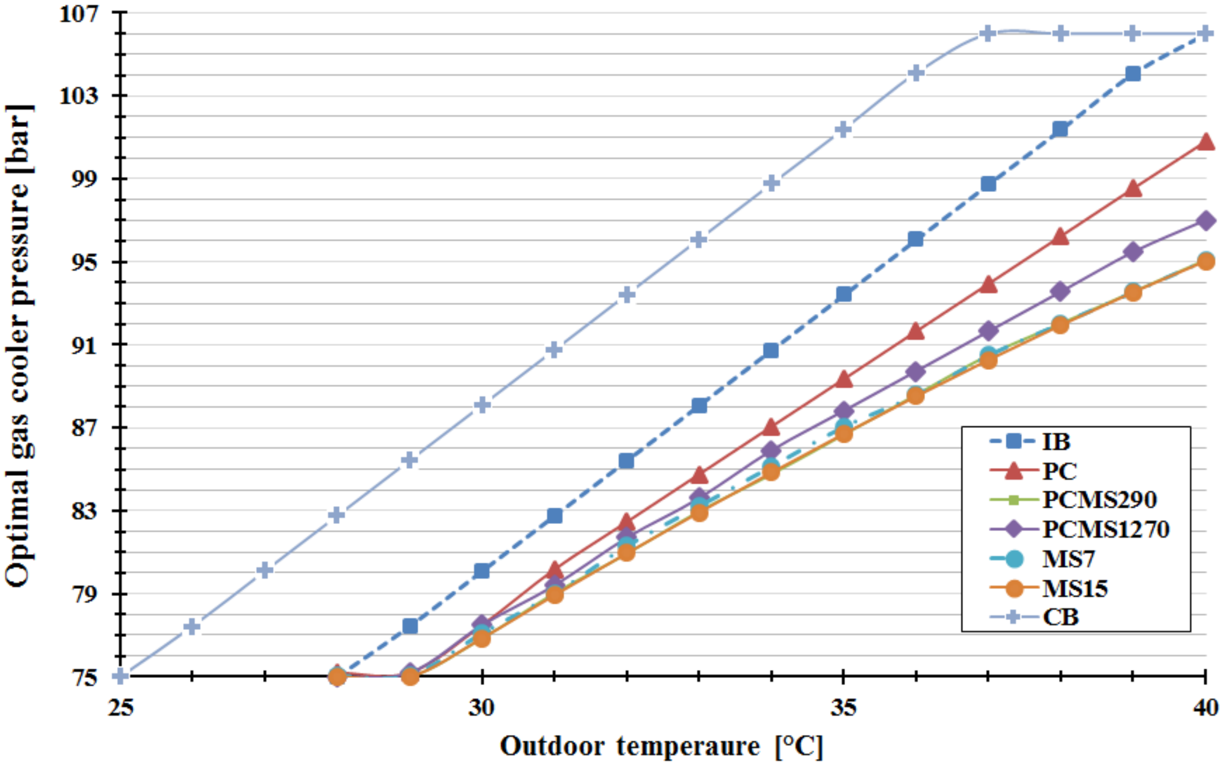

Fig. 16 - Optimal gas cooler pressure of the evaluated solutions as a function of the outdoor temperature in transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

According to Table 3, CB started operating in transcritical conditions at t_{ext} = 25 °C, whereas the other evaluated systems reached them at t_{ext} = 28 °C (Table 4). Furthermore, MS7 and MS15 had a similar trend in optimal gas cooler pressure. IB was characterized by values on average about 6% higher than those of both the systems with mechanical subcooling. This difference went up with the increment in the outdoor temperature, underling that these solutions are even more efficient at very high temperatures, in accordance with Llopis et al. (2015a). The maximum level of gas cooler pressure showed in Fig. 16 was due to the operating envelopes of the selected compressors. Furthermore, this outcome was markedly connected to the CO_2 quality values of CB at temperatures over 37 °C (Fig. 12). PC pointed out values on average 4.2% lower than those showed by IB, whereas PCMS1270 achieved on average a reduction by 5.3%. It is important to notice that, due to technological limits of the selected R1270 compressor, the performance of this system was penalised by a lower maximum evaporating temperature in comparison with R290 compressor. PCMS290 exhibited the lowest optimal gas cooler pressures, which were on average 6.3% smaller than the ones associated with the IB.

3.2. Performance comparison among the evaluated refrigeration systems

Fig. 17 makes a comparison in terms of COP among all the selected solutions. It was possible to notice that below the outdoor temperatures of 14 °C, all the solutions (except for CB) performed better or similarly than CS. As soon as transition operations arose and thus the mechanical subcooling and/or the auxiliary compressor began operating, MS15, PC, PCMS290, PCMS1270 and MS7 had the best COPs among the alternatives to CS. Although they could obtain the benefits associated with the subcooling and/or those of the parallel compression, their COP could not exceed the one of the baseline at high outdoor temperatures. PCMS1270 showed COPs similar to the ones of PCMS290 and MS7 and an increase in COP difference as the external temperature increased due to a lower maximum evaporating temperature of the subcooler could be noticed.

PCMS290 presented the highest COPs and thus it can be claimed that either a large subcooling or the combination of the parallel compression with the mechanical subcooling is necessary for a CO_2 booster refrigeration system which was run in warm climates. PC was also an optimal alternative especially for external temperatures up to 33 °C. In fact, at very high outdoor temperatures, the adoption of a mechanical subcooling was much more efficient than PC.

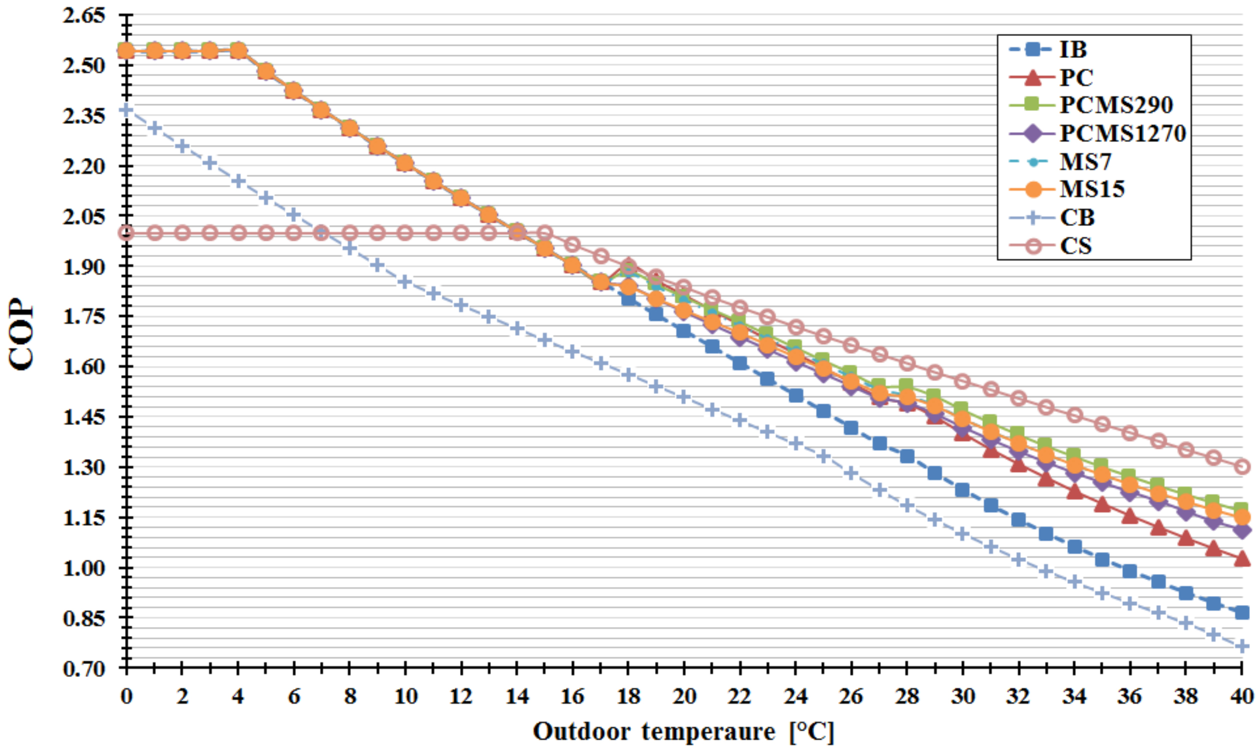

Fig. 17 - COP of the selected solutions at outdoor temperatures from 0 to 40 °C (t_{MT} = -10 °C, t_{LT} = -35 °C).

At the outdoor temperature of 28 °C, CB, IB, MS7 and MS15 exhibited a COP which added up to 1.19, 1.33, 1.52 and 1.51, respectively. At very high outdoor temperatures, the difference in COP between MS15 and MS7 brought down since the subcooler outlet temperatures in the latter case reached values close to 15 °C.

In comparison with IB, MS15 and MS7 had on average about 23.2% and 23.3% higher values of COP in transcritical conditions. The difference in performance was smaller in transition operations since the benefit associated with the optimal high pressure vanished. This implied that MS7 and MS15 could only obtain the advantage in terms of the increment in the refrigerating effect thanks to the drop in the refrigerant quality entering the liquid receiver. At these running conditions, IB highlighted on average 7.7% and 6.2% lower COP values than those exhibited by MS7 and MS15. These outcomes underlined the importance of the adoption of a mechanical subcooling loop for CO_2 booster refrigeration systems operating in warm weathers. It is important to notice that MS7 and PCMS290 were slightly penalised at outdoor temperatures ranging from 18 °C to 22 °C due to the operating envelope of the R290 compressor. A similar phenomenon occurred for PCMS1270 at external temperatures over 37 °C. In transition conditions, the difference in COPs between MS7 and MS15 was on average equal to 1.4%. As transcritical conditions took place, it got negligible (0.12%) since they had similar subcooler outlet temperatures. At outdoor temperatures over 27 °C, the adoption of an auxiliary compressor led to an average enhancement of COP by 16% beside IB.

PCMS1270 showed COPs similar to the ones of PCMS290 and MS7 and an increase in COP difference with the rise in the external temperature due to a lower maximum evaporating temperature of the subcooler in comparison with that using R290. PCMS290 had the highest COPs which were on average 25% higher than the ones presented by IB. In transition operations, PC, PCMS290 and PCMS1270 had an increase in COP respectively by 7.5%, 8.2% and 5.6% beside IB.

In all the evaluated cases, the effect of improvements in the systems were more appreciable as the outdoor temperature increases. Furthermore, PC featured higher COPs for external temperatures up to 33 °C and thus it exhibited a more marked decreasing trend than the one associated with MS15.

The use of a mechanical subcooler can improve the performance of a conventional R744 booster system considerably. On the other hand, total investment costs would increase and the adoption of an operating strategy similar to that with variable subcooler outlet temperature (MS7) would make more difficult the control and the management of the overall system. As shown in Fig. 18, the required cooling capacity of the mechanical subcooling loop ranged from 12.9 kW to 67.5 kW for MS15, PCMS290 and PCMS1270 and from 24.1 kW to 67.9 kW for MS7. This means that the mechanical subcooler would have run at part load for the most of the time. Setting the subcooler outlet temperature to a prefixed value would allow facilitating the control strategy of the overall system. Furthermore, the difference in required cooling capacity of the subcooler was almost negligible at high outdoor temperatures, whereas it dropped in transition conditions due to lower required CO_2 subcooler outlet temperature for MS7 beside MS15, PCMS290 and PCMS1270.

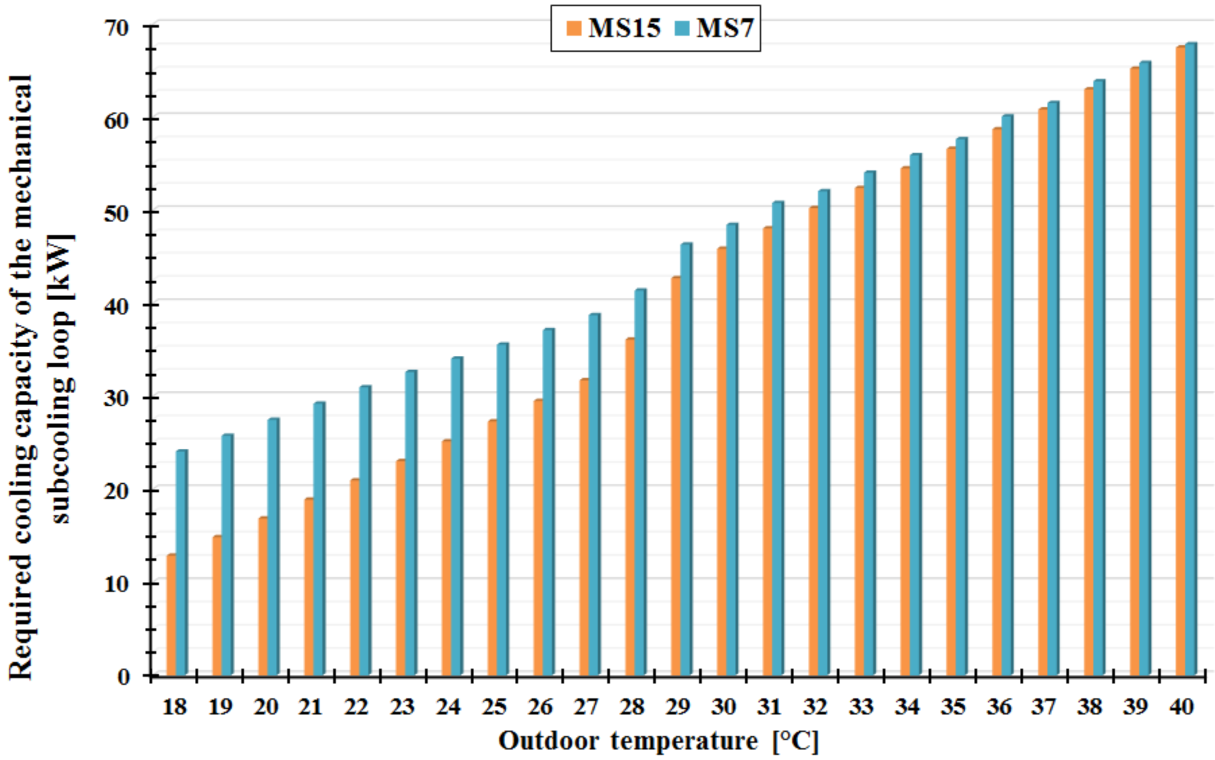

Fig. 18 - Required cooling capacity of the subcooler in transition and transcritical conditions (t_{MT} = -10 °C, t_{LT} = -35 °C).

Neither the trend of the cooling capacity nor the one of the compressor power input are shown in any figures since the former is constant and the latter has the same trend as that of COP.

3.3. Comparison in terms of annual energy consumption

The annual energy consumption of the selected solutions performing in Athens and Valencia are compared in Table 8. The conventional booster (CB) had an energy consumption 20% higher than the reference cascade system (CS). The decrement in the approach temperature of the gas cooler/condenser led to an energy saving by approximately 5% on the part of CS beside IB. On the other hand, it was possible to achieve comparable energy consumption with that associated with CS by employing one of the other investigated solutions. This points out that carbon dioxide can efficiently be used even in warm climates. Furthermore, it is possible to claim that the adoption of a dedicated mechanical subcooling is even more beneficial at very high external temperatures.

Table 8 – Annual energy consumption [MWh] and percent difference [%] in comparison with CS.

	CB	IB	MS7	MS15	PC	PCMS290	PCMS1270
Athens	656.6 (+20.2%)	574.8 (+5.2%)	544.3 (-0.4%)	547.5 (+0.2%)	547.2 (+0.2%)	541.9 (-0.8%)	550.1 (+0.7%)
Valencia	644.4 (+19.2%)	563.8 (+4.3%)	539.1 (-0.2%)	543.0 (+0.5%)	539.9 (-0.1%)	537.2 (-0.6%)	545.0 (+0.8%)

3.3. Comparison in terms of Total Equivalent Warming Impact

The values of TEWI of the selected solutions are compared in Table 9. It is worth noticing that the use of the European average indirect emission factor would have led to percent differences in TEWIs for the R744 solutions similar to those regarding the annual energy consumption presented in Table 8.

Although R744 is characterized by a low value of GWP, CB had a higher TEWI than CS in Athens due to its higher indirect CO_2 emissions. In Valencia, TEWI of the latter was larger than that of CB. The reduction in approach temperature of the gas cooler/condenser and in the minimum condensing temperature drove to a drop in terms of environmental impact. In fact, IB showed on average 5.1% in Athens and 21.6% in Valencia lower TEWI compared with that of the cascade refrigeration. This is due to the high GWP of R134a and thus to its larger direct emissions.

All the enhanced solutions featured lower TEWI than that of the cascade system. This result could be associated with lower direct emissions impact than CS due to the lower GWP of carbon dioxide in comparison with that of R134a. On the other hand, the impacts in terms of indirect emissions were similar to one another as the difference in annual energy consumption among the selected enhanced solutions was almost negligible. Interesting results were obtained for both the systems with dedicated mechanical subcooling. As far as MS7 is concerned, TEWI dropped by about 10.2% in Athens and by 25% in Valencia beside that of CS, whereas it brought down on average by about 9.6% and by 24.4% for MS15, respectively. The lowest TEWI values were achieved by PCMS290 which were 10.6% in Athens and 25.3% in Valencia lower than those of the baseline. Similar values were found for PCMS1270 and PC, which were able to reduce TEWI by 24.2% and 24.9% in the Spanish locality and 9.2% and 9.7% in the Greek one in comparison with CS.

It could be concluded that the adoption of MS15 allowed reducing the environmental impact by 420.6 t_{CO_2} in Athens and by 423.6 t_{CO_2} in Valencia over a period of time of 10 years. Considering MS7, these values were as high as 443.8 t_{CO_2} and 432.9 t_{CO_2}, respectively. Furthermore, PC, PCMS290 and PCMS1270 led to a reduction in the environmental impact by 422.8, 460.8 and 402.1 t_{CO_2} in Athens and 431.1, 437.5 and 418.9 t_{CO_2} in Valencia over the plant life time.

Table 9 – TEWI [tonnes] and percent difference [%] in comparison with CS.

	CB	IB	MS7	MS15	PC	PCMS290	PCMS1270
Athens	4727.6 (+8.4%)	4139.1 (-5.1%)	3919.1 (-10.2%)	3942.4 (-9.6%)	3940.1 (-9.7%)	3902.1 (-10.6%)	3960.9 (-9.2%)
Valencia	1553.3 (-10.3%)	1359.0 (-21.6%)	1299.6 (-25.0%)	1308.9 (-24.4%)	1301.4 (-24.9%)	1295.0 (-25.3%)	1313.6 (-24.2%)

4. Discussion and conclusions

A comparison in terms of energy and environmental performance among different commercial refrigeration solutions has been performed. The following configurations, which have the MT cooling capacity of 97 kW at -10 °C and the LT cooling capacity of 18 kW at -35 °C, have been taken into account:

- a R744/R134a cascade refrigeration system (baseline indicated as CS);
- a conventional R744 booster refrigeration system (CB);
- a R744 booster refrigeration system characterized by lower approach temperatures of the gas cooler/condenser and lower minimum condensing temperature than those of the previous case (IB);
- R744 booster system with dedicated mechanical subcooling and subcooler outlet temperature down to 7 °C (MS7);
- R744 booster system with dedicated mechanical subcooling and subcooler outlet temperature set to 15 °C (MS15);
- R744 booster system with parallel compression (PC);
- R744 booster system which combines the parallel compression and a dedicated R290 mechanical subcooling loop (PCMS290);
- R744 booster system which combines the parallel compression and a dedicated R1270 mechanical subcooling loop (PCMS1270).

The comparison has been made considering the weather trends in Valencia (Spain) and in Athens (Greece). Performance of CB, IB and MS15 have been optimized in terms of gas cooler pressure in transcritical conditions, whereas the subcooler outlet temperature has been considered as an additional optimization variable for MS7. As for PC, the optimization procedure has also involved the auxiliary compressor mass flow rate and the intermediate pressure in transition conditions. In transcritical operations, gas cooler pressure becomes an additional optimization key parameter. Taking into account the solutions which integrates the parallel compression and the mechanical subcooling, performance has been evaluated as a function of the intermediate pressure in transition operations, whereas high pressure has also been considered in transcritical conditions.

All the solutions have been compared in terms of both annual energy consumption and TEWI. The outcomes obtained show that either a large subcooling or the combination of the parallel compression with a mechanical subcooling is necessary in order to accomplish comparable performance with the one of the CO_2/R134a cascade refrigeration system in both the evaluated locations. Both of them are more efficient as the outdoor temperature rises, whereas the configuration with parallel compression (PC) performs better for outdoor temperatures up to 33 °C. In fact, it shows similar consumption to that of the baseline (CS) and it is more beneficial in Valencia than in Athens due to less warm weather trend in the first case than in the second one. MS7 and PCMS290 exhibit similar energy consumption to that of the baseline in both the investigated weather conditions. Due to the technological constraints of R1270 compressor, the solution which uses such refrigerant has the worst performance among the enhanced solutions.

As far as PC is concerned, the vapour mass flow rate has a growing trend with the increment in the external temperature. Additionally, the percentage of vapour sucked by the additional compressor is almost constant and it adds up to 99.38% of the total vapour mass flow rate in both transition conditions and transcritical ones.

An additional benefit associated with the adoption of the subcooler and/or an auxiliary compressor is the reduction in environmental impact mainly due to the achievement of consumption which is comparable with the one of the cascade system. This leads to similar indirect CO_2 emissions to each other. Furthermore, the use of a refrigerant with low GWP, such as R744, also decreases direct CO_2 emissions markedly. Reduction in TEWI of the enhanced solutions ranges from 9.2% to 25.3% beside the baseline.

The adoption of a subcooler loop presents some drawbacks, such as:

a) the investment cost of a CO_2 booster system is 48% higher than the one of a R404A system, according to Shilliday (2012) and the adoption of a dedicated mechanical subcooling would drive to an additional growth in it;

b) the subcooling loop needs to be run at part load conditions for most of the time and a variable subcooler outlet temperature would make its control system difficult to be implemented.

The latter issue could be partly worked out by setting the CO_2 subcooler outlet temperature to a fixed value. Furthermore, MS15 has similar results in terms of both energy consumption and TEWI to MS7 in both the selected locations.

From the results obtained, it can be claimed that:

- PCMS290 shows the best performance in both the selected locations, followed by MS7;
- the solution with parallel compression performs well for outdoor temperatures up to 33 °C and then its performance worsens due to the large amount of vapour which needs to be sucked.
- it is important to notice that a control system can more easily deal with the auxiliary compressor rather than with the mechanical subcooling. This limit could be partly overtaken by adopting a fixed set-point temperature at the subcooler outlet;
- at very high outdoor temperatures, it is necessary to adopt a mechanical subcooling and/or an auxiliary compressor to achieve performance similar to that of the baseline CS;
- an economic analysis should be carried out in order to evaluated the payback period for all the evaluated R744 solutions and establish whether they could also be competitive in economic terms or not;
- an experimental campaign should be implemented in order to validate the results computed in this paper.

Furthermore, the outcomes have also underlined that at very high outdoor temperature, the adoption of a mechanical subcooling is much more efficient than that of the parallel compression.

The only paper in which a similar comparison to the one attained in this study has been made by Polzot et al. (2015). Considering the difference in the assumptions made and in the operating conditions, the results are quite consistent.

The improvement by utilising CO_2 is to large extent related to avoidance of the direct emission, as the power consumption is similar for all the systems. The exception is the conventional booster, which shows relatively high emissions in Athens. Improved design of the CO_2 may make them even more competitive with HFC systems.

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