# R744 Booster integrated system for full heating supply to supermarkets



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# R744 BOOSTER INTEGRATED SYSTEM FOR FULL HEATING SUPPLY TO SUPERMARKETS

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## ABSTRACT

In warm climates carbon dioxide (R744) refrigeration systems are known to perform worse because of the transcritical operation which occurs for a large part of the year, increasing substantially their energy use. On the other hand, the high discharge temperature potentially allows recovering a large amount of heat at various temperature levels. This paper investigates the energy performance of a R744 refrigeration system which provides a medium-sized supermarket with DHW and space heating, besides satisfying the cooling load required by the chilled and frozen food storage and display equipment. The system is controlled to meet the full heating demand of the building and is equipped with an additional air-cooled evaporator, which can be used as a supplemental heat source. Dynamic simulations of the refrigeration system, including mutual interactions with the building and the HVAC, are carried out at three climate conditions. The results are compared to alternative refrigeration systems not controlled by the heating demand, such as the R744 booster system itself, a R404A direct expansion system and a R134a/R744 cascade system, where heat pumps are employed for heating. Energy saving was predicted both when compared to the $CO_2$ system (up to 4.5 %), and to the HFC systems (up to 16 %). Furthermore, the heating COP of the integrated system appears to be always higher than that of standalone heat pumps.

The comparison shows that heat recovery can fully supply refrigeration and heating needs of the supermarket with reduced running and possibly investment costs when compared to separate systems.

## KEYWORDS:

R744; COP; refrigeration; modelling; heat recovery; heating

## NOMENCLATURE

| | |
|---|---|
| COP | coefficient of performance |
| DHW | domestic hot water |
| EER | Energy Efficiency Ratio |

| | |
|---|---|
| EV | expansion valve |
| HP | heat pump |
| HR | heat recovery |
| HS | high stage |
| HX1 | heat exchanger for domestic hot water heating |
| HX2 | heat exchanger for space heating |
| LE | load evaporator |
| LS | low stage |
| LT | low temperature level |
| MT | medium temperature level |
| *p* | pressure, bar |
| PC | parallel compression |
| q | heating load, kWh |
| SOL | solenoid valve |
| *t* | temperature, °C |
| W | electrical energy, kWh |

| | |
|---|---|
| β | pressure ratio |

*Subscripts*

| | |
|---|---|
| *c* | condensing |
| *e* | evaporating |
| *ext* | outdoor conditions |
| GC | gas cooler |
| INT | intermediate |
| *opt* | optimized |
| *out* | outlet |

## 1. INTRODUCTION

Carbon dioxide (R744) is recently being considered a suitable refrigerant in supermarket refrigerating plants thanks to its negligible environmental impact and its favourable thermo-physical properties. The R744 refrigerating systems perform better than the traditional hydrofluorocarbon ones at low outdoor temperature, while their energy consumption remains comparable as long as the outdoor temperature keeps below around 25 °C, thus allowing subcritical operation (Cavallini and Zilio, 2007, Finckh et al., 2011, Sawalha 2017). At higher outdoor temperature transcritical operation takes place, with a substantial decrease in the energy efficiency and the need to optimize the gas cooler pressure, which shows to be affected not only by the gas cooler outlet temperature (Liao et al, 2000), but also by the effectiveness of the internal heat exchanger, if any, and the isentropic efficiency of the high-stage compressors (Ge and Tassou, 2011). Various technical solutions can be adopted to improve the energy efficiency of the basic "booster" $CO_2$ system, to achieve at least a similar energy performance as the traditional HFC plants. In the literature various papers are available with comparisons among different solutions (Gullo et al., 2016a, 2016b; Hafner et al., 2014; Polzot et al., 2015, Sawalha, 2008, Wiedenmann et al, 2014). The most widespread is the adoption of a parallel compressor devoted to remove the flash gas generated in the liquid receiver at transcritical conditions, which would otherwise be expanded and supplied at the low-stage compressors (Sarkar and Agrawal, 2010, Minetto et al, 2005, Chiarello et al 2010). All authors underline that this solution suggests the optimization not only of the gas cooler pressure but also of the intermediate pressure. This solution has demonstrated its effectiveness in mild and warm climates (e.g. Minetto et al. 2014a against a single stage system), and is now adopted as the baseline at such climates. One of the drawbacks resides in the extra investment costs for the parallel compressors which operate only at the highest outdoor temperature conditions. Gullo et al. (2015) estimated that at an evaporating temperature of -10 °C and cooling medium temperature in the range from 30 °C to 50 °C, the adoption of an auxiliary compressor allows an energy saving of around 19 % at a 23 % increase in the capital cost of the whole plant. Polzot et al (2015) concluded that a parallel compressor allows to attain similar energy use to that of a R744/R134a cascade refrigeration system.

Significant improvements in the COP can also be obtained by performing subcooling of the refrigerant at the gas cooler exit. In this way the specific cooling capacity of the system is increased and the optimum gas cooler pressure at transcritical conditions can be reduced, with beneficial effects on the COP (Llopis, 2015, Llopis, 2016, Yang, 2011). Qureshi and Zubair (2012) investigated the increase in performance achievable using a dedicated refrigerating unit to perform subcooling in HFC and R717 systems and lately (Qureshi et al, 2013) in a R22 system, always getting positive results. Llopis (2015) applied a R290 mechanical subcooling to single and double stage transcritical $CO_2$ systems, and estimated that at 30 °C outdoor temperature and with 5 K subcooling degree, the maximum expected increments in COP are 13.7 % at -5 °C evaporating temperature (single stage $CO_2$ system) and 13.7% at – 30 °C evaporating temperature (double stage). Experimental tests (Llopis, 2016) revealed even higher improvements on the COP and especially the refrigerating capacity.

Overfeeding the evaporators is another solution to improve the efficiency of refrigerating plants, thanks to a better exploitation of the heat transfer area. A small amount of liquid exits the evaporator, without the need for liquid recirculation. Minetto and Fornasieri (2011) investigated an innovative system for controlling the refrigerant flow rate and they measured an energy saving up to 13 % due to the higher evaporating temperature (Minetto et al (2014b). They also checked that most of the excess liquid at the evaporator exit can be evaporated at the internal heat exchanger, thus preserving the compressor. However, an ejector has been installed and successfully used to suck the liquid from a liquid receiver downstream the evaporator. Ejectors can in fact be used both to suck liquid or vapour and to recover expansion work thus increasing the COP. Pardinas et al. (2018), Gullo et al (2017), He et al (2017) among the most recent ones showed that ejectors allow a significant energy saving especially at high outdoor temperature, even if at the expense of a more complex plant configuration, thus resulting in possible control issues.

In spite of all the above mentioned solutions, the efficiency of a $CO_2$ plant remains a critical issue at warm and mild climate conditions, giving place to a yearly energy consumption which can be higher than that of a conventional and simpler HFC plant. However this disadvantage can be strongly mitigated or even turned to advantage by performing heat recovery in favour of space heating and domestic hot water (DHW) production (Tambovtsev et al, 2011, Nidup, 2009, Cortella et al, 2014). In fact heat recovery, together with flooded evaporation, parallel compression and integration of air conditioning, is one of the most promising features of integrated $CO_2$ system (Karampour and Sawalha, 2018, Polzot et al, 2016a). In conventional HFC systems heat recovery is performed switching from floating condensing operation to gas cooler pressure control, in order to raise the condensing temperature at the desired level (Arias and Lundqvist, 2006). With $CO_2$ systems the high temperature reached at the compressor discharge side and the high heat capacity of the gas at supercritical conditions has opened to new opportunities for heat recovery solutions. Heat rejected by the refrigeration cycle can be recovered at different temperature levels, placing the heat exchanger(s) at the compressor discharge before the gas-cooler, or in place of the gas-cooler (or condenser in subcritical mode), or at the exit of the gas-cooler, acting as a sub-cooler in subcritical mode (Sawalha, 2013, Karampour and Sawalha, 2017). Typically the first heat exchanger recovers heat for domestic hot water (DHW) at 70-50 °C, the second one for space heating (50-40 °C) and the last one for pre-heating DHW or for other low temperature applications. As an example, heat at low temperature can be rejected to a heat pump which, in turn, supplies the HVAC system (Minea, 2010). Extended modelling of $CO_2$ system configurations from the simplest one, without receiver (Sawalha 2013), to the improved ones with parallel compression (Karampour and Sawalha, 2015) have been performed, showing that with a proper control strategy heat recovery can cover the whole heating demand of an average size supermarket in relatively cold climate with slightly lower annual energy use in comparison to a conventional R404A system and a heat pump for the heating load. While $CO_2$ plants with full heat recovery are appreciated in the cold Northern Europe countries (Karampour and Sawalha, 2014, Karampour and Sawalha, 2017, Karampour and Sawalha, 2018), the application of heat recovery in mild-warm countries is still an open issue and may become essential for increasing the appeal of $CO_2$ solution or for reducing the operating costs where the use of natural refrigerants is an inescapable choice, because of taxation on HFCs or laws (Polzot, 2017, Polzot et al, 2017). In fact, “all-in-one” integrated $CO_2$ energy systems have the advantage of being compact in size and environmentally acceptable, probably free from any restrictions in the future. On the other hand the challenge is on their global cost, due to the (still) high price of some components and to the possibility of showing a lower efficiency. Both heat for space heating and cooling capacity in favour of air conditioning can be provided by such a system (Cortella et al, 2018), which can be eventually coupled with a heat storage (Polzot et al, 2016b, 2016c). However most applications still involve just heat recovery, mainly for its simplicity. Furthermore, when excess heat is available, it can be successfully supplied to a district heating network. Adrianto et al (2018) estimated that giving priority to space heating and supplying the remaining quota to district heating is the most effective scenario in terms of energy and cost, when compared to other configurations and control rules with different priorities.

In this paper a R744 refrigeration system is considered, to satisfy the cooling load required by chilled and frozen food storage and display equipment in a medium-sized supermarket, while supplying DHW and space heating. The system is controlled to meet the full heating demand of the building and is equipped with an additional outdoor air-cooled evaporator, which can be used as a supplemental heat source. In this way the refrigerating unit acts as a heat pump, with the only drawback that in humid and cold climate the external evaporator needs frequent defrost operations, the same as with air to water heat pumps. An alternative could be the insertion of a third heat exchanger on the high stage pressure line, to provide heat while performing further subcooling of the refrigerant (Sheehan et al, 2018). Dynamic simulations of the refrigeration system, including mutual interactions with the building and with the HVAC, have been carried out at three mild/warm climate conditions. The results in terms of energy consumption are compared to alternative refrigeration systems without heat recovery, such as the R744 booster system itself, the R404A direct expansion system and the R134a/R744 cascade system. For all these schemes, heat pumps are considered to take care of space heating and DHW loads.

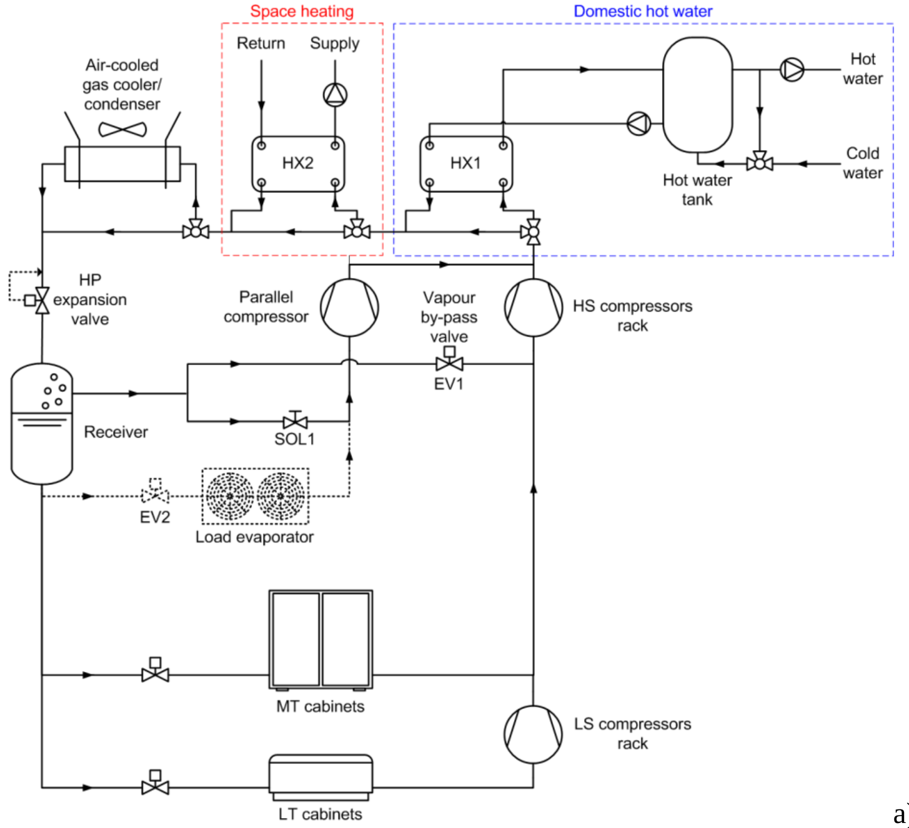
## 2. SYSTEM DESCRIPTION

Commercial refrigeration units are characterized by the presence of two levels of evaporating temperature: the Medium Temperature (MT) level, around -8 °C, in order to supply the cooling demand from the chilled food display cabinets and the Low Temperature (LT) one, around -35 °C, for frozen food storage equipment. The schematic of the investigated system (Booster with Heat Recovery – BHR) and the corresponding thermodynamic diagram are depicted in Fig. 1. The baseline is a booster system with parallel compressor (Booster – B), characterized by a receiver, at the intermediate pressure $p_{INT}$, which separates the liquid from the flash gas. The liquid is expanded to the $p_{MT}$ and to the $p_{LT}$ evaporating pressure levels, corresponding to the MT and LT evaporating temperatures respectively. In subcritical operation valve SOL1 is closed and the flash gas is expanded in valve EV1 to the MT pressure level to be finally compressed by the High Stage (HS) compressors. In transcritical operation, when a predefined production of flash gas is reached, the valve SOL1 is open, the by-pass valve EV1 is closed and thus the flash gas is compressed to the high stage pressure ($p_{HS}$) by a parallel compressor. In order to perform heat recovery at two temperature levels, separate heat exchangers are introduced in series at the exit of the high stage compressors rack. The first one (HX1) is used for the production of domestic hot water (DHW) while the second one (HX2) supplies hot water for space heating purposes. When the system is running in heat recovery mode, an air-cooled gas cooler/condenser allows further cooling the refrigerant exiting the heat exchangers, otherwise the two heat exchangers are by-passed and the whole heat is rejected into the ambient.

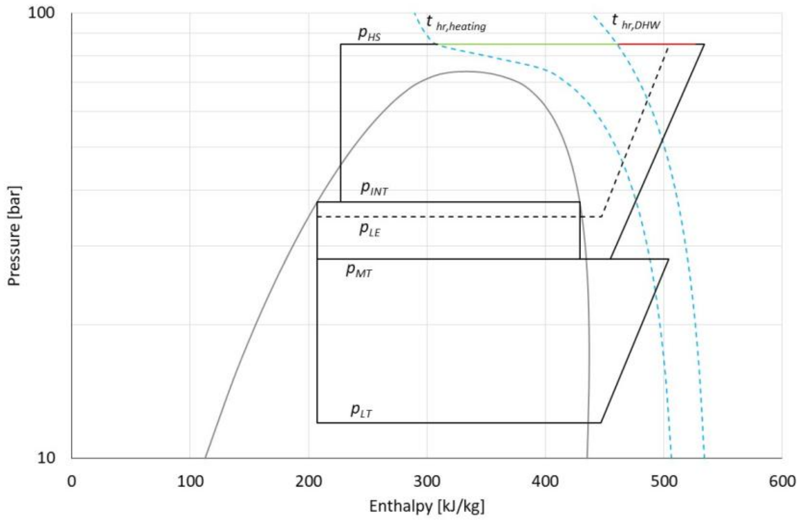
If there is no demand for heat recovery from the supermarket, the system runs in floating condensing mode i.e. the high stage pressure follows the ambient temperature; thus subcritical operation takes place when outdoor temperature is lower than 20 °C, transcritical operation when outdoor temperature is higher than 26 °C and transitional mode in between. The rules for high stage and intermediate pressure control in transcritical and transitional modes are described in Section 3.1.

On the contrary, when recovery is required, the high stage pressure of the system may be increased according to the demand, up to forcing the system to transcritical operation. Furthermore, it is possible to increase the amount of thermal energy recoverable from the two heat exchangers by reducing the subcooling in the gas cooler (Sawalha, 2013, Tambovtsev et al., 2010), thus increasing the mass flow rate of refrigerant and consequently the recoverable heat. This is done by reducing fan speed and by-passing the gas cooler.

Finally the system is also provided with an air-cooled “load evaporator” which can be used as a supplemental heat source and is activated during low refrigeration duty periods, when the heat rejected by the HX2 heat exchanger is not sufficient to meet the heating demand of the building. The pressure at the additional evaporator $p_{LE}$ can be further controlled by valve EV2 below $p_{INT}$ depending on the outdoor temperature, in order to allow for heat recovery from outside. In this case, superheated vapour at its outlet is collected by the parallel compressor, while the possible small amount of flash gas from the liquid receiver is expanded to $p_{MT}$ and finally sent to the HS compressors rack.




a)




b)

Figure 1. Sketch of the investigated Booster refrigerating system with Heat Recovery (BHR): a) schematic b) thermodynamic diagram at low outdoor temperature with heat recovery (solid line) and load evaporator activated (black dashed line).

For the sake of comparison, the following configurations are investigated: a state-of-art R744 booster system with parallel compression and heat recovery for DHW only (B), the same R744 booster system with heat recovery for both DHW and heating (BHR), supplied with load evaporator, a traditional two-temperature

R404A direct expansion system (DX) and a R134a/R744 cascade system (C). In order to fully supply the DHW and heating demands, two heat pumps (HPs) for DHW and heating are operated when DX and C systems are considered. Table 1 summarizes the configurations investigated.

Table 1. Configurations considered

| ***Acronym*** | ***Configuration*** |
|---|---|
| **B** | R744 Booster system with parallel compression and heat recovery for DHW |
| **BHR** | System B with full Heat Recovery (DHW and space heating) |
| **B+ HPs** | System B and Heat Pump for space heating |
| **DX+ HPs** | R404A Direct expansion system and Heat Pumps for DHW and space heating |
| **C+HPs** | Cascade R134a/R744 system and Heat Pumps for DHW and space heating |

## 3. MODEL DESCRIPTION

The mathematical models of the refrigeration systems and of the heat pumps have been developed in the TRNSYS environment (Klein et al., 2010), as it allows carrying on dynamic simulations of a complex system, which includes mutual interactions with the building and with the HVAC system.

As an example, the heating and cooling demands of the supermarket are affected by the presence of open display cabinets and, in turn, their cooling load profile is strongly dependent on the indoor thermal and humidity levels and on the ratio LT/MT refrigerating capacity (Mylona et al, 2018). Thus, the refrigerated display cabinets and the cold rooms in the food store have been modelled as described in Polzot et al. (2015); the total cooling capacity at rated conditions of each display cabinet is adjusted taking into account realistic and time-dependent working conditions in a supermarket. The influence of indoor air temperature and humidity on the sensible and latent fractions of the cooling load are considered as well as the time schedule for auxiliary devices (Cortella et al, 2011, Polzot et al. 2015). On the other hand the contributions due to the sensible and latent heat transfer by air infiltration from the open cabinets and to conductive and radiative heat transfer through both display cabinets and cold rooms contours are computed as credits to the HVAC system.

Hereafter the models of the refrigeration systems and of the heat pumps are described.

### 3.1. Commercial refrigeration unit

The simulations of the different commercial refrigeration systems are based on sub-hourly profiles of the cooling load at the evaporators and of the outdoor temperature. A detailed description can be found in Polzot et al. (2015 and 2016c). The thermodynamic and thermophysical properties of the refrigerants are calculated by linking our in-house routines in the Trnsys environment to the CoolProp libraries (Bell et al., 2014). Correlations from the manufacturer (Bitzer, 2017) have been used to define the global efficiency of the compressors (ratio of the theoretical compression work to the electrical power use) as a function of the pressure ratio. In detail, such correlations are (Polzot, 2017):

$$\eta_{HS,glob} = -0.0509\beta_{HS}^2 + 0.2883\beta_{HS} + 0.2550 \quad (1)$$

$$\eta_{LS,glob} = -0.0378\beta_{LS}^2 + 0.1796\beta_{LS} + 0.4553 \quad (2)$$

$$\eta_{PC,glob} = -0.0434\beta_{PC}^2 + 0.2058\beta_{PC} + 0.4815 \quad (3)$$

where $\beta$ is the pressure ratio.

Several control rules are implemented for the BHR system depicted in Fig.1. If the refrigeration system is meant to cover the full heating demand, the high stage pressure is controlled as follows:

- when the heat recovery demand is null the high stage pressure $p_{HS}$ is driven by the outdoor temperature. In particular in transcritical and transitional operation ($t_{ext}$>20 °C) the parallel compressor is activated and both the high stage pressure $p_{HS}$ and the parallel compressor suction pressure $p_{INT}$, are set according to the following correlations, which have been inferred through the optimization of the specific design cycle in order to maximize the COP:

$$p_{HS,opt} = 1.998\ t_{GC.out} + 17.058 \quad \text{[bar]} \quad (4)$$

$$p_{INT,opt} = -0.01\ t_{GC.out}^2 + 1.244 t_{GC.out} + 17.058 \quad \text{[bar]} \quad (5)$$

  where $t_{GC,out}$ (°C) is the temperature of the refrigerant exiting the condenser/gas cooler.

- when there is heat recovery demand for DHW, the high stage pressure $p_{HS}$ is increased to 49 bar. When the demand is for heating (BHR only) the high stage pressure $p_{HS}$ is iteratively increased between 75 and 85 bar until the heat recovered in HX2 meets the heating demand. When the maximum value of the high stage pressure $p_{HS}$ is reached, the system starts to reduce the gas cooler capacity by regulating the fans speed to the limiting case of switching them off (Sawalha (2013), Shi (2017)). Finally, if the heat available from heat recovery is not yet sufficient, the load evaporator is activated, the fans speed velocity is set back at the maximum value to perform subcooling and the system starts to increase the mass flow rate flowing through the load evaporator until the whole heating demand is matched.

The heat exchangers HX1 and HX2 are modelled simply by assuming appropriate approach temperature values. HX1 provides hot water at 60 °C to avoid growth of legionella. The supply temperature and the minimum return temperature for heating purposes at HX2 are fixed at 40 °C and at 30 °C, respectively.

The DX scheme is made of two independent R404A systems, for the LT and MT temperatures.

The C scheme is made of a cascade system, where a R744 plant covers the LT load, while a R134a one covers the MT load and the heat rejection from the LT system. Heat available from de-superheating is recovered for pre-heating when the heat pumps are operating to supply heating and/or DHW.

The values of the main design parameters considered for the simulations of the commercial refrigeration units are listed in Table 2

Table 2. Main design parameters for the commercial refrigeration units

| *Parameter of Commercial Refrigeration Unit* | *Unit* | *Value* |
|---|---|---|
| LT Evaporating temperature | °C | -35.0 |
| MT Evaporating temperature | °C | -8.0 |
| Useful superheating | K | 5.0 |
| Superheating in the suction lines | K | 5.0 |
| Approach temperature at the condenser | K | 5.0 |
| Subcritical subcooling | K | 2.0 |
| | | |
| *Specific for BHR* | | |
| Minimum condensing temperature | °C | 8.0 |
| Liquid receiver pressure $p_{INT}$ | bar | 37.7 |
| Maximum value of high stage pressure $p_{GC}$ in heat recovery mode | bar | 85.0 |
| Approach temperature at the gas cooler | K | 3.0 |
| Approach temperature at HX1 and HX2 | K | 5.0 |
| Approach temperature at LE | K | 5.0 |
| LE capacity | kW | 200.0 |
| Gas cooler electric power at rated conditions | kW | 3.2 |
| | | |
| *Specific for DX* | | |
| Minimum condensing temperature | °C | 25 |
| | | |
| *Specific for C* | | |
| Approach temperature at the cascade condenser | K | 5 |
| Minimum condensing temperature | °C | 25 |
| Approach temperature at the desuperheater | K | 5.0 |

### 3.2. Heat pumps

In-house models for the heat pumps suppling thermal energy for heating and for DHW production have been developed. Vapour compression cycles for R410A and R134a have been implemented in Trnsys linked to CoolProp libraries. The global efficiencies of the compressors are obtained as a function of the pressure ratio by using BITZER Software (Bitzer, 2017). More details on compressors selected in all the investigated solution and the correlations of the global efficiency can be found in Polzot (2017).

In the air/water heat pumps of the reference solution the impact of the defrost operations has been taken into account through a reduction of 10% in COP whenever the outdoor temperature goes below 0 °C .

The values of the main design parameters considered for the simulations of the heat pumps are listed in Table 3.

Table 3. Main design parameters for heat pumps.

| *Parameter of Heat Pumps* | *Unit* | *R410A (heating)* | *R134a (DHW)* |
|---|---|---|---|
| Useful superheating | K | 4.0 | 4.0 |
| Superheating in the suction lines | K | 4.0 | 4.0 |
| Subcooling in heating mode | K | 3.5 | 3.5 |
| Approach temperature at the source heat exchanger | K | 10.0 | 10.0 |
| Approach temperature at the load heat exchanger | K | 5.0 | 5.0 |
| Minimum condensing temperature in cooling mode | °C | 25 | - |
| COP reduction during defrost | % | 10 | 10 |

Simulations have been carried out and correlations for the estimation of COP and EER have been inferred as a function of the evaporating and condensing temperatures respectively. As an example, for a load temperature value of 45 °C and 2 °C respectively, the correlations for the R410A heat pumps are:

$$COP_{R410\text{A}} = 0.124\ t_e + 4.34 \tag{6}$$

$$EER_{R410\text{A}} = -0.179\ t_c + 11.23 \tag{7}$$

which are typical of high performance commercial products.

## 4. LOADS ON THE HVAC AND REFRIGERATION SYSTEMS

The system has been considered for a supermarket of 6352 m$^2$ vending area located at the ground floor of a larger modern shopping mall. The supermarket is rectangular in plan, the roof and one of the longest sides (north oriented) are insulated external walls, while the other is a single glazed window construction facing an internal hallway. The floor is attached to the underground parking and the other sides to warehouse areas. There are refrigerated display cabinets for a total length of 208 m and 10 cold rooms for the Medium Temperature (MT) and frozen food display cases for a total length of 86 m and 2 cold rooms for the Low Temperature (LT). The refrigeration plant has a capacity of 140 kW at the Medium Temperature and 28 kW at the Low Temperature evaporative level.

In this study three different locations characterized by mild climate conditions have been considered. The weather files are extrapolated from Meteonorm (Remund, 2014). In Table 4, synthetic values of daily and annual external air temperature and the value of Heating Degree Days (estimated for 20°C indoor temperature and outdoor temperature lower than 14 °C) are reported for different climates.

Table 4. Climate conditions considered.

| *Climate conditions* | *Unit* | *Climate 1* | *Climate 2* | *Climate 3* |
|---|---|---|---|---|
| Annual average temperature | °C | 18.6 | 16.4 | 11.6 |
| Annual variation of the monthly mean temperature | K | 13.4 | 15.8 | 22.2 |
| Diurnal temperature variation, maximum value over the year | K | 10.6 | 10.4 | 23.7 |
| Heating Degree Days | HDD/year | 751 | 1435 | 2404 |

The whole shopping mall has been simulated in Trnsys in order to assess the heating demand of the supermarket. For the sales area, the heating set-point temperature and the heating set-back are 20 °C and 15°C respectively, the lighting and electrical equipment load is 35 W/m$^2$ , the occupancy 0.25 person/m$^2$ and the ventilation 2 air changes per hour. The load of the refrigerated display cabinets is dynamically calculated as stated above.

Simulation are carried out with a 15 minutes time step. The monthly heating demands of the supermarket are compared in Figure 2 for the various climates, together with the demand for domestic hot water, whose usage is estimated at a maximum value of 250 dm$^3$ per hour during the opening hours.

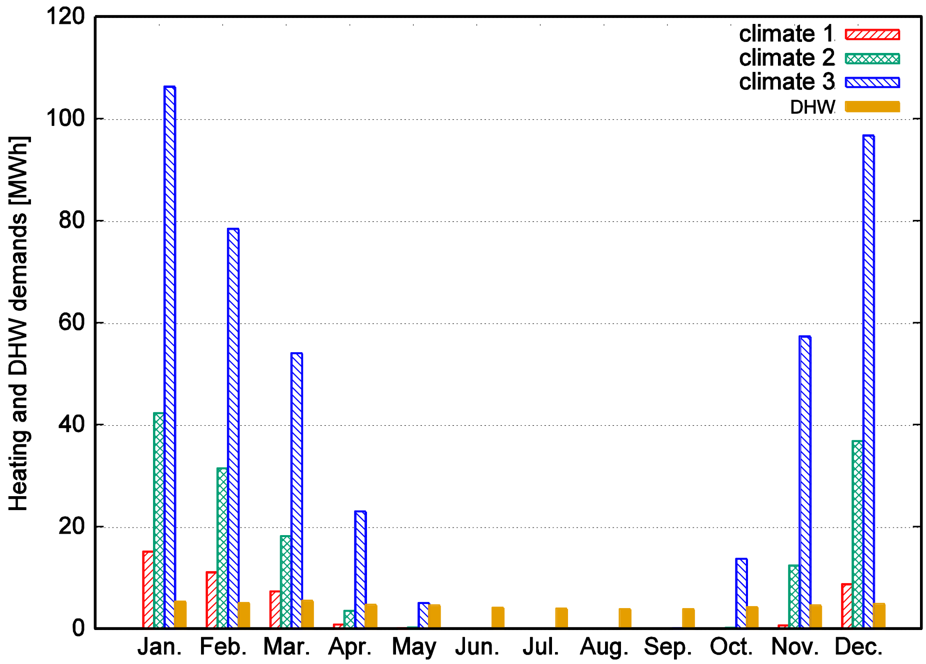


Figure 2. Heating and DHW monthly demands for different climates

As stated above the cooling load profiles on the evaporators of the refrigerating unit are evaluated on the basis of a detailed simulation of the display cabinets and cold rooms and their interaction with the indoor ambient with a time step of 15 min. An example of annual and weekly profiles of the cooling load can be found in (Polzot et al. 2015). In Figure 3, the monthly values are reported for climate 2, separated for the chilled food (MT) and for the frozen food (LT). Due to the equipment installed in the supermarket, the load on LT evaporators is much lower than on MT ones. The demand changes over the year because of the different set point values for the indoor temperature in the opening hours and because of the influence of outdoor conditions when the HVAC is switched off in the closing hours. The difference on annual basis on the cooling demands is of the order of +5% for LT and +8% for MT at climate 1 with respect to climate 2 and -5% for LT and -8% for climate 3 with respect to climate2.

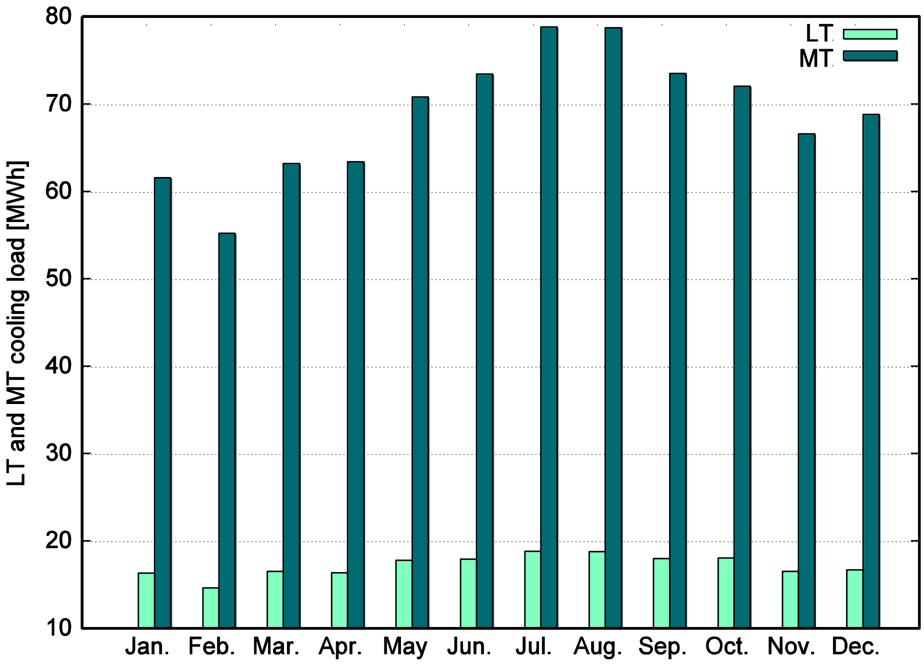



Figure 3. Monthly cooling load on LT and MT evaporators of the refrigeration system for climate 2.

## 5. SYSTEM SIMULATION AND COMPARISON

Various comparisons are made to investigate the feasibility and convenience of full heat supply by means of the R744 booster system with Heat Recovery (BHR). Firstly the effect of heat recovery is investigated on the operating conditions and performance of the baseline R744 booster (B). Then the advantages of heat recovery from the refrigeration system are assessed by comparison to a baseline solution where the refrigeration demand and thermal energy demand are independently supplied by two R404A direct expansion refrigeration system and by two air to water heat pumps respectively (DX+HPs). Finally, regarding the possibility of heat recovery from a refrigeration system other than a R744 booster, a R134a/R744 cascade system coupled with air to water heat pumps (C+HPs) has been considered.

### 5.1. Comparison among the R744 booster systems without and with heat recovery (B vs. BHR)

According to the control rules described above, in heat recovery mode the high stage pressure may be increased to meet the full heating demand of the supermarket. Figure 4 makes a comparison in terms of frequency over the year of the high stage pressure $p_{HS}$ among the R744 transcritical booster system with parallel compression (B) and the same system with heat recovery (BHR) for both the hottest (climate 1) and the coldest (climate 3) conditions considered. It may be noticed that, in climate 1, the high stage pressure in heat recovery is quite close to the distribution for the basic booster, except for a small peak at the maximum value of 85 bar (0.8%) and slightly lower frequency values in the subcritical range between 37.7 and 55 bar. On the contrary, for climate 3, the frequency peak shifts from the subcritical value of 49 bar (17.6%), which is the minimum value for recovering heat for DHW, to the maximum value of 85 bar, which occurs for 13% of working hours. Thus, while in climate 1 the heat recovery from the refrigeration system covers all the heating demand without forcing the high stage pressure to values much higher than the optimal ones, in the coldest climate the BHR system operates more frequently at higher pressure than the B system, with a detriment in its performance as refrigerator.

As stated above, when the maximum value of the $p_{HS}$ is reached, the subcooling at the gas cooler is reduced by reducing the fan velocity or bypassing the heat exchanger. Figure 5 shows the comparison in terms of frequency over the year of the gas cooler outlet temperature between the two systems. In climate 1 the temperature of carbon dioxide $t_{GC,out}$ exiting the gas cooler is not increased to meet the full heating demand of the supermarket, which is satisfied just by occasionally increasing $p_{HS}$ . In climate 3, the gas cooler is by-passed mainly when the outdoor temperature is low, thus the frequency of minimum $t_{GC,out}$ decreases by half (from almost 25% to 12,5%), i.e. the system does not always take advantage of the low outdoor temperature to condense at the minimum value. The peak at the minimum temperature is still present because, when the load evaporator is activated to meet the full heating demand, the subcooling is turned back on.

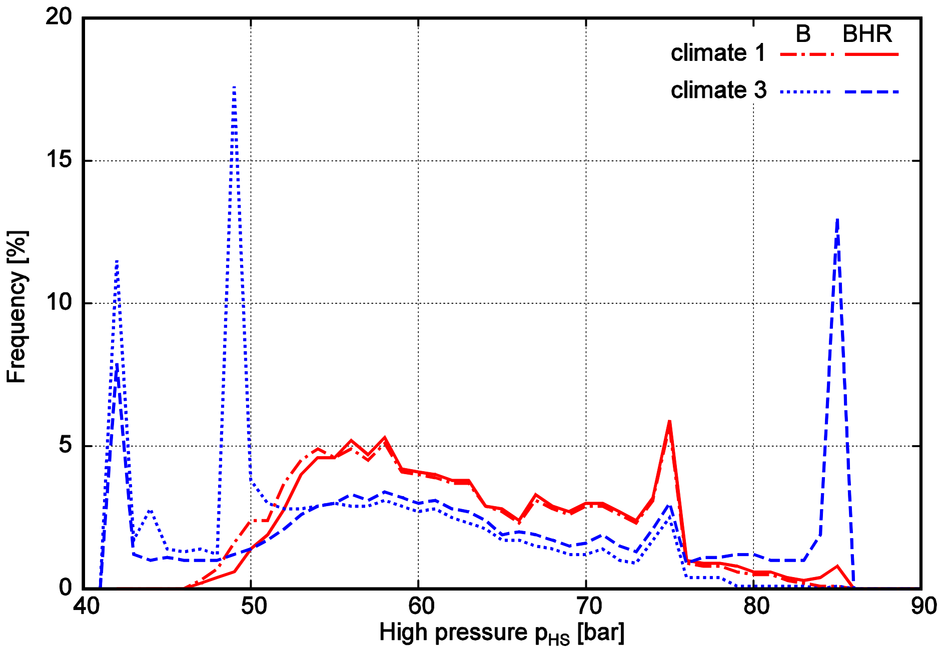



Figure 4. Annual frequency of the high stage pressure value $p_{HS}$ in B and BHR systems in climate 1 and climate 3.

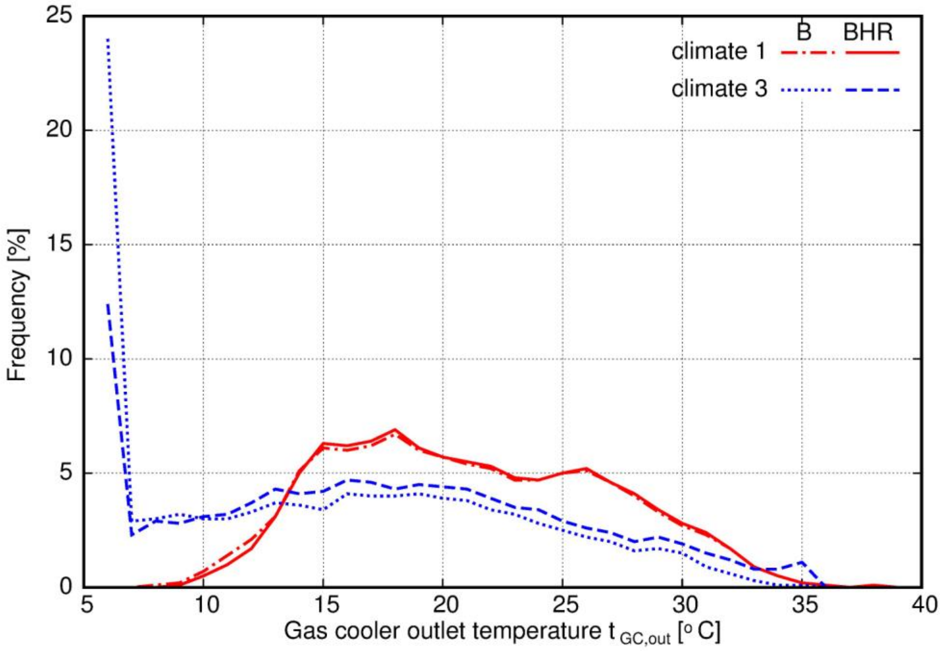



Figure 5. Annual frequency of the gas cooler outlet temperature value in B and BHR systems in climate 1 and climate 3.

Figure 6 shows the percentage of time over the year at which the load evaporator is activated for the BHR solution. In the coldest climate it works up to 9% of the year. In the hottest climate considered, the load evaporator works for a very few hours over the year, thus the installation of this device is not to be suggested at all climates conditions, allowing a significant reduction in the investment cost.

In Figure 6 the working hours of the parallel compressor in B and in BHR configurations is also reported . In the B solution, the additional compressor is activated 3493 hours per year in climate 1, 2859 h in climate 2, 1705 h in climate 3 i.e. whenever the outdoor temperature exceeds 20 °C. In the BHR system, the parallel compressor is activated also in the cold season when the high stage pressure is increased. Thus the difference between B and BHR values measures the fraction of time over the year when the system operation is controlled by the heating demand. It may be noticed that with heat recovery the working hours of the parallel compressors for the coldest climate increase by 130% reaching a value close to that for the hottest climate.

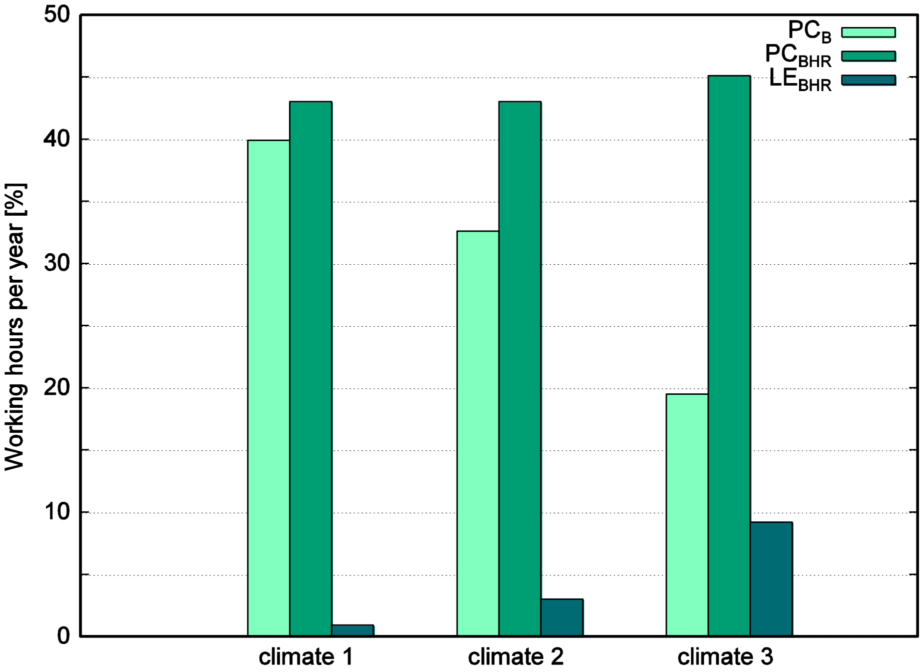



Figure 6. Parallel compressor (PC) and load evaporator (LE) working hours per year in the evaluated climate conditions.

In the winter season, the sequential control on the high stage pressure, on the subcooling and on the load evaporator drives to an increment in the energy use due to the high stage and the parallel compressors. The colder the climate is, the higher the increment during the heating season is. Figure 7 shows the daily energy use profiles over a whole year for the BHR system (refrigeration, space heating and DHW) compared with the B system (refrigeration and DHW) in climate 3. Due to heat recovery, the daily electric consumption in December reaches the summer peak value, while in January it exceeds it by 20%.

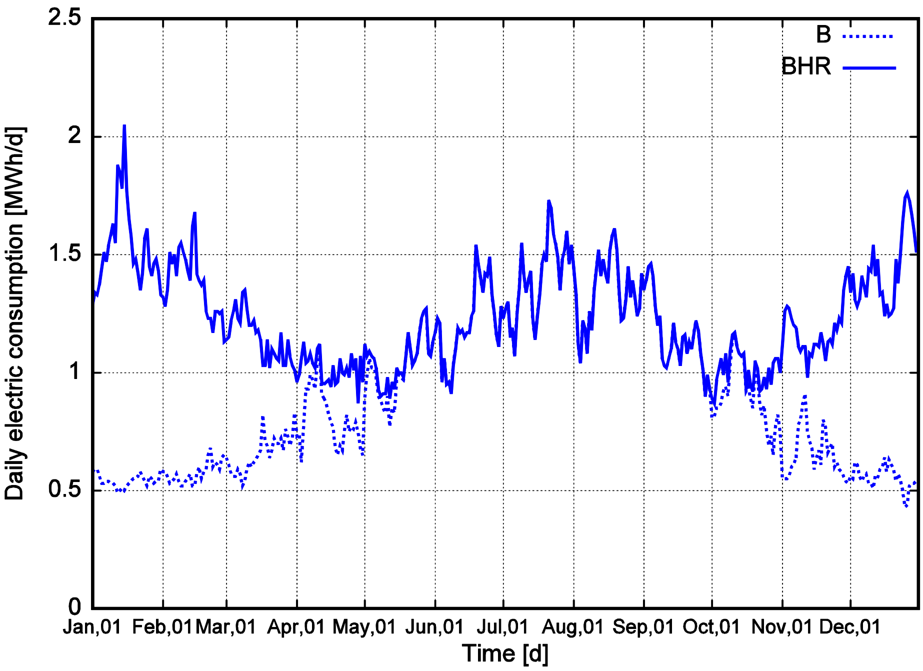



Figure 7. Daily electrical energy consumption profiles of the BHR and B systems in climate 3

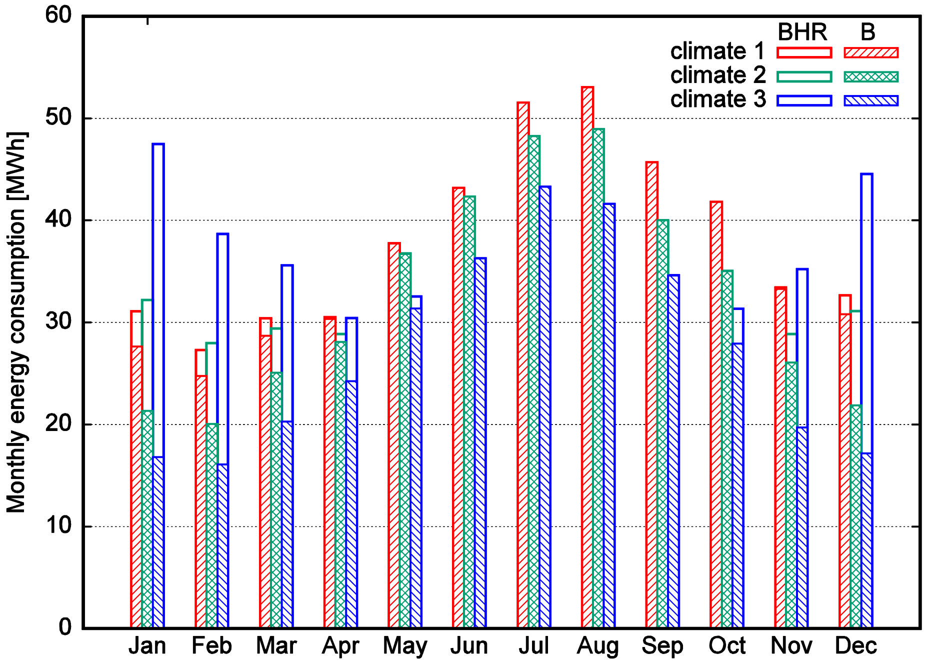


Figure 8. Monthly electrical energy consumption values of the BHR and B systems

The comparison between the BHR and B configurations in terms of monthly electrical energy consumption is reported in Figure 8 for the selected climate conditions. R744 refrigeration systems are known to perform worse in mild climates than in cold climates (Cavallini and Zilio, 2007, Finckh et al., 2011, Sawalha 2017). When space heating is not needed, the BHR coincide with B and the performance of the system is highly sensitive to the gas cooler exit temperature. In climate 3, which is characterized by low ambient temperatures for a long time over the year, the electrical energy consumption of B system is lower than in the other two climate conditions, up to 19% less than climate 1 between June and September. On the other hand, when heat recovery for space heating is activated, the BHR consumption is much higher than B system for climate 3 and just slightly higher for the other two climate conditions.

On an annual basis, the electrical consumptions are reported in Table 5. The R744 booster system controlled to cover the total heating demand of the supermarket (BHR) consumes just 2% more in climate 1, 9% in climate 2 and up to 37% more in climate 3 than the B system. It should be noticed that the heat recovery levels off the annual consumptions among the three climates.

A more complete comparison can be done adding to the electrical consumption of the refrigeration system B the consumption of a separate system supplying the space heating demand (B+HPs).

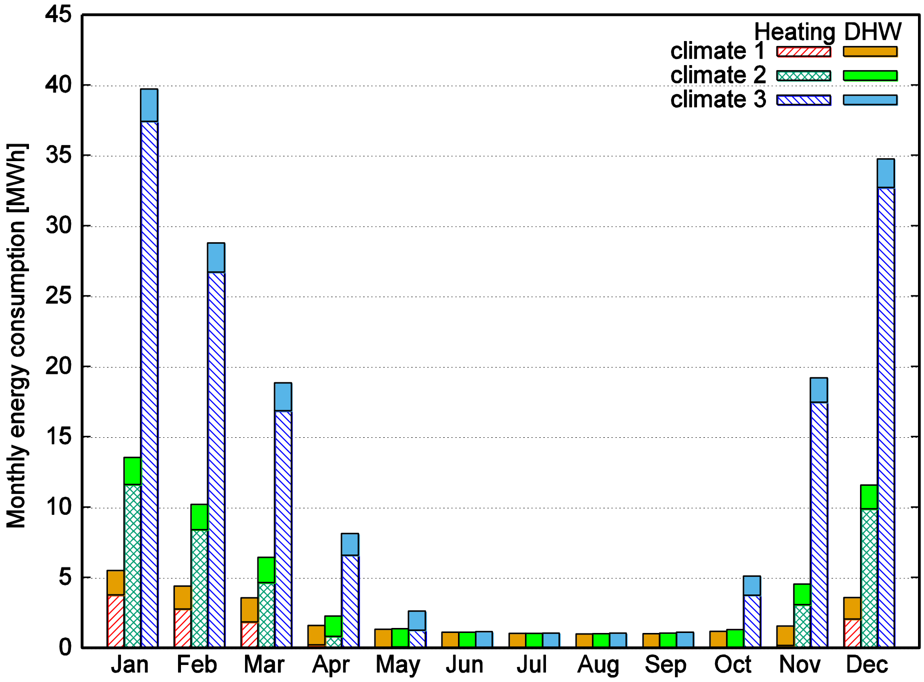


Figure 9: HPs energy consumption for space heating and DHW production at the selected climate conditions.

In case the integration is given by heat pumps, their monthly energy consumption is depicted in Figure 9 for the selected climate conditions. The total energy consumption on annual basis is reported in table 5 (B+HPs). The increment is then negligible for climate 1 and climate 2 while it is around 4.5% in climate 3 with respect to the BHR system. At mild climate conditions the choice among BHR and B+HPs is then driven mainly by investment costs, while at cold climate conditions also some reduction in the energy consumption has to be accounted for when heat recovery is chosen.

The heating COP of the integrated system has been estimated as the ratio of the heat recovered in favour of space heating ($q_{heating}$) to the energy used in excess for heat recovery, i.e. the difference between the energy used by BHR system ($W_{BHR}$) and B system ($W_B$) (Karampour M., Sawalha S., 2018):

$$\mathrm{COP_{HR}} = \frac{q_{heating}}{W_{BHR} - W_B}$$

Its monthly average value is compared to the monthly average COP of a standalone heat pump. Figure 10 reports this comparison, and shows that the integrated solution performs better at all climates, especially at the cold climate in the coldest months, when the heat pump cannot take advantage of higher outdoor temperature.

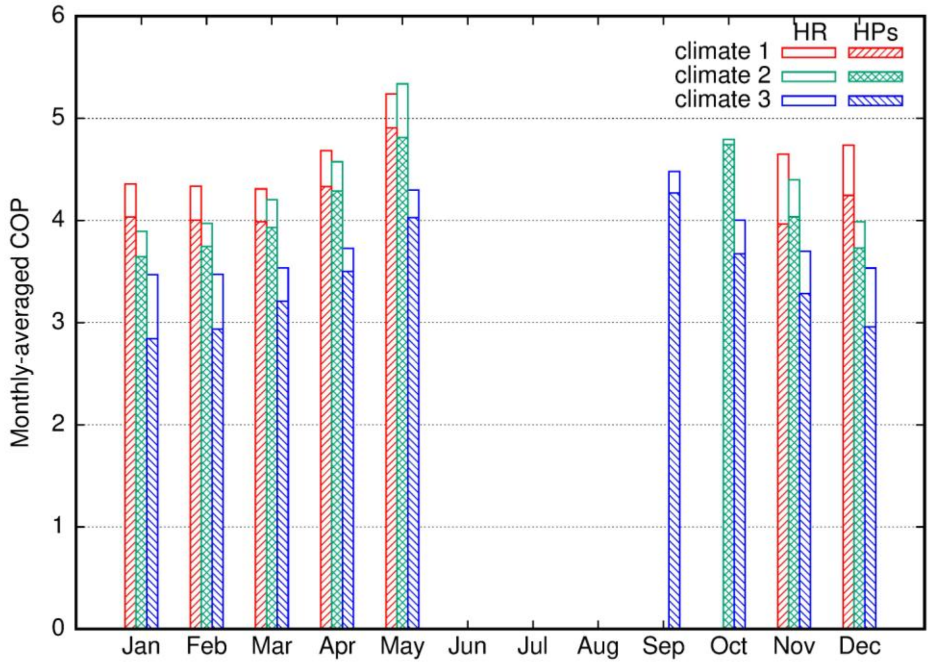



Figure 10. Heating COP of the integrated system (HR) compared to that of a standalone heat pump (HPs).

Table 5. Annual electrical energy consumption for the selected configuration [MWh/year]

| *Configuration* | *Climate 1* | *Climate2* | *Climate 3* |
|---|---|---|---|
| **B** | 448.5 | 393.7 | 329.5 |
| **BHR** | 458.4 | 429.8 | 451.7 |
| **B+HPs** | 459.3 | 432.2 | 472.2 |
| **DX+ HPs** | 496.4 | 480.5 | 539.4 |
| **C+HPs** | 487.0 | 475.6 | 538.9 |

### 5.2. Comparison between BHR and alternative systems

As stated above, the performance of the integrated R744 system (BHR) is compared also to traditional stand-alone systems which totally (DX+HPs) or partially (C+HPs) make use of hydrofluorocarbon refrigerants. All these systems are intended to fully meet refrigeration, DHW and heating loads, with the support of heat pumps.

In Figure 11 the monthly energy consumption of the DX+ HPs and C+HPs configurations are compared to the BHR system at intermediate climate conditions. It can be noticed how efficient is heat recovery from R744 booster system in wintertime, as the total energy consumption for the configuration with stand-alone refrigeration and heating systems (DX+HPs) exceeds up to 25% the BHR one. Furthermore, the cascade refrigeration system integrated by heat pumps (C+HPs) seems less promising as it is more efficient only in the summertime and leads to a total energy consumption slightly higher than DX+HPs. The influence of outdoor conditions on the performance of the R744 system is clear looking at the comparison in the summertime, where the C+HS system is better performing even if the heat pump contribution is required to supply DHW at 60 °C.

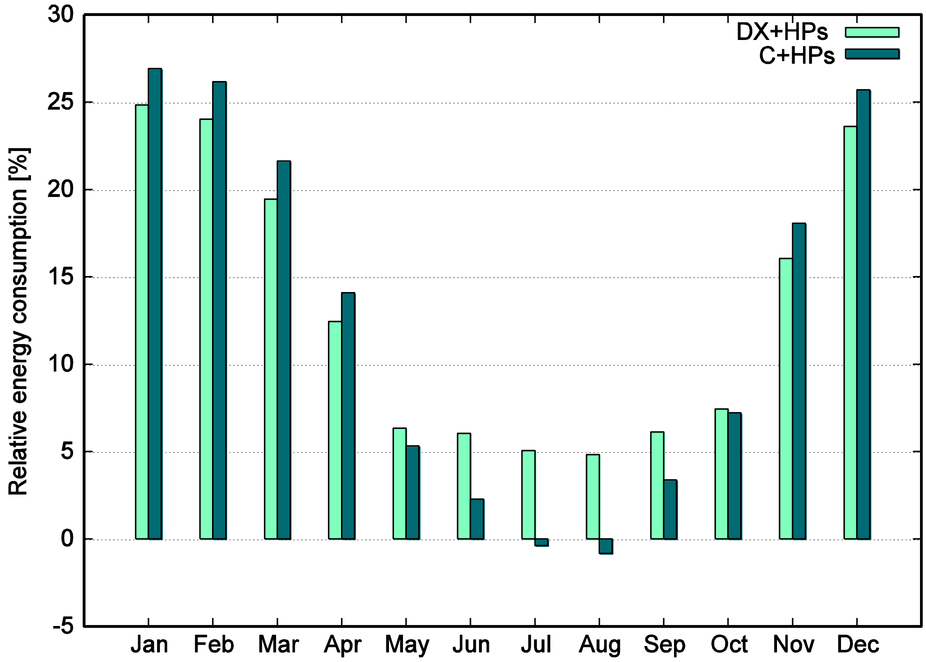



Figure 11 Relative energy consumption (%) of the DX+HPs and C+HPs configurations compared to BHR system on monthly basis at climate 2.

The comparison on annual basis is reported in Table 5 for all climates. The integrated system BHR consumes from 7.7% (in climate 1) to 16.3% (in climate 3) less than the stand-alone system DX+HPs. The availability of heat recovered at high temperature in the transcritical booster system gives better savings, from 5.9% (in climate 1) to nearly 16% (in climate 3), also with respect to the cascade system C+HPs.

It is once again shown that the R744 systems are more advantageous in the cold climates even if heat recovery is performed, forcing transcritical operation when it could be avoided. The all-in-one configuration offers further advantages among which a final solution to the need of reducing GWP of refrigerants, the reduction in the footprint of the plant and in its investment cost.

## 6. CONCLUSIONS

Dynamic simulations performed with a validated model on a commercial refrigeration system interacting with the heating plant proved the feasibility of a synergic solution, where heat from the refrigerating plant is recovered to fully satisfy the DHW and heating needs of the building. Carbon dioxide was chosen as the fluid for the refrigerating plant, whose operation had to face prolonged transcritical conditions also in wintertime, to supply heat while guaranteeing the cooling load for food preservation, An additional evaporator has been installed outdoors to act as a supplemental heat source when recoverable heat is not sufficient. Significant energy saving (up to 16 %) was predicted when comparing this solution with common current configurations like a HFC direct expansion cycle or a cascade cycle both supported by heat pumps to supply heating. A minor energy saving (up to 4.5 %) was estimated when comparing a $CO_2$ systems with full heat recovery to one supported by heat pumps for heating purposes. In this case an “all-in one” solution could be considered, to supply not only heating in the wintertime but also air conditioning in the summertime, with a significant cut in the investment cost.

The main conclusion is that $CO_2$ systems have some peculiarities which can be exploited to reduce the gap in energy use when compared to traditional HFC systems in mild/warm climates. Enlarging the boundaries of the investigation to include the heating systems allows a wider point of view and suggests synergic solutions where the total running costs can be effectively reduced.

## 7. ACKNOWLEDGMENT

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