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Energy analysis of a transcritical CO_2 supermarket refrigeration system with heat recovery

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Abstract

Carbon dioxide (R744) is widely used as refrigerant in supermarkets located in cold weather sites thanks to its negligible environmental impact and its favourable thermo-physical properties. Due to its low critical temperature, transcritical operations in CO_2 refrigeration systems can commonly take place, increasing the energy consumption substantially. On the other hand, the high temperatures reached by the CO_2 in the high pressure heat exchanger (gas cooler) potentially allow recovering a large amount of heat at different temperature levels according to the supermarket needs.
The paper deals with the energy performance evaluation of a R744 refrigeration system, which provides the selected supermarket with DHW and heating, besides satisfying the cooling load required by the refrigerated and frozen food storage equipment. The system is equipped with an additional air-cooled evaporator which can be used as a supplemental heat source, to increase the amount of heat recovered and meet the full heating demand of the building. Different control strategies are examined in order to minimize the electric consumption and, contemporary, maximize the heat recovery.

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Keywords: R744 booster system; DHW; HVAC; systems integration; control strategy.

1. Introduction

Many efforts are being recently devoted to reduce the environmental impact of refrigerating systems in commercial refrigeration applications. Both the choice of low Global Warming Potential (GWP) refrigerants and the increase in

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the energy efficiency of plants are being considered, to contain the direct and indirect contributions to the greenhouse effect.

Further opportunities for energy saving emerge when the boundaries of the system into consideration are extended to the whole supermarket or shopping mall. In fact, heat recovery from the commercial refrigeration systems in favor of HVAC and DHW production can be effectively performed. Up to now the prevalent solution with conventional HFC refrigeration systems consists in using waste heat rejected by condensers directly for the whole space heating [1], in parallel to a heat pump to support space heating [2], or for supplying the pre-heating coil of the air handling unit [3]. However heat recovery affects negatively the performance of the refrigeration system, because the condensing temperature must be raised for an effective recovery, thus preventing from taking advantage of a floating condensing control rule.

Carbon dioxide (refrigerant R744) employed as the only working fluid is proving to be a viable choice as replacement of halogenated hydrocarbons. Its negligible GWP almost zeroes the direct environmental impact, but its low critical temperature (31 °C) forces transcritical operations when heat rejection cannot be performed below the critical point, resulting in a detriment of the efficiency of the system in mild and hot climate conditions. However various measures can be adopted to force subcritical operation or to improve the efficiency of the system in all operation modes, thus reaching the same COP levels typical of the HFC refrigeration systems. Furthermore, the high temperature reached with CO_2 at the compressor discharge allows heat recovery at various temperature levels, e.g. for domestic hot water (70-50 °C), direct space heating (50-40 °C), space heating through a heat pump, fresh air preheating or snow melting (40-30 °C). Bearing in mind that with CO_2 in transcritical mode heat is rejected through a gas cooler where CO_2 undergoes a single phase gas cooling, the three temperature levels mentioned above can be met through three heat exchangers: the first placed before the gas cooler (acting as a desuperheater in subcritical operations), the gas cooler itself (acting as a condenser in subcritical operations) and the third downstream of the gas cooler (acting as a subcooler in subcritical operations) [4]. At all these conditions the discharge pressure has to be optimized in order to achieve the maximum COP of the system.

Sawalha [4] estimated that heat recovery from the desuperheater is able to cover the entire heating demand of an average size supermarket in relatively cold climate, leading to slightly lower annual energy consumption when compared to a conventional R404A refrigeration system with separate heat pump for heating needs. Furthermore, improvements in the energy efficiency can be achieved by means of a mechanical subcooler [5, 6], i.e. a separate refrigerating unit aimed at cooling down the refrigerant leaving the gas cooler, and/or parallel compression [5, 6], i.e. an auxiliary compressor aimed at compressing directly to high pressure level (HP) the vapour exiting the receiver. The latter technology was proved to be beneficial to CO_2 refrigeration systems integrated with AC and operating in cold/mild climates [7]. Heat recovery can be improved through an additional evaporator placed outdoors and acting as a supplemental heat source [8]. Heat recovery affects in some way the performance of the refrigerating system and appropriate devices and control rules are required. As an example, a gas-cooler by pass conveniently operated could be adopted to attain the optimal compromise between heat recovery and efficiency. Tambovtsev et al. [9, 10] discussed such aspects and tested some control strategies.

In this paper a R744 booster refrigeration system operating in mild climates is theoretically investigated. Heat recovery from two heat exchangers connected in series at the exit of high stage compressor is used to provide a selected supermarket with DHW and space heating. The employment of an additional evaporator to guarantee full supply of both the DHW and heating demand is taken into account. Each component of the system is modelled in Trnsys environment [11] by developing in-house Types, based on thermodynamic relations. The cooling load profiles at the refrigeration system evaporators and the heating demand of the building result from the dynamic simulation of the commercial building and of the refrigerated food storage equipment. The results are compared to the baseline i.e. a R134a/CO_2 cascade refrigeration system together with a R410A heat pump for space heating and hot water production.

Nomenclature

AC	air conditioning
BS	baseline system: R134a/CO_2 cascade refrigeration system with R410A heat pump
COP	coefficient of performance
DHW	domestic hot water
$\dot{E}_{EL}$	power consumption, kW
GWP	global warming potential
HFC	hydrofluorocarbon
HP	high pressure
HRS	investigated system: R744 booster system with heat recovery
HS	high stage
HVAC	heating, ventilating, and air conditioning
HX1	domestic hot water heat exchanger
HX2	space heating heat exchanger
LS	low stage
LT	low temperature
MT	medium temperature
$\dot{Q}_{CO}$	cooling demand, kW
$\dot{Q}_{HR}$	heat recovered, kW

2. Methods

2.1. System description

The investigated system, a R744 booster system with heat recovery and auxiliary compressor, is depicted in Fig. 1. The liquid exiting the receiver is expanded from the intermediate pressure to two different pressure levels, according to the temperature levels of the chilled and frozen display cabinets: at 26.5 bar for Medium Temperature (MT) and at 12 bar for Low Temperature (LT), corresponding to -10 and -35 °C evaporating temperature respectively. The whole amount of flash gas generated by the high-pressure (HP) expansion valve is compressed by the auxiliary compressor or, in subcritical operations, it is expanded to the MT pressure level by the vapour by-pass valve and is compressed by the High Stage (HS) compressors.

Two heat exchangers at the exit of HS compressors recover heat at two different temperature levels. 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 cooling down the refrigerant exiting the heat exchangers, otherwise the two heat exchangers are by-passed and the whole heat is rejected into the ambient.

The system is also provided with an air-cooled "load evaporator" which 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. In this study the pressure at the additional evaporator is not optimized but is fixed at the same pressure level of MT display cabinets.

The discharge pressure of the system is controlled according to the heating demand so that when the supermarket does not need heat the high pressure follows the ambient temperature, while when heating energy is required it is increased and more heat is available. In heat recovery mode the discharge pressure and the auxiliary compressor suction pressure, i.e. the intermediate pressure, are optimized to maximize the overall efficiency of the system (COP_{H+R}) defined as the ratio between the sum of the heat recovered ($\dot{Q}_{HR}$) and the cooling demand ($\dot{Q}_{CO}$) to the power consumed ($\dot{E}_{EL}$):

$$COP_{H+R} = \frac{\dot{Q}_{HR} + \dot{Q}_{CO}}{\dot{E}_{EL}} \tag{1}$$

In heat recovery mode the gas cooler acts as a subcooler improving the performance of the system. On the other hand the total mass flow rate of the refrigerant flowing through the refrigerating unit and, consequently, the amount of energy recoverable from the two heat exchangers reduce as the degree of the subcooling increases. In order to provide the supermarket with all the heating demand the gas cooler can be by-passed thus making more heat available for recovery [4, 9].

The above mentioned system configuration (HRS) is compared with a baseline system (BS) where a typical R134a/CO_2 cascade refrigeration system satisfies the cooling load required by the display cabinets and the cold rooms and a R410A heat pump provides space heating and domestic hot water.

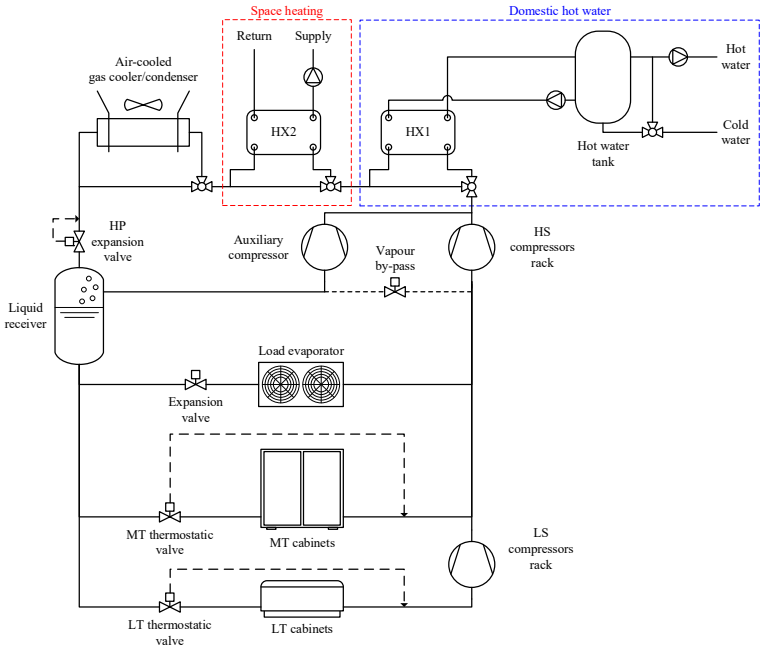


Fig. 1. Schematic of R744 refrigeration system with two different heat recovery temperature levels and an additional air-cooled evaporator.

2.2. Operating conditions

The R744 booster refrigeration system and the baseline cascade refrigeration system are designed to supply 140 kW at MT and 20 kW at LT.

The hot water usage is estimated at a maximum value of 180 dm^3/h during opening hours. The supply water set-point temperatures are 60 °C for DHW and 35 °C for the space heating, while the minimum return temperature is assumed to be 30 °C.

Table 1 and Table 2 summarize the operating conditions used in the evaluated systems.

Table 1. Operating conditions of the HRS system.

R744 booster system operating conditions	Unit	Value
Useful superheating	K	5
Superheating in the suction line	K	5
Approach temperature of the air-cooled gas cooler/condenser	K	3
Approach temperature of the HX1 and HX2 heat exchangers	K	5
Subcooling in subcritical operations	K	2

Table 2. Operating conditions of the BS system

R134a/CO_2 cascade system operating conditions	Unit	Value
Useful superheating	K	5
Superheating in the suction line	K	5
Pinch point temperature of the cascade condenser	K	5
Approach temperature of the condenser	K	10
Minimum condensing temperature	°C	25
R410a heat pump operating conditions	**Unit**	**Value**
Useful superheating	K	4
Superheating in the suction line	K	4
Approach temperature of the condenser	K	5
Approach temperature of the evaporator	K	10
Subcooling	K	3.5
Defrost COP reduction	%	10

2.3. Models

Each component of the system has been modelled in Trnsys environment [11] by developing in-house Types, based on thermodynamic relations. BITZER Software [12] has been used to define the global efficiencies of the compressors as a function of the pressure ratio, whereas the thermodynamic properties have been obtained by using CoolProp libraries [13].

The cooling load profiles of the refrigeration system and the heating demand of the building result from the dynamic simulation of the commercial building and the refrigerated food storage equipment. 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 [14]. 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 [15]. A detailed description on how the components of the refrigeration system have been modeled can be found in Polzot et al. [16].

The energy consumption of the system is evaluated in three different Northern Italy climate conditions: “climate A” which corresponds to Genoa, with a mild climate (at the seaside), “climate B” which corresponds to Udine, a location at almost the same latitude but 100 km far away from the sea and “climate C” which corresponds to Turin.

Table 3 and Fig. 2 report the most significant values of the three climates.

Table 3. Climate conditions considered.

	climate A	climate B	climate C
Annual average temperature	16 °C	14 °C	12 °C
Average annual temperature fluctuation	16 K	20 K	21 K
Maximum daily temperature fluctuation	10 K	19 K	18 K
Heating Degree Days	1435 HDD	2323 HDD	2617 HDD
Average temperature during the heating season	11°C	8 °C	6 °C

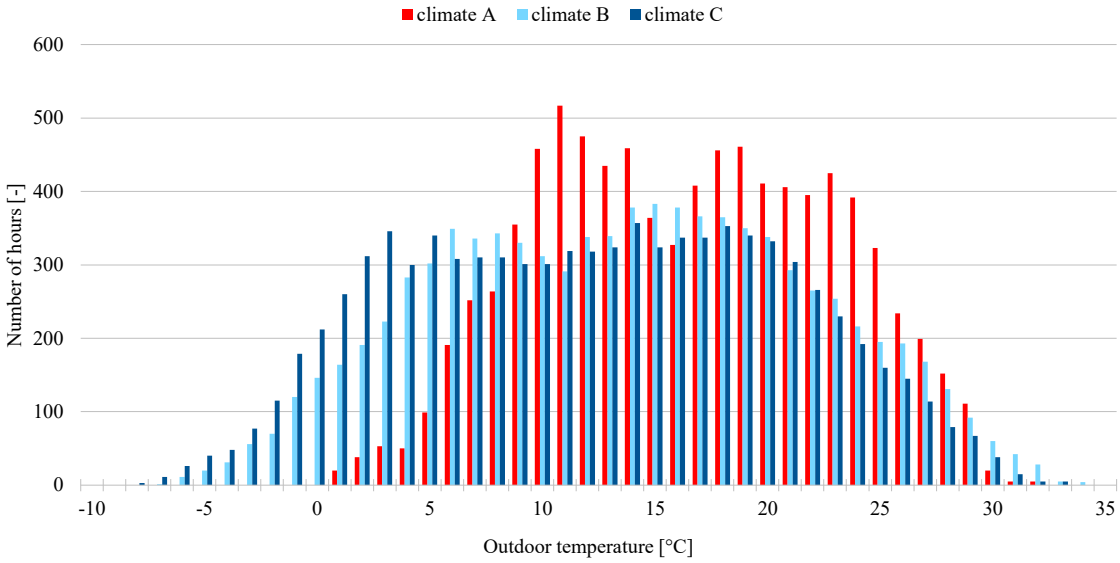


Fig. 2. Temperature bins for the selected climate conditions.

In Fig. 3 the heating demands with respect to the three different climate conditions are shown together with the DHW demand. As can be noticed the DHW demand is negligible if compared to the heating one.

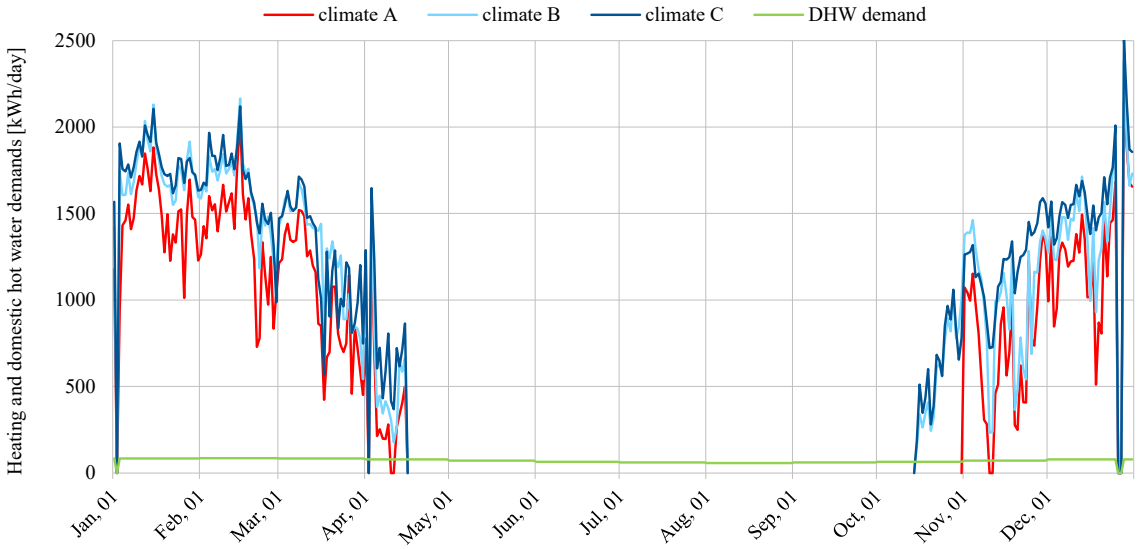


Fig. 3. Heating and domestic hot water daily demands for the selected climate conditions.

3. Results

The HRS system, i.e. the R744 booster system with heat recovery, is designed to cover the total heating demand of the supermarket, while in the BS system a dedicated R410A heat pump provide the DHW and the hot water for the space heating. Fig. 4 compares the daily profiles of the electrical energy consumption of the refrigeration system and the heating system in the BS configuration for climate B. In comparison with climate B, the heating demand of the building is 20% lower for climate A and 14% higher for climate C, while the heat pump consumes 32% less energy in climate A and 12% more in climate C. This is due to the larger number of defrost operations in the colder climates.

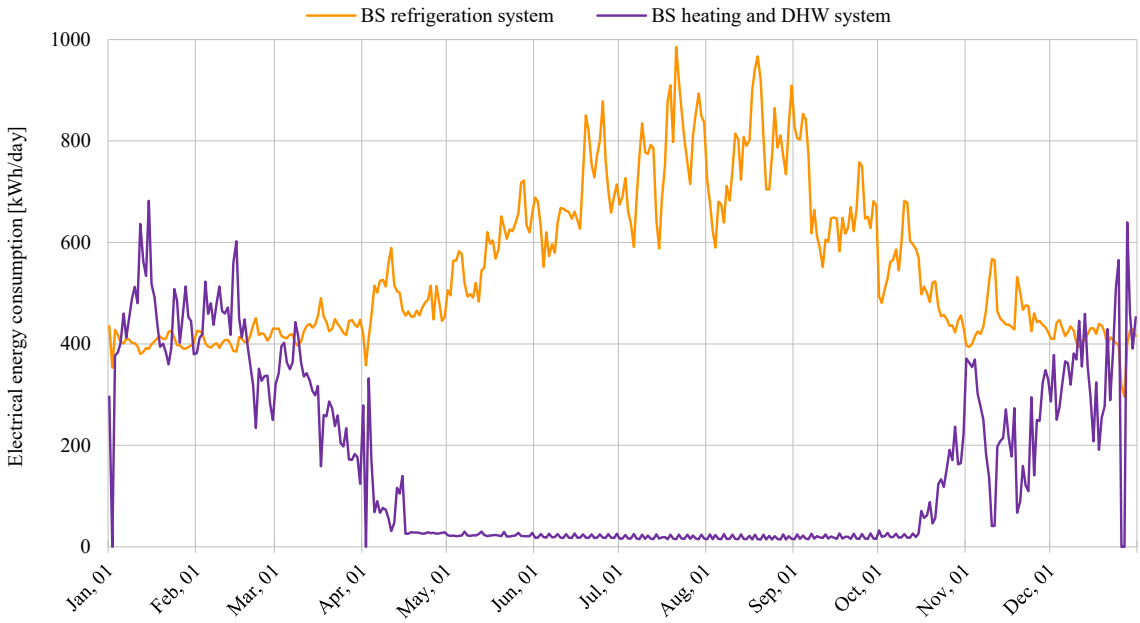


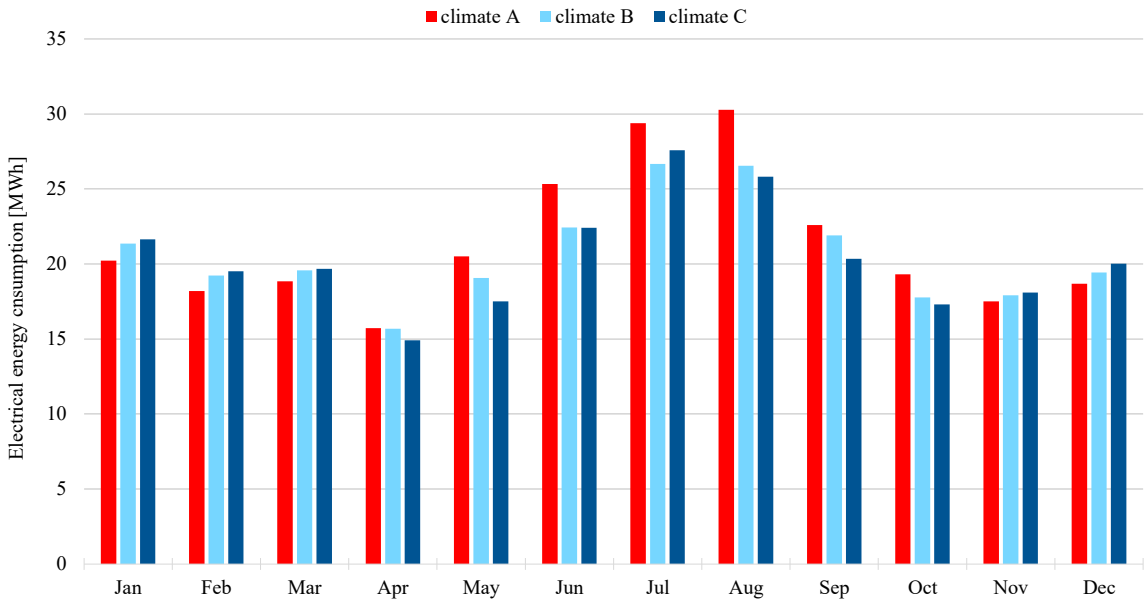
Fig. 4. Baseline electrical energy consumption at climate B.

The adoption of the HRS system leads to the elimination of the electrical energy consumption associated with the heat pump, whereas the increase in the high pressure in the R744 booster system drives to an increment in the power input related to the HS compressors. In climate B and climate C the larger number of hours in which the gas cooler is by-passed to recover more energy, leads to a further increment in the energy consumption. This is due to the larger amount of flash gas, which causes an increase in the power input associated with the auxiliary compressor. Running the air-cooled "load evaporator" has the main effect of increasing the mass flow rate of refrigerant, allowing to cover the heating demand of the building in the coldest days.

Fig. 5 compares the monthly electrical energy consumption of the HRS system for the various climate conditions. CO_2 refrigeration systems are known to perform worse in mild climates than in cold climates [14]. In the hot season, when heating the supermarket is not needed, the performance of the system is highly sensitive to the gas cooler exit temperature. In climate A, which is characterized by high ambient temperatures for a long time over the year, as shown in Fig. 2, the electrical energy consumption in the hot season is 12% higher than in the other two climate conditions.

The results in terms of relative monthly energy consumption of the HRS system in comparison with the baseline reveal that the investigated system leads to high energy savings in the cold season, especially in the climates conditions characterized by low outdoor temperatures. The energetic benefits related to the adoption of HRS configurations bring down the electricity consumption in the heating season by 8% in climate A, by 14% in climate B and by 17% in climate C.

Fig. 5. HRS system electrical energy consumption at the selected climate conditions.



Comparing HRS with BS (Fig. 6) it appears how in the summer season the CO_2 booster refrigeration system consumes about 10% more energy than the R134a/CO_2 cascade system in all the investigated climate conditions, while it is greatly advantageous in the heating season.

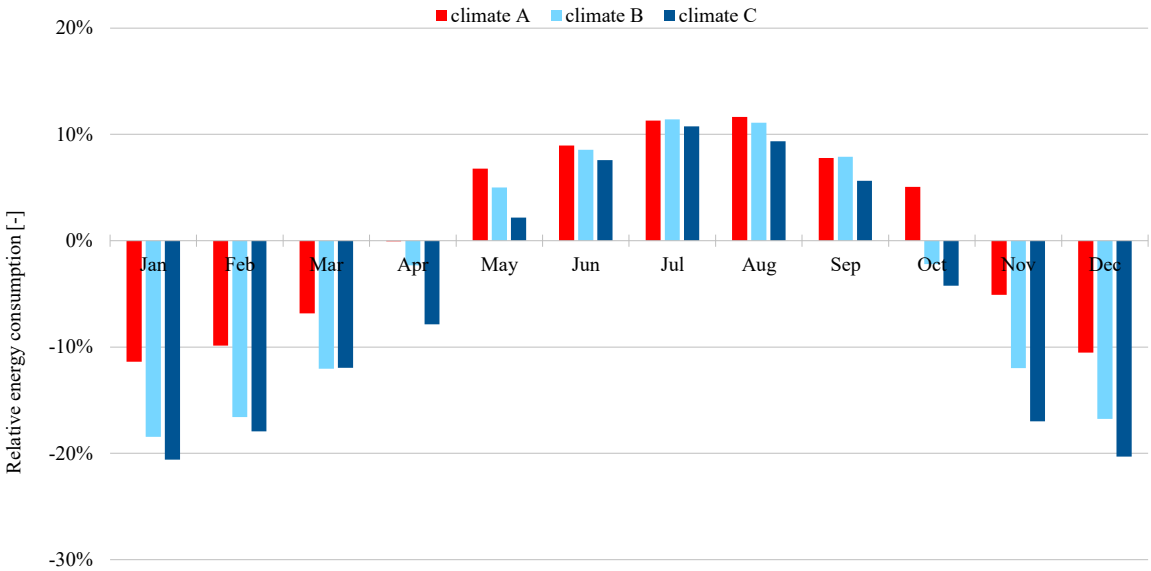


Fig. 6. Relative energy consumption of the HRS system in comparison with the BS system at the selected climate conditions.

Table 4 reports the relative annual energy consumption of the HRS system in comparison with the BS configuration. The high annual average temperature and the low daily and average annual temperature fluctuations, which characterize climate A, does not allow achieving energy savings. The global energy consumption of the HRS

solution in climate B is comparable to that performed in climate C (equal to 247.6 MWh and 244.6 MWh respectively), accomplishing an energy saving ranging from about 3.6% to more than 6.5%.

Table 4. Annual energy consumption and relative annual energy consumption in comparison with the BS system.

		climate A	climate B	climate C
BS refrigeration system	[MWh]	216.1	201.2	198.6
BS heating and DHW system	[MWh]	37.6	55.2	61.8
HRS system	[MWh]	256.6	247.6	244.6
Relative annual energy consumption	[%]	+1.1	-3.6	-6.5

4. Conclusions

A low environmental impact solution based on a CO_2 system with heat recovery has been evaluated for the reduction of global energy consumption for heating, refrigeration and domestic hot water production in a typical medium size supermarket. As the baseline a typical R134a/CO_2 cascade refrigeration system together with a R410A heat pump for space heating and hot water production has been considered. Different control strategies and the employment of an additional air-cooled evaporator have been taken into account to guarantee the full supply of the DHW and heating demands.

The CO_2 transcritical system with heat recovery has demonstrated to be an efficient solution in mild climates characterized by a high average annual and daily temperature fluctuations where energy savings range from about 3.6% to more than 6.5% in comparison with the baseline system. In mild climates with low fluctuations and characterized by high outdoor temperatures for a long time over the year the system has shown a slight increase in energy consumption due to the sensitivity of the efficiency of CO_2 systems to the gas cooler exit temperature.

Following the control strategy investigated in this study, the employment of the additional air-cooled evaporator is not to be suggested in all climates. The gas cooler by-pass strategy allows covering the total DHW and heating demand in supermarkets operating in mild climates and/or in well-insulated buildings with a possible reduction in the investment costs.

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