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SIMULATION AND MONITORING OF A CO_2 COMMERCIAL REFRIGERATION SYSTEM SERVING A SUPERMARKET IN NORTHERN ITALY

Marco Azzolin[(a)], Gianluca Cattelan[(a)], Corrado de Gioia-Carabellese[(b)], Simone Dugaria[(c)], Silvia Minetto[(d)], Davide Del Col[(a)]

[(a)] University of Padova, Department of Industrial Engineering
Padova, 35131, Italy, e-mail: marco.azzolin@unipd.it
[(b)]Frascold SpA, Rescaldina (MI), 20027, Italy, e-mail: corrado.dgc@frascold.it
[(c)] Faculty of Science and Technology, Free University of Bolzano-Bozen,
Bolzano, 39100, Italy, e-mail: simone.dugaria@unibz.it
[(d)]National Research Council of Italy (CNR), Construction Technologies Institute (ITC),
Padova, 35127, Italy, e-mail: silvia.minetto@itc.cnr.it

ABSTRACT

In the present paper, the simulation and the monitoring of a CO_2 commercial refrigeration unit serving a supermarket located in Northern Italy are presented. The unit consists of a booster compressor rack with parallel compression that provides both the cooling load (at medium and low temperature) and all the thermal functions in one single unit: it integrates a heat exchanger for the air-conditioning demand and two-stages heat recovery, for sanitary hot water production and space heating. The monitoring data are used as boundary conditions to develop a numerical model of the refrigeration cycle, operating in steady-state conditions, and calculate the hourly performance of the system for the whole year. The model is then used to evaluate the performance of alternative configurations and another possible management of the heat recovery system.
Keywords: Refrigeration, Carbon Dioxide, Compressors, COP, Heat Recovery

1. INTRODUCTION

In the last years, the climate change has forced the introduction of some decisions aimed to phase out the traditional high GWP fluids, like the European F-Gas regulation (EU 517/2014). The use of natural refrigerants is steadily increasing, and the high performance of CO_2 systems in cold climates makes carbon dioxide a viable solution for the replacement of HFC-based plants in the field of commercial refrigeration (Gullo et al. 2018). In warm climates, CO_2 plants have to work for long periods in trans-critical conditions with high energy consumption, so it is important to assess possible solutions to improve the performance, like the use of parallel compression (Sarkar and Agrawal, 2010), overfed evaporators (Karampour and Sawalha, 2018) and ejectors (Gullo et al. 2017). To improve the overall efficiency of a CO_2 system, it is also possible to exploit the wide temperatures range of the plant to adopt an all-in-one solution, with the integration of a heat recovery system for the hot water and space heating production and air-conditioning.
In this paper, various solutions have been theoretically studied for the improvement of the performance both in cold and warm climate conditions: overfed evaporators, vapor ejectors and dedicated mechanical subcooling. The simulations have been performed using a model derived by the monitoring of an existing plant for the boundary conditions and efficiency of the compressors. The monitored system has been used as a benchmark configuration and it is composed by a booster system with parallel compression and the integration of two-stage heat recovery and air-conditioning.
Each configuration has been assessed with the climatic data of two cities characterized by different yearly average outdoor temperature: Bratislava and Athens, in order to analyze the influence of the climate conditions on the system performance.

2. MONITORED CO_2 SYSTEM

The base configuration considered in this study is that of an existing booster plant, installed in a supermarket in Northern Italy, which has to satisfy the cooling demands at two different temperature levels, the hot water production to feed the hydronic circuits for sanitary hot water (SHW) and space heating (SH) and the air-conditioning system. The system schematic (named B-PC), along with the installed sensors, is reported in Fig. 1.

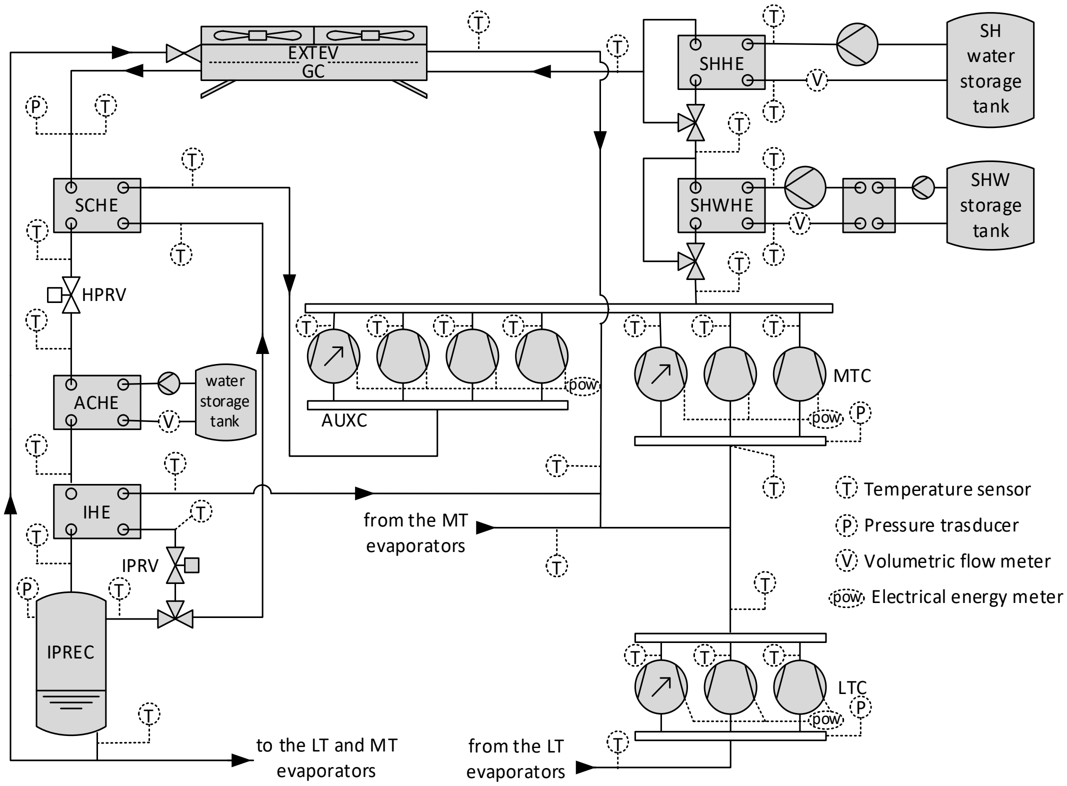

Figure 1: Scheme of the booster system with parallel compression, heat recovery and air-conditioning

The plant is composed by four pressure levels. The high pressure line connects the discharge of the medium temperature compressors (MTCs) and the auxiliary compressors (AUXCs) to the gas cooler (GC). The SHHE and SHWHE heat exchangers at the compressors outlet allow to recover heat for space heating and sanitary hot water production. Carbon dioxide leaving the GC is throttled by the high pressure regulating valve (HPRV), and then sent to the intermediate pressure receiver (IPREC), where liquid and vapor phases are separated. The ACHE heat exchanger, which is placed right after the high-pressure regulation device, is used when it has to satisfy the cooling demand of the air conditioning system. At the IPREC outlet, liquid CO_2 is laminated to the medium and low pressures using dedicated expansion valves. The fluid at the medium temperature feeds the cabinets, while that at the low temperature is sent to the freezers. The low temperature compressors (LTCs) are then used to elaborate the outgoing vapor from the low temperature evaporators. The pressure inside the tank IPREC is controlled through the IPRV throttling valve or the AUXCs. If the vapor amount is not enough to be processed by the AUXCs, it is throttled to the medium pressure, superheated in the IHE heat exchanger, and then mixed with the vapor discharged by the LTCs. Thus, the total mass flow rate is compressed by the MTCs to the high pressure level. If the amount of vapor in the IPREC is enough to be processed by the AUXCs, it flows through the SCHE heat exchanger, where it is superheated by the CO_2 at the outlet of the GC. This avoids the presence of liquid droplets at the suction of the compressors. During the non-heating season the high pressure is regulated based on the external air temperature. In the coldest hours of the year, to increase the amount of heat available for space heating, the system is forced to work in trans-critical mode. In this case, if the heat absorbed by the cabinets and freezers cannot match the thermal demand of the heating system, an additional evaporator (EXTEV) is used to evaporate the excess of liquid at the medium pressure.

The LTC rack is composed by 3 compressors of the SK3 series (Frascold, 2017a), the MTC rack by 3 compressors of the TK (Frascold, 2017b) series and the AUXC by 4 compressors of the TK series. In all the compressors racks, one compressor is equipped with an inverter that works in the frequency range of 30 – 60 Hz, to achieve a better regulation of the CO_2 mass flow rate.
Field measurements have been carried out to monitor the performance of the CO_2 system and some of the results have been used to develop and validate the numerical model used in this study, such as the vapor superheating at the outlet of the evaporators, the regulation of the high pressure level, and the effectiveness of the internal heat exchangers (Dugaria et at. 2019). Some results of the field tests are reported in Fig. 2. It can be seen that as the outdoor temperature increases, heat flow removed by the low temperature evaporators is constant and its average value is 11 kW. The cooling demand at the medium temperature evaporators shows an increase during the warmest period; this is due to the higher environmental temperature and humidity. The heat flow rate at the external evaporator reaches its maximum at an external temperature of about 1 °C and it decreases for higher temperatures. For lower external temperatures, since they mainly occur at night, the heating demand decreases and so the heat flow rate at the EXTEV.

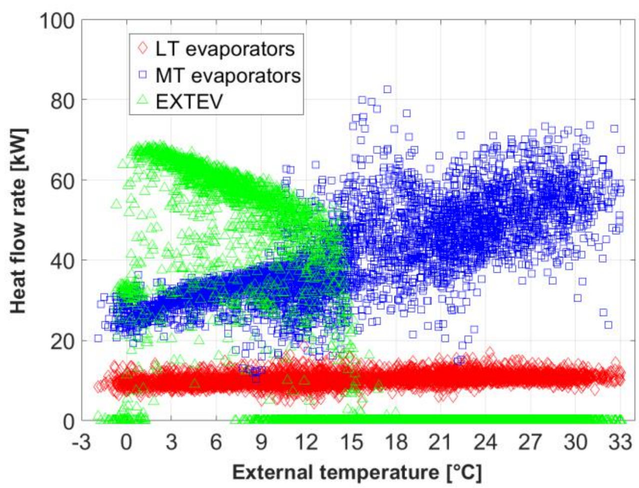

Figure 2: Hourly heat flow rates vs. outdoor temperature

3. MATHEMATICAL MODEL DESCRIPTION

To assess the performance of a CO_2 system under different climate conditions and plant configurations, a mathematical model has been developed in Matlab, based on the operative information derived from the monitoring of the system described above (Dugaria et at. 2019). In the model, the number and type of compressors are the same of the monitored system. Each configuration is described in the following section, and has been assessed with the weather data of Bratislava and Athens (EnergyPlus), in order to study the influence of the outdoor environment on the performance of the plant.

3.1. Boundary conditions and assumptions

The hourly cooling, heating and air-conditioning demands have been evaluated by means of a two-steps method, based on the study made by Karampour and Sawalha (2018). Regarding the cooling capacity, a weekly load profile that express the hourly fraction of the nominal capacity has been defined, then the hourly load fraction is corrected by 0.5% for each kelvin of difference in the external air from its design value (32 °C). When the external temperature is lower than 4 °C, a constant correction factor is considered. The nominal cooling capacities at the MT and LT are set to 65 kW and 15 kW respectively.
The sanitary water is supposed to be taken at 60 °C from the hot storage tank and heated up to 65 °C, and the maximum capacity of the SHWHE is equal to 10 kW. At the SHHE, a water return temperature equal to 50 °C is assumed. The SHHE has been designed with a nominal heating capacity equal to 35 kW for external temperature equal to -5 °C. In both the SHWHE and the SHHE, an approach point temperature difference between CO_2 inlet

and water outlet equal to 5 K has been considered. The ACHE cools the water taken from the cold water storage tank, from 12 °C to 5 °C, and the system has a cooling capacity equal to 35 kW for external temperature of 35 °C. In this case, the approach point temperature difference at the CO_2 inlet of the ACHE is assumed equal to 4 K. With dry expansion evaporators, a superheating of 10 K has been considered, neglecting the effect of the external ambient, considering an optimal insulation, as made by Karampour and Sawalha (2017). The low and medium pressures have been fixed equal to 13 bar and 27 bar respectively, in agreement with the field-measurement described above. All the expansion processes in the system are assumed isenthalpic, and the fluid outgoing the IPREC is in saturation conditions. The effectiveness of the internal heat exchangers IHE and SCHE of Fig. 1 are respectively 0.75 and 0.70.

3.2. System regulation

The control strategy of the CO_2 plant is made depending on the outdoor temperature. The high pressure level is determined dividing all the working conditions in 4 zones, similar to that adopted by Gullo et al. (2016). The first one is the sub-critical zone, where the system operates if the environment temperature is below 20°C and no heat recovery is needed. In this case the high pressure is fixed to 41 bar, to ensure feasible working conditions for the compressors. When the external temperature is above 27 °C, the system has to work in trans-critical mode, and the high pressure is regulated following a correlation derived by the monitoring of the plant described in section 2. For external temperatures between these two cases, the high pressure changes linearly with the external temperature, in order to avoid discontinuous regulations between the sub-critical and trans-critical zones. When space heating is needed, the high pressure level is fixed to 75 bar, to ensure a sufficiently high temperature of the CO_2 at the discharge of the compressors. The pressure can be further increased up to 86 bar to enhance the heat recovery production. If in these conditions it is not possible to entirely satisfy the heating demand, the mass flow rate elaborated by the MTCs is increased and the EXTEV is used to evaporate the excess of liquid.

3.3. Alternative system configurations

Beside the benchmark B-PC configuration, some alternative solutions to increase the performance of the system have been analyzed with the climate data of Bratislava and Athens.

- Booster system with parallel compression and overfed evaporators (B-PC-OE). In this case, the evaporation temperatures are increased by 5 K with respect to the case of dry expansion evaporators, and a vapor quality at the outlet equal to 0.95 has been considered. The refrigerant at the outlet of the medium temperature evaporators is sent to a medium pressure receiver, where liquid and vapor are separated. The vapor is sent to the MTC, while the liquid is expanded to the low pressure level, where it is mixed with the mass flow rate at the outlet of the low temperature evaporators. To avoid that liquid droplets enter the LTC, the refrigerant mass flux is superheated before the suction of the LTC using the refrigerant at the medium temperature. The following alternative configurations will include overfed evaporators at both the medium and low temperature levels.
- Booster system with parallel compression, overfed evaporators and vapor ejectors (B-PC-OE-EJ). In this configuration, which is represented in Fig. 3a, the vapor at the outlet of the medium pressure receiver is sucked by the ejectors, which reject it at the intermediate pressure level recovering part of the work for the expansion process. The liquid mass flux is instead expanded to the low pressure level, where it is mixed with the mass flow rate flowing through the low temperature evaporators and then the total mass flow rate is superheated using an internal heat exchanger. The ejectors are modelled basing on the correlations used by Gullo et al. (2018) for the evaluation of the entrainment ratio, and are enabled when the outdoor temperature is above 20 °C.
- Booster system with parallel compression, overfed evaporators and dedicated mechanical subcooling (B-PC-OE-DMS). The scheme of this configuration has been reported in Fig. 3b. In this configuration, a dedicated R290 system is installed in order to decrease the refrigerant temperature at the outlet of the GC below the outdoor level. The degree of subcooling supplied by the DMS is calculated basing on the work of Catalan et al. (2019), and the system is enabled starting from outdoor temperatures equal to 20 °C. The thermal effectiveness of the heat exchanger used to cool the CO_2 is assumed equal to 0.6. With these assumptions it is possible to evaluate the evaporation pressure of the R290 system. The isentropic efficiencies of the compressors, which allow to calculate the real work spent for the compression process, are evaluated using the following correlation, used by Llopis et al. (2015):

$$\eta_{is} = 1 - 0.04 \cdot r \qquad \text{Eq. (1)}$$

where r is the ratio between condensation and evaporation pressures. In the R290 cycle the superheating at the compressor and the subcooling at the condenser have been supposed equal to 5 K and 3 K, respectively. Thus, the R290 mass flow rate can be determined considering the heat balance in the heat exchanger where CO_2 and R290 flow, since the heat flux is known on CO_2 side.

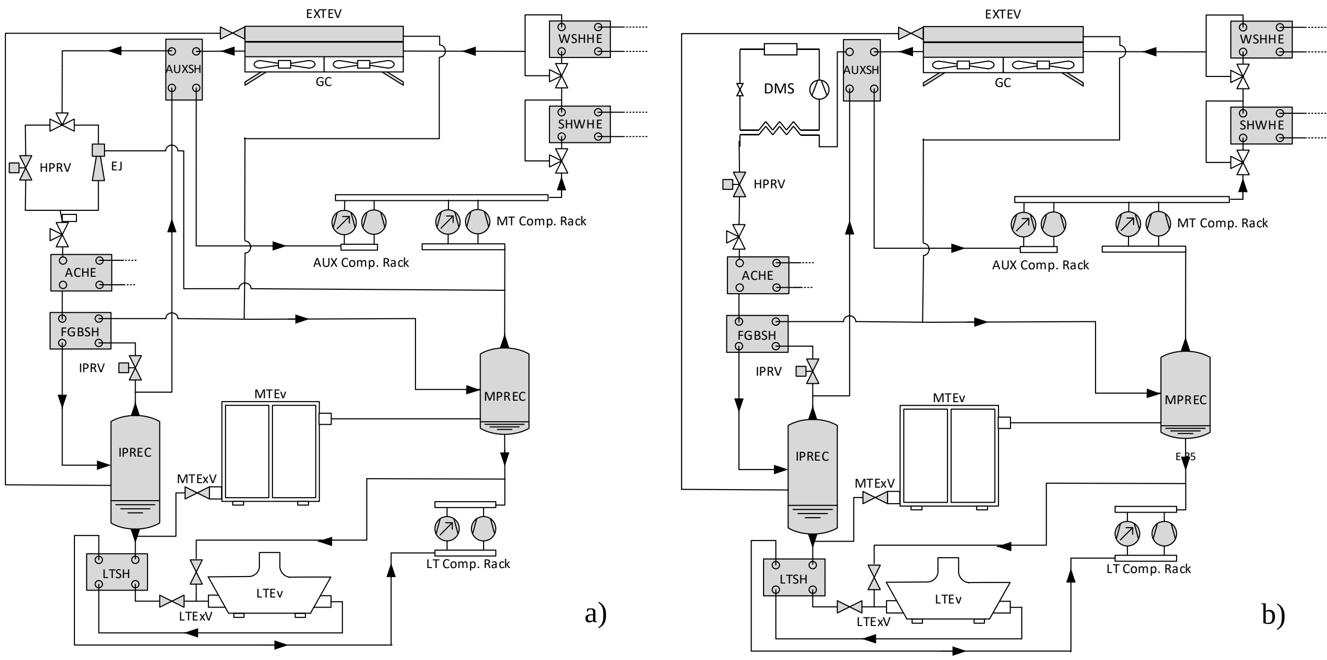

Figure 3: Schemes of the B-PC-OE-EJ (a) and B-PC-OE-DMS (b) configurations

4. MODEL RESULTS

The performance of the system configurations listed in section 3 have been assessed using the climate data of two cities: Bratislava (LAT. 48.2° N, LONG. 17.2° E, yearly average temperature equal to 10.4 °C) and Athens (LAT. 37.9° N, LONG. 23.7° E, yearly average temperature equal to 17.9 °C). The results of the simulations in terms of yearly energy consumption of the compressors are reported in Table 1.

Table 1. Yearly energy consumption of the compressors for each studied configuration and location. (B-PC: Booster with Parallel Compression, OE: Overfed Evaporators, EJ: Vapor Ejectors, DMS: Dedicated Mechanical Subcooling. E_{LTC}, E_{MTC}, E_{AUXC} and E_{DMS} are the electrical power consumptions of the LTC, MTC, AUXC and of the compressors of the dedicated mechanical subcooling system respectively).

	Configuration	E_{LTC} [kWh]	E_{MTC} [kWh]	E_{AUXC} [kWh]	E_{DMS} [kWh]	E_{TOT} [kWh]
Bratislava	B-PC	17023	151906	3181	\	172109
	B-PC-OE	15565	146718	3354	\	165638
	B-PC-OE-EJ	15566	142672	6846	\	165083
	B-PC-OE-DMS	15566	143948	102	4829	164445
Athens	B-PC	17568	151715	14333	\	183615
	B-PC-OE	16085	138804	14948	\	169837
	B-PC-OE-EJ	16085	121313	28719	\	166117
	B-PC-OE-DMS	16085	133637	1891	14138	165751

Overfeeding the evaporators allows to reduce the electrical energy consumption of the MT and LT compressors regardless of the outdoor temperature and can therefore ensure higher energy savings during the year. With the installation of this solution it has been possible to decrease by 7.5% and by 3.8% the yearly energy consumption for Athens and Bratislava respectively. The overfed evaporators reduce the discharge temperature of the compressors, so when space heating is needed, a higher mass flow rate has to be processed in order to satisfy the demand. For this reason, in Bratislava, which is characterized by a higher space heating demand in the winter, the achievable energy saving is lower.

The combined usage of overfed evaporators and vapor ejectors helps to further reduce the MTC energy consumption, at the expense of an increase of the operating hours of the AUXC. The overall effect has been a reduction of the total energy consumption equal to 9.5 % for Athens, and to 4.1 % for Bratislava. The energy saving is higher in the case of Athens, due to the highest outdoor temperatures in the summer period, which allow a larger use of the vapor ejectors (that are operative only when the environmental temperature is above 20 °C). The outdoor temperature is higher than 20 °C for 3560 hours in Athens and for 1452 hours in Bratislava.

The installation of a dedicated mechanical subcooling system, along with the overfed evaporators, allowed the highest reduction of the total energy consumption for both of the cities (equal to 9.7% for Athens and 4.5% for Bratislava with respect to the B-PC case). This solution is particularly effective in warm climate conditions when, considering a standard system, the high refrigerant temperatures at the outlet of the GC produce a large amount of flash vapor in the IPREC, and the consequent increase of the AUXC energy consumption. Since the city of Athens is characterized by higher outdoor temperatures in the summertime, the energy savings achievable with this solution are larger.

Fig.4 shows the winter, summer and yearly COP obtained by each assessed configuration. Winter period goes from 1st October to 31st March, summer period lasts from 1st April to 30th September. The hourly COP has been defined as the ratio between the summation of the cooling loads, the heat fluxes provided for SHW, SH and air-conditioning purposes and the total power consumption of the compressors.

$$COP = \frac{Q_{LT} + Q_{MT} + Q_{SHW} + Q_{SH} + Q_{AC}}{P_{LTC} + P_{MTC} + P_{LTC} + P_{DMS}} \quad \text{Eq.(2)}$$

The configuration with overfed evaporators allowed to increase by 13.4% the yearly COP in Bratislava, and by 11.8% in Athens with respect to the case of dry expansion evaporators, since this solution is effective for the overall outdoor temperature range. The B-PC-OE-EJ configuration produced higher benefits in Athens, where the yearly COP has been increased by 13.4%; for Bratislava the mild outdoor temperatures reached during summertime have limited the yearly COP increase, equal to 13.7% with respect to the B-PC system.

It is interesting to observe that for the city of Bratislava in summertime the COP of the B-PC-OE-DMS configuration is higher than that achieved in wintertime thanks to the introduction of the mechanical subcooling, which allows to increase by 5.6% the COP during summertime with respect to the solution without DMS. This is due to the fact that in wintertime the low ambient temperatures reached in Bratislava lead to high space heating demands, so the system has to work in trans-critical conditions (with high energy consumption) for a large number of hours in order to produce enough heat recovery. Also for the city of Athens this solution allows to operate with almost the same average COP during winter and summer period, although this city is strongly penalized by the high outdoor temperatures reached. The yearly COP with this configuration is increased by 21.2% in Athens and 16.5% in Bratislava.

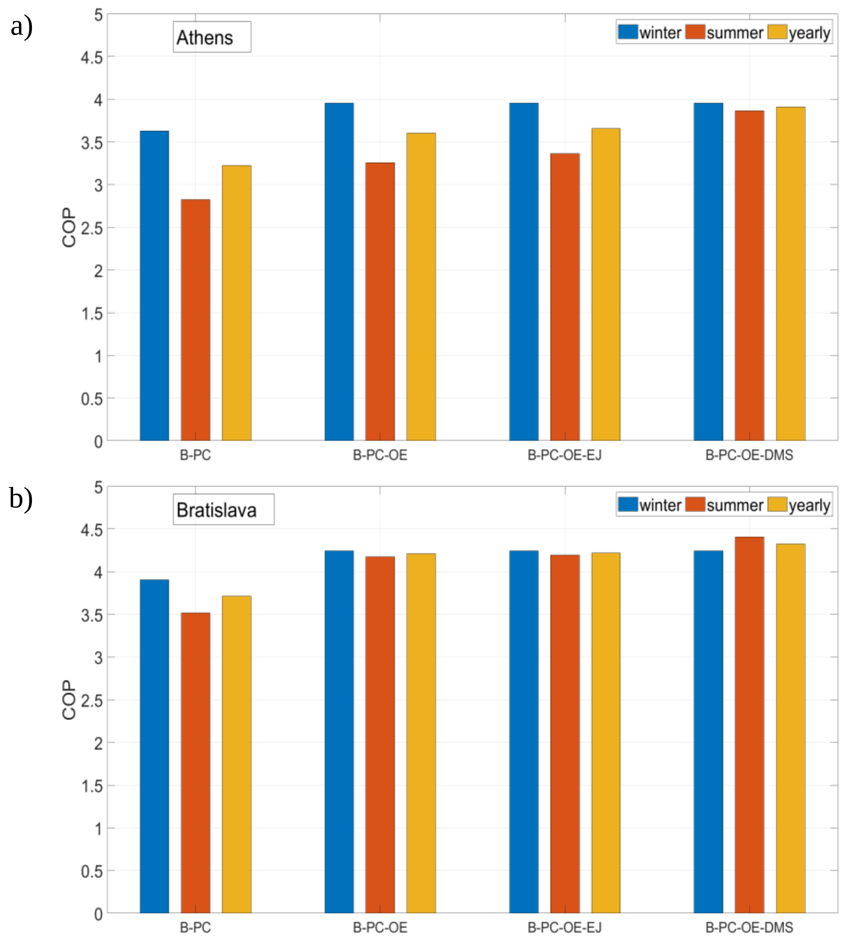

Figure 4. Winter, summer and yearly COP obtained by the simulations for Athens (a) and Bratislava (b). (B-PC: Booster with Parallel Compression, OE: Overfed Evaporators, EJ: Vapor Ejectors, DMS: Dedicated Mechanical Subcooling)

5. CONCLUSIONS

This study was based on the analysis of all-in-one CO_2 systems for commercial refrigeration, designed to satisfy the cooling demands along with that of the sanitary hot water, space heating and air-conditioning.
The results obtained from field measurements on an existing supermarket located in Northern Italy have been presented. The main operative parameters of the monitored system have been used to develop a mathematical model, which allowed to simulate the performance of different solutions and to compare the results with that obtained considering a standard system, which is the booster with parallel compression. The alternative system configurations analyzed in this studied were the overfed evaporators, vapor ejectors and dedicated mechanical subcooling system, which used an inverse cycle based on the hydro-carbon R290. All these solutions have been simulated using the climate data of two cities characterized by different yearly average temperatures, Bratislava and Athens, in order to analyze the effectiveness of these configurations both in mild and warm regions. The configuration with overfed evaporators allowed a yearly energy saving equal to 7.5% for Athens and 3.8% for Bratislava, with respect to the B-PC system. The vapor ejector system (B-PC-OE-EJ) allowed to further reduce the energy consumption. For Athens this solution achieved an energy saving equal to 9.5%, while for Bratislava it was equal to 4.1%. The system with the integration of a dedicated mechanical subcooling (B-PC-OE-DMS) allowed to achieve the highest effectiveness. The energy savings obtained with this solution have been equal to 9.7% for Athens and to 4.5% for Bratislava.

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NOMENCLATURE

AC	air conditioning	*LT*	low temperature
AUXC	auxiliary compressors	*LTC*	low temperature compressors
B-PC	booster with parallel compression	*MT*	medium temperature
DMS	dedicated mechanical subcooling system	*MTC*	medium temperature compressors
E	Compressor energy consumptions[kWh]	*OE*	overfed evaporators
EJ	vapor ejectors	*P*	electrical power consumption [kW]
EXTEV	external evaporator	*Q*	heat flux [kW]
GC	gas cooler	*r*	compression ratio
HE	heat exchanger	*SH*	space heating
HPRV	high pressure regulating valve	*SHW*	sanitary hot water
IPREC	intermediate pressure receiver	*TOT*	total