Modelling commercial refrigeration systems coupled with water storage to improve energy efficiency and perform heat recovery

Published
DOI:10.1016/j.ijrefrig.2016.06.012

MODELLING COMMERCIAL REFRIGERATION SYSTEMS COUPLED WITH WATER STORAGE TO IMPROVE ENERGY EFFICIENCY AND PERFORM HEAT RECOVERY

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ABSTRACT

A basic CO_2 transcritical/subcritical commercial refrigeration system is considered, applied to cold rooms and display cabinets in a supermarket. Subcooling of the refrigerant or heat recovery from condensation can be performed, taking advantage of a large fire prevention water tank. The whole refrigeration system is modelled in a TRNSYS environment, taking into account the hourly weather data and calculating the hourly cooling load demand from display cabinets and cold rooms equipment. New *types* have been written to describe display cabinets and cold rooms, CO_2 refrigerating units and a particular water store.

Simulations consider a simple double compression cycle with liquid receiver, and other options among which an auxiliary compressor. Results show that CO_2 plants are feasible and energetically acceptable in mild climates, provided that improvements to standard cycle are adopted. Furthermore, heat recovery can be effectively performed through the employment of a heat storage.

KEYWORDS:

CO_2; commercial refrigeration; efficiency; modelling; heat recovery

NOMENCLATURE

AC	auxiliary compressor
COP	coefficient of performance, -
CRU	commercial refrigeration unit

CR	cold room
HX	heat exchanger
LT	low temperature
MT	medium temperature
p	pressure, MPa
q	heat flux, W
r	pressure ratio, -
RDC	refrigeration display cabinet
t	temperature, °C
ΔT	temperature difference, K
η_{is}	isentropic efficiency

Subscripts

ap	approach point
aux	auxiliary devices
def	defrost
GC	gas cooler
inf	infiltration
lat	latent
out	outlet
sen	sensible
set	set point
tank	water tank
w	wall

1. INTRODUCTION

Commercial refrigeration is being especially affected by the phase down schedule for HFCs, recently forced in Europe through the European "F-gas regulation" (EU Fgas, 2014). In fact, as of 1st January 2022 placing on the market is prohibited for fluorinated greenhouse gases with a GWP of 150 or more used in multipack centralised refrigeration systems for commercial use, with a rated capacity of 40 kW or more. Exception is made for the primary refrigerant circuit of cascade systems, where fluorinated greenhouse gases with a GWP lower than 1500 (like R134a) may be used. This directive represents a step towards a possible full ban of HFCs, as it has already happened in some countries (IIR, 2015), thus giving rise to the need for finding alternate solutions (Cavallini et al, 2014). In spite of its low critical temperature as well as high operating pressure levels, carbon dioxide (R744) is receiving growing attention, due to its favourable thermophysical properties, non–toxicity, non–flammability and to its very low GWP, which leads to a negligible direct contribution to the greenhouse effect. On the contrary the indirect contribution could be negatively affected because of the lower efficiency of R744 systems when compared with those operated by traditional HFCs. This is particularly true for applications in mild and warm climate, where the CO_2 systems operate for a long period of time at transcritical conditions, with a significant decrease in their energy performance. Research is going on to face design issues and improve the energy efficiency of CO_2 systems in such conditions (Kim, 2004), and various solutions have been identified and tested (Cavallini and Zilio, 2007, Cecchinato et al, 2009, Hafner et al, 2014, Sawalha, 2009, Gullo et al, 2016) also with more in-depth analyses involving system irreversibilities (Gullo et al, 2015).

The main concern at mild/warm climates is related to the high temperature of the refrigerant at the gas cooler exit, which is very effective on the performance of the system (Pettersen, 1997). For this reason many efforts are devoted to investigate configurations where gas cooling can be promoted to the highest level, by means of internal heat exchangers (Ge and Tassou, 2011c, Sawalha, 2008, Yang and Zhang, 2011, Cavallini et al , 2007) with an average increase in the COP of up to 10 %, or by means of the "mechanical subcooling", i.e. performing subcooling thanks to another refrigerating unit (Llopis 2015, Qureshi and Zubair, 2012, Hafner et al, 2014). The effectiveness in terms of global energy consumption of the solution with mechanical subcooling is strictly related to the COP of the additional refrigerating system, and its evaluation on a yearly basis is not so easy. As an example, Wiedenmann et al. (2014) investigated the effect of subcooling performed by an adsorption refrigerator driven by waste heat. Because of the additional investment cost, high payback periods were estimated, with a rather low market potential.

It comes out that an effective improvement could be safely reached while performing subcooling by means of an external device at no or very low expense. Looking for other cold sources for subcooling, Ferrandi and Orlandi (2013) investigated the effect of a cold storage used to cool down the refrigerant at the exit of the liquid receiver in a "booster" cycle. With a water store size from 12 to 27 m^3, they predicted an average reduction in summer daily power consumption around 5% compared to a traditional plant, and a peak refrigerating capacity reduction around 28 %.

In this paper a similar solution is investigated, where a fire prevention water tank is employed as a cold sink for performing subcooling. Of course an application has to be identified for the heat released to the water tank, and this has been found in the HVAC plant. In fact, every supermarket is located in a building or a shopping mall where heating and air conditioning have to be performed, which could take advantage from a heat storage device. The idea of recovering heat from a refrigeration plant in favour of other systems is well known and widespread, especially for hot water production (Nidup 2009, Sawalha 2013, Hafner et al, 2012). Much less common is heat recovery for heating purposes (Cortella and Saro, 2010, Cortella and D'Agaro, 2016), both for the great amount of energy required and for the troubles in matching needs and availability. Heat pumps are commonly used for heating purposes in commercial buildings. Their seasonal performance not only depends on their design, but it is definitely affected by the temperature of the secondary fluid at the inlet of the evaporator (Tammaro et al, 2015). For this reason outdoor air is not the best choice as a heat sink, and geothermal (aquifers or boreholes) or any great capacity heat sources are employed whenever feasible and convenient. A great capacity heat storage like the fire prevention water tank can be a solution to perform subcooling of the refrigeration plant and allow heat recovery in favour of heat pumps for heating purposes, while acting as a buffer and disconnecting heat demand and supply (Polzot et al., 2015).

A thorough evaluation of both the systems is required, by means of a transient state simulation, in order to perform an accurate estimation of their thermal loads and energy use all through the year, and to investigate the effectiveness of various heat recovery strategies, with the aim of reducing the global energy consumption without affecting the performance. Various software codes are available in the literature, able to simulate both the refrigeration and HVAC systems in supermarkets, which can be employed to perform such evaluations. Among all, we mention here *Cybermart* (Arias, 2005, Arias and Lundqvist, 2005), *EnergyPlus* (Stovall and Van Baxter, 2010), and *SuperSim* (Ge and Tassou, 2011a and 2011b). Only *Energy Plus* is freely available, and can be used for the simulation of systems with some complexity but made of conventional components (Cortella et al., 2011). When special components or specific control rules have to be implemented, the adoption of a more flexible tool is envisaged. In the framework of the CommONEnergy project funded by the European Community 7th Framework Programme the authors contributed to the development of an Integrative Modelling Environment based on TRNSYS. This tool aimed at the evaluation of retrofitting energy solutions that involve the building, with its passive and active solutions for heat gains control, the HVAC plant, the refrigeration plant, the lighting systems, heat storage and recovery devices and finally the energy generation systems when available. It has been used to evaluate various solutions where the synergy between the HVAC and refrigeration plant is exploited (Cortella et al., 2014). In this paper the authors describe the use of this tool to simulate a transcritical CO_2 commercial refrigeration plant, in the particular configuration with a water storage to allow refrigerant subcooling and/or condensation and heat recovery in favour of heat pumps for heating purposes.

2. THE REFRIGERATION SYSTEMS

A supermarket is considered in this paper, where a refrigeration plant is installed with a capacity of 140 kW in the Medium Temperature (MT) and 22 kW in the Low Temperature (LT) systems. It supplies 17 refrigerated display cabinets and 10 cold rooms from the Medium Temperature and 4 frozen food display cases and 2 cold rooms from the Low Temperature.

Several options could be adopted for this plant, among which CO_2 plants are preferred. As a baseline for the comparison of different solutions, a cascade configuration is chosen, with CO_2 as refrigerant for the direct expansion LT display cabinets, and R134a as refrigerant for the MT portion of the system (Figure 1).

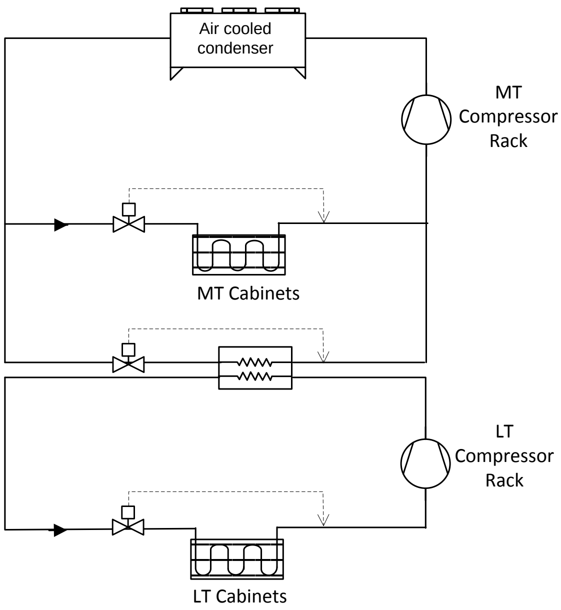

Figure 1. Sketch of the cascade refrigeration system considered as a baseline.

Among the options for carbon dioxide systems, this paper deals with a CO_2 cycle with mid pressure receiver and flash gas compression, with an innovative option. In the basement of the building a 950 m^3 water reservoir is available, for the fire prevention system, which has been considered as a possible cold storage to perform subcooling of the refrigeration system. Figure 2 reports the schematic of the CO_2 plant studied.

The refrigeration system operates both in subcritical and transcritical conditions, depending on the external temperature. Water is circulated by means of a pump from the tank to heat exchangers 1 or 2 alternatively. Heat exchanger 1 (HX1) performs subcooling of the refrigerant exiting the condenser or gas cooler, while heat exchanger 2 (HX2) is used to cool water in the reservoir. Flash gas removal is performed by the MT compressor rack or, alternatively, by an auxiliary compressor. In a first instance the system was simulated in the absence of the auxiliary compressor, for the sake of cost reduction and simplification in terms of control and oil return. Then parallel compression was included, to evaluate its effect on global energy consumption. No internal heat exchangers were considered at the moment.

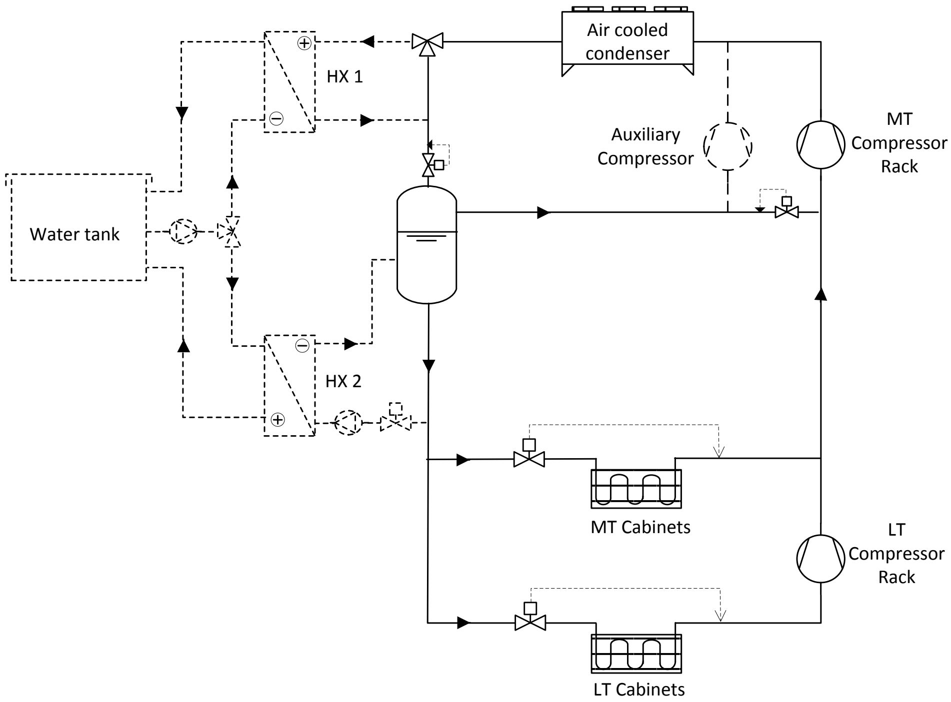

Figure 2. Sketch of the CO_2 refrigeration system with the water reservoir.

The most important design parameters considered in the simulation are summarized in Table 1.

Table 1. Main design parameters

Parameter of Commercial Refrigeration Unit	***Value***
LT Evaporating temperature	-35 °C
MT Evaporating temperature	-10 °C
Superheating at evaporators	5 K
Superheating in the suction lines	5 K
ΔT approach of the condenser/gas cooler	3 K
Specific for Cascade Configuration	
ΔT approach of the cascade condenser	5 K
Minimum condensing temperature	25°C
Specific for Booster Configuration with water reservoir	
Minimum condensing temperature	8 °C
Liquid receiver temperature	3 °C
ΔT approach of HX1 and HX2	3 K
Parameter of water tank	
Water tank volume	950 m^3
Height	2.5 m
R-value at the walls/bottom	0.16 m^2 K W^{-1}
R-value at the top	3.26 m^2 K W^{-1}

3. MODEL DESCRIPTION

Supermarket refrigeration systems include a large number of components. The core component is the commercial refrigeration unit (CRU), made up by evaporator coils, air-cooled condensers, compressor racks,

etc. The cooling load profiles at LT and MT evaporators depend upon the performance of the refrigerated food storage equipment (food display cabinets and cold rooms) whose heat and mass transfers from the indoor environment must be modelled according to the different typology (i.e vertical or horizontal display cabinet), temperature class, protection from ambient air infiltration, etc.
The mathematical models of single components have been developed in the TRNSYS environment (Klein S.A. et al., 2010), as it allows to run dynamic simulations of a complex system by easily implementing controls and interconnections between components.
In the following the description of each component is given in detail.

3.1. Refrigerated display cabinets (RDCs) and cold rooms (CRs)

Regarding refrigerated display cabinets, the cooling load q_{RDC} (i.e. the heat extraction rate at the cabinet evaporator) is made up of three parts:

$$q_{RDC} = q_{sen} + q_{lat} + q_{aux} \tag{1}$$

where q_{sen} is the sensible load, which includes sensible heat transfer by ambient air infiltration in the refrigerated volume through the air curtains (in open-fronted RDCs) or via door openings, and both the conductive and radiative heat transfer through the RDC envelope; q_{lat} is the latent heat transfer by ambient air infiltration in the refrigerated volume and q_{aux} is the sensible load from auxiliary devices (lighting, fans, defrost and anti-mist heaters) as a fraction of their electric power which has to be removed from the refrigerated volume.
The total cooling capacity at rated conditions (as it is stated by the manufacturer according to ISO 23953) is adjusted taking into account the actual and time-dependent working conditions in a supermarket (off-rated conditions). 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. The estimation of the cooling load is based on the model proposed by Faramarzi (Faramarzi, 1999, Cortella et al., 2011).
The total cooling load of the cold rooms q_{CR} is made up of the following parts:

$$q_{CR} = q_w + q_{inf} + q_{def} + q_{aux} \tag{2}$$

where q_w is the conductive heat transfer through the cold room walls, q_{inf} is the heat load due to sensible and latent heat transfer by air infiltration in the refrigerated volume via door openings; q_{def} is the heat load due to defrost and q_{aux} is the contribution to the sensible heat load due to fans, heaters and lighting in the cold room.
In more detail, the mass flow rate infiltrating into the cold room is calculated according to the analytical model by Gosney and Olama, 1975, which has proven to fit the experimental data better than other models in the comparison carried out in Foster et al., 2003. The mass flow infiltration is then corrected taking into account the cold room operation and the kind of door protection.

3.2. Commercial refrigeration unit (CRU)

Cascade refrigeration cycle

A typical cascade refrigeration cycle as sketched in Figure 1 has been implemented. The input data for the cycle definition are the LT and MT evaporating temperatures, the refrigerant superheating at the inlet of both the LT and MT compressor (split into superheating in the evaporators and superheating in the suction lines), the approach temperature of the cascade condenser. The MT condensing temperature is set depending on the outdoor air temperature, considering an appropriate approach temperature, down to a minimum condensing level (see Table 1).

The expansion in all valves is considered isenthalpic and the pressure drop in the suction lines is neglected.

A commercial compressor rack is chosen at design conditions. Data from the manufacturer are used in order to identify the thermodynamic conditions at the discharge port in off-design conditions and to infer the isentropic efficiency of the compressor as a function of the pressure ratio *r*. As an example, in the present case the isentropic efficiency of the high stage compressor is given by

$$\eta_g = -0.0053r^2 + 0.0674r + 0.4802 \,. \qquad (3)$$

The computation of thermodynamic properties of the refrigerants is based on CoolProp libraries (Bell et al., 2014).

Booster refrigeration cycle

A booster refrigeration cycle, as sketched in solid line in Figure 2, is the basic one. The procedure for the definition of the cycle in subcritical mode and the compressor model are the same as described above for the cascade system.

In transcritical mode, the gas cooler discharge pressure is optimized to maximise the COP. Correlations are available in the literature (Liao, 2000, Ge and Tassou, 2011c). The authors performed an optimization for this specific design cycle, and the correlation

$$p_{GC,opt} = \max(75;\ 0.256\,t_{GC,out} - 0.1247)\ \text{[MPa]} \qquad (4)$$

has been inferred where $t_{GC,out}$ [°C] is the temperature at the exit of the condenser/gas cooler.

The transition from subcritical to transcritical operation is actually performed by means of an electronic controller which conveniently operates the back pressure valve at the inlet of the liquid receiver. A smooth change of the gas cooler pressure is obtained from the subcritical operation mode (condensing pressure related to condensing temperature) to the transcritical one. The pressure of the liquid receiver is kept constant (see Table 1) and only saturated liquid and saturated vapour are assumed to exit the receiver.

The basic model has been integrated with the water tank for the fire prevention system (Fig.2). A TRNSYS model (Type) of a constant volume stratified store has been modified in order to take into account the heat exchanges with the refrigeration system and the calculation of conductive heat losses to the ground for totally or partially buried tanks. A simplified calculation method for ground losses has been implemented based on

ASHRAE, 2001. Ground temperature is assumed to be depending on the external temperature, the soil thermal properties and the depth to ground surface, but independent from the tank temperature. The contribution from solar radiation has been ignored because the tank is buried in the basement of the building.

Heat exchangers HX1 for subcooling refrigerant and HX2 for recharging i.e cooling the water in the tank are modelled simply by assuming appropriate approach temperature values. Because HX2 is fed with liquid CO_2 which evaporates while cooling water, the pressure in the liquid receiver has to be kept at 3.8 MPa corresponding to a saturation temperature of 3°C, so as to prevent water from freezing.

The control strategy is the following:

$$\begin{cases} \text{if } t_{tank} < t_{GC,out} - \Delta t_{ap} \wedge \vartheta \in daytime \Rightarrow \text{subcooling active} \\ \text{if } t_{tank} > t_{tank,set} \wedge \vartheta \in nighttime \Rightarrow \text{recharging active} \end{cases} \quad (5)$$

thus subcooling is made available during daytime, whenever water temperature t_{tank} is lower than the refrigerant temperature at the exit of the condenser/gas cooler $t_{GC,out,}$, taking into consideration an appropriate temperature approach Δt_{ap} at HX1. Refrigeration through HX2 (recharging) might take place at nighttime, when subcooling is not necessary, the cooling load from refrigerated food storage equipment is lower and refrigeration power is available with the highest COP. It is operated whenever water temperature is above a set point temperature $t_{tank,set}$. For this solution, the optimized transcritical pressure $p_{GC,opt}$ is stated as a function of the CO_2 temperature at the exit of HX1.

Finally, the option of an auxiliary compressor for flash gas removal has been introduced in the model. The compressor size is chosen in order to perform a total removal of the flash gas at design conditions. The auxiliary compressor is activated as soon as transition operation takes place, and the pressure of the liquid receiver has to be optimized. In transcritical running mode, the high pressure is an additional independent variable and the system is optimized as a function of the gas cooler exit temperature in the absence of heat exchange with the water tank, otherwise this parameter is replaced by the HX1 exit temperature. The lower and upper limits were set to 3.8 MPa and 5.5 MPa for the intermediate pressure and to 7.5 MPa and 10.6 MPa for the discharge pressure. In all the evaluated cases, the maximization of the COP was chosen as the objective function (Gullo et al., 2016).

4. SYSTEM SIMULATION AND COMPARISON

Simulations have been carried out for the cascade cycle and for CO_2 cycles with different levels of interaction with the cold storage, i.e. with a passive heat storage (only HX1 operating) or with a heat storage recharged by the system itself (HX1 and HX2 conveniently operating). The configurations are summarised in Table 2.

Table 2. CO_2 cycle configurations considered

Acronym	*Configuration*
B	Basic: liquid receiver, no subcooling, no cold storage, no auxiliary compressor
BHS	Basic with subcooling (HX1 operating)
BHSR	Basic with subcooling, cold storage recharged (HX1 and HX2 operating)
BHS+AC	Basic with subcooling (HX1 operating), auxiliary compressor
BHSR+AC	Basic with subcooling, cold storage recharged (HX1 and HX2 operating), auxiliary compressor

The cooling load profiles on both the LT and MT evaporators are the result of the simulation of the refrigerated food storage equipment installed in the supermarket and operating at off-rated conditions. Profiles are depicted in Figure 3. The refrigerating energy to supply the MT and LT systems is 752 and 149 MWh/year respectively.

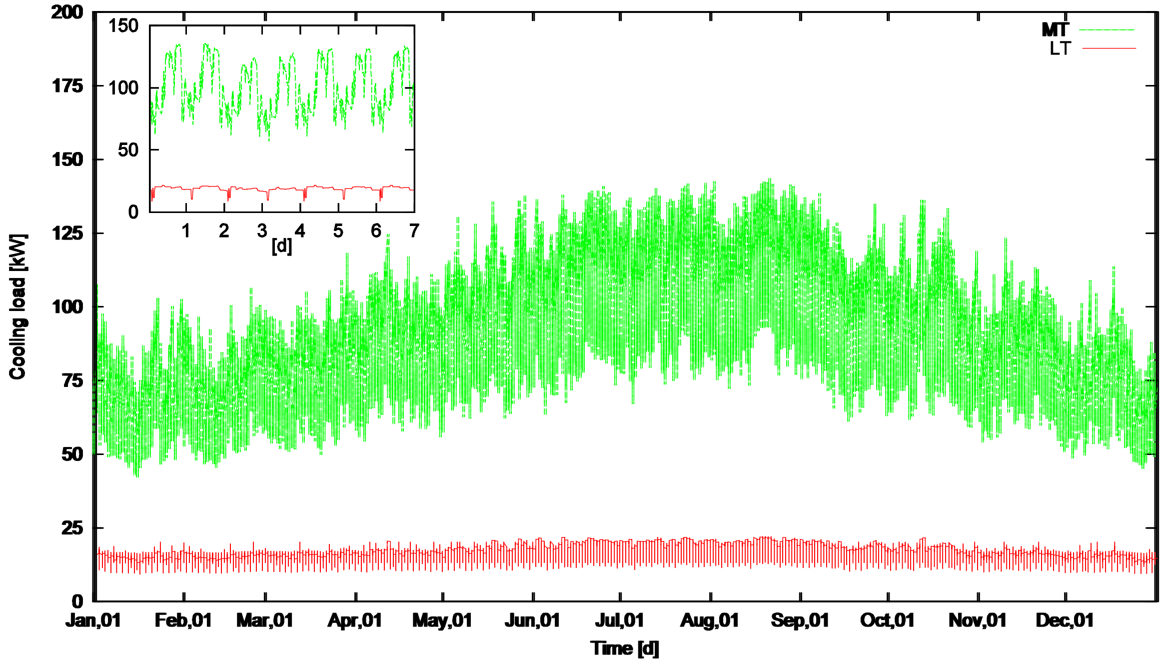

Figure 3. Yearly profile of the cooling load from the LT and MT systems and zoom on the first week of July

The mild climate of a seaside town in Northern Italy (Genua) has been considered. Outdoor temperature distribution (Meteonorm weather data, Meteotest, 2015) is plotted in Figure 4. Since in the booster system the transition from subcritical to transcritical mode occurs in a range from 6 to 7.5 MPa, the corresponding outdoor temperature, for the given approach temperature at the heat exchangers, is in the range from 17 to 26 °C. Thus transition occurs 40% of the time on an annual basis in the climate conditions considered and full transcritical mode occurs only 5% time/year. This fact underlines the importance for an effective control of operating conditions in the transition from subcritical to transcritical mode.

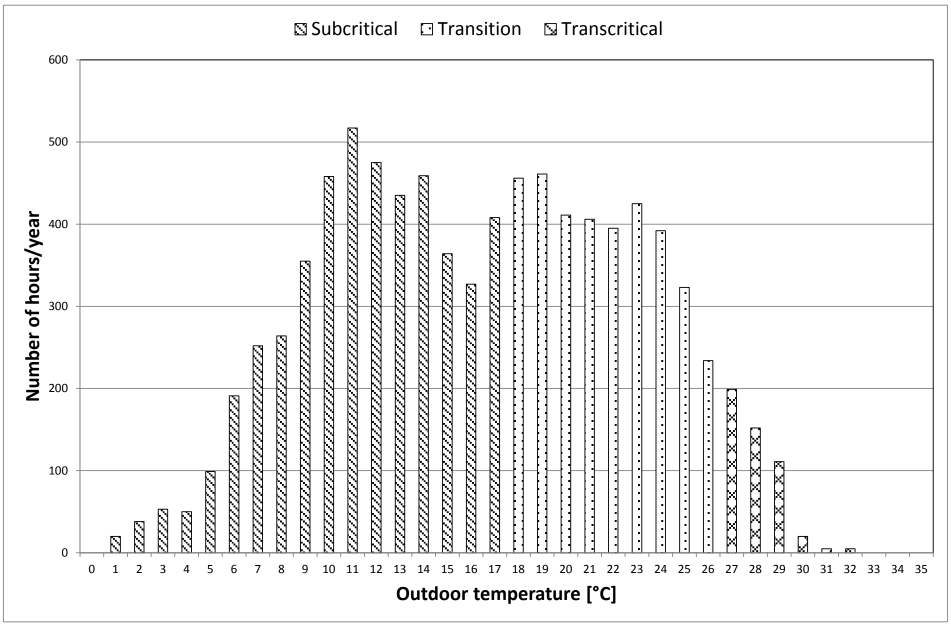

Figure 4. Outdoor temperature distribution on annual basis

4.1. Basic CO_2 cycle (Cycle B)

Carbon dioxide refrigeration systems are known to perform worse than standard HFC systems in mild/warm climate. This is also the outcome from our simulation (Figure 5). In the cold season the favourable thermodynamic properties of carbon dioxide and the absence of the intermediate heat exchanger between the LT and MT system favour the CO_2 cycle (daily electrical energy consumption down to -25÷-33%), while in the hot season the situation is opposite with peak energy demand definitely higher (daily electrical energy consumption up to +30÷35%) for the Cycle B solution (Figure 5).

On a yearly basis the cascade cycle demands 303.9 MWh electrical energy, the basic CO_2 cycle (Cycle B) demands 331.0 MWh (+8.9 %).

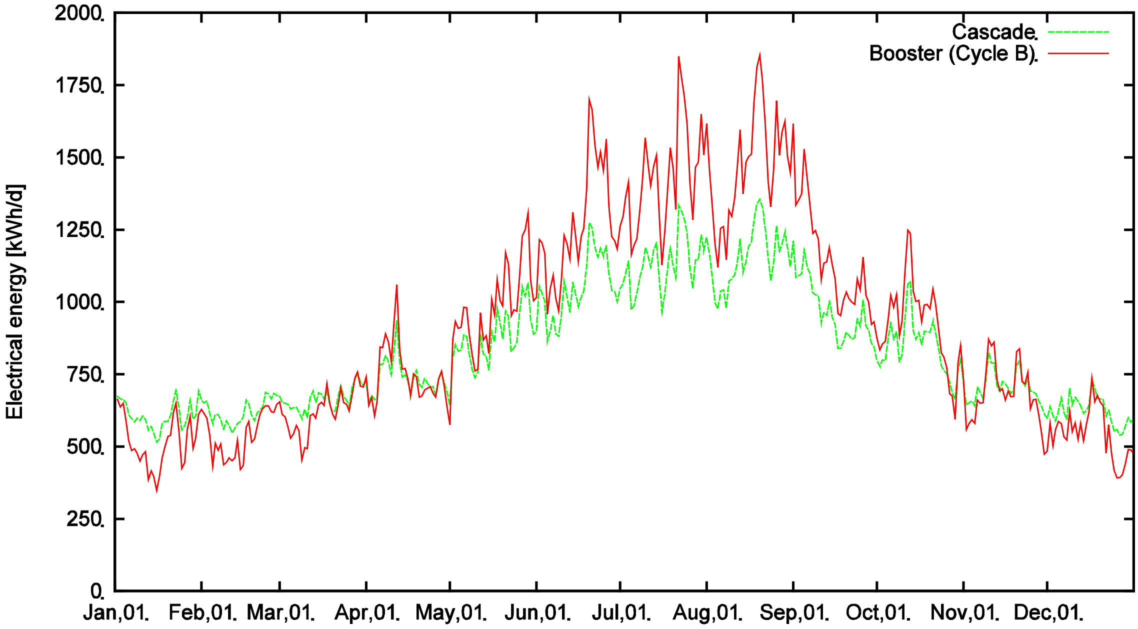

Figure 5. Daily electrical energy consumption of cascade cycle and basic CO_2 cycle (Cycle B)

4.2. CO_2 cycle with subcooling (Cycle BHS)

The main concern for a CO_2 cycle in the warm season is related to the condensing pressure which is very close to critical conditions or to high gas cooler exit temperature. Both these occurrences give way to high quality of refrigerant at the liquid receiver. The heat exchanger HX1 operates to cool the high density gas (or to cool down the liquid below its saturation temperature) taking advantage of cold water stored in the tank, thus reducing the flash gas flow rate.

The water in the reservoir is then subject to a temperature rise which has been depicted in Figure 6. The maximum temperature is 27°C while the equilibrium temperature during inactivity is around 17°C. It is clear that heat losses from the tank to the ground play an important role, and can effectively perform water cooling during the year.

The daily energy rejected to the reservoir during subcooling is plotted in Figure 6 as well. Some sporadic subcooling takes place in February and in March and increases during spring and summer. The total heat transferred to the tank by subcooling amounts to 25.8 MWh/year while the electrical energy demand is reduced to 323.0 MWh/year, i.e. a reduction of 2.4% compared to the Cycle B. Doubling the volume (2V) of the reservoir, the heat rejected increases to 35.6 MWh/year, leading to a reduction on the electrical energy demand of 3.6% in comparison with the Cycle B. The limiting case of a reservoir volume equal to ten times the existing one has been simulated as well. The effect of the higher heat capacity on the water temperature is shown in Figure 6 (Tank 10V) but the corresponding annual electrical energy consumption does not get lower than 311.5 MWh, with 52.0 MWh/year thermal energy discharged to the reservoir.

In order to cool down the water, dry cooling has been considered, but in the climate conditions here considered it is not feasible. In fact the temperature difference between water in the tank and outdoor air almost never exceeds 5 K in the period when subcooling is operated (Figure 6). The same conclusion has been drawn comparing the wet bulb temperature for the employment of an evaporative condenser.

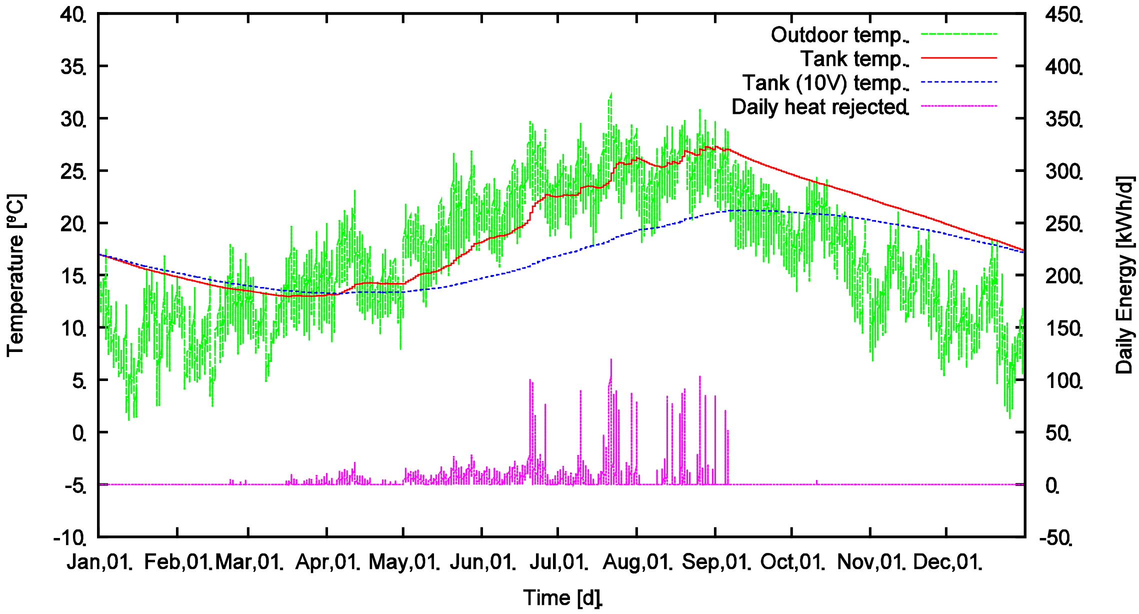

Figure 6. Outdoor temperature and volume averaged water temperature in the reservoir with subcooling operation for a tank volume V=950 m^3 and a tank volume 10 times V. Daily energy rejected to the reservoir during subcooling for tank volume V.

4.3. CO_2 cycle with subcooling, cold storage recharged (Cycle BHSR) and auxiliary compressor (Cycle BHSR+AC and Cycle BHS+AC)

Subcooling showed to be effective, and it could be even more successful when performed at a lower temperature. Given that heat losses from the heat storage to the ground are not sufficient, cooling of water is required when the climate conditions are more favourable and excess refrigerating power is available, i.e. typically at night. Recharging the cold storage by means of HX2 has been considered (see Figure 2). HX2 is a flooded evaporator at 3 °C and HX1 and HX2 operate alternatively according to the control strategy described in Equation (5).

The main control parameters for this system are the temperature set point in the water tank, the refrigerating power made available for water cooling and the operation period. The energy consumption of the system BHSR has been estimated at several configurations, but for the sake of simplicity Table 3 reports the results obtained with $t_{tank,set}$ = 17 °C. During the recharge phase the system is operated at 70 to 90 % of its global refrigerating power; 65 % in the average is required by display cabinets, the remaining part is available for recharging the storage.

Table 3. Energy consumption of BHSR at $t_{tank,set}$ = 17 °C

		Recharging period [h]			
		4	***6***	***8***	***10***
Refrigerating power [%]	**70%**	324,7	325,1	325,4	325,6
	80%	325,3	325,7	326,2	326,4
	90%	325,8	326,3	326,7	327,2

The effectiveness of BHSR is strictly related to the operating conditions at which recharging can be performed. Values in Table 3 show that at the climate conditions considered the global energy usage is almost unaffected by the control parameters considered for recharging. The advantage of subcooling is counterbalanced by the energy spent for recharging at all conditions, and this is due to small outdoor temperature excursion. The configuration (refrigerating power = 70%, 4 h) is considered as reference for Cycle BHSR. Figure 7 shows the daily energy rejected to the storage by subcooling and the daily energy taken when recharging. The respective yearly absolute values are 28.4 MWh for subcooling operation and 9.1 MWh for recharging operation. The difference among the two is due to heat loss to the ground.

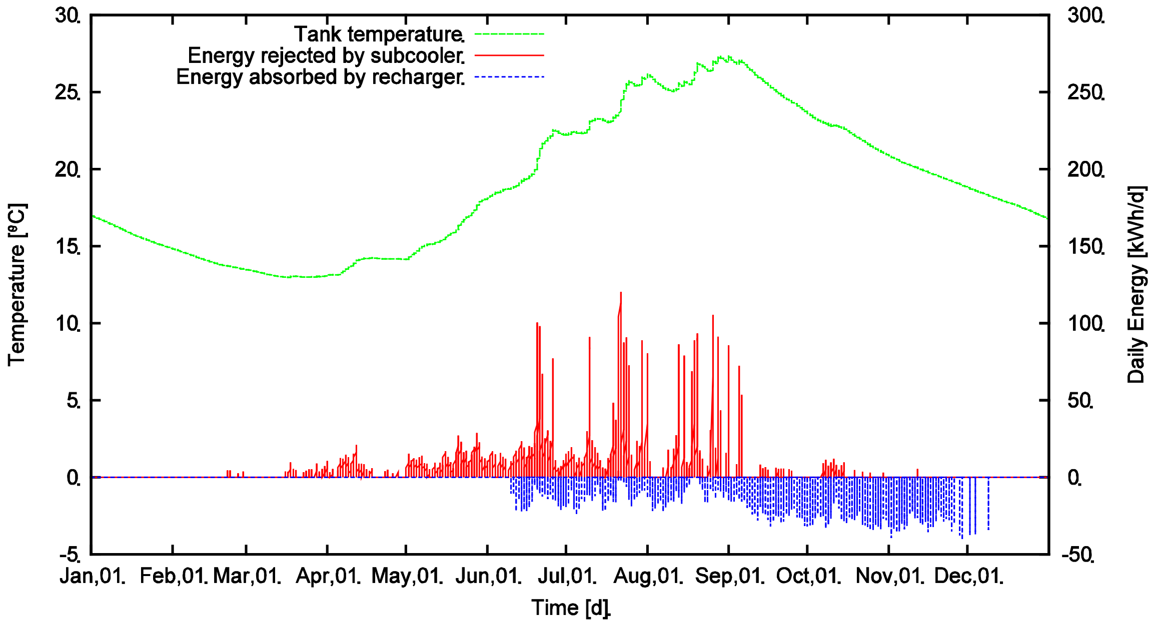

Figure 7. Daily energy rejected by subcooling and absorbed by recharging (cycle BHSR 70%, 4 hours night time)

One more reason for the low efficiency of recharging is possibly related to flash gas compression. Actually this operation pours a great flow rate of vapour into the receiver, which is throttled down to MT pressure before being compressed. Therefore, the adoption of the auxiliary compressor has been evaluated to elaborate the whole mass flow rate of vapour generated at the liquid receiver and keep it at the desired pressure (Cycle BHSR+AC). With this system the energy consumption decreases to 304.1 MWh, around 6.6 % lower than BHSR (324.7 MWh) and comparable to the reference cascade cycle.

Figure 8 reports the Coefficient of Performance (COP) of the cycles BHS and BHSR with and without auxiliary compressor. The COP is calculated as the ratio between q_{RDC} and the electrical energy consumption. The effectiveness of flash gas removal through an auxiliary compressor is clear in the warm season.

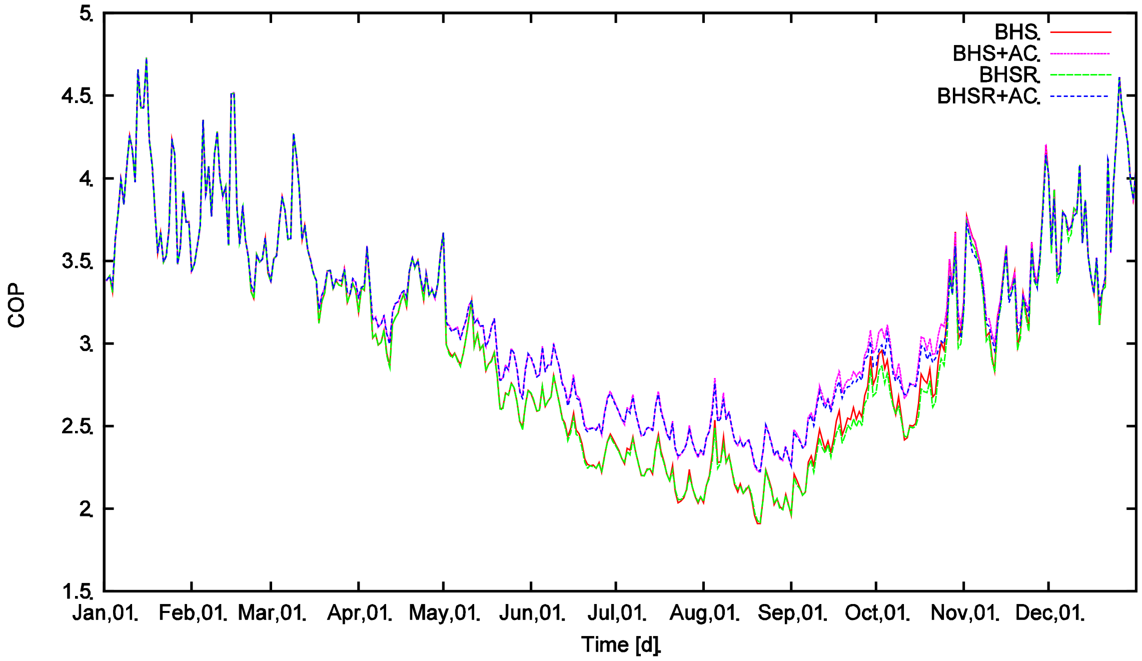

Figure 8. Daily averaged COP of CRU for CO_2 cycles BHS and BHSR with and without auxiliary compressor

In Table 4 the annual electrical energy consumption is then compared for the cycles considered in the simulations. By using the cold water storage for subcooling and the auxiliary compression (BHS+AC), it is possible to achieve the same electrical consumption as a common HFC cascade cycle.

Table 4. Annual electrical energy consumption for cascade and CO_2 systems

Configuration	*Cascade*	*B*	*BHS*	*BHSR*	*BHS+AC*	*BHSR+AC*	*BHS*+AC*
Annual electrical energy consumption (MWh)	303.9	331.0	323.0	324.7	302.9	304.1	276.3

BHS*: BHS configuration with water storage at 10 °C

Further considerations should be made about the heat recovery option. The availability of a heat storage allows performing a significant heat recovery in favour of other systems like HVAC or hot water production, while further improving the performance of the refrigerating unit. An optimal heat storage temperature has been identified at 10 °C, which allows heat recovery from both condensation and subcooling, and makes energy available to heat pumps for space heating in wintertime and HVAC reheating in summertime.

Table 5 reports the yearly energy consumption for the BHS+AC Cycle estimated with the water storage at 10 °C. In the period from October to April heat is recovered from condensation while in the period from May to September heat is recovered from subcooling. The yearly energy consumption decreases to 276.3 MWh, while 763.1 MWh are made available from heat recovery, mostly in wintertime when they can be fruitfully employed for space heating.

Table 5. Energy consumption and heat recoverable with water storage at 10 °C

	Oct-Apr	*May-Sep*	*Year*
Energy consumption (MWh)	122.6	153.7	276.3
Heat recoverable (MWh)	638.3	124.8	763.1

It is worth mentioning that thanks to the high discharge temperature encountered with CO_2 cycles, a portion of the recoverable heat can be supplied to hot water production by means of a devoted heat exchanger.
It clearly appears that the employment of water storage allows a significant heat recovery, especially because it uncouples supply and demand profiles thus making the option of heat recovery much more feasible.

5. CONCLUSIONS

Environmental impact and energy consumption are two of the main drivers in the development of new solutions for commercial refrigeration systems. Carbon dioxide is being more and more promoted, due to its negligible environmental impact. However up to now its application as the only refrigerant is especially devoted to cold climates, which allow for the best exploitation of this fluid. Various solutions are available in the literature to extend a convenient application of CO_2 also to mild climate conditions, through modifications of the refrigeration system and the adoption of various innovative configurations. The development of a comprehensive simulation tool has made available the comparison of various scenarios where different configurations of CO_2 refrigerating cycles linked with a water store are considered. In fact, water tanks for the fire prevention system are often available and they can be profitably used as heat storage devices.
Results showed that at the climate conditions of a mild seaside North Mediterranean location, the basic CO_2 “booster” cycle requires about 9 % more energy on a yearly basis, when compared to a baseline HFCs cascade configuration. The employment of the heat storage as cold sink for performing subcooling allows a slight reduction in the energy consumption (about 2 to 4 %), which could be significantly improved with a lower heat sink temperature. However, at the selected location, neither dry cooling is feasible nor cooling via an auxiliary evaporator during night time is effective, due to a limited daily temperature excursion. The employment of the auxiliary compression shows to be the sustainable solution to achieve similar energy consumption as the baseline cascade cycle.
The availability of the heat storage allows a considerable heat recovery in favour of a heat pump for space heating or other purposes, with advantages also for the refrigerating unit. In fact, performing both auxiliary compression and subcooling, if water temperature in the store is kept at 10 °C not only the energy consumption of the refrigerating system is 9 % lower than the baseline, but also the whole amount of heat rejected all through the year can be effectively collected. This option is of great interest because heat storage allows to uncouple supply and demand profiles and makes the option of heat recovery much more feasible. Heat recovery in favour of heat pumps is the best choice, nonetheless a portion of the desuperheating or gas cooling heat can be directly devoted to hot water production or other high temperature purposes.
In general it is demonstrated that CO_2 plants in mild climates are feasible and energetically acceptable, provided that improvements to standard cycles are adopted. Furthermore, heat recovery can be improved through the employment of a heat storage.

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7. ACKNOWLEDGMENT

The research leading to these results has received funding from the European Community Seventh Framework Programme (FP7/2007-2013) under grant agreement n° 608678.

Paride Gullo would like to thank the "Fondo Sociale Europeo in Friuli Venezia Giulia" for its support.