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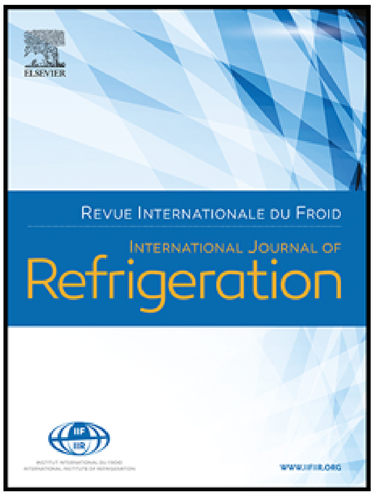

TRANSCRITICAL CO_2 COMMERCIAL REFRIGERATION PLANT WITH ADIABATIC GAS COOLER AND SUBCOOLING VIA HVAC: FIELD TESTS AND MODELLING

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Highlights

- Evaporative pads applied to gas cooler are very effective to reduce energy use
- Subcooling by chilled water from HVAC is as effective as parallel compression
- Evaporative gas coolers are a promising solution also in hot and humid climates

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TRANSCRITICAL CO_2 COMMERCIAL REFRIGERATION PLANT WITH ADIABATIC GAS COOLER AND SUBCOOLING VIA HVAC: FIELD TESTS AND MODELLING

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ABSTRACT

Subcooling methods at the exit of the gas cooler in transcritical CO_2 commercial refrigeration systems have been studied in the recent years showing that overall remarkable improvements can be obtained. Another strategy that results efficient is the use of evaporative systems at the gas cooler (adiabatic cooling) as it allows to significantly reduce the refrigerant quality at the liquid receiver and to lower the heat rejection pressure. In this work, a fully instrumented CO_2 transcritical booster system with parallel compression, in operation in a small size supermarket in northern Italy, made available measured data of its performance when subcooling and/or adiabatic cooling are active. The plant operates in a mild climate, where it suffers operation at transcritical conditions for most of the year. Subcooling in this plant is performed by coupling the refrigeration system with the HVAC system. Taking advantage of experimental measurements, a model in the TRNSYS environment is validated and allows the prediction of the annual plant performance when these strategies are adopted. The adiabatic cooling showed to allow a significant reduction (about 10%) in the energy use, and makes unnecessary the use of a parallel compressor. Subcooling by the HVAC gives rise to a reduced saving (2.9 %) due to the absence of a dedicated mechanical subcooler, however it is almost comparable to parallel compression. These trends are confirmed in two other hot and humid climates.

Keywords: CO_2, Field measurements, Modelling, Subcooling, Adiabatic Cooling, Commercial Refrigeration

NOMENCLATURE

AC	Air Conditioning	GC	Gas Cooler
AD	Adiabatic Cooling	LS	Low Stage
COP	Coefficient Of Performance	LT	Low Temperature
CRU	Commercial Refrigeration Unit	MT	Medium Temperature
EER	Energy Efficiency Ratio	PC	Parallel compression
HPV	High Pressure Valve	Q	Heat flow rate [kW]
HS	High Stage	SC	Subcooling via HVAC system
HVAC	Heating, Ventilation, Air Conditioning	*t*	Temperature [°C]

1. INTRODUCTION

CO_2 is being increasingly employed as refrigerant in many applications, among which commercial refrigeration, with the aim of removing any direct greenhouse effect derived from the use of high Global Warming Potential (GWP) refrigerants. Among the available natural refrigerants, CO_2 turned out to be a viable candidate, especially for supermarket applications, because it combines favourable environmental properties (GWP = 1) and high safety characteristics (non-flammable and non-toxic, A1 ASHRAE classification) along with excellent thermo-physical properties.

However, in warm and hot climates the energy efficiency of CO_2 plants suffers their operation at transcritical conditions, due to the low critical temperature (approximately 31°C) of CO_2, thus partly invalidating the efforts to reduce the global warming effect.

Therefore, in order to extend the convenience of use of CO_2 in warm climates, several technologies and alternative plant schemes have been studied and analysed in the last decades, showing that improvements of the performance of transcritical CO_2 refrigeration plants can be achieved. Among the available solutions, the improvements associated with subcooling of the fluid exiting the gas cooler are being recently investigated by many researchers. Subcooling permits to reduce the amount of flash gas production at the liquid receiver on booster systems. It can be performed by an internal heat exchanger, by chilled water from air conditioning chillers or reversible heat pumps, by coupling the refrigeration and the HVAC systems (Cortella et al. 2014a, Polzot et al., 2017), or by benefitting of cold water storage (Polzot et al., 2015, Polzot et al., 2016).

In addition to the solutions just mentioned, Dedicated Mechanical Subcooling (DMS), which consists in using an additional vapour compression cycle, sometimes with a secondary fluid, to provide subcooling, is being increasingly used even if not yet completely analysed (Llopis et al., 2018). On the other side, if we focus on the improvement of the performance of the gas cooler, another method is to adopt an evaporative cooling system with evaporative pads, designed for dry coolers both for refrigeration and HVAC system (De Angelis et al., 2017). Outdoor air is forced to pass through a wet pad where it is adiabatically cooled before flowing through the finned coil ("adiabatic" gas cooler).

Several studies are available in the literature on DMS systems, and Llopis et al. (2018) have recently presented a detailed review work on subcooling techniques.

Focusing the attention on simulations in transcritical regime, Hafner and Hemmingsen (2015) evaluated the theoretical performance of a direct expansion DMS system with R290 as refrigerant in a CO_2 plant with internal heat exchanger. With a DMS capacity 30% of the system refrigerating capacity, they estimated a reduction from 3% to 23% of the energy demand if compared to a R404A direct expansion plant. Also Llopis et al. (2015) simulated the use of a R290 DMS in single and double stage CO_2 plants with intercooler, at -30, -5 and 5 °C evaporating temperature. They predicted increments up to around 20% at outdoor temperature higher than 25°C and maximum capacity gain of 28.8%. In their computation they considered a maximum value of 7.5 K for the subcooling. Gullo et al. (2016) simulated a booster refrigeration system with a direct expansion R290 DMS system for a small supermarket in warm European climate (Valencia and Athens). Two cooling capacities for the DMS were considered, to achieve 7°C or 15°C at the CO_2 subcooler exit. They obtained a COP increase around 23% at transcritical conditions for both configurations. Similar configurations were investigated by Catalan-Gil et al (2019) who obtained comparable results and concluded that DMS is more beneficial than parallel compression at high outdoor temperatures. Also Liu et al. (2019) made theoretical comparisons among various CO_2 plant configurations with mechanical subcooling, performing also an exergy analysis to support the adoption of subcoolers. Finally, Yang and Zhang (2010, 2011) investigated the effect of subcooler size and control, which shows an optimal point and should be considered.

With regard to experimental results, lab tests were performed by Llopis et al. (2016) on a single stage double-throttling CO_2 system (4 kW) with a direct expansion R1234yf subcooler (0.7 kW). The system operates at constant compressor speed and 0°C evaporation temperature, with gas cooler and DMS condenser cooled by water at 24, 30.2 and 40°C. With evaporating temperature of 0°C and -10°C and cooling water at

24, 30.2 and 40 °C the authors measured on the same plant COP increments from 6.9 to 30.3%. Beshr et al. (2016) and also Bush et al. (2017) performed tests on a prototype plant but with indirect DMS. The subcooler operates with R134a and a water-glycol mixture is used as heat transfer fluid. They found a COP increase of 33.5% at 29°C rejection temperature and 36.7% at 35°C.

On the contrary, Mazzola et al. (2016) worked on real applications, and tested different supermarket CO_2 refrigeration systems with DMS and with subcooling performed by chilled water or groundwater. DMS was tested in a supermarket in a hot climate, switching it on only at transcritical conditions. They verified the chance to reduce the gas cooler pressure by 10 bar with outdoor temperature at 40 °C, allowing the system to operate below 100 bar. A reduction in the electrical power peak was estimated at around 30 % at such condition, with an annual energy saving of around 25% compared to the same system without subcooling.

As regards adiabatic coolers, some authors dedicated their research to study their performance and integration in practical applications. In particular, Camargo et al. (2005) first presented a mathematical model for an adiabatic saturation system, named "DEC-direct evaporative cooling" for thermal comfort purposes. Then they carried out laboratory experimental performance tests on such system applied to an air conditioning plant, by means of an evaporative pad with wetted area of 11.5 m^2. They measured its effectiveness and observed that it is almost linearly dependent on the outdoor dry bulb temperature; they achieved a reduction up to 7.4 °C of the air temperature after saturation.

Filippini et al. (2018) compared the performance of a single stage CO_2 system with flash-gas valve equipped with two technological solutions on the gas-cooler, i.e. adiabatic cooling (cellulosic wet panels on the cooling air path of the gas-cooler) and evaporative cooling (water sprayed directly on the gas-cooler) together with the combination of the two. The performance was analysed in reference to the base case with dry cooler and in three different climates. They defined an activation set of the adiabatic panels at 17 °C and the choice was due to economic optimization, i.e. water costs balanced with electricity cost saving. The analysis showed that the use of gas coolers with wet systems makes the CO_2 cycles efficient also in areas with average warm annual temperature and that the simultaneous use of both systems is a worthwhile solution.

Finally, Malli et al. (2011) experimentally investigated the thermal performance of cellulosic evaporative cooling pads made from corrugated papers. They underlined the need for an accurate choice of the pad, to find a good match between heat transfer characteristics, pressure drop and water consumption. An optimum point may occur in the selection of the pad thickness depending on air speed velocity.

In this paper, measurements taken on a CO_2 transcritical plant located at a small size supermarket in Italy, at "humid subtropical climate" conditions (Peel 2007), are considered. The measurements were taken when adiabatic cooling and indirect mechanical subcooling were active (Cortella et al., 2019). In this application, for model validation purposes, subcooling is performed by the reversible heat pump of the HVAC plant. The field data allowed to complete the calibration of a comprehensive model built in the Trnsys environment (Klein 2010) of the Commercial Refrigeration Unit (CRU), already widely validated in D'Agaro et al. (2019) without subcooling and adiabatic cooling. The system has been described by in-house routines. Time dependent simulations are performed, with a 1 min time step based on actual weather and room climate conditions, for the validation of the model on a weekly basis.

Then, annual simulations with an hourly time step are performed for the estimation of power profiles and annual energy consumption, in order to evaluate the convenience of each solution and compare the overall performance of the entire system when these strategies are adopted in such kind of refrigeration system.

2. REFRIGERATION SYSTEM

The Commercial Refrigeration Unit (CRU) analyzed in this work is an existing monitored transcritical CO_2 booster system with parallel compression and heat reclaim at two different temperature levels. Such system is installed in a supermarket which has been refurbished in the framework of the FP7 European Project

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CommON*Energy* (Cortella 2014b, Commonenergy 2017). It is a small supermarket with a selling area of approximately 1200 m^2, located in Modena (IT), that has undergone renovation during summer 2016, where the retrofitting process involved several aspects such as plants, envelope, solutions for both artificial and natural lighting. In particular, the solution set which involved the refrigeration system consists of the installation of the new CRU and of an array of new generation closed refrigerated display cabinets. The solution was also focused on its integration with the HVAC. The CRU is equipped with a heat exchanger used for subcooling where the cooling energy is provided by the chilled water of the HVAC system.

2.1 Refrigeration System

The refrigeration system considered is composed by a transcritical CO_2 booster system, and closed refrigerated display cabinets (RDC), both for chilled and frozen food. The nominal compressor capacity is 70.5 kW for High Stage (HS) and 10.8 kW for the Low Stage (LS), at -10°C and -35°C respectively (Figure 1). The HS discharge pressure p_{HS} is driven by the outdoor and operating conditions. In addition, there is a parallel compressor to process the flash gas coming from the liquid receiver placed after the first expansion valve HPV (High Pressure Valve) at the receiver pressure of 35.0 bar.

Both compressor racks, LS and HS, are composed by two compressors, distinguished as master (variable speed) and slave (ON/OFF type) compressor. The high stage nominal electrical power is of 23.7 kW, i.e. 8.1 kW for the master compressor and 15.6 kW for the slave; similarly, the low stage has a nominal electrical power of 2.4 kW, i.e. 1.05 kW for the master and 1.35 kW for the slave. Other 9 kW of nominal electrical power must be considered for parallel compressor.

The gas cooler is equipped with an Adiabatic Cooling (AD) system. Outdoor air that flows through the gas cooler is made to pass through cellulosic pads where an evaporative cooling process takes place when water is supplied. Even if the adiabatic cooling system is particularly efficient in hot and dry climates, it is widely used in less severe climate conditions, where a reduced usage of water is encountered.

Furthermore, it is possible to activate a subcooling heat exchanger in order to perform subcooling at the exit of the gas cooler, at the expense of the HVAC system, to further lower down the gas cooler outlet temperature in transcritical mode. Such plant allows for several configurations of the system. In Figure 2 the thermodynamic cycle in the two configurations with parallel compressor or SC is depicted.

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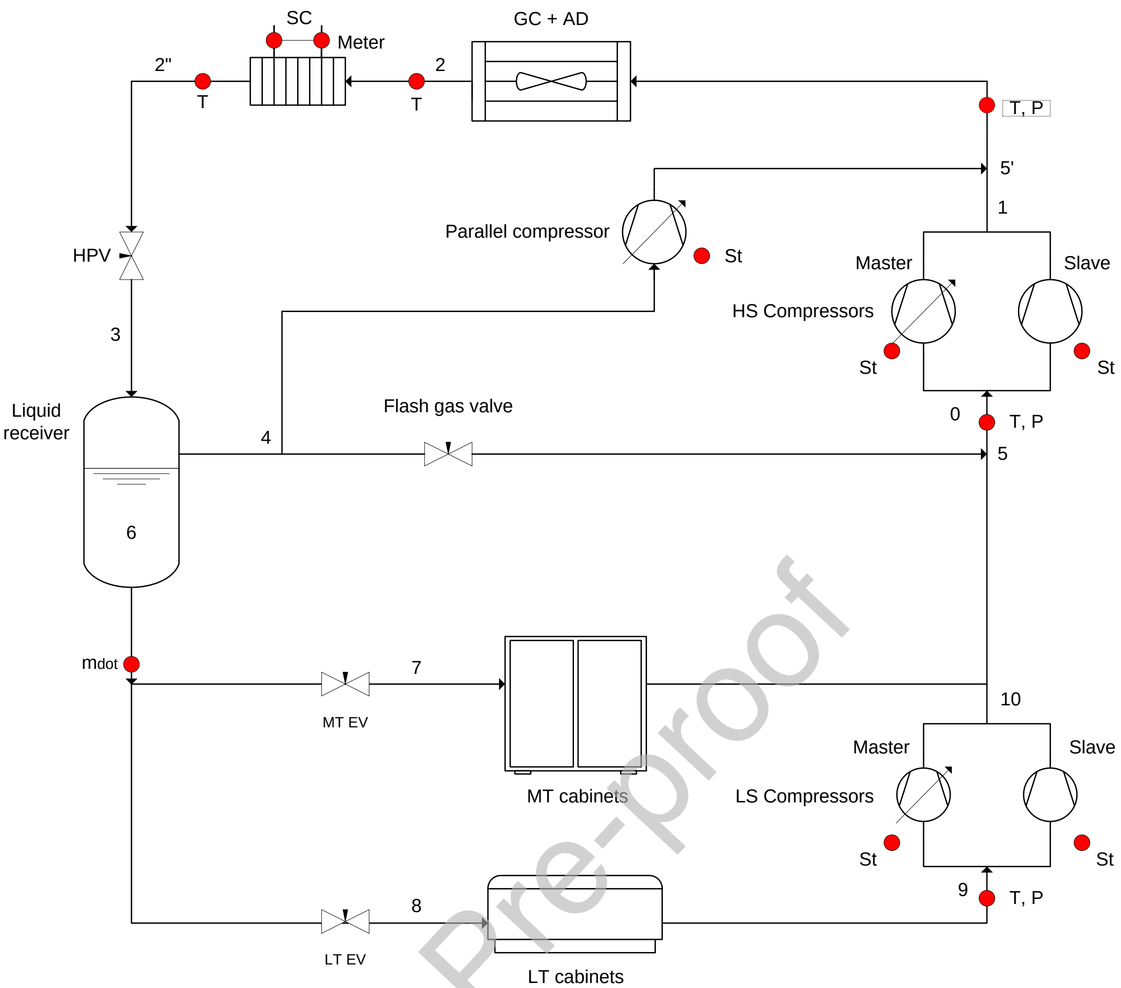

Figure 1: Schematic drawing of the CO_2 refrigeration system. The approximate position of the probes is indicated by the red flags. T=temperature probe, P=pressure gauge, Meter=energy meter, m_{dot}= mass flow meter, St=compressor status sensor.

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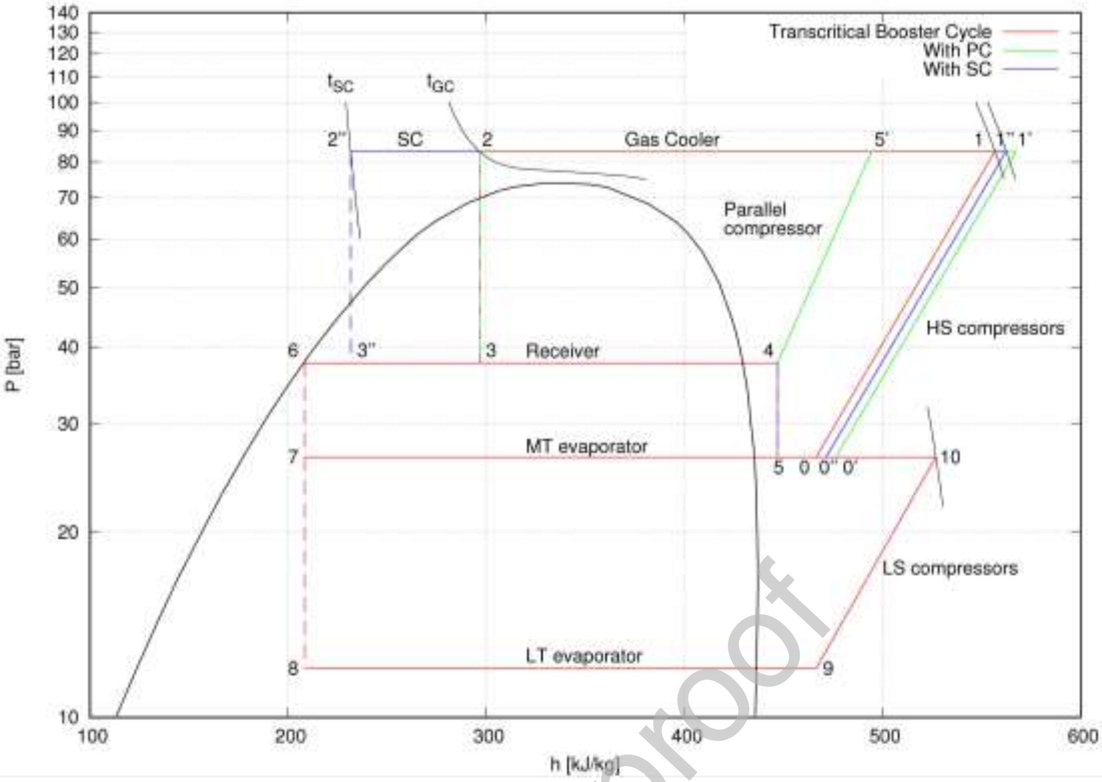

Figure 2: (p-h) diagram of the transcritical CO_2 Booster system cycle (red) with parallel compression (green) or with SC (blue).

2.2 Thermodynamic model of the CRU

A mathematical model of all the components of the refrigeration system, from the final users (display cabinets, cold rooms) to the refrigeration unit, has been developed in the TRNSYS environment, to carry out time dependent simulations of this system. To consider the mutual interactions with the building and with the HVAC system, the model has been extended to the whole building, allowing the computation of thermal loads on the HVAC system and of mutual interactions between cabinets and indoor environment, at any climate condition.

The models of the display cabinets and of the cold rooms are described in detail in Polzot et al. (2016) and D'Agaro et al. (2019). They estimate the total cooling capacity by tuning the performance at rated conditions of cabinets and cold rooms according to the time-dependent indoor conditions in the supermarket. The sensible and latent contributions from the refrigerated equipment to the HVAC system are also calculated and considered for the HVAC loads.

The sub-hourly profiles of the cooling load at the evaporators are an input for the model of CRU, along with the outdoor temperature and the heating demand profiles. The model of the CO_2 transcritical booster system with parallel compression is also described in detail in Polzot et al. (2016) and more recently in D'Agaro et al. (2019), where heat recovery is also considered, and a thorough validation process with field data from the plant aforementioned can be found.

The values of the main design parameters are reported in Table 1 and reflect the settings of the existing plant. The minimum condensing temperature is needed to allow both the MT expansion valves and the HS compressor working within their operating range at subcritical conditions. The liquid receiver pressure is kept constant at 35 bar, even if it could be beneficially raised in the presence of subcooling. However it was kept constant in the plant we monitored, for the sake of improving control stability. The superheating values at both evaporating levels are inferred from the experimental data and count for useful (a few degrees) and not-useful superheating, mostly due to heat loss, especially in the long suction lines.

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Table 1. Main design parameters for the commercial refrigeration unit.

Parameter	***Unit***	***Value***
LT evaporating temperature	°C	-35
MT evaporating temperature	°C	-10
Minimum condensing temperature	°C	6
Liquid receiver pressure	bar	35
Subcritical subcooling	K	3
Gas Cooler/Condenser approach temperature difference	K	4
Subcooler approach temperature difference	K	7
LT superheating	K	30
MT superheating	K	20

3. FIELD TESTS AND VALIDATION

The model has been validated with monitored data collected in the hot season, integrating the previous validation (Cortella et al. 2018, D'Agaro et al. 2019) with the plant operation by SC (subcooling by HVAC) and AD (adiabatic cooling). The system is instrumented with sensors placed as in Fig. 1, to measure temperature (Pt500, ± 0.5 K), pressure (pressure gauges 0-80 bar for the evaporating pressure, 0-140 bar for the high stage and intermediate pressure, ± 1% f. s.), refrigerant mass flow rate (Coriolis mass flow meter, ± 0.1 % reading), heat meters for the secondary fluids at the heat exchangers (ultrasonic mass flow meter, Pt500 temperature sensors, ± 0.8 %), electrical power (energy analyser, ± 1%).
In the following sections the comparison between the results from the simulations and the monitored data is reported for four selected weeks, that cover all the possible operating conditions under exam, and are the following:

- AD week 1 (from the 1st to the 7th of June, 2018);
- AD week 2 (from the 15th to the 21st of June, 2018);
- SC week 3 (from the 7th to the 13th of September, 2018);
- SC week 4 (from the 14th to the 20th of September, 2018);

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3.1 Adiabatic cooling validation

When it comes to Adiabatic Cooling, the thermal process is the adiabatic saturation. An interesting characteristic of this process is that it is more efficient at high temperature, allowing to enhance the annual performance of those processes that require the transfer of heat to environment.
Due to manufacturer constrains only a portion of the air flow is subject to adiabatic cooling. A global effectiveness of the overall process is defined as

$$\varepsilon = \frac{t_{db} - t_{air}}{t_{db} - t_{wb}} \quad (1)$$

where t_{db} and t_{wb} are the dry and wet bulb temperature values for outdoor air, and t_{air} is the temperature of the air after its cooling. With an appropriate approach temperature difference at the gas cooler (Table 1) from the experimental tests the effectiveness value showed to be ε = 0.6.
The adiabatic saturation of air is described in the model using the *CoolProp* libraries (Bell et al., 2014) for humid air.
The adoption of AD allows a remarkable reduction in the discharge pressure and in turn yields a significant reduction of the compressor power, especially at transcritical regime. Figure 3 shows the difference between the gas-cooler pressure in dry gas-cooling conditions (gas-cooler pressure rules described in Polzot et al., 2016) and with adiabatic cooling, for two typical weeks in transcritical regime. An average difference of 10 bar can be noticed. The outdoor temperature and relative humidity are the inputs of the adiabatic cooler model, and are taken from the monitored data.

In particular, Figure 3 shows the comparison of the pressure values at the outlet of the HS compressors for the two selected weeks, as well as the computed pressure profiles, which align quite well with the measured data confirming that the adiabatic cooler, with the assumptions above, is accurately described for a rather wide range of outdoor conditions typical of the summer season.

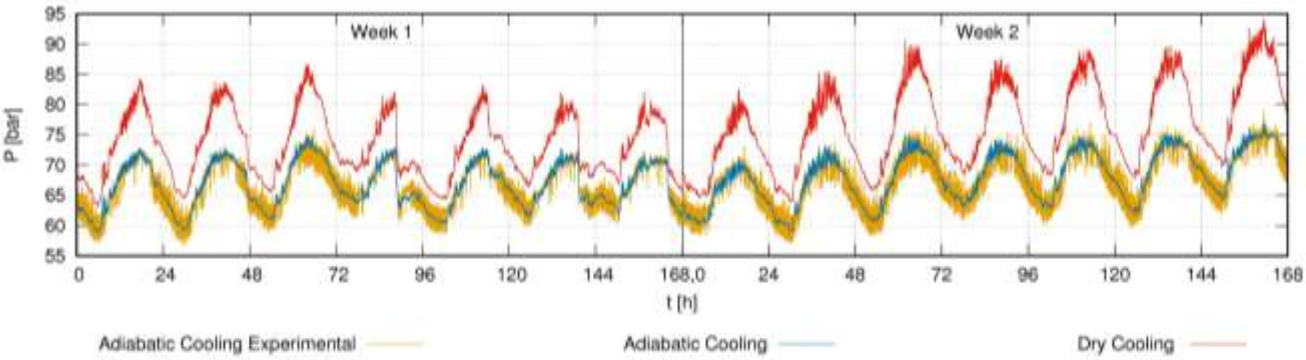

Figure 3: Pressure at the outlet of HS compressors: comparison between adiabatic and dry cooling, and validation of the adiabatic cooling model.

The model has been widely validated in terms of the estimation of energy use at the compressors (D'Agaro et al, 2019), and the introduction of the AD system does not affect its accuracy. Table 2 shows a comparison between measured and simulated weekly electrical energy use for the two weeks in Figure 3.

Table 2. Comparison between monitored and simulated electrical energy demand for the selected weeks.

Week	*Operating conditions*	*Monitored Electrical Energy Demand [kWh]*	*Simulated Electrical Energy Demand [kWh]*	*Error simulated vs monitored [%]*
1	AD	4034	3966	-1.7
2	AD	3931	3793	-3.5

3.2 SC validation

Subcooling is based on the use of a subcooling system at the exit of the gas-cooler/condenser and prior to the high pressure valve. The beneficial effects of subcooling in a transcritical booster system is mainly a reduction of the vapour quality in the liquid receiver, with a consequent reduction in flash gas production. Also a reduction of the optimum discharge pressure could be considered. Of course, all this is obtained at the expense of an energy input to the subcooling system.

Dedicated Mechanical Subcooling (DMS) consists of an auxiliary vapour compression refrigerating system especially devoted to cool the refrigerant, in this case CO_2, at the exit of the gas-cooler/condenser. This function can also be performed by the air conditioning reversible heat pumps, which is our case of study, by means of a chilled water loop and a heat exchanger (SC; Figure 1).

Figure 4 shows an example of a few measurements of the quantities related to the subcooler in a typical week.

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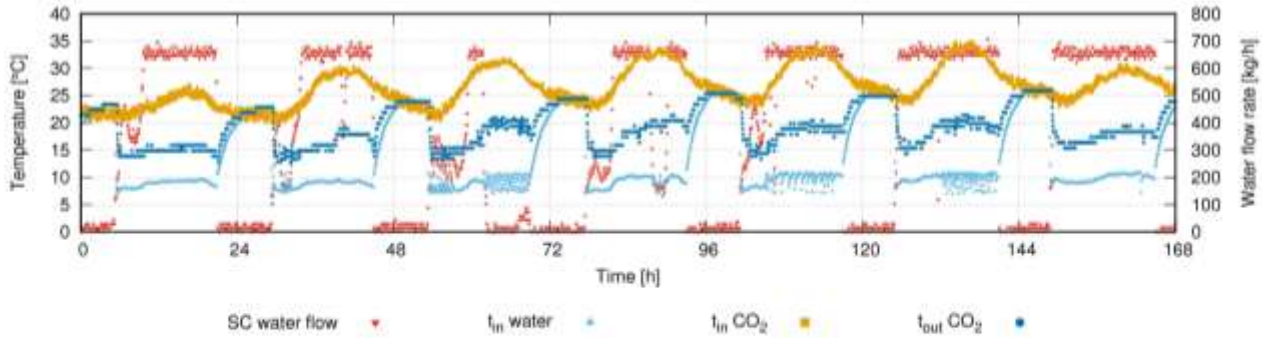

Figure 4: week 3, SC behaviour, experimental values.

SC operation is controlled by the water flow rate, based on the gas cooler outlet temperature and subject to chilled water availability. Chilled water enters the subcooler at 8 °C on average when the system is activated, and the approach temperature difference appears to be around 7 K. The maximum heat flow rate exchanged is around 18 kW in this plant, corresponding to the heat exchanger design size. No gas cooler pressure optimisation has been introduced in this system.

The CO_2 outlet temperature t_{outSC} and the heat flow rate Q_{SC} have been selected as reference quantities for the validation of the SC model. The inputs are the monitored water inlet temperature and mass flow rate. Figure 5 shows the comparison between experimental and simulated data for two weeks when SC is operating.

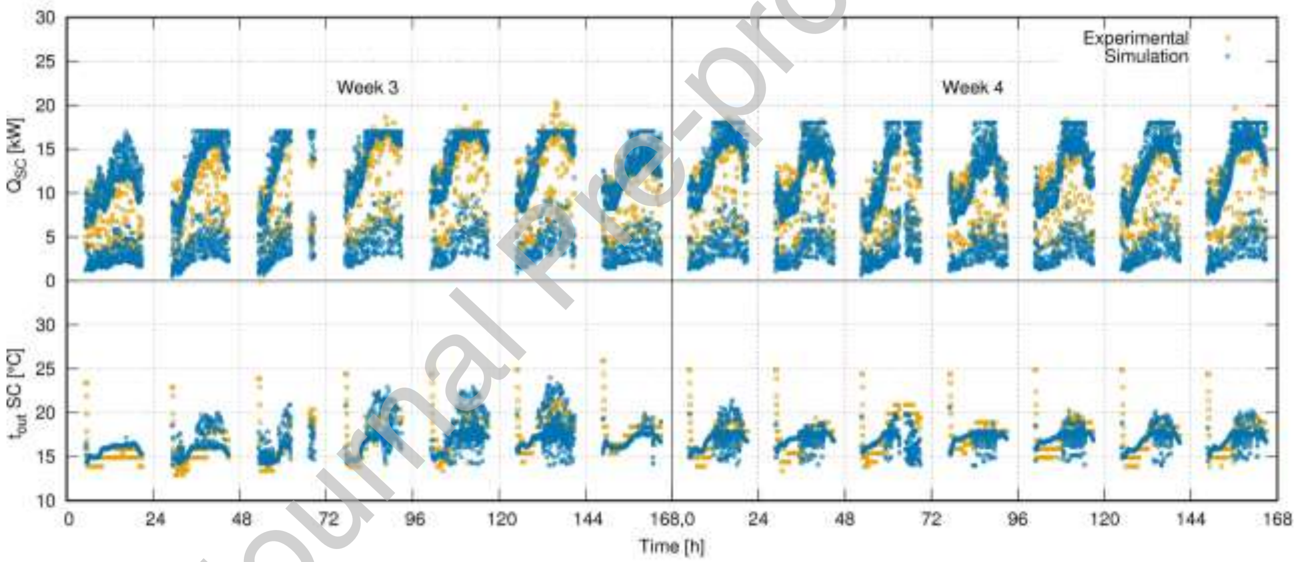

Figure 5: weekly profiles of the CO_2 outlet temperature and of the exchanged heat rate, experimental vs simulation, weeks 3 and 4.

The experimental data are characterized by some scattering probably due to the reversible heat pump control system. The model tries to reproduce such behaviour through the variation in the water inlet temperature. However, as the model features a steady state simulation at each time step, it cannot catch the actual transient behaviour with the typical time constant of the system, due to the heat capacity of pipes and of the heat exchanger. However, the average value of the CO_2 outlet temperature can be considered reproduced with a rather good approximation. As regards the heat flow rate, the average magnitude is respected although the simulated data show values that align on two distinct bands. This can be explained by the fact that in the model, at each time step, the CO_2 flow rate follows the instantaneous compressor activation status, with the result of yielding two bands depending on the activation of the slave compressor (D'Agaro et al. 2019).

4. YEARLY PERFORMANCE SIMULATIONS

Through validation, the model showed to yield reliable predictions of the annual electrical energy demand in different scenarios and permits to evaluate the performance of the plant scheme under analysis.

4.1 AD and SC performance comparison

In this analysis, the model is used to compare the benefits given by the gas cooler adiabatic cooling (AD) and SC methods. Moreover, the control rules, which basically are the rules used to determine when to activate each component, are sought. The results apply for a mild weather typical of northern Italy, more precisely of Modena, where the supermarket under exam is located. Simulations are carried out for a whole year with an hour time step.

The indoor conditions of the supermarket are obtained by a time dependent simulation of the thermal behaviour of the building. The refrigeration cooling load for LT and MT are obtained from the simulation of the display cabinets and the cold rooms. The integration between the refrigeration system and the building allows capturing the demand variation over the day and 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. In this way, the comprehensive model of the building developed in the framework of CommONEnergy (Dipasquale et al, 2016) has been integrated in the TRNSYS environment with the in-house routines of the display cabinets and CRU.

The cases analysed in this work are the following:

- REF (reference case 1): Booster system with flash gas valve;
- PC (reference case 2): Booster system with parallel compression scheme;
- AD: adiabatic gas cooler applied to REF.
- PC + AD: adiabatic gas cooler applied to PC;
- SC: sub-cooler heat exchanger applied to REF. The sub-cooler outlet temperature is set at 15°C and the maximum heat extraction rate is 18 kW. No gas cooler pressure reduction is considered and the parallel compressor is not activated.

In all cases the CRU is set to operate only to cover the refrigeration duty of the display cabinets and cold rooms of the supermarket, i.e. no heat recovery or AC integration is considered.

The adiabatic gas cooler activates when the outdoor temperature is equal or greater than 19°C, which is the lower limit of the transition zone (described in Polzot et al. (2016), temperature band placed between subcritical and transcritical conditions) and the parallel compressor has the same activation temperature. For instance, Filippini et al. (2018) defined an activation rule of AD at 17 °C, which resulted from an economic optimization. Our choice, based on the operation of the system, is not far from this option. In order to obtain a fair comparison, the same activation rule is adopted for SC.

As already mentioned, the subcooling capacity for SC is provided by the reversible heat pump of the HVAC system, which has an EER calculated with a model of its thermodynamic cycle tuned after the actual functioning of the system. For an evaporating temperature of 2°C, the EER expression as a function of the outdoor temperature t_{ext} is the following:

$$EER = 5.629 - 0.0886\, t_{ext} \tag{2}$$

Chilled water at the inlet of the sub cooler is set at 8°C (experimental data).

Figure 6 shows an example of the yearly profile of the CRU power use and depicts a comparison between the REF and AD cases. A remarkable reduction of the power demand of the system can be observed when AD is active.

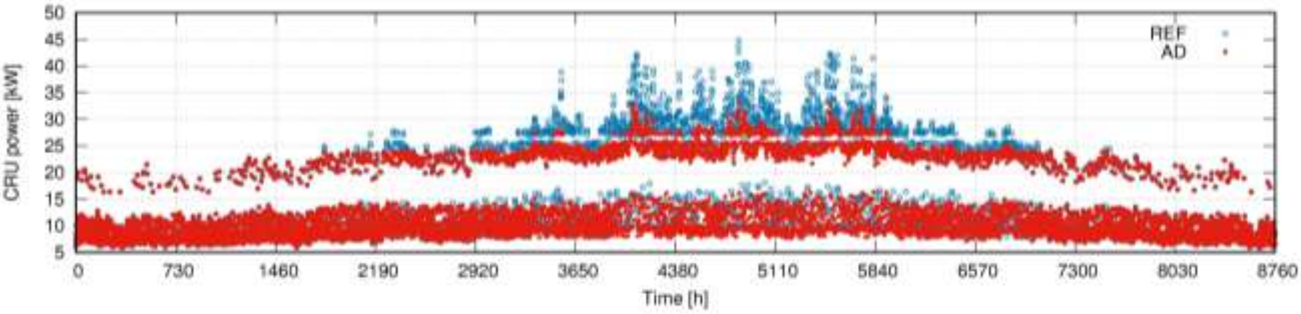

Figure 6: CRU yearly electrical power utilization: comparison between the reference case and the AD one

The electrical energy utilization of the CRU for a whole year, in the analysed cases, is reported in Table 3.

Table 3. Comparison between the annual electrical energy demand of the cases analysed.

Annual Electrical Energy Demand [MWh]					
Scheme	***CRU***	***Sub Cooler***	***Total***	***Energy saving [%]***	***Energy saving vs PC [%]***
REF	128.7	-	128.7	0.0	-
PC	123.5	-	123.5	4.0	0.0
AD	116.2	-	116.2	9.7	5.9
PC + AD	114.7	-	114.7	10.9	7.2
SC	117.1	7.9	125.0	2.9	-1.2

The results, in energy terms, are also presented on a monthly basis in the histogram of Figure 7.

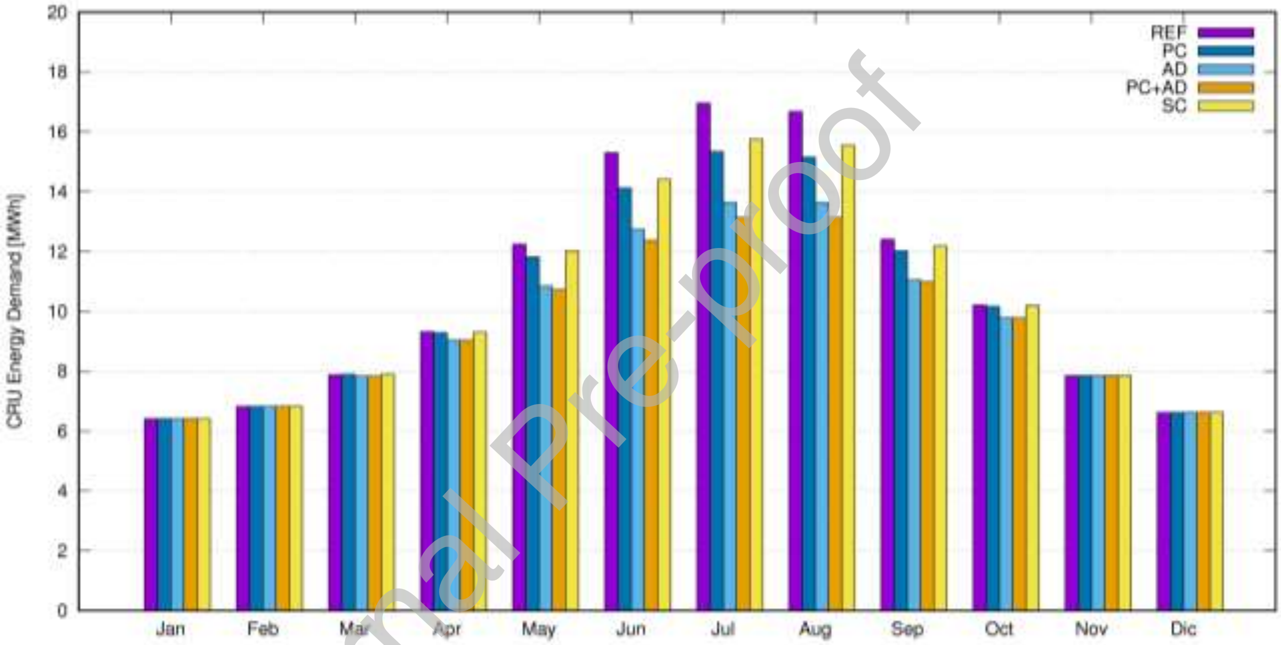

Figure 7: Monthly electrical energy demand for all cases examined.

Firstly, we can see that the performance enhancement methods (AD and SC) yield benefits in the warm and hot months, when higher outdoor temperature appears and the system suffers transcritical operation. The results have been compared both to the REF case and to the PC scheme, in order to offer a direct confrontation with these two widely used configurations.
The PC scheme with AD offers the highest energy saving in this example of CO_2 transcritical booster system. More in general, it can be observed that AD plays a fundamental role in boosting the performance of the plant. In fact, in both cases where AD is active it yields a total energy saving that is almost equal or higher than 10%.

For completeness, a further analysis has been carried out in two example locations of hot climate, such as Doha (Qatar) and New Delhi (India), in order to verify the influence on the energy savings and obtain new performance comparisons between PC, AD and SC schemes. Table 4 summarizes the results, which can be compared to those of Modena reported in Table 3.

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Table 4. Comparison between the annual electrical energy demand of the cases analysed for Doha and New Delhi.

Climate	*Scheme*	*CRU*	*Sub Cooler*	*Total*	*Energy saving vs PC [%]*
Doha	PC	169.3	-	169.3	0.0
	AD	155.5	-	155.5	8.2
	SC	150.2	22.3	172.5	-1.8
New Delhi	PC	162.2	-	162.2	0.0
	AD	150.9	-	150.9	6.9
	SC	145.0	20.3	165.3	-1.9

First of all it appears that in these two climates the CRU and the subcooler, when applicable, require a significantly higher amount of energy. However when we go to the comparison among the configurations, in both climates AD offers the highest system improvement with respect to the PC scheme, while SC is slightly unfavourable (around 2% higher energy use when compared to PC). Furthermore, when compared to the Modena case, the higher number of hours with outdoor temperature greater than 19 °C leads to an additional saving when the AD is used.

As already explained, the adiabatic cooler has the capability of lowering the heat rejection temperature on the gas cooler with the result of setting a lower gas cooler pressure in transcritical conditions. Figure 8 shows the effect of the AD on the reduction in the air temperature during the year.

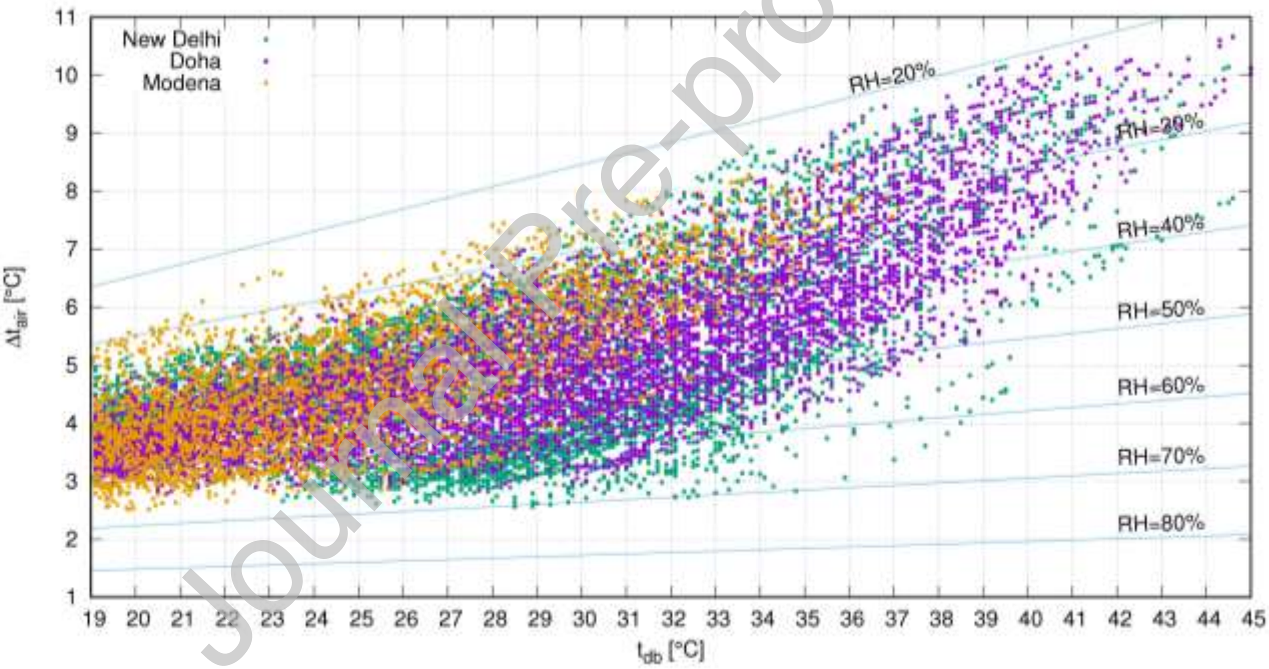

Figure 8: Air temperature decrease due to the adiabatic cooler, results from simulation.

Another important feature that is worth to examine regarding AD is the activation temperature. In Figure 9, the annual energy saving of the CRU for the Modena case are represented as a function of the AD activation temperature in a range that spans from 14°C to 22°C outdoor temperature. As it could be expected, the lower is the activation temperature the higher are the saving, even at low temperature values. This fact should be further investigated with an economical optimisation approach.

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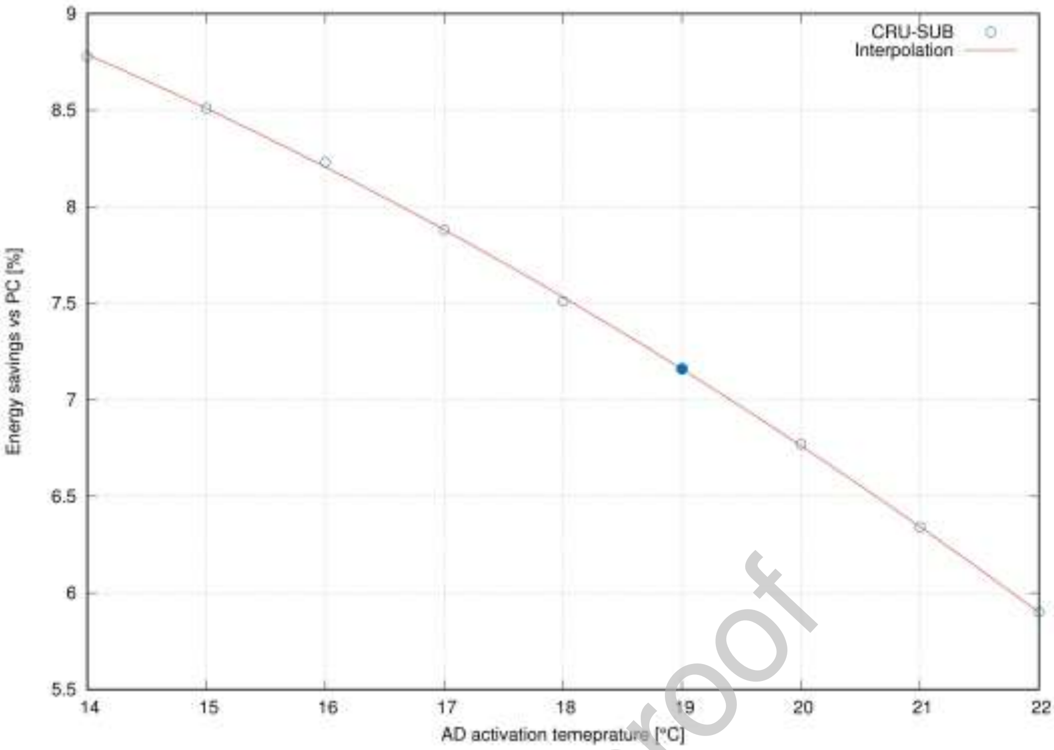

Figure 9: CRU annual energy saving vs PC for the Modena case as a function of AD activation temperature.

Although fairly performing, the SC scheme in this example is slightly less effective than the PC scheme, by roughly one percent. This is mainly due to the relatively low evaporating temperature of the AC reversible heat pump and to the employment of an indirect subcooling system, which involves a secondary loop of water at 8 °C and consequently introduces a cascade heat exchanger with its irreversibility.
A further analysis on a better exploitation of subcooling techniques, considering optimisation and sizing, is worth to be carried out in a further work using the tools available.

5. CONCLUSIONS

Performance enhancement methods for a CO_2 transcritical booster system have been analysed both experimentally (field tests) and with simulations. Field data allowed to complete a thorough validation of a model built in the TRNSYS environment with in-house routines. The techniques investigated are adiabatic cooling on the gas cooler and mechanical subcooling at the exit of the gas cooler, which is performed by means of the HVAC system.
Different plant schemes have been compared. Such comparison showed that, in this example of application and climate, adiabatic cooling by itself remarkably enhances the performance, showing to be an effective solution even when compared to parallel compression. The activation temperature is an important design variable, and its choice should be based on an economic trade-off between running and maintenance costs and energy benefits. The adoption of both adiabatic cooling and parallel compression is the most effective solution, but the contribution of parallel compression is minor, thus suggesting a thorough evaluation at the design stage. Subcooling via a mechanical subcooler should be as well effective, but when it is performed at the expense of the HVAC system, the low evaporating temperature of the reversible heat pump and the water loop introduce significant penalties which reduce its advantages.
The analysis has been extended to two other examples of hot and humid climate, revealing that AD alone is still the most advantageous solution in terms of energy use, while subcooling via HVAC remains slightly unfavourable.
The analysis suggests a further investigation of dedicated mechanical subcooling with a particular focus on the size of the unit and on its optimised control rules, extended to other examples of supermarket applications.

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Declaration of interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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