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MONITORING OF A COMMERCIAL REFRIGERATION CO_2 SYSTEM AND COMPARISON WITH SIMULATIONS

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ABSTRACT

The demand for natural refrigerants is growing in commercial refrigeration systems. In the recent years, carbon dioxide has increased its market share in the field of commercial refrigeration and has proven to be a viable solution for the replacement of hydrofluorocarbon (HFC) systems. This success is mainly due to the ongoing technological evolution of carbon dioxide refrigeration systems.
In the present paper, a CO_2 commercial refrigeration unit serving a supermarket located in Northern Italy is presented. The unit consists of a booster compressor rack with parallel compression, with ten compressors arranged to provide around 20 kW cooling capacity at low temperature and 90 kW cooling capacity at medium temperature. Four out of ten compressors are arranged in parallel. The installation, in addition to the cooling load, provides all the thermal functions in one unit: it integrates a heat exchanger for the air conditioning and the possibility of two stage heat recovery, for sanitary hot water production and for space heating. The refrigeration unit is equipped with pressure and temperature sensors, power consumption and load analysers for the compressors. A computer model has been developed to evaluate the acquired data of the system and to analyse the key parameters. The preliminary results from the monitoring of this unit are presented in this paper and used to calibrate the model of the system. Afterwards, simulations have been performed at variable operating conditions in a cold month to evaluate the performance of the unit. The results of the model have been compared to an independent set of monitoring data.

Keywords: Refrigeration systems, Carbon Dioxide, Compressors, CO2 booster system with parallel compression

1. INTRODUCTION

In the past years, the refrigeration sector has made a big effort to meet the hydrofluorocarbon (HFC) phase down requirements from the European F-Gas Regulation (No 517/2014) and the global agreement to phase down HFCs - the Kigali Amendment to the Montreal Protocol (UNEP, 2016). In this context, the use of carbon dioxide as environmentally friendly refrigerant is one of the most promising and long-lasting solution in food retail sector. CO_2 booster systems has been widely adopted in the northern European Countries for supermarket applications, since their performance at ambient temperature lower than about 25°C is comparable with conventional HFC systems. Although the efficient use of CO_2 in transcritical booster system presents some difficulties at higher ambient temperature, it offers the opportunity of heat recovery and air conditioning integration that should be considered to evaluate the application of this system to warmest climates of south Europe. Karampour and Sawalha (2018) identified two-stage heat recovery and air conditioning integration as two of the most promising features for CO_2 transcritical booster systems. In transcritical operations, the compressed carbon dioxide at the discharge section of the MT compressors can typically reach temperatures over 100 °C. These temperatures, in combination with non-isothermal heat rejection, allow to efficiently recover most of the heat normally rejected at the gas cooler for space heating and sanitary hot water production. On the other hand, the pressures at which the flash tank receiver normally works allow to reach saturation temperatures close to the typical values adopted to feed water into the hydronic network of air conditioning system. The integration of air conditioning and heat recovery in a CO_2 refrigeration system is a compact solution capable to satisfy both the cooling and heating demands in supermarkets. Compared to a complete stand-alone system, the integration of all thermal functions in a single unit requires less space and fewer components, with a reduced risk of service interruption. Karampour and Sawalha (2017) highlighted two main challenges that need to be considered in the use of integrated CO_2 system solutions: the first is the higher prices that CO_2 systems have in comparison with conventional heat pump or air conditioning systems; the second regards the energy performance of the system.

Despite the installation of CO_2 refrigeration systems for supermarket has rapidly increased, only few publications on field measurements of CO_2 system with integrated heat recovery and air conditioning system has been presented. Tambovtsev et al. (2011) investigated heat recovery strategy involving gas cooler bypass by monitoring a supermarket located in Germany. Their results show that the adoption of the gas cooler bypass can increase the total COP of the system up to 20%. In the work of Funder-Kristensen et al. (2013), a CO_2 refrigeration system integrating heat recovery for a supermarket in Denmark has been studied. The heat recovery unit replaced a conventional gas heating system providing the entire heating demands of the supermarket saving 20% of running cost. A field measurements analysis of a supermarket CO_2 refrigeration system with heat recovery installed in Sweden was presented by Abdi et al. (2014). The monitoring of the working conditions shows that only 30–70% of the heat theoretically available for recovery was utilized, while the rest was rejected outdoor. Hafner et al. (2016) presented the integration of heat recovery and air conditioning units in CO_2 refrigeration system for a supermarket in Italy. The performance of the refrigeration and air conditioning system with and without a multi-ejector unit have been compared. This study show that ejector can lead to a total energy power consumption saving between 15% and 30% with respect to the configuration without the multi-ejector unit. Karampour and Sawalha (2017) monitored the performance of an integrated CO_2 trans-critical booster system including space and water heating, air conditioning, and parallel compression installed in a supermarket in Sweden. Their results show that the system was capable to provide the entire air conditioning demands and recover a great fraction of the available heat, with high COP values.

In the present study, the performance of a CO_2 commercial refrigeration system providing all the thermal needs of a supermarket located in Northern Italy is investigated. The analysis involves field measurements and the development of a model of the system. The objective of this study is to investigate the system working conditions and performance while providing refrigeration and heating.

2. SYSTEM DESCRIPTION

The considered CO_2 commercial refrigeration unit is a transcritical booster system with parallel compression serving a supermarket located in Monticello Brianza (45°43′ N - 9°19′ E), near Milan in Italy, and has been in operation since August 2017. The system is capable to provide around 20 kW cooling capacity at low temperature (LT) and 90 kW cooling capacity at medium temperature (MT). In addition to the cooling capacity, it provides all the thermal needs: it integrates a heat exchanger for the air conditioning at the intermediate pressure level and a two stages heat recovery system on the high pressure line. The recovered heat satisfies the demands for sanitary hot water and space heating of the supermarket. The schematic of the integrated CO_2 system is shown in Figure 1. The CO_2 exiting the gas cooler (GC) enters an internal heat exchanger (SUBHE) where it is cooled by the cold vapor stream flowing in the suction line of the auxiliary compressors. Although the sub-cooling effect has been found to be very small, the presence of this internal heat exchangers ensures that only vapor is present at the suction line of the auxiliary compressors. The sub-cooled liquid is throttled by the high-pressure valve (HPV) to the intermediate-pressure. Here, depending on the demand of the air conditioning system, the two-phase CO_2 can flow through the air conditioning heat exchanger (ACHE) or it can enter directly an internal heat exchanger (IHE) where is partially condensed by the cold vapour stream coming from the flash gas valve (FGV). Vapour and liquid are separated in the flash receiver (FR) connected to the outlet of the internal heat exchanger (IHE). The liquid is used to feed the MT and LT evaporators, while the vapours stream can be alternatively directed to the flash gas by-pass valve (through the internal heat exchanger IHE) or to the suction line of the auxiliary compressors (through the internal heat exchanger SUBHE). The superheated vapor exiting the LT evaporators (EVLT) is compressed by the LT compressor rack and directed to the suction line of the MT compressors where it mixes with the vapor coming from the evaporators working at medium pressure (EVLT and EXTEV). After being compressed to the high pressure, the hot gas discharged from the MT and auxiliary compressors passes through the two stages heat recovery system. In the first heat recovery stage (SHWHE), sanitary how water is heated from 60 °C to 65 °C, while the second stage of the heat recovery system (SHHE) is dedicated to space heating, providing the hydronic system with water at 55°C forward and 50°C return temperature. Two independent three-way valves are used to completely bypass these heat exchangers when no space heating nor hot water are requested. A gas-cooler (GC) installed on the roof of the supermarket is used to reject the remaining heat to the outdoor ambient. The LT compressors rack (CLT) consists of three compressors A1.5-3SK3 (Frascold, 2017a), while three compressors S20-12TK (Frascold, 2017b) compose the MT compressors rack (CMT). In both compressors racks, one compressor is equipped with a speed regulator to fill the capacity gaps between the on/off compressors. Four additional compressors are installed as parallel compression unit (CAUX): three of four compressors S20-12TK are operating in on/off mode while the last compressor is a S15-10TK (Frascold, 2017b) equipped with a speed regulator. The lubricant is a fully miscible POE oil. The lubricating oil is collected in a separator installed downstream of the discharge line of the MT and auxiliary compressors. An electronic regulator of the oil level is installed on each compressor and the oil is distributed from the separator to each compressor on demand.

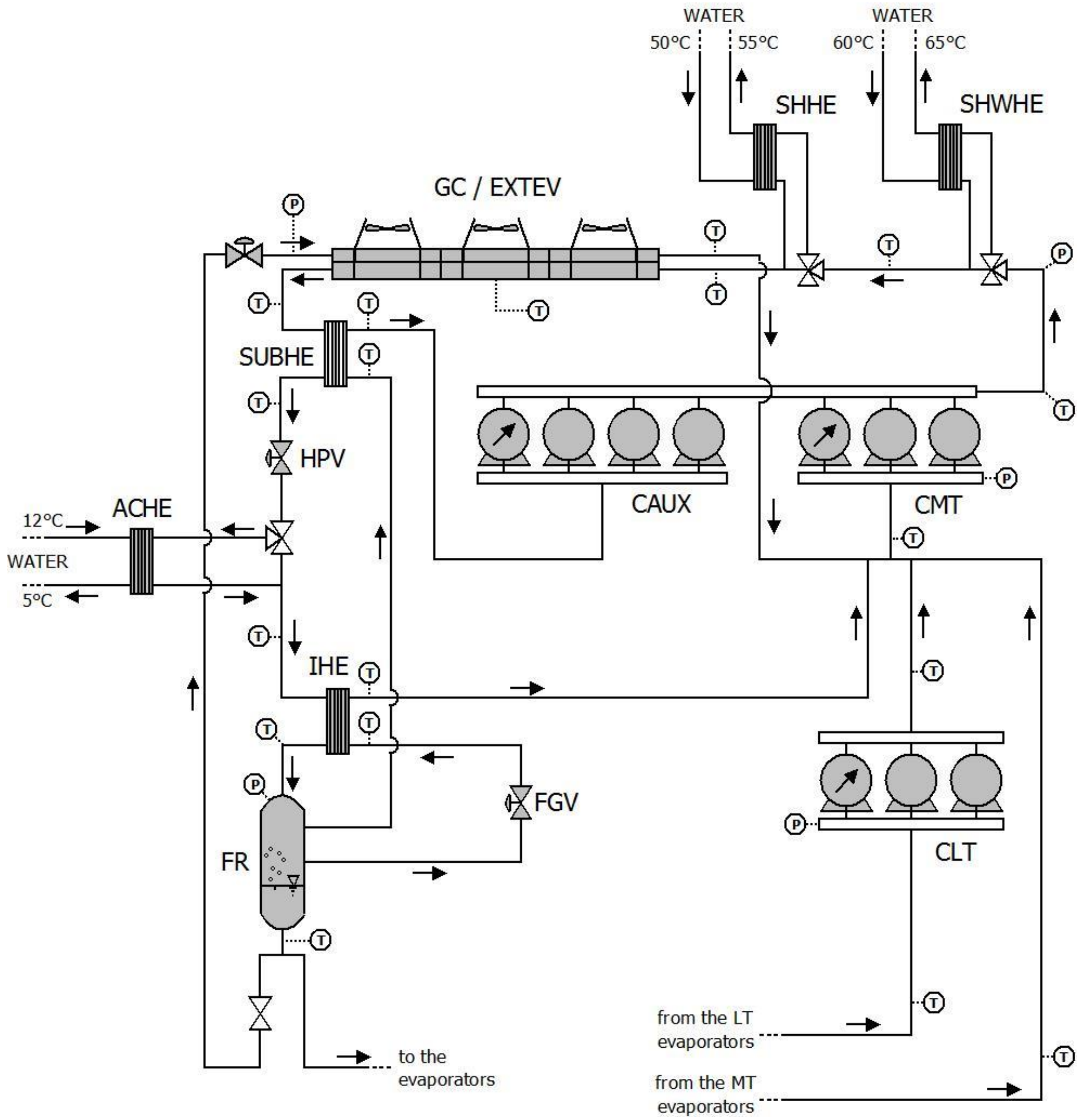


Figure 1: Schematic of the booster system with parallel compression, two stages heat recovery and air conditioning system.

2.1 System regulation

In this system, the outlet temperature from the gas cooler and the high pressure are regulated by a control algorithm for the optimization of the COP in summer transcritical mode. The minimum and maximum discharge pressures of the MT and auxiliary compressors set the limits to high pressure regulation. Although the use of a constant intermediate pressure affects negatively the performance of the system during transcritical operations, the pressure at the flash tank receiver is kept constant to allow the integration of the air conditioning system. The auxiliary compressors are activated only when the flash vapor flow exceeds the capacity of the flash gas valve for a certain time frame. That means enhanced reliability and less maintenance because of the reduced wear and tear on the compressors.

During the winter period, to increase the amount of heat available for heat recovery, the high pressure is increased over its critical value forcing the system to work in transcritical mode. When the heat absorbed by the cabinets (medium temperature evaporators) and freezers (low temperature evaporators) cannot satisfy the thermal demands of the heating system, an additional evaporator (EXTEV) is used to evaporate the excess of liquid. This is an air-cooled heat exchanger, equipped with a dedicated expansion valve, where the liquid CO_2 coming from the flash tank receiver evaporates at the same pressure as the MT cabinets. The superheated vapor exiting the EXTEV is sent to the suction line of the MT compressors. The additional evaporator EXTEV is placed in the same framework of the gas cooler and served by the same fans.

3. DATA COLLECTION AND EVALUATION OF NON-MEASURED PARAMETERS

Real time monitoring of the operating conditions in the considered refrigeration system is done remotely using the *boss* system developed by CAREL (2016). CAREL *boss* is a web-based supervision software which allows the centralized management through automatic data and alarm synchronization. The minimum variable sampling time is 5 s but at every time step data is recorded only if a value variation from the previous time step is detected. This results in a database with data stored at different time resolution (greater or equal to 5 s). The stored data are available in an online database and can be downloaded with different time interval.

The results from the monitoring of this unit are used to calibrate a steady-state model of the refrigeration CO_2 cycle. The model is developed in Matlab® and used to estimate the parameters which were not directly measured, such as mass flow rates and thermal loads.

3.1 Pressure and temperature measurements

As shown in Figure 1, pressure transducers are installed at each pressure level. The low and medium pressures are measured at the suction line of the LT and MT compressors, respectively. The intermediate pressure is measured in the upper side of the flash tank receiver. The high pressure is measured at the discharge line of the MT and auxiliary compressors and at the outlet of the gas cooler. An additional pressure transducer is installed right after the expansion valve of the external air-cooled MT evaporator.

The CO_2 temperatures are measured at the inlet and outlet of the main components in the systems using NTC sensors. The useful superheating at the MT and LT are obtained by averaging the outlet temperature of the MT cabinets and LT freezers. The external ambient temperature, which affects the regulation of the high pressure and the heat rejection to the external ambient, is measured on the air side inlet and outlet of the gas cooler. The outlet air from the gas cooler can be sent to the additional MT evaporator when it is on service during the winter period.

3.2 Mass flow rates and electrical power consumptions

The absence of mass flow meters requires an indirect method to evaluate the mass flow rates flowing in the refrigeration unit. The method adopted here is based on the power and mass flow rate polynomial expressions for the compressors series SK3 and TK given by the compressors manufacturer. In these expressions, the mass flow rate and the power absorbed by the compressor are given as a function of the evaporating temperature at the suction and discharge pressure, at the nominal frequency of 50 Hz and 10 K superheating. The coefficients of the polynomial expressions are obtained from laboratory tests according to the European Standard EN 12900 (2013) and verified through field-measurements. Corrections to account for different superheating and frequency are applied as specified by the manufacturer. Power consumption analyzers were installed in January 2018. Before this date the running capacities of the compressor units were acquired as percentage of their full capacities. By measuring the pressure and temperature of the CO_2 at the suction lines of the compressors, it is possible to evaluate the number of compressors in use and their running frequency. Once the running frequency and the suction conditions of the

compressors are known, it is possible to calculate the power absorbed and the mass flow rate elaborated by these compressors. The measurements of the discharge temperatures are then used to verify the estimated parameters.
Once the mass flow rates elaborated by the compressors are determined, the mass flow rates in the other components can be calculated through mass and energy balances.

3.3 Cooling and heating loads

The cooling loads of the systems are determined by estimating the CO_2 mass flow rate and the inlet and outlet enthalpies of the LT and MT lines. The enthalpies are calculated by measuring temperature and pressure at the outlet of flash tank receiver and at the suction line of the compressors. By measuring the temperature variation occurring in the pipelines from the evaporators inlet to the suction stage of the compressors, it is possible to determine the heat absorbed at the evaporators and the heat exchanged in these pipelines.
The water and CO_2 temperatures in the heat recovery system were not measured. Hence, the thermal load of the space heating and sanitary hot water production are evaluated by assuming 5 K approach point temperature difference at the CO_2 outlet in the counterflow heat exchangers. Since the actual water temperatures values at the inlet and outlet of these heat exchangers were not acquired, their nominal values are used instead.

4. RESULTS AND DISCUSSION

In order to understand how the considered transcritical booster system operates, the running conditions of the main components are evaluated. The monitoring activity has been conducted from August 2017 to January 2018. The monitored key parameters are: outdoor ambient temperature, pressure and outlet temperature of the gas cooler, pressure of the flash tank receiver, pressure and outlet temperature of the evaporators, pressure and temperature of the suction line of the compressors, power consumption and load fraction of the compressors. The acquired data are analyzed and filtered to eliminate meaningless data points. In order to calibrate a steady-state model of the system, which is necessary to estimate the non-measured parameters, the points acquired during non-stable conditions are not considered. The model is used to evaluate the performance of the unit under variable operating conditions.

4.1 Evaluation of the working conditions

Figure 2 reports the gas cooler, LT and MT evaporators and flash tank receiver pressures as function of the external air ambient temperature. The gas cooler pressure is the key parameter to understand when the system is operating in subcritical or transcritical mode. When no heat recovery is required the gas cooler pressure is regulated as a function of the external temperature. As expected, when the ambient temperature exceeds about 25°C the system works in transcritical conditions and the high pressure is optimized by a control algorithm in order to maximize the COP. For ambient temperature between 20°C and 5°C the pressure displays two trends: it varies between the saturation pressure corresponding to the gas cooler outlet temperature and a pressure level equal to 85 bar. In fact at low external temperature the heat recovery system can be enabled and the system is forced to work in transcritical conditions at around 85 bar. The wide variation of the pressure in this zone is due to the intermittent activation of the heat recovery system. When the external temperature is below 5°C the system works only in transcritical conditions since the heat recovery is always required. The regulation control operates in order to keep constant the pressures at the MT and LT evaporators and at the receiver. Their fluctuations are due to the variation of the boundary conditions for the system. The LT evaporators show pressure variation within 12% around 13 bar (average value), which corresponds to a saturation temperature around -33°C. The average pressure at the MT evaporators is 27 bar (corresponding saturation temperature around -9°C) with variations within 18%. The flash tank receiver works at an average pressure of 36 bar and its variation is around 12%.

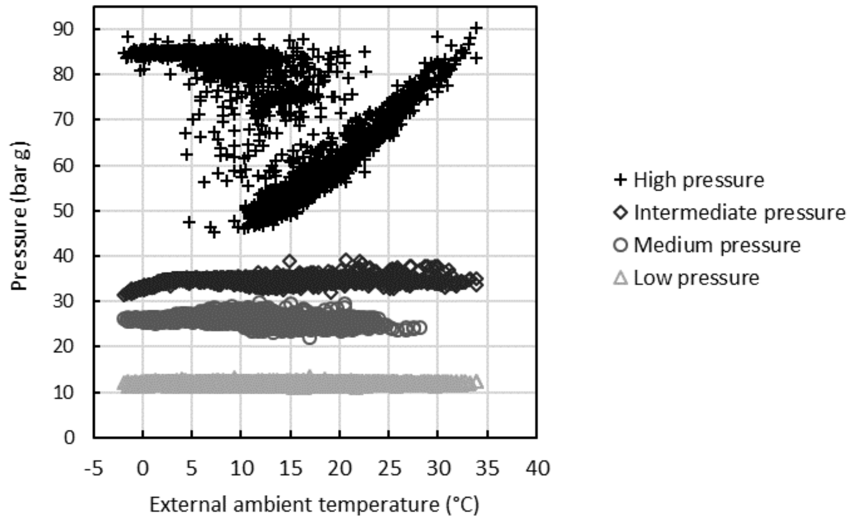


Figure 2: Monitored pressure levels as a function of the external ambient temperature. The points were acquired with a time step of one hour between 21st Aug. 2017 and 21st Dec. 2017.

The temperature approach point at the gas cooler is calculated as the temperature difference between the CO_2 outlet and the air inlet. The approach point is mainly affected by the rotational speed of the gas cooler fans. At high ambient temperatures, in order to reduce the vapor fraction entering the receiver, the outlet gas cooler temperature should be as low as possible. Hence, the gas cooler fans will work at their full speed. At low ambient temperature, the heating demand is generally high and the fans are still working at their full speed in order to evaporate through the additional evaporator (EXTEV) the excess of liquid present in the receiver. Figure 3 depicts the CO_2 gas cooler outlet temperature and the external ambient temperature.

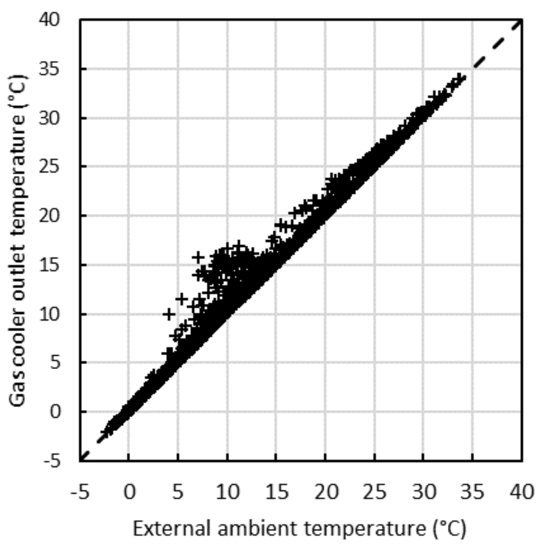


Figure 3: Monitored gas cooler outlet temperature as a function of the external ambient temperature. The points were acquired with a time step of one hour between 21st Aug. 2017 and 21st Dec. 2017.

For ambient temperatures lower than 5°C and higher than 25°C the measured gas cooler outlet temperatures are very close to the external temperatures and the average approach point temperature difference is less than 1 K. Between 5°C and 25°C, the gas cooler outlet temperature diverges from the external ambient temperature due to the continuous pressure variations caused by the intermittent activation of the heat recovery system. In subcritical conditions, the fans speed is regulated to obtain a constant subcooling in the liquid at the gas cooler outlet. Most of the data acquired between 21st Aug. 2017 and 21st Dec. 2017 report a subcooling between 1 K and 5 K for subcritical conditions.

Each evaporator is controlled by a dedicated expansion valve, which varies its aperture in order to maintain a constant superheating at the evaporator outlet. The average (named useful) superheating at the LT freezers results about 9 K, whereas for the MT cabinets it results about 12 K. These values are calculated as the average temperature differences between the monitored outlet and saturation temperatures for the LT and MT evaporators. In the pipelines connecting the evaporators outlet to the suction of the compressors, the CO_2 flow presents temperature variation due to the heat transfer with the surrounding ambient. These variations are usually named non-useful superheating. In the considered system, the non-useful superheating in the LT line results to be dependent from the external ambient temperature following the relationship reported in Equation (3):

$$\Delta T_{nu-sh,LT} = 13.5 + 0.24 \cdot t_{ext} \tag{3}$$

The non-useful superheating in the MT line results almost independent from the external ambient temperature, with values ranging between 2 K and -2 K. As for the MT cabinets and the LT freezer, a dedicated expansion valve is installed on the MT external evaporator. The superheating in this line follows Equation (4):

$$\Delta T_{sh,EXTEV} = 7.8 + 1.03 \cdot t_{ext} \tag{4}$$

It must to be mentioned that, while data for the LT line analysis are available from 21st August 2017 and 21st December 2017, the temperature readings of the sensors on the MT lines were reliable only for the month of December 2017, where the external temperature was limited between -1°C and 12°C.
Until January 2018, the running capacities of the LT, MT and auxiliary compressor units were acquired as percentage of their full capacities. **Figure 4** reports the measured percentage capacities of the LT compressor as a function of the inverter frequency installed on the speed regulated compressor. The capacities variation at a constant frequency depends on the number of compressors in use and the CO_2 conditions at their suction line. It can be clearly observed that the installed compressors do not allow a continuous modulation of the capacity. This is mainly due to the limited frequency regulation. In the lowest band only the speed-regulated compressor is working, while in the other two bands, one or two fixed-speed compressors are added. The acquired data are used to establish a relationship between percentage capacity and the working frequency. Similar considerations can be drawn for the MT compressors.

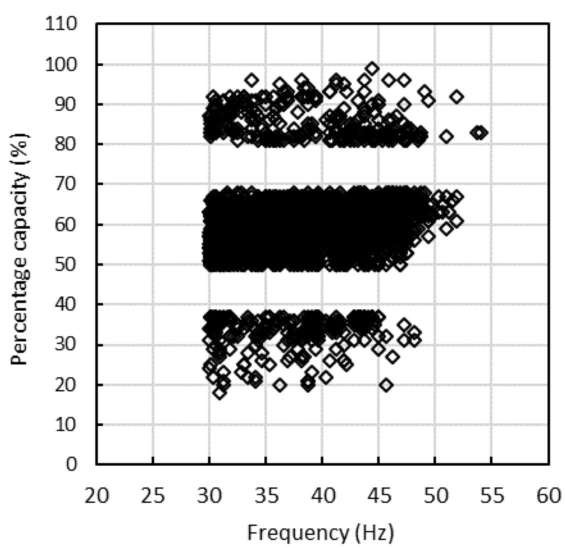


Figure 4: Monitored running capacities percentage of the LT compressors as a function of the inverter frequency. The points were acquired with a time step of one minute in Jan. 2018.

4.2 Performance prediction

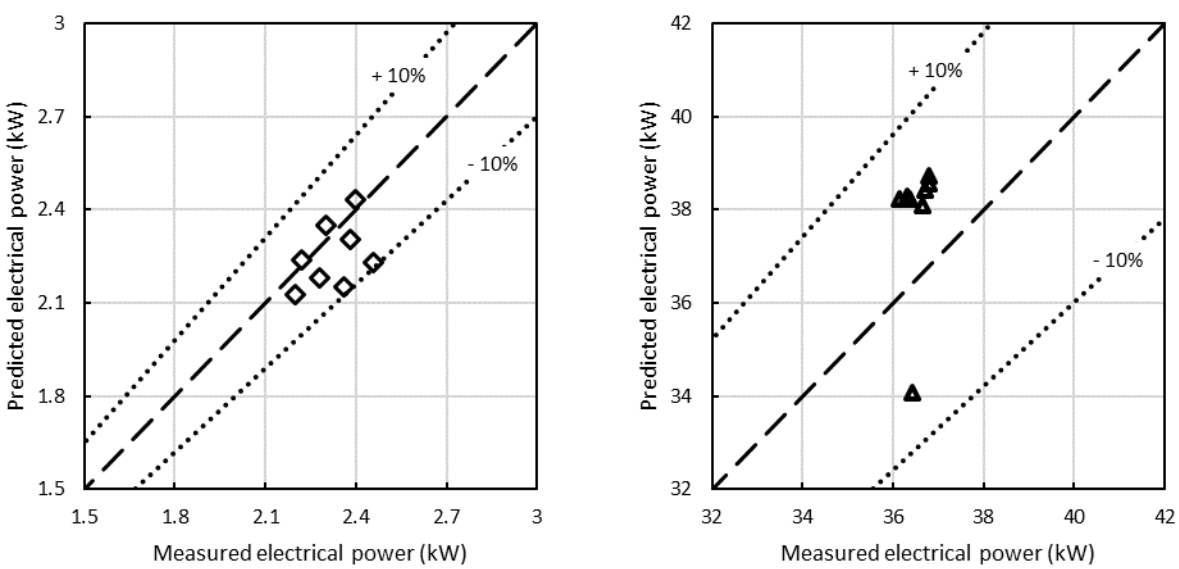
A steady-state model thermodynamic of the refrigeration cycle is developed in Matlab® in order to evaluate the parameters which were not directly measured. Under the assumptions presented in Section 3, the model is capable to determine thermal loads, mass flow rates and compressors' power, as well as the inlet and outlet conditions for the main components in the system. To this purpose, the inputs required by the model are: working pressures, useful and non-useful superheating, gas cooler approach point temperature difference, percentages of compressor capacities and external ambient temperature. A first set of monitoring data are used to calibrate the model. The calibrated model is then used to simulate the system operations under the boundary conditions of a second set of data. This dataset is independent from the one used in the calibration and is composed by 5 minutes averaged points taken at the end of January 2018. The data are filtered to select only the points representing stable operation. Table 1 reports the main results of these simulations.

Table 1: Predicted compressors power, thermal load and coefficients of performance.

Date (dd/mm/yy h:m)	External ambient temperature (°C)	LT compressors power (kW)	MT compressors power (kW)	LT freezers (kW)	MT cabinets (kW)	MT external evaporator (kW)	Sanitary hot water* (kW)	Space heating* (kW)
29/01/18 19.25	8.0	2.3	38.4	11	48	54	11	37
29/01/18 19.35	8.5	2.1	38.7	10	32	71	11	36
29/01/18 19.40	8.4	2.2	38.6	10	38	64	11	37
31/01/18 08:50	4.5	2.2	38.2	10	78	28	10	37
31/01/18 08:55	4.4	2.3	38.3	11	67	39	10	37
31/01/18 09:05	4.4	2.2	38.2	10	86	20	10	37
31/01/18 10:00	4.5	2.4	38.1	11	55	50	10	37
31/01/18 10:05	4.6	2.2	34.1	10	44	49	9	33

* Prediction based on the assumption that the heat recovery system works at its full capacity.

The accuracy of the model is then tested by comparing the predicted parameters with the measured data for example the power absorbed by the compressors. In **Figure 5** the predicted electrical power of the LT and MT compressors is plotted against their measured values. The model is able to predict the electrical power within 10%.



(a) (b)

Figure 5: Predicted electrical power against measured electrical power for the LT (a) and the MT (b) compressor units. The monitored data are acquired in January 2018.

The predicted values of the gas cooler outlet temperature and the MT compressors discharge temperature are compared in Table 2. In the reported cases, the agreement in the prediction of the MT compressors discharge temperature results within 5 K. The model tends to overestimate the gas cooler outlet temperature: the predicted values differs by less than 1 K from the measured data.

Table 2: Predicted and measured values at the gas cooler and MT compressors discharge.

Date (dd/mm/yy h.m)	Outlet gas cooler temperature		MT compressors discharge temperature	
	Predicted value (°C)	Measured value (°C)	Predicted value (°C)	Measured value (°C)
29/01/18 19.25	8.5	7.7	115.6	113.8
29/01/18 19.35	9.0	8.5	114.3	115.5
29/01/18 19.40	8.9	8.4	115.1	114.8
31/01/18 08:50	5.0	4.7	116.3	111.9
31/01/18 08:55	4.9	4.7	116.1	111.5
31/01/18 09:05	4.9	4.7	116.3	112.3
31/01/18 10:00	5.0	4.8	116.7	112.9
31/01/18 10:05	5.1	4.9	116.8	112.4

5. CONCLUSIONS

The preliminary results from the monitoring of a CO_2 transcritical booster system with parallel compression, heat recovery and air conditioning integration have been presented. The monitored key parameters were: outdoor ambient temperature, pressure and outlet temperature of the gas cooler, pressure of the flash tank receiver, pressure and outlet temperature of the evaporators, pressure and temperature of the suction line of the compressors, power consumption and load fraction of the compressors. The monitoring data were useful to understand how the system operates.
A steady-state model of the refrigeration cycle was developed to evaluate the parameters which were not directly measured. The acquired data has been used to calibrate this model. The calibrated model has then been used to simulate the system operation under variable condition in a cold month. The model shows a good agreement with the measured data. This model will be a useful tool to suggest modification in the working conditions of the monitored system aimed to improve its performance.
Future works will address the extension of the monitoring to the summer period, in order to investigate the system operation and performance while refrigeration and air conditioning are provided simultaneously.

NOMENCLATURE

$\Delta T_{nu-sh,LT}$	non-useful superheating in the low temperature line	(K)
$\Delta T_{sh,EXTEV}$	superheating at the medium temperature external evaporator	(K)
ACHE	air conditioning heat exchanger	
CAUX	auxiliary compressor rack	
CLT	low temperature compressor rack	
CMT	medium temperature compressor rack	
EVLT	low temperature evaporators	
EVMT	medium temperature evaporators	
EXTEV	medium temperature external evaporator	
FGV	flash gas bypass valve	
FR	flash tank receiver	
GC	gas cooler	
HFC	hydrofluorocarbon fluids	
HPV	high-pressure valve	
IHE	internal heat exchanger	
LT	low temperature	
MT	medium temperature	
SHHE	space heating heat exchanger	
SHWHE	sanitary hot water heating heat exchanger	
SUBHE	internal heat exchanger (subcooler)	
t_{ext}	external ambient temperature	(°C)

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