A NEW APPROACH TO PERFORMANCE DATA HARVESTING AND PROCESSING IN COMMERCIAL REFRIGERATION SYSTEMS

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ABSTRACT

In the last years, new lay-outs for commercial refrigeration systems, mainly based on CO_2, have been progressively replacing HFC ones. Their performances are normally calculated with thermodynamic evaluations and therefore fair comparison amongst different systems is difficult. The performance of the refrigeration unit could in principle undergo a certification process similar to those adopted for other appliances; however technology is rapidly evolving, adapting to different climatic conditions and specific customers' requirements, such as integration with AC and heating system, and therefore systematic laboratory testing would be a challenge.

In addition, plant design and its management heavily affect the overall efficiency. Therefore, the evaluation of the entire system efficiency requires collection of data from the field. Despite commercial refrigeration systems are fully equipped with monitoring devices, providing enormous amount of data, easily accessible information about performance and energy efficiency are often lacking and not available, though in an anonymous way, for database built up. Together, there is increasing necessity for identifying key parameters that can help in forecasting performance degradation of the entire system or of single components, to prevent costly failures.

A new approach to data collection and processing is presented and its potentialities illustrated by preliminary field data.

Keywords: remote monitoring, data harvesting, commercial refrigeration, CO_2

1. INTRODUCTION

Since 1992, European Commission has been constantly working towards environment safety, with special emphasis on energy efficiency, providing consumers with information that allows them to choose more efficient products while ensuring the free movement of energy-related products in the European Union.

As recently reported by the EU Commission to the European Parliament and the Council (COM(2015) 345), these objectives remain as relevant as they were more than 20 years ago.

As far as standard HFCs based solutions are declining, mainly because of the F-gas Regulation (EU No 517/2014), manufacturers are proposing different alternatives, mostly based on natural fluids. According to the Commission Report C(2017) 5230, as far as no synthetics with GWP<150 are available for direct expansion into the shopping area, from 2022 CO_2 is actually the only alternative to avoid cascade systems and/or indirect fluids for serving both MT and LT.

Carbon dioxide is therefore proposed as the long-term option for commercial refrigeration. Systems working with CO_2 look and behave differently than traditional ones. Gaining confidence in these solutions is crucial for a fast transition towards environmental friendly long term options.

At the same time, extensive field measurements are necessary to ensure the market about new systems reliability and energy performance.

Local control and monitoring systems are largely used. All major suppliers provide integrated control systems that support the interconnected management of HVAC&R, air quality, lighting, etc.

The main purpose of remote monitoring is typically to control system operations. In addition alarm conditions are detected, possibly at an early stage. Once that an alarm has occurred, experts can analyze monitored parameters to understand the impact of the failure on either the system operability or the food safety and quality; readings can also be used to go in deep the causes of the failure and to adopt countermeasures for the future. This is, however, a challenging approach, which requires detailed knowledge of the plant and high professionalism to interpret signals coming from the field and mostly a lot of time is needed. Additional

measuring devices and a deeper analysis compared to state of the art might also be required for full understanding of failures.
Processing such a huge amount of data requires however an automatic analysis.
Together with operations and failures monitoring, evaluation of energy performances is gaining relevance. While energy consumption is often recorded, evaluation of energy performance, typically described by first thermodynamic low application with Coefficient of Performance (COP) or Energy Efficiency Ratio (EER), requires the knowledge of the useful effect of the refrigeration system, i.e. cooling capacity and, if the case, heating capacity from the heat recovery system. The evaluation of cooling capacity basically entails knowing the refrigerant mass flow through cabinets and cold rooms and their inlet and outlet conditions.
The knowledge of energy performance of refrigeration in the system is the first requirement for comparing different solutions and help end users in performing the right choice for their sites.
This paper will present the general architecture of the new data acquisition system and highlight parameters that are preliminary investigated to focus on system performance and reliability status.
The operational purposes of the proposed approach are:
-the measurement of system energy efficiency, based on data acquired on the field which are then centrally processed;
-the analysis of the degradation of operational parameters measured in the field, not necessarily related to the unit control, to forecast and prevent future-be failures.
The data are to be available to systems manufacturers, for an immediate solution of potential problems and a rapid improvement of products.
The adoption of not-in-kind measuring instruments in the HVAC&R business will be promoted to use as far as possible measured values instead of calculation. .
The proposed approach also supports the building up of a database, which contains data about different layouts, in different places and with different load profiles. These data, likely in anonymous way, might be available for comparison and energy evaluation for investment planning. Availability of data at governmental level might also support energy and environmental policies.

2. DATA ACQUISITION SYSTEM ARCHITECTURE

The proposed system architecture is presented in Fig. 1.
Signals coming from different sensors located in the refrigeration unit are converted to modBUS protocol and then transmitted to a data logger, which is a single board, low cost computer. Input/Output modules are used to convert different signals coming from transducers (for instance analogic 4-20 mA or 0-10 V signal) or from refrigeration unit electrical board (on/off signals and inverter output) into a modBUS protocol and then information is delivered by a RS485 cable to a modBUS RTU/TCP converter. An Ethernet cable connects this converter to the computer. The computer is equipped with a USB/3G stick that allows to store all the data in a cloud server.
By means of an open source database it is then possible to manage data and download them from the server. The time step of signal acquisition is selectable; while 1 minute seems to be a reasonable interval, it is possible to intensify the sampling if any specific phenomenon dares detailed investigation.
By monitoring and processing the data, early detection of potential failures is feasible, together with continuous performance monitor.
Data can be downloaded in table format so that calculation is no more located to the unit controller, but is remotely performed. In such a way, calculation tools, system lay-outs and refrigerant properties can be easily managed and updated if required and outputs are fully transparent as source measure data are available for any further verification.
CO_2 technology is rapidly evolving and plants can look very different one from each other, especially if integration with air conditioning and heating system is implemented. When dealing with COP, it is then necessary to correctly implement the system layout to evaluate useful effects and attribute them the right measured power input.

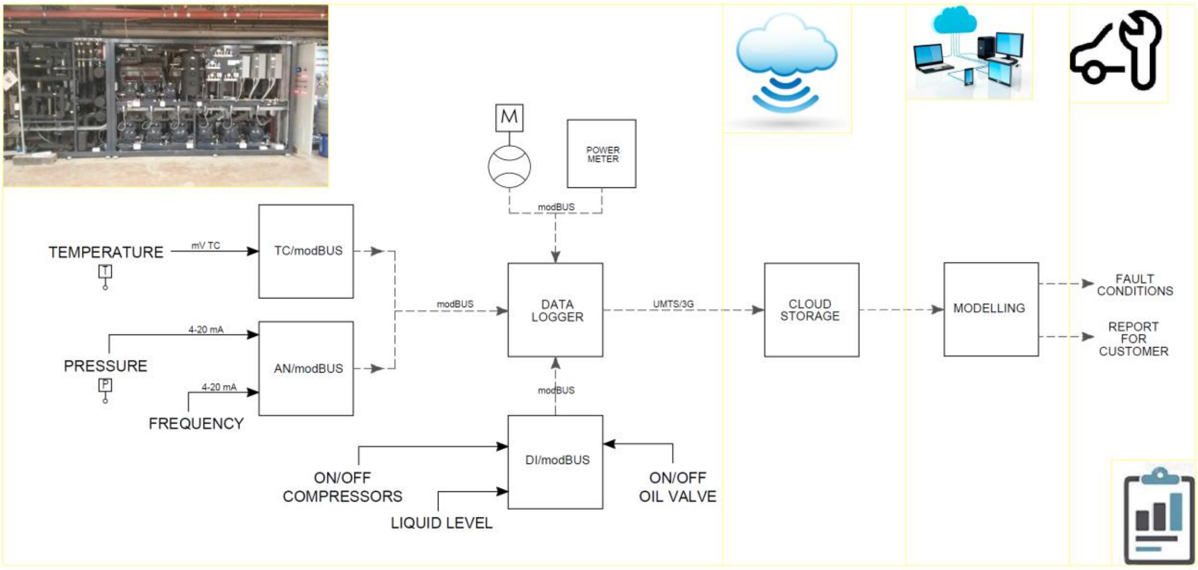

Figure 1. Architecture of the system

2.1 Sensors

The new data acquisition approach intends to measure operational parameters of a refrigeration or heat pump system to identify key parameters that can give information on the performance and health status of the entire circuit or of specific components.

There are parameters, which are normally acquired with traditional DAQ systems, such as temperature, pressure, status of some components, i.e. compressors or valves on/off. On the other hand, analogue signals, such as frequency output of inverters for compressors or fans, as well as modulating valves position are available from the unit controller or the electrical board; they can be therefore utilized also as read variables.

Energy or power meters are gaining popularity, as annual energy monitoring is typically registered for technical and commercial purposes. They are naturally providing gross values including many components.

The proposed system intends to read signals from the electrical board, while autonomous sensors are also installed whenever necessary, so that there is no need for any communication or signals transfer with the unit controller, which can be of whatever type and brand. As far as autonomous sensors are concerned, individual measurement of power input to each compressor is crucial to forecast failures.

To this extent, the identification of affordable cost-effective sensors is crucial.

Coherently, in present state-of–the–art refrigeration systems or heat pumps, the refrigerant mass flow rate is not measured as it would require invasive measurements with extremely high cost meters. Refrigerant mass flow rate is however a key parameter able to provide precious information on both system performance and compressors status. With the increasing complexity of commercial refrigeration layouts, adopting parallel compression, liquid and vapour ejectors and internal heat exchangers, the mass flow rate estimated using compressors performance data might not be sufficient to assess the cooling load. In addition, the calculated mass flow rate is unaffected by possible compressors performance deterioration and therefore can’t be used for forecasting any anomalous behavior.

One of the purposes of the project is to identify direct low-cost methods to measure mass flow rate at crucial branches of the circuit, such as liquid line to MT and LT cabinets, to AC or heat recovery, when applicable, or simply evaporator and gas cooler in the case of heat pumps.

3. FIELD TEST

A plant adopting the proposed DAQ system was installed in autumn 2017 in Italy. It is based on a booster layout with parallel compression and expansion work recovery by ejectors for both vapor pre-compression and liquid recirculation. The nominal cooling capacity at 35° C outdoor temperature is 72 kW (@ -8°C evaporation temperature) for MT, 21 kW for LT (@ -32°C evaporation) and 120 kW for AC.

Air condition and heat recovery are integrated in the system.

The layout of the refrigeration system with all the installed sensors is presented in Fig. 2. Temperature is currently measured using seven commercial type NTC sensors, identified in Fig. 2 with T, while pressure is measured with four commercial type piezoresistive pressure transmitters (p in Fig. 2).
In order to evaluate the total electric power input, three-phase electric energy meters are installed. Each energy meter is placed before each compressors rack to measure the power input to low temperature (LT), Medium temperature (MT), and parallel (AUX) compressors. The total power input to the system, including also auxiliaries (fans, pumps,valve motors, etc) is monitored.
Each compressor has its own oil valve, which opens whenever the oil level sensor in the crankcase is activated by low level signal; the oil valves on/off signal is collected. The inverter frequency is also acquired, being one compressor for each rack inverter driven.
Eventually, the level of the liquid in the low pressure liquid separator is monitored by means of on/off digital signal sensors.
The installation of refrigerant mass flow meters in the liquid line to LT and MT compressors is planned as soon as suitable components are identified, together with water mass flow meters and temperature sensors in the AC and heating loops.

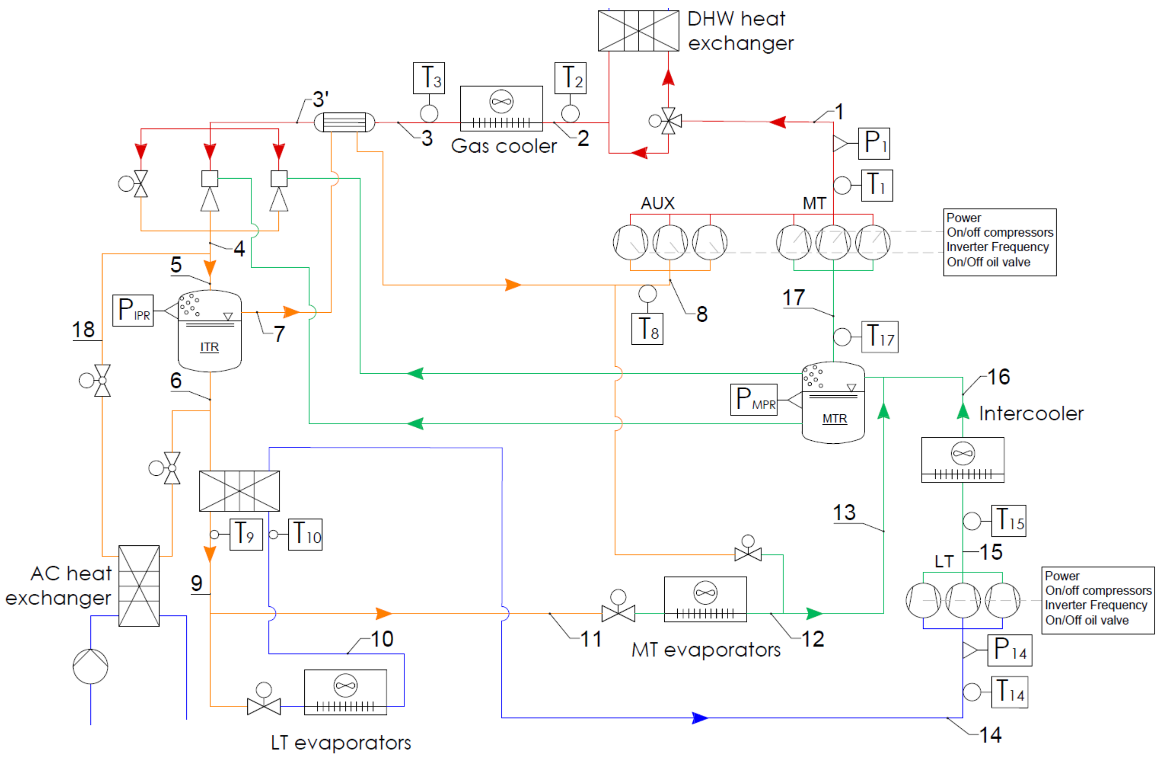

Figure 2: layout of the field test plant and sensor location.

The system was commissioned in the late autumn and, being a refurbishment, it is still not under full operations; for instance, evaporators are still running in the dry-expansion mode. In addition, the cold season results in subcritical operations, mainly at the lowest admitted condensation temperature, i.e. 10°C. For this specific situation, preliminary data are presented. They currently intend to show the idea behind and the outcome of the proposed approach in data harvesting and processing from commercial refrigeration units.

3. ENERGY PERFORMANCE

Refrigeration systems energy consumption is typically monitored and registered. It depends on both cooling demand and how efficiently cooling load is provided. It is extremely difficult to categorize cooling demand as it depends on the number and type of cabinets and cold rooms installed, on the indoor and outdoor conditions, the shopping habits, customer density, opening schedule etc. In addition, external temperature affects the system energy efficiency. As a result, the comparison of energy consumptions of different stores is a hard task. However, manufacturers and contractors are more and more interested in proving the performance of their plants, as it becomes crucial information for the customer to compare different alternative layouts and to drive future investments.

A huge effort has been taken in the years to develop figures able to normalize energy consumption in commercial refrigeration. For example, IEA annex 44 (Van der Sluis et al., 2015) has proposed figures considering energy intensity for heating and cooling in the food retail stores with respect to store area, collecting a huge amount of data in Europe and America.

The evaluation of the cooling load is critical to correctly weight the effect of outdoor conditions on the energy consumption. The measurement of refrigerant mass flow rate through cabinets and cold rooms is the straight forward way to cooling load. However, it is not yet common practise to install flow meters in the refrigerant lines. Actually, in simple independent MT and LT vapour compression cycles, the estimation of mass flow rates with compressor performance data provided by manufacturers offers the possibility of estimating mass flow rate through cabinets in a satisfactory way. The advent of carbon dioxide and the development of booster systems has slightly complicated the matter; however, as far as no ejectors and parallel compressors are involved, the evaluation of cooling load via compressor performance data is still feasible in a simple way, as demonstrated by some commercially available data monitoring systems. However, state-of-the art CO_2 systems propose ejector supported parallel compression and overfed evaporators, together with integration with air conditioning and heating; in addition layouts can vary according to specific requirements of the customers or system designer experience. The system presented in Fig. 2 is an example of what is currently proposed to the market. As a consequence, the evaluation of cooling load, and the heating one, if the case, is no more possible by simply using compressor performance data; the knowledge of the state of the refrigerant in many points of the circuit is needed and some simplifying assumptions eventually needed. In any case, a detailed model of the system is required.

The model developed for the field test system of Fig. 2 is described in the following.

Mass flow rate through MT, LT and AUX compressors can be calculated using compressor manufacturer's data, at measured boundary conditions. In the case of LT compressors, the mass flow rate through compressors is exactly equal to $\dot{m}_{e,LT}$, i.e. mass flow through evaporators, thus giving direct access to LT cooling power.

Under the assumption of saturated or superheated vapour at the MT evaporator exit, no mass flow is to be considered in the liquid ejector line. The unknown quantities are then five, i.e. $\dot{m}_{e,LT}$, $\dot{m}_{ej}, \dot{m}_{bp}, \dot{m}_{e,AC}$ and fluid status at AC evaporator exit, identified by h_{18}, that can be determined with five equations.

Namely, with reference to fluid status numbered as in Fig.2, the system equations are:

$$\dot{m}_{bp} + \dot{m}_{e,MT} + \dot{m}_{c,LT} = \dot{m}_{c,MT} + \dot{m}_{ej}$$

$$\dot{m}_{bp} h_8 + \dot{m}_{e,MT} h_{12} + \dot{m}_{c,LT} h_{16} = \dot{m}_{c,MT} h_{17} + \dot{m}_{ej} h_{vap,MTR}$$

$$\dot{m}_{c,MT} + \dot{m}_{c,Aux} + \dot{m}_{ej} = \dot{m}_{e,MT} + \dot{m}_{e,LT}$$

$$\left(\dot{m}_{c,MT} + \dot{m}_{c,Aux} + \dot{m}_{ej}\right) h_4 + \dot{m}_{e,AC} h_{18} = \left(\dot{m}_{c,Aux} + \dot{m}_{bp}\right) h_8 + \left(\dot{m}_{e,MT} + \dot{m}_{e,LT} + \dot{m}_{e,Aux}\right) h_6$$

$$\dot{Q}_{e,AC} = \dot{m}_{e,AC}\left(h_{18} - h_6\right)$$

They correspond to energy and mass balance at MTR and energy and mass balance at IPR and energy balance at AC heat exchanger, once that power meter is installed on water side to measure the cooling power at AC evaporator $\dot{Q}_{e,AC}$.

Not to introduce any ejector modelling in the calculation, to the purpose of this analysis it seems reasonable to assume that expansion through ejectors is isenthalpic, i.e. $h_{3'} = h_4$.

The global COP is then defined as $COP = \frac{\dot{m}_{e,MT}(h_{12}-h_9)+\dot{m}_{e,LT}(h_{10}-h_9)+\dot{Q}_{e,AC}+\dot{Q}_{DHW}}{P_{el,MT}+P_{el,Aux}+P_{el,LT}}$, where the heating power delivered and DHW system $\dot{Q}_{DHW}$ is measured with standard power meters on water side.

The above equations result to be very much simplified during the present operations, being no ejectors or auxiliary compressors in operation or AC demand because of the winter time, and no heating required so far because refurbishment is still undergoing.

It is also relevant to calculate individual COP$_s$, which require to distribute MT and AUX compressors power input to each level of temperature at which refrigeration or AC is provided. In fact, both MT and AUX compressors serve all pressure levels.

$$COP_{LT} = \frac{\dot{m}_{e,LT}(h_{10} - h_9)}{P_{el,LT} + P_{el,MT}\frac{\left(\dot{m}_{c,LT} + \dot{m}_{bp}\frac{\dot{m}_{e,LT}}{\dot{m}_{e,LT} + \dot{m}_{e,MT}}\right)}{\dot{m}_{c,MT}} + P_{el,Aux}\frac{\dot{m}_{e,LT}(h_{10} - h_9)}{\dot{m}_{e,LT}(h_{10} - h_9) + \dot{m}_{e,MT}(h_{12} - h_9) + \dot{Q}_{e,AC}}}$$

$$COP_{MT} = \frac{\dot{m}_{e,MT}(h_{12} - h_9)}{P_{el,MT}\frac{\left(\dot{m}_{e,MT} + \dot{m}_{bp}\frac{\dot{m}_{e,MT}}{\dot{m}_{e,LT} + \dot{m}_{e,MT}}\right)}{\dot{m}_{c,MT}} + P_{el,Aux}\frac{\dot{m}_{e,MT}(h_{12} - h_9)}{\dot{m}_{e,LT}(h_{10} - h_9) + \dot{m}_{e,MT}(h_{12} - h_9) + \dot{Q}_{e,AC}}}$$

$$COP_{Aux} = \frac{\dot{Q}_{e,AC}}{P_{el,Aux}\frac{\dot{Q}_{e,AC}}{\dot{m}_{e,LT}(h_{10}-h_9)+\dot{m}_{e,MT}(h_{12}-h_9)+\dot{Q}_{e,AC}}}$$

However, by the time ejectors and parallel compressors will come to work, mass flow maters are planned to be installed in the LT and MT refrigerant liquid lines, thus avoiding any simplification due to modelling or measurements in fluid conditions in not steady-state operations.
On the other hand, the application of a standardised methodology to measure the seasonal performance of commercial refrigeration centralised units might offer an opportunity to label refrigeration units, but a huge effort is needed to meet the requirements of new technologies offered by the market (Minetto et al, 2017). In any case, units labelling can't take care of any inefficiency or improvement caused by the plant lay-out and its management, as it compares refrigeration units at the same boundary conditions.
In Fig. 3a evaluated COP as function of pressure ratio is shown for LT and MT system of Fig. 2.
Fig. 3b shows the total COP, i.e. the sum of MT and LT cooling loads, calculated as above described, divided by the measured total power input to MT and LT compressors, as a function of condensation temperature.

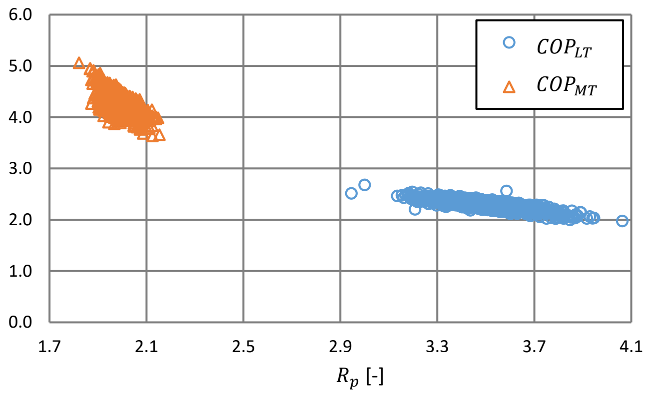

Fig 3a: COP for MT and LT of system in Fig.2 (December)

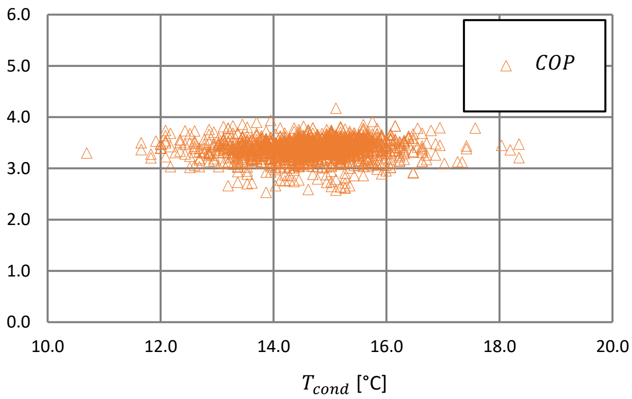

Fig.3b. Total COP system in Fig.2 (December)

When COP will be available for one year long, it will be fully illustrate the efficiency of the system of Fig. 2. MT and LT COP values will be then available to compare this layout to other options at whatever cooling load and outdoor temperature profile, if the evaporation temperature is the same. Of course, the COP calculation must be transparently shown and detailed to independent check.

4. IDENTIFICATION OF ANOMALOUS WORKING CONDITIONS

Early identification of anomalous working conditions is crucial for preventing costly faults. Commercial monitoring systems are typically provided with alarms, which occur when well-defined limits are overcome, and are normally intended to promote timely service.
While well-established alarms are still demanded to the unit controller, the new system intends to analyze some operational parameters in order to precociously identify deviations from standard operations that would either lead to future alarms or simply affect the refrigeration system performance.
The major effort is dedicated to analyze compressors. However, the analysis of water flow in heat recovery heat exchangers is considered to be worth investigation to detect scaling and prevent system inefficiency or failure.

4.1 Compressors operations

The deterioration of compression and volumetric efficiency impacts on the system energy efficiency and, more seriously, it might represent an early warning that a problem is occurring.
We are recalling the general definition of volumetric efficiency and overall compression efficiency for piston semi-hermetic compressors, as used in CO_2 applications:

$$\eta_v = \frac{\dot{m}}{\rho_s \dot{V}}$$

$$\eta_c = \frac{\dot{m}}{P_{el}}\Delta h = \frac{\eta_v \rho_s \dot{V} \Delta h_{is}}{P_{el}}$$

The definition can be applied to each compressor individually but it can also describe the entire compressor racks, i.e. medium temperature, low temperature and auxiliary compressors, as if they were variable displacement compressors. Both volumetric and compression efficiencies depend on pressure ratio and absolute suction conditions.
Any deviation in volumetric efficiency affects the processed mass flow through the compressor rack. Therefore the measurement of the actual mass flow rate and its comparison to the expected values, that can be derived from compressors' manufacturer data, might give an index of the volumetric efficiency deterioration.
At the same time, when measuring the electrical power input, any drop in compression efficiency can be evaluated dividing the electrical power input by the actual mass flow rate.
The parameter that has been identified to globally detect anomalous compressor operations is $\frac{P_{el}}{\dot{V}}$, where P_{el} is the electrical power input to each rack of compressors and $\dot{V} = \sum \dot{V}_i \frac{f_i}{50}$ is the actual swept volume, depending on the swept volume per unit time of each compressor in operation and its individual frequency (f), if inverter driven. The required measured parameters are P_{el} , compressor status (on/off) and inverter frequency.
$\frac{P_{el}}{\dot{V}}$ depends on operating conditions, i.e. pressure ratio and suction conditions. Therefore another index has been defined, in order to account for actual operating conditions, $\frac{P_{el}}{\dot{V}}$ has been referred to its nominal values and the identity $\left(\frac{P_{el}}{\dot{V}}\right)_T = \frac{P_{el}}{\dot{V}} \frac{\eta_c}{\rho \eta_v \Delta h_{is}}_T$ is to be verified, with ρ and Δh_{is} calculated at suction conditions, volumetric and isentropic efficiency evaluated according to manufacturer's catalogue data at real working conditions. The deviation from identity tells us how far we are from expected values. As calculation is performed in a server, there is no practical limit to accessibility and manageability of refrigerant properties.
Figures 4a and 4b represent the above described indexes for MT and LT compressor racks as functions of pressure ratio; the plant of Fig.2 was only run in December, the pressure ratio R_p is quite limited and parallel compressors were not operating at all. For LT compressors, R_p will be maintained throughout the year, due to the system design (booster).
Any deviation of $\left(\frac{P_{el}}{\dot{V}}\right)_T$ from the unit is representative of imperfect matching between theoretical and real compressor performance, which might depend either on measurement or calculation accuracy either on appropriateness of nominal data. For example, in the calculation of $\left(\frac{P_{el}}{\dot{V}}\right)_T$ as in Figures 4a and 4b , the impact of the inverter in the compression and volumetric efficiency was not accounted for. In addition, if single compressor power measurement is implemented, individual compressor analysis might help in understanding if a specific deviation is imputable to any of them.
The identification of anomalous operations can be highlighted by long term drift from initial value of $\left(\frac{P_{el}}{\dot{V}}\right)_T$.
In Figures 4a and 4b, the actual status of compressors partialisation is represented by the parameter $(\frac{V}{V_n}) = \frac{\sum \dot{V}_i \frac{f_i}{50}}{(\sum \dot{V}_i)_n}$, which tells us which is the actual swept volume with reference to the nominal one.
It is therefore clear that a reliable, cost-effective and practically implementable in the field method for measuring refrigerant mass flow rate in operations would add in useful information.
The amount of lubricant oil that is carried over by compressors is a key parameter to evaluate the system status. Discrepancies in behaviour or anomalous operations with respect to lubricant carryover can be analysed by evaluating each compressor requirement of oil refill from the reservoir with respect to its own on-time. This evaluation can be simply performed thanks to two digital signals, i.e. compressors status and oil return valves status.
Vibrational frequency analysis is a well established methodology to analyse compressors behaviour and forecast costly failures, however it is not yet applied to HVAC&R systems. In the next future the possibility of installing accelerometers will be analysed in the project

5. CONCLUSIONS AND FUTURE DEVELOPMENTS

Data collected from the field can be utilised to monitor refrigeration systems performance and to identify key indexes, which can help in early detection of anomalous operations. In complex lay-outs, like CO_2 ejector supported parallel compression systems, which are spreading to South Europe, often integrating AC and heating, system modelling is required to evaluate COP. One of the purposes of the project is to identify direct low-cost methods to measure mass flow rate at crucial branches of the circuit, such as liquid line to MT and LT cabinets, to AC or heat recovery, when applicable, or simply evaporator and gas cooler in the case of heat pumps. In such a way, some assumptions can be skipped for the sake of accuracy.

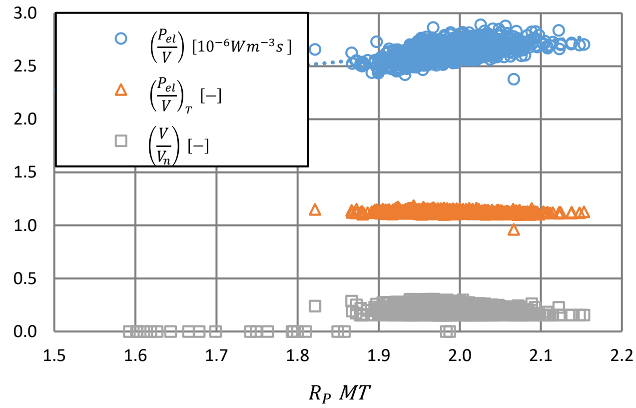

Figure 4a: MT compressor rack operations

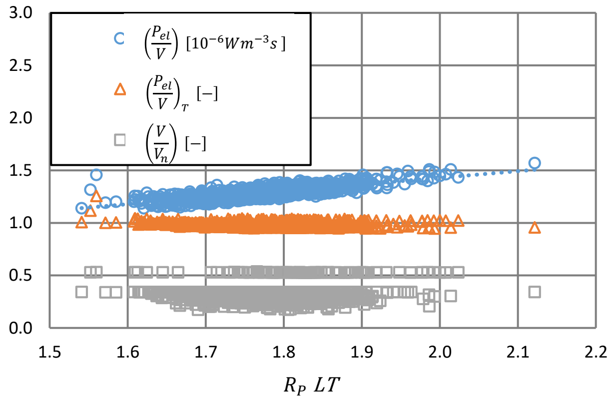

Figure 4b: LT compressor rack operations

Performance indicators for compressors can easily describe their behaviour and highlight drifts that can be symptomatic of future-be failures.

Vibrational frequency analysis is a well established methodology to analyse compressors behaviour and forecast costly failures, however it is not yet applied to HVAC&R systems. In the next future the possibility of installing accelerometers will be analysed in the project

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NOMENCLATURE

	f	frequency	P_{el}	electrical power (kW)
	h	enthalpy (kJ kg^{-1})	$\dot{Q}$	thermal power (kW)
	$\dot{m}$	mass flow rate (·kg·s–1)	R	ratio (-)
	p	pressure (10^5 Pa)	T	temperature (°C)
			V	swept volume per unit time (·m^3·s^{-1})
Greeks	η	efficiency (-)	ρ	density (kg m^{-3})
Subscripts and Acronyms	AC	air conditioning	IPR	intermediate pressure receiver
	Aux	parallel compressors	LT	low temperature
	bp	by pass valve	MT	medium temperature
	c	compression	MTR	medium temperature receiver
	cond	condensation	n	nominal
	DHW	domestic hot water	o	out
	e	evaporator	p	pressure
	ej	ejector	s	suction
	is	isentropic	v	volumetric
			T	target

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