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Not-in-kind approach to remote monitoring in CO_2 refrigeration systems

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ABSTRACT

The complexity of CO_2 layouts for commercial refrigeration installed in warm climates, integrating also air conditioning and heating, with different load ratios, poses challenges in evaluating and comparing energy performances. Both analysis based on first or second law of thermodynamics requires the measurement of refrigerant mass flow rate to straight forward evaluate useful effects. A low-cost, reliable and low intrusive option to measure CO_2 volume flow will be illustrated, demonstrating the use of not-in-kind sensors in commercial refrigeration systems. Signals coming from the field might be used to evaluate the system performance, i.e. Coefficient of Performance and exergetic efficiency, and potentially to promptly identify anomalous working conditions leading to inefficient operations or failure. Field data will validate the possibility of identifying the system performance and its health status using a limited amount of signals and filtering them through proper figures.
Keywords: Refrigeration, Carbon Dioxide, Compressors, COP, Evaporators, Energy Efficiency.

1. INTRODUCTION

Commercial refrigeration with CO_2 has rapidly and widely spread in the last years, reaching near 14.000 installation in Europe in early 2018, according to Shecco, 2018. Different layouts have been developed, to provide the market with suitable solutions, according to the climate or utililsation. In particular, integrated solutions providing refrigeration, air conditioning and heating are currently required from the market, due to their cost and performance competitiveness, as demonstrated by Karampour and Sawalha(2018).
Although detailed and reliable simulation models are developed by many research groups and companies, there is an increasing demand for performance data collected in the field. Field Data can assure the market about feasibility, reliability, energy efficiency of CO_2 integrated systems (Multipack, 2019) and promote fast transition to low environmental impact solutions in the food retail sector by awareness raising (SuperSmart, 2019).
Monitoring Systems and Monitoring Solutions are commonly used to refer to a wide range of different systems, with different characteristics and features. Based on their capabilities and scopes, three different categories can be identified, i.e.Monitoring Systems, Remote Control and Planning Systems and Integrated and Optimized Control Systems. Monitoring systems centralize the data, make non conformal or fault data available to head quarters, which can identify the right procedure to fix the problem. Control systems allow remote operation reducing the need for on-site service; remote control can be used as planning tool and set-ups can be changed or reset autonomously by the system under given conditions. Integrated control systems can handle more complex planned operations and they can take advantage of sub-system interactions.

Monitoring systems are powerful tools for management, providing a centralized database and representing an effective tool to improve food quality (continuous temperature monitoring and recording) and reduce food waste (prompt identification of non-conformal system behaviours).
Control systems include all the functionalities of monitoring systems, they allow to act on the system parameters in the local controllers and permit to define a virtual control room accessible form multiple platform (PC, Tablet…). Access to this environment can be granted to specialized technicians, consultants, suppliers, thus reducing service time and improving the overall system performance. Control systems represent the simplest type of Smart Systems and their natural development is represented by integrated and optimized control systems; they can positively impact on different aspects such as energy saving, food quality, system reliability, nevertheless they require high integration and high overall complexity.
Monitoring & Control Systems are themselves subsystem that need maintenance and constant attentions as all the other subsystems. They can provide significant improvements, but only if used and interpreted by trained and professional personnel with expertise in data analysis and HVAC&R systems.
The huge amount of data coming from the field can actually represent an obstacle to the obtainment of the benefits deriving from remote monitoring and control, if data are not properly filtered and processed by experienced people.
In addition, the increasing complexity of CO_2 system layouts calls for even higher expertise level and by the wide range of designs offered by the market.
According to the past gained experience, it seems relevant to try to provide commercial refrigeration with reliable and sustainable monitoring systems and to offer an insight into some parameters that could offer a future sharable way of interpreting and comparing system performances, as well as highlighting operations that might not follow design requirements.

2. CASE STUDY

For the present study a supermarket included in the Multipack project has been selected. MulitPACK [2019] is an EU project demonstrating and building up confidence for standardised integrated cooling & heating packages installed in high energy demanding buildings. The overall aim of MultiPACK is devoted to demonstrate performances of the integrated CO_2 vapour compression packs at a commercial level through full-scale applications at the end-users in Southern Europe. Namely, the concept of multifunctional Heating, Ventilation, Air Conditioning and Domestic Hot Water (HVAC & DHW) units are demonstrated for a group of various commercial and public high-energy-consumption buildings throughout Southern Europe, while refrigeration prioritized packs for supermarkets are proven in three medium size shops located in Portugal and Italy. The supermarket which is considered in this paper is located in central Italy.

2.1. System Layout

The system overall layout is presented in Fig. 1. The unit is based on a booster concept with parallel compression and expansion work recovery by means of ejectors for both vapour pre-compression (10) and liquid recirculation (9).
Semi-hermetic compressors are installed, three each rack of compression level: Medium Temperature MT(1), Low Temperature LT(3), Auxiliary (2). For each rack, one out of the three compressors are inverter driven to allow a smoother capacity modulation.
The unit also satisfies the heating and the cooling loads of the supermarket by means of an Air Handling Unit AHU (11), in which CO_2 directly flows in to heating and cooling coils. In winter time, a three way valve (12) deviates CO_2 to the AHU coil if requested by the building temperature control, consequently high pressure is increased. If the heating demand exceeds the heat recovered from the refrigeration system, the heat pump mode is activated, i.e.the external evaporator (6) is fed with liquid from the receiver (7). The heat pump mode, supported by dedicated auxiliary compressor, can work independently from those removing flash gas, thanks to the solenoid valve (16) In summertime, the AHU cooling coil is fed with two-phase refrigerant expanded through the electronic expansion valve (17), which is thermostatically controlled. Post –heating is also provided by CO_2 in summertime.
A liquid receiver (7) is located after the expansion valve, able to manage charge variations in the circuit and to provide sufficient liquid head to circulate CO_2 into the two refrigeration lines. In case of

liquid flow back from MT evaporators, due to the adopted Electronic Expansion Valve EEV control, the liquid accumulated in the suction liquid receiver (13) is pumped back to the receiver (7) by liquid ejectors.

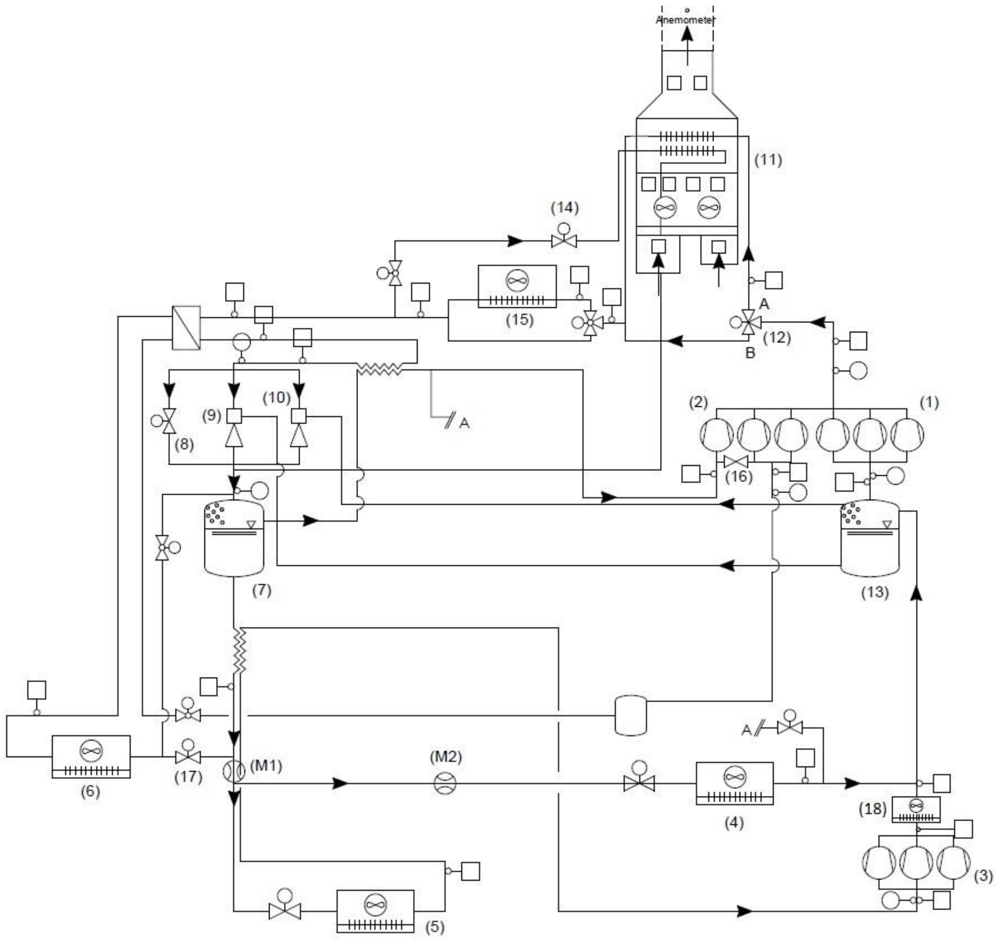

Figure 1: System lay-out

2.2. Measuring Devices

Following the requirements from the MultiPACK project, the system is fully equipped with measuring instruments, for pressure, temperature, refrigerant mass flow and compressor input power measurements. The location of the instruments is referenced in Figure 1, where temperature probes are indicated with a square and pressure sensor with a circle.

There are 18 commercial type NTC 10 kΩ ±1% at 25 °C Beta 3435 sensors, whose manufacturer's declared precision is ±0.5°C at 25°C and ±1.0°C in the range -40°C to +90°C. Pressure is measured with six commercial type piezoresistive pressure transmitters (circle in Figure 1), with accuracy ranging from ± 1%FS to ± 4 %FS, depending on temperature level pressure sensors have FS 60 10^5 Pa, except those on high pressure side (150 10^5 Pa).

In order to evaluate the total electric power input, three-phase electric power meters are located before each compressors rack to measure the power input to Low Temperature (P_{LT}), Medium Temperature (P_{MT}), and Auxiliary (P_{AUX}) compressors. Their accuracy is ± 0.5 % FS (FS is 24 kW for LT compressor rack and 120 kW for MT and AUX compressor racks). The status of every single compressor and the inverter frequency are also acquired. The total power input to the system, including auxiliaries (fans, pumps, valve motors,..) is monitored.

Two Coriolis mass flow meters (M) are placed on the liquid lines, the first (M1) one measures both MT and LT CO_2 mass flow, the second one (M2) is dedicated to MT flow. Their accuracy is 0.1% of the actual flow. On the AHU side, eight temperature and relative humidity sensors are installed to measure enthalpy differences before and after the coils, together with building return air and external air temperature and humidity. A hot wire anemometer (accuracy ± 0.2 m/s+3% of measured value) is installed on the main air duct, to measure air velocity of the whole air volume flow, so it is possible to calculate the total cooling and heating power provided by the unit. The accuracy of the RH probes is ± 3% and of the reading (temperature range 0 -40°C, RH up to 90%) and ± 0.5 °C for temperature from+10°C to+30 °C.

3. PERFORMANCE EVALUATION

The variety of technical solutions offered by manufacturers as an alternative to F-gases banned from 2020 (like R-404A) and the availability of very new integrated solutions for refrigeration and air conditioning based on CO_2 require a comparison of systems in terms of energy efficiency, rather than energy consumption. It claims for experimental methodologies, as consolidated and validated models for those systems are not yet available or affordable or, if available, they are not easily applicable to different systems, as they are typically tailored and validated on a specific site.

To this extent, the performance evaluation is based on measured values of power input and cooling capacity and remote monitoring and subsequent data processing play a major role.

Both first and second law of thermodynamics efficiencies are presented, in order to evaluate the best approach.

The second law approach in commercial refrigeration systems analysis, which compares the actual power input to the ideal one, i.e. the Carnot one, to provide the same useful effect, has been explicitly proposed by the EU project SuperSmart.

To the Authors' knowledge, it has been first applied, based on experimental data from the field, by Blust et al, 2018.

When considering the MT cycle, the first law Coefficient of Performance COP follows the definition of equation (1), taking into consideration all useful effects with respect to the power input to get it. With reference only to refrigeration, COP has been defined for the MT cycle, considering also the heat rejected to the MT circuit from the LT one, i.e. $\dot{Q}_{LT} + P_{LT} - \dot{Q}_{ds}$, being $\dot{Q}_{ds}$ the heat rejected to the desuperheater (18).

$$COP_{MT} = \frac{\dot{Q}_{MT}+(\dot{Q}_{LT}+P_{LT}-\dot{Q}_{ds})}{P_{MT}+P_{Aux}} \qquad \text{Eq. (1)}$$

The above definition is not valid when there is cooling at AHU ($\dot{Q}_{AC}$).In heating season, if ($\dot{Q}_H > 0 \wedge P_{Aux}>0$), Eq (1) is not applicable; however, if the status of the auxiliary compressors removing vapour from the flash tank (7) is off, Eq (1) can be used imposing $P_{Aux}=0$ because auxiliary compressors are only working for heating purposes.

For the LT cycle, only the power input to the LT compressors has been considered, thus defining the COP of the low stage cycle:

$$COP_{LT} = \frac{\dot{Q}_{LT}}{P_{LT}} \qquad \text{Eq. (2)}$$

Finally, a total COP_{tot}, considering useful effects at every temperature level follows the equation:

$$COP_{tot} = \frac{\dot{Q}_{MT}+\dot{Q}_{LT}+\dot{Q}_{AC}+\dot{Q}_{H}}{P_{MT}+P_{LT}+P_{Aux}} \qquad \text{Eq. (3)}$$

On the other hand, secondo law efficiency relates the actual power input to the ideal minimum value to get the same useful effect, i.e. the Carnot Power, which results in taking care of the temperature level of each heat sink and source.

Equation (3) might then be reformulated as:

$$\eta_{tot} = \frac{P_{C,MT}+\dot{P}_{C,LT}+P_{C,AC}+P_{C,H}}{P_{MT}+P_{LT}+P_{Aux}} = \frac{\dot{Q}_{MT}\left(\frac{T_{amb}}{T_{MT,ref}}-1\right)+\dot{Q}_{LT}\left(\frac{T_{amb}}{T_{LT,ref}}-1\right)+\dot{Q}_{AC}\left(\frac{T_{amb}}{T_{AC,ref}}-1\right)+\dot{Q}_{H}\left(1-\frac{T_{amb}}{T_{H,ref}}\right)}{P_{MT}+P_{LT}+P_{Aux}} \qquad \text{Eq. (4)}$$

Where the reference temperatures $T_{MT,ref}$, $T_{LT,ref}$, $T_{AC,ref}$, $T_{H,ref}$ for MT, LT, AC and space Heating might be respectively 273.15 K, 251.15 K, 299.15 K, and for heat recovery 294.15 K, considering the standard prescriptions of 26°C and 21°C air temperature during cooling and heating seasons, respectively.

In general terms, while the second law approach mitigates the effect of outdoor temperature with respect to COP approach, which shows huge variations of the same system performance at equal Load Ratio ($LR = \dot{Q}_{MT}/\dot{Q}_{LT}$) when changing outdoor temperature, it completely reverses the effect of load ratio variations at the same operating conditions. This means that, while the COP decreases when decreasing the load ratio, the exergetic efficiency might have a different trend depending on MT and LT performances. Similar considerations can be applied also to AC and heating loads, in the case of heating pump mode.
The parametric analysis of the impact of boundary conditions, including load ratio, on the first and second law efficiencies dares a dedicated effort; this manuscript just aims at presenting the alternative approaches, to be eventually implemented in remote monitoring, and at showing the calculated values of the reference system.
The thermal loads to be used equally in equations 1, 2, 3 and 4, are calculated, using measured values of total refrigerant mass flow rates to the MT, LT and AC sections, taken at the inlet of each distribution line and the enthalpy of the refrigerant before and after the MT, LT and AC evaporators. Power inputs are also measured at compressor racks.

3.1. Performance Measurements in the Field

The field test started in December 2018, thus providing us, so far, with winter operations, i.e. subcritical conditions or transcritical with heat recovery or heat pump mode.
The main peculiarities of the registered working conditions are the high MT evaporation temperature (-5°C), together with the quite low load ratio $LR = \dot{Q}_{MT}/\dot{Q}_{LT}$, with respect to similar installations in Italy (D'Agaro et al., 2019, Dugaria et al., 2019).
Fig. 2 reports the trend of thermal loads during a 24hours period, clearly showing that, during the nightime the AHU is switched off. Although the data acquisition time step was set to 1 min, due to connection issues, some acquisitions were not stored; nevertheless, the presented data are obtaining by moving average over a 60 min. period.

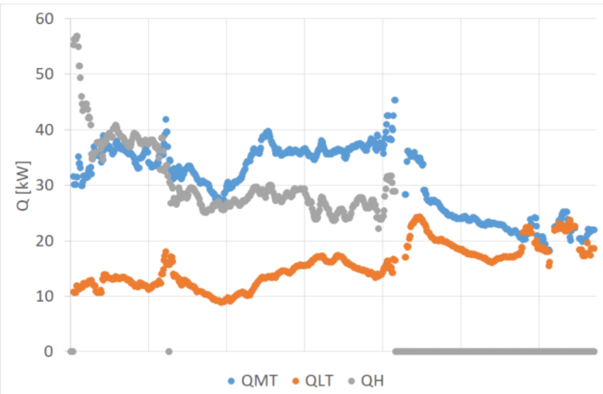

Figure 2: Cooling and heating loads during a 24h period

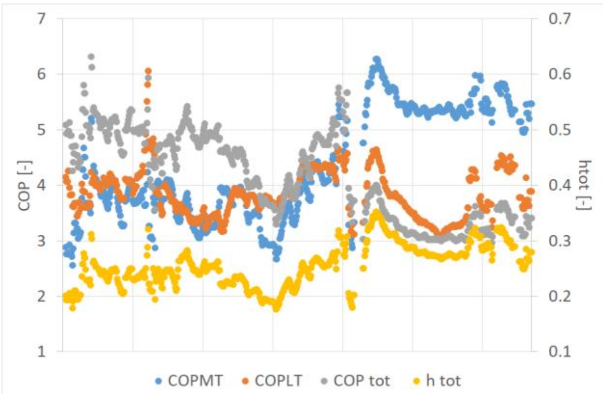

Figure 3: COP and η_{tot} during a 24h period

First law COPs, as defined in equations (1-3) are reported in Fig. 3, together with total efficiency (equation 4).
Table 1 shows the average values of the working conditions and measured performance (useful effect) values for data illustrated in Figs. 3 and 4, over the all stable period; COP and efficiency is calculated considering the total energy values over the stable period.
It is worth observing how the COP_{tot} and η_{tot} can provide different information when considering the proposed boundary conditions. While the first law approach is commonly used, the second law one requires more use to get acquainted to it.
It is worth reminding that both first and second law approaches require the values of refrigerant mass flow rates in the involved lines. As pointed out by Minetto et al.(2018), the evaluation of refrigerant mass flow rates with compressor curves, despite sufficiently accurate in the case of simple systems, might not be easily applicable in complex layouts, such as those involving ejectors.

Table 1. Measured values of boundary conditions and performances during winter operations

-	-	22/12/2018	22-23/12/2018	12/12/2018	12-13/12/2018
T_{amb}	(°C)	14.5	14.2	10.6	9.9
T_{evMT}	(°C)	-4.6	-5.0	-4.9	-5.1
T_{evLT}	(°C)	-31.3	-30.2	-30.4	-30.9
T_{evaux}	(°C)	2.6	-	-0.8	-
p_{gc}	(10^5 Pa)	70.3	55.8	77.3	53.7
Q_{MT}	(kW)	35.0	24.2	31.4	22.4
Q_{LT}	(kW)	13.4	19.4	16.9	14.5
Q_H	(kW)	30.9	-	49.3	-
LR	(-)	2.6	1.2	1.9	1.5
COP_{MT}	(-)	2.7	5.3	3.2	6.3
COP_{LT}	(-)	3.8	3.7	4.0	4.1
COP_{tot}	(-)	3.6	3.3	4.4	4.0
η_{tot}	(-)	0.20	0.29	0.23	0.27

4. FLOW METER: A PROPOSAL

As it emerges from the previous sections, the measurement of refrigerant mass flow is crucial to define the useful effects of the system and classify it according to first or second law approach.
Installing flow meters in commercial refrigeration systems is not common practice; therefore, there have been no specific developments for providing the market with reliable, sufficiently accurate, small and low cost sensors, to be finally included in remote monitoring systems.
Flow meters for fluids based on the hot wire anemometer principle are available on the market; however their size and their cost does not make them suitable for large scale application, especially when many sensors are required in the same system, as in the case of complex layouts as the one of Fig. 1.
In the following section, the theoretical analysis of an in-house flow meter dedicated to the application is presented.

4.1. Schematic Description

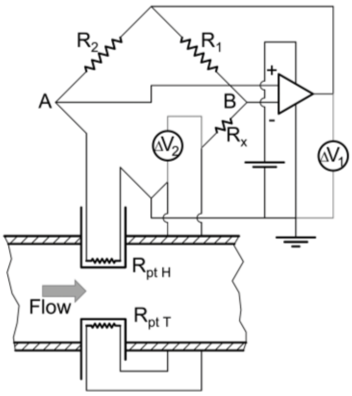

Figure 4: System sketch

Table 2. Circuit sizing

Resistor	Nominal Resistance
R_1	500 Ω
R_2	50 Ω
R_x	15 Ω
R_{ptH}	100 Ω
R_{ptT}	1000 Ω

The proposed sketch in Fig. 4 presents the sensor electrical circuit, which is based on the constant temperature hot wire anemometer principle. The anemometer is composed of two sensors: The first is the velocity probe ($R_{pt\,H}$ in Figure 4) and the second is the temperature compensation sensor ($R_{pt\,T}$ in Figure 5). In the proposed set-up both sensors are platinum resistors. Due to the high pressures and possible mechanical stress given by oil droplets or pressure fluctuations, both the probes are assumed to be enclosed at the top of cylindrical stainless steel housing.
The Voltage signal ΔV_1 represents the velocity signal. The second output Voltage channel ΔV_2 is used to obtain a reading on the temperature compensation probe, which expected to be more precise than the temperature measures available on the field, being inside the fluid flow.

4.2. Expected performance

In order to correctly design the Wheatstone bridge, and to assess the overall precision of this sensor, numerical simulation were run. The system is resolved imposing the energy equilibrium of the probes heating element (Equations 5 and 6) and the Wheatstone bridge equilibrium between points A and B (Equation 7):

$$i_H^2 R_{ptH} = \left(\frac{\Delta V_1}{R_2+R_{ptH}}\right)^2 R_{ptH} = \alpha A\left(T_{ptH} - T_{CO_2}\right) \qquad \text{Eq.(5)}$$

$$i_T^2 R_{ptT} = \left(\frac{\Delta V_1}{R_1+R_x+R_{ptT}}\right)^2 R_{ptT} = \alpha A\left(T_{ptT} - T_{CO_2}\right) \qquad \text{Eq.(6)}$$

$$i_H R_2 = \frac{\Delta V_1\, R_2}{R_2+R_{ptH}} = \frac{\Delta V_1 R_1}{R_1+R_x+R_{ptT}} = i_T R_1 \qquad \text{Eq.(7)}$$

Equations 5-7 allow to solve the three unknown ΔV_1, T_{ptT} and T_{ptH}, once the dependency on temperature of the resistors R_{ptT} and R_{ptH}, and the convection coefficient α are defined. The Callendar–Van Dusen equation was used to describe the Pt sensor sensitivity to the temperature. Convective heat exchange coefficient was computed applying the flat-plate correlation, using the probe diameter as reference length.
The output signal was then computed, assuming four different installation points: the MT line (subcooled liquid), the LT return line (superheated vapour), the compressors rack outlet line (GL_{in} and GC_{in}^{sup}) and the gas cooler return line (GL_{out} and GC_{out}^{sup}). As the refrigeration unit can work in both super critical and subcritical cycle, a total of six reference conditions has been studied.

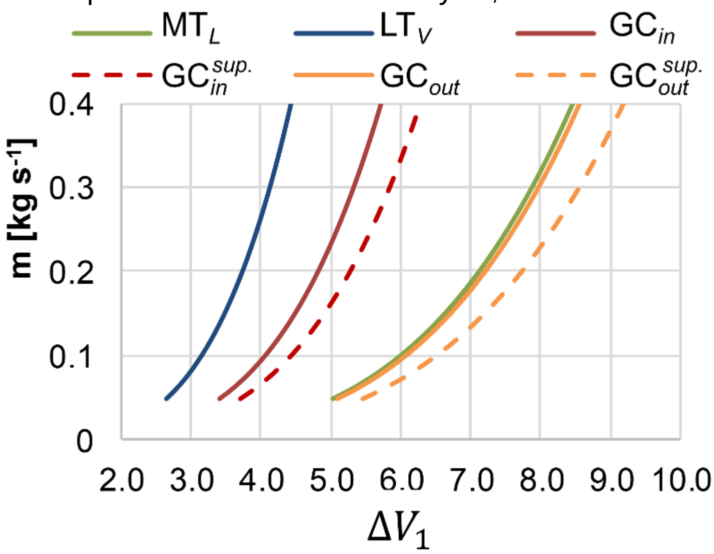

Figure 5: Expected output voltage difference ΔV_1 for in different measuring points in the system

Table 3. Reference conditions and equilibrium sensors temperatures

MT$_L$ $p = 38.0$ bar		LT$_V$ $p = 13.8$ bar	
T_{CO_2}	-2.3 °C	T_{CO_2}	3.2 °C
T_{ptT}	-1.8 °C	T_{ptT}	3.6 °C
T_{ptH}	2.0 °C	T_{ptH}	7.4 °C
GC$_{in}$ $p = 64.8$ bar		GC$_{out}$ $p = 64.8$ bar	
T_{CO_2}	69.4 °C	T_{CO_2}	16.5 °C
T_{ptT}	69.9 °C	T_{ptT}	16.9 °C
T_{ptH}	73.8 °C	T_{ptH}	20.8 °C
GC$_{in}$ $^{sup.\ crit.}$ $p = 100.0$ bar		GC$_{out}$ $^{sup.\ crit.}$ $p = 100.0$ bar	
T_{CO_2}	120.0 °C	T_{CO_2}	40.0 °C
T_{ptT}	120.4 °C	T_{ptT}	40.4 °C
T_{ptH}	124.4 °C	T_{ptH}	44.3 °C

The expected output signals ΔV_1 are presented in Fig. 5, while in Table 3 reports used reference conditions, as well as the equilibrium temperature for the sensors. The highest signals can be obtained in the liquid lines (or the dense vapour when supercritical condition is considered), thanks to the higher heat exchange coefficient. The Value of resistance Rx in the Wheatstone bridge defines the temperature difference between the sensors. The use of a 15 Ω resistor results in a temperature difference $T_{ptH} - T_{ptT} \cong$ 4 K, while the difference between the temperature sensor and the actual flow temperature is $T_{ptT} - T_{CO_2} \cong$ 0.4 K. In order to avoid phase change in the high temperature sensor, the difference $T_{ptH} - T_{CO_2}$ (approximatively 4.4 K) should be lower than the subcooling of the liquid, while no restriction is present when the vapour or the supercritical condition are considered.
As result of the sensitivity study in Figure 6, the proposed sensor seems a feasible alternative to the much expensive flow meter M1 and M2 (Figure 1), when sufficient subcooling is present The

measurement taken in the supercritical region ($GC_{in\ sup.}$) might pose some challenges for the determination of Q_H.

5. CONCLUSIONS

In this manuscript, data collected from remote monitoring of a fully instrumented Integrated Refrigeration unit based on CO_2 have been processed to evaluate energy performance both on a first and second law of thermodynamics approach. The second law approach can offer the opportunity of weighting the different useful effects, based on their temperature level and on load ratios. The two methodologies can propose a different insight and dare a further detailed sensitivity analysis. Refrigerant mass flow measurement is the straightforward way to evaluation of cooling and heating loads and should be included in remote monitoring. However, reliable, small and low-cost sensor for large scale implementation is a key issue. A new sensor based on the hot wire anemometry, potentially small and low cost, is presented and its performances theoretically evaluated according to the actual location it might have in the real plant. According to the proposed arrangement, location in the liquid line to MT and LT cabinets offers encouraging results.

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