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# An integrated $CO_2$ unit for heating, cooling and DHW installed in a hotel. Data from the field.

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# An integrated $CO_2$ unit for heating, cooling and DHW installed in a hotel. Data from the field.

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**Abstract.**
This paper presents a new heat pump unit intended to provide heating, cooling and hot water to a hotel and using $CO_2$ as the working fluid. The heat pump was installed in mid-2018 in a hotel located in a touristic area in North Italy and open nearly all over the year. The unit can benefit from ground water as heat source or heat sink. The heat pump features a two-phase multi-ejector as expansion device. An original two-evaporator lay-out is implemented, where the first one is gravity driven and the second one is ejector driven. It is fully equipped with measurement instruments, data acquisition system and cloud data storage, developed and tailored for the unit. The unit was developed, instrumented, installed and monitored within the H2020 MultiPACK project, which aims at building confidence in integrated heating, ventilation, air conditioning and refrigeration packages based on $CO_2$ technology in high energy-demanding buildings.
In this paper, operational parameters and performance data will be presented, related to winter season, thus to heating and domestic hot water service.

## 1. Introduction
In recent years, the use of $CO_2$ as a refrigerant in different application has been successfully investigated and implemented; due to the tough revision of the refrigerants use consequent to the F-gas regulation implementation in Europe and later on to the Kigali amendment to the Montreal Protocol. $CO_2$ has been rising interest also in the civil buildings sector, especially where Domestic Hot Water (DHW) demand is high, like in hotels, gyms, spas. Transcritical $CO_2$ cycle represents an efficient solution for warming up tap water through a high temperature lift, achieving the best energy performances by using once-through heat exchangers and high stratification water storage [1]. As demonstrated by different authors [2,3,4] $CO_2$ units providing cooling, heating and DHW may present energy performances that are at least comparable to the same state of the art HFC's systems. When cooling and hot water production at high temperature occur simultaneously, $CO_2$ transcritical cycle can very efficiently serve the purpose [4]. For residential applications, Minetto et al. [3] proposed a water side reversible $CO_2$ heat pump, able to provide heating, cooling and DHW, that underperformed the state of the art R410A system, but could lead to the same seasonal efficiency as the R410A layout by the simple implementation of a two-phase ejector in place of the expansion valve. The same authors showed that the considered $CO_2$ heat pump is always more energy efficient in DHW production but penalised during heating and cooling, according to the selected boundary conditions, i.e. water return temperature from the heating system and outdoor temperature profile during cooling. As a general trend especially in Central and South European

Countries, it must however be considered that, while domestic hot water demand from buildings is continuously increasing, due to the better thermal insulations of the envelope and the increasing internal heat loads, as well as the cooling energy demand, the energy requested to the heating system is decreasing. When moving to public buildings, such as hotels, spas, gyms, the DHW demand becomes preponderant and, in the summertime, it is often simultaneous to cooling demand: consequently, the use of $CO_2$ is rapidly gaining interest.
Recently, data from the field of a first combined $CO_2$ heat pump, chiller and tap hot water system installed in a Scandinavian hotel were presented, showing good results in the first working period, with an average COP of the system of 2.98 [5].
The MultiPACK project aims at increasing confidence and proving the benefits of $CO_2$ systems by installing three reversible heat pump units with DHW production in high energy demanding buildings and to monitor and evaluate their performances.
In this paper, the installation and operations of the first $CO_2$ water side reversible heat pump with an original two phase multi ejector layout implemented will be fully documented.

## 2. Unit layout and instrumentation

The heat pump is installed in a hotel located in a touristic area in North Italy: it is intended to provide heating, cooling and sanitary hot water to a hotel, which is open nearly all over the year. The unit can benefit from ground water as heat source or heat sink.
The layout of the system is presented in Figure 1, in heating configuration mode.
The heat pump features a two-phase multi-ejector as the expansion device, which works in parallel to a high pressure valve. An original two-evaporator lay-out is implemented, where the first one is gravity driven and the second one is ejector driven. Water flows firstly through the gravity driven heat exchanger, where $CO_2$ evaporates at the compressor suction pressure, and then through ejector driven heat exchanger, which benefits from a lower evaporation pressure, accordingly to the pressure lift provided by the ejector. The unit is reversible on water side, by means of a hydronic module made by three way valves that can switch to the ground water or the HVAC plant according to the building request. The unit can work in heating or DHW mode alternatively in winter time, giving priority to DHW, using ground water as the heat source. In summer, it can provide cooling and DHW at the same time using HVAC water loop as the heat source, thus providing a double useful effect. If cooling or DHW is not requested by the building, the hydronic module switches to groundwater heat exchanger acting as a heat source or sink according to the working mode.
Fan coils are installed in the hotel rooms, providing heating and cooling, and they are fed by the hydronic water loop.
Domestic hot water is produced and accumulated in two water tanks that are connected in series to allow stratification; in such a way only cold water flows to the heat pump DHW heat exchanger.
Two compressors are installed, one of them is inverter driven, and the capacity regulation of the compressors is based on the suction receiver pressure variation, which is influenced by the heating and cooling demand.

### *2.1. Data acquisition system*

The evaluation of the performances of the unit under different conditions is carried out by collection and processing of operating conditions that are relevant for the calculation of key performance indicators. The measuring instruments are shown in Figure 1. Seven PT 100 temperature probes are placed before and after the three $CO_2$/Water heat exchangers, on the water side identified in Figure 1 with $PT_1..PT_7$; together with four magnetic mass flow meters ($M_1..M_4$), they can measure the heat exchanger performance and the unit useful effect. Pressure probes ($P_1..P_3$), with 160 bar FS, are located at the suction pressure level, high pressure side, and ejector suction nozzle.
Several NTC probes ($T_1..T_{15}$) are placed in the system to monitor temperature on both the $CO_2$ and water loop, including the hydronic module represented in the Figure 1. The DHW tank is fully equipped with NTC sensors to provide information about water stratification and storage temperature.
An energy meter measures electrical power absorbed by the compression rack. The frequency of inverter driven compressor and the on/off status of the compressors are also detected. A magnetic mass flow

meter is installed in the city water side to measure the hotel DHW water consumption. The accuracy of instruments is summarised in Table 1.
All the the instruments are linked to an analog/digital converter. The modules are then connected in RS-485 mode and the Modbus protocol is used to deliver data information. A data logger gets the data from the modules with a minute time step and stores them on a cloud database. The data logger is a single board computer Raspberry Pi, that is equipped with a USB/3G stick that connects to the cloud server. Data are downloaded from the cloud server and then processed to assess energy performances; to identify critical operations and thus possibly to prevent malfunctions of the unit and, in general, of the HVAC plant.

**Table 1.** Accuracy of the instruments.

| Instruments (S) | Accuracy |
|---|---|
| Magnetic flow meter (M) | ± 0.8% reading |
| PT 100 (PT) | Class A IEC 60751 |
| NTC (T) | Class accuracy 1% |
| Pressure (P) | ± 0.5% FS |
| Energy meter | Class accuracy 0.2% |

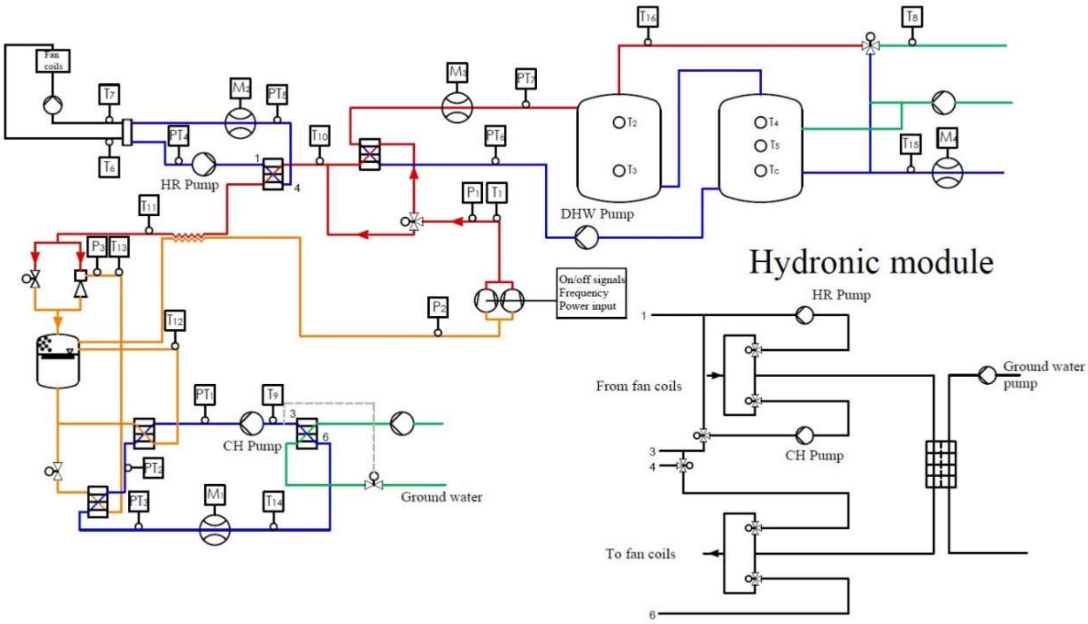


**Figure 1.** The layout of the $CO_2$ unit and the layout of the hydronic module. Instruments and sensors are also shown. The unit is represented in heating configuration.

*2.2. Calculation of performance indicators*
In order to assess the performances of the unit, the following parameters are calculated. Thermal power at each Brazed Plate Heat Exchangers BPHEs is calculated as:

$$Q = M_j \, Cp \left| T_{PT_{in}} - T_{PT_{out}} \right| \tag{1}$$

Where, $M_j$ is the mass flow measured with the magnetic mass flow meter and flowing through the heat exchanger; $T_{PT_{in}}$ is the temperature of the water entering the heat exchanger, and $T_{PT_{out}}$ the temperature of the water exiting the heat exchanger. According the working mode of the unit and to the specific heat exchanger for which $Q$ is calculated, the parameters utilized in the formula are different.
Instant $COP_{it}$ in both DHW and Heating working mode of the unit, calculated as the heat load on the heat exchanger with respect to electrical power absorbed by the compressor, minute by minute is:

$$COP_{it} = \frac{Q}{P_w} \tag{2}$$

While $COP_{av}$, is the average $COP$ calculated in a defined time period $t_1 .. t_2$ as the ratio between the total thermal energy exchanged in the period and the total electrical energy absorbed in the same time.

$$COP_{av} = \frac{\sum_{t_1}^{t_2} Q}{\sum_{t_1}^{t_2} P_w} \tag{3}$$

The maximum uncertainty calculated for both domestic hot water power $Q_{DHW}$ and heating power $Q_H$ are respectively 1.8% and 5.2%, while the for the cooling powers $Q_{ev\_fl}$ ,$Q_{ev\_ej}$ and $Q_{ev}$ that are the heat transfer rate at the flooded gravity driven evaporator, ejector driven evaporator and the total cooling power, maximum uncertainties are 1.5%, 1.0% and 0.8%.

## 3. Domestic Hot Water DHW working mode

It is widely known that in transcritical cycles the secondary fluid inlet temperature to the gas cooler needs to be as low as possible to achieve best possible performance; this is of course valid also in the case of DHW production, where large temperature lift can be achieved at high COP provided that the water temperature that enters the gas cooler is low. The water tank has then to promote stratification; it is common practice to connect several tanks in series, so that natural convection is limited and stratification is improved.
Dedicated components for DHW production in the analysed unit is a brazed plate heat exchanger (gas cooler), with a design power of 30 kW (water delivered at 65°C) ; two water storage tanks connected in series, with a volume of 750 litres each and a water pump. The speed of the pump is regulated in order to vary the mass flow rate to keep the outlet water temperature at the design temperature, so to guarantee that the temperature of water entering the hottest storage tank is constant, so to preserve stratification. City cold water pipe coming from the net is connected to the bottom of the coldest storage tank, to refill it when hot water is drained.
When the tanks are fully charged, the hottest water is accumulated in the first tank. If there is hot water demand from the building, hot water is drawn from the top of the first tank; it is then mixed with city water by means of a three-way valve, to assure safe temperature delivery. The bottom of the first tank and the second water tank become progressively colder, as water from the net fills the coldest tank from its bottom. During the recharge phase, water is transferred from the bottom part of the cold tank to the upper part of the hot tank, after heating up through the BPHE. Temperature at the bottom of the cold tank ($PT_6$) needs to be kept as low as possible: it is very important that the stratification is preserved during the whole cycle to achieve best performances.
A water loop recirculation allows to keep the water ready to the taps at the desired temperature; the recirculating pump is activated only during specific periods of the day, following the hotel manager's experience. The management of mild water back from the recirculation loop, indicated with the green lines at the right top corner of Figure 1, represents the major challenge to preserve stratification, as water at an intermediate temperature flows back to the receiver and finally influences the water inlet temperature to the gas cooler.
In DHW mode, on/off of the compressors and consequently of the pump are controlled by a temperature probe that is placed in the bottom part of the cold tank (Tc).

Considering the stable operating conditions of the unit, as the heat source temperature is not depending on outdoor temperature, high pressure is fixed at a set point of 105 bar, and no optimum pressure value is seeked [6].

### *3.1. Results*

Results from the field, during the DHW mode, are presented in this section.
A charging cycle period is first analysed and discussed.
Figure 2 shows profile temperatures of the water ($PT_6$) that enters the gas cooler and of the water temperature out of the gas cooler ($PT_7$); mass flow during the charging period keeps nearly constant, until $PT_6$ starts raising. Water mass flow then increases to maintain the water discharge temperature $PT_7$ constant at the set point value 62 °C. The water temperature to the gas cooler ($PT_6$), at the beginning of the cycling period stays between 10°C and 14 °C, increasing during time and reaching the peak of 32°C before the tank charging stops. During the charging period, temperatures of water in the tank ($T_2$, $T_3$, $T_4$, $T_5$) raise, keeping the stratification (Figure 3). The power provided by the unit, as well as the electrical power absorbed by the compressor are shown in Figure 4. Instant values of $COP$ are also represented. During the first 2.5 hours, when $PT_6$ is nearly constant, the unit provides a thermal power that is about 20 kW, while it provides 30 kW in the last part of working period. Electrical power coherently increases from 6 to 10 kW in the last part of the on period. This behaviour is related to the specific unit control, which is based on the suction pressure; when $PT_7$ increases, the outlet temperature of $CO_2$ ($T_{10}$) increases, thus decreasing the heating and cooling power, which determines the suction pressure to increase. Then the controller starts the second compressor to re-establishes the design suction pressure, consequently increasing the unit power. COP however decreases, due to the higher gas cooler outlet temperature.

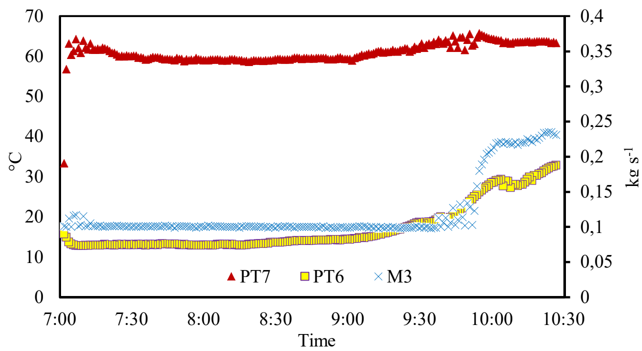


**Figure 2.** $PT_6$ and $PT_7$ temperature profile, and $M_3$ profile.

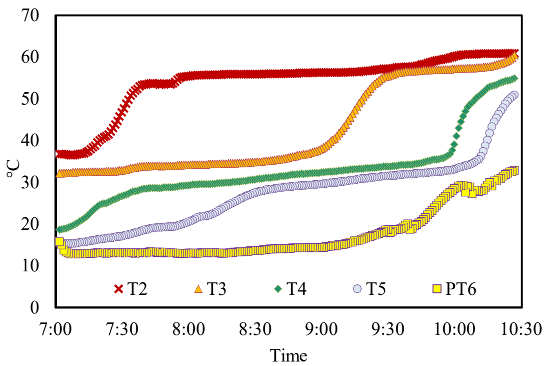


**Figure 3.** Temperatures inside the two tanks connected in series and $PT_6$.

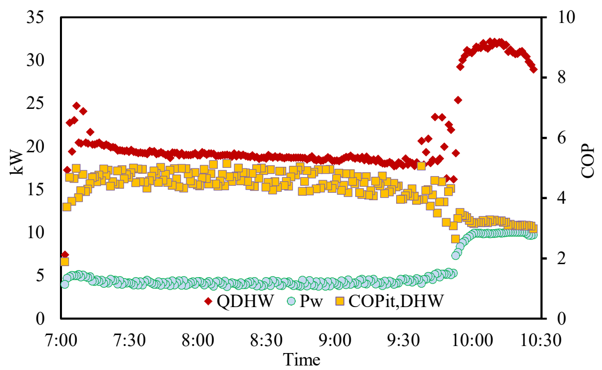


**Figure 4.** Power delivered by the unit, power absorbed by compressor and instant COP.

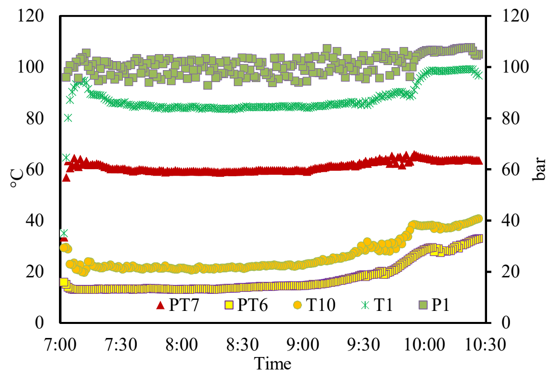


**Figure 5.** Temperature profile at the BPHE and high pressure profile.

Figure 5 shows temperature of $CO_2$ and water in the gas cooler, and the working pressures profile. High pressure stays in the range 100 to105 bar.
During the first period, $COP_{it,DHW}$ reaches 5.0, due to the low water temperature inlet, while the average $COP_{av,DHW}$ during the total working period is 4.1. The last part of the charging period negatively affects the overall performance, as the instant $COP_{it,DHW}$ is lower and the energy provided in this condition is quite high. During the analysed charging cycle, the total DHW energy is 72.0 kWh, and total electrical energy absorbed is 17.6 kWh.
In Figures 6, 7 and 8, the behaviour of the two evaporators during the working period is highlighted. Figure 6 represents water $PT_1$, $PT_2$, $PT_3$ profiles, together with $CO_2$ evaporation temperatures calculated as the saturation temperature at the receiver (corresponding to $T_{ev\_fl}$) and at the suction ejector nozzle (corresponding to $T_{ev\_ej}$). In Figure 7, the same evaporation temperatures are displayed with the corresponding temperatures of $CO_2$ measured out of the evaporators, $T_{13}$ and $T_{12}$. The Figure helps in understanding how the cooling load is distributed between the two evaporators: when the unit is working at low power demand, most of the heat power is exchanged at the flooded evaporator, being almost double with respect to the one transferred at ejector driven evaporator, which is working with a high superheat. When the second compressor switches on, the cooling power of the ejector driven evaporator increases (Figure 8), with a near null superheating, because of the increased recirculated mass flow by the ejector, while the cooling power of the gravity fed evaporator decreases. The reason might lay in the impact on the natural circulation of the gravity driven evaporator caused by the increased mass flow rate in the ejector driven evaporator.
Further investigations are planned to explain the two evaporators mutual influences and the individual contribution at full and partial load.

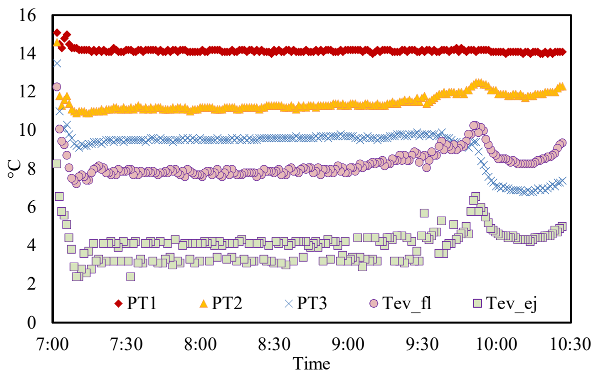


**Figure 6.** Temperature profile at the gravity driven and ejector driven evaporator.

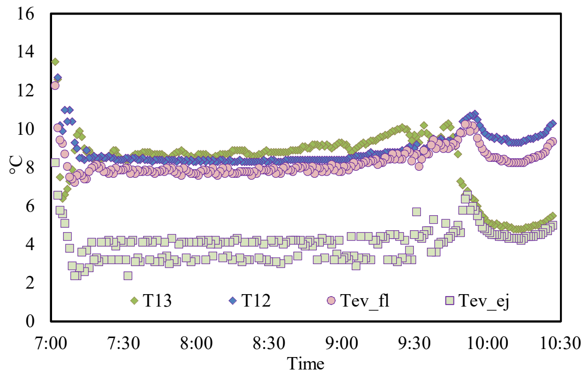


**Figure 7.** Superheating profile of $CO_2$ at the gravity driven and ejector driven evaporator.

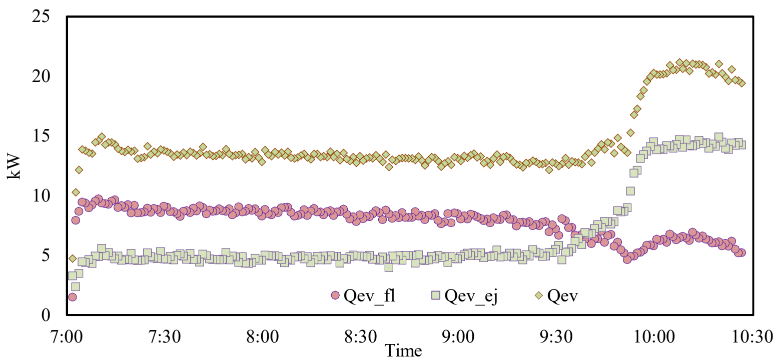


**Figure 8.** Thermal loads at the gravity driven and ejector driven evaporator.

The domestic hot water demand profile is measured with $M_4$ magnetic mass flow meter. The total energy requested from the building for DHW service can consequently be estimated as

$$E_{DHWd} = \sum_{t_1}^{t_2} M_4\, Cp\, (T_8 - T_{15}) \tag{4}$$

Considering four days in April 2019, when the DHW demand was quite low, the total energy given by the unit $E_{DHWq}$ was also calculated together with the total energy demanded by DHW $E_{DHWd}$. They were respectively $E_{DHWq}$ 225 kWh and $E_{DHWq}$ 125 kWh, meaning that lot of energy was wasted by thermal losses in the recirculation loop.
Some field tests are now planned to improve performance by acting on the set point/hysteresis which determines the unit on/off cycles and to adapt the total charging time to the building energy demand.

## 4. Heating working mode

In this section the heating working mode of the unit is presented, considering a time period of 5.5 hours in April 2019. Due to the short data collection period, only limited data sets are available. Full season data will be collected during the project.

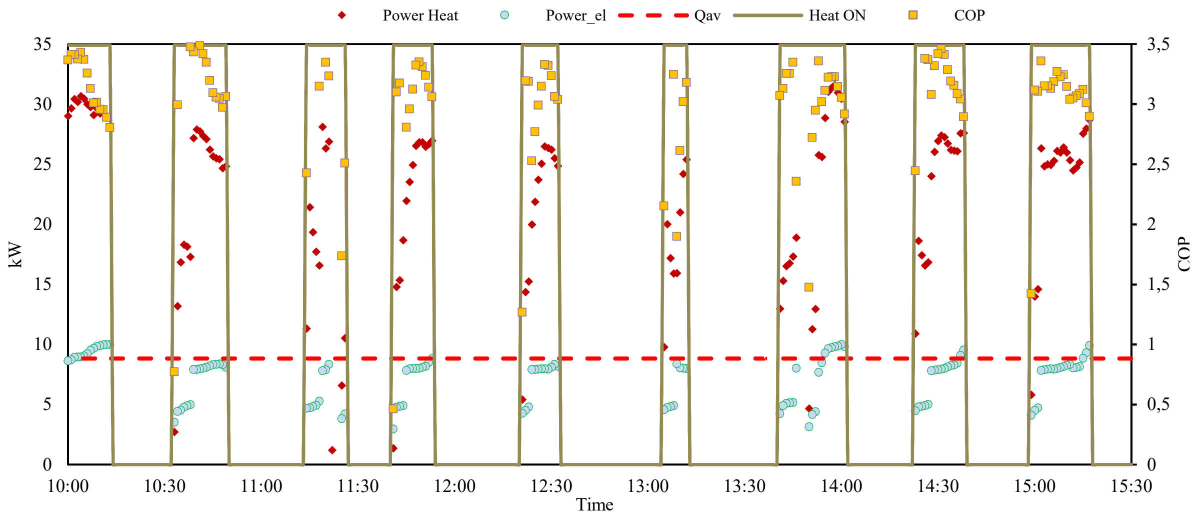


**Figure 9.** $Q_H$ provided by the unit, $COP_{it,H}$, P and $Q_{av,H}$ during Heating working mode

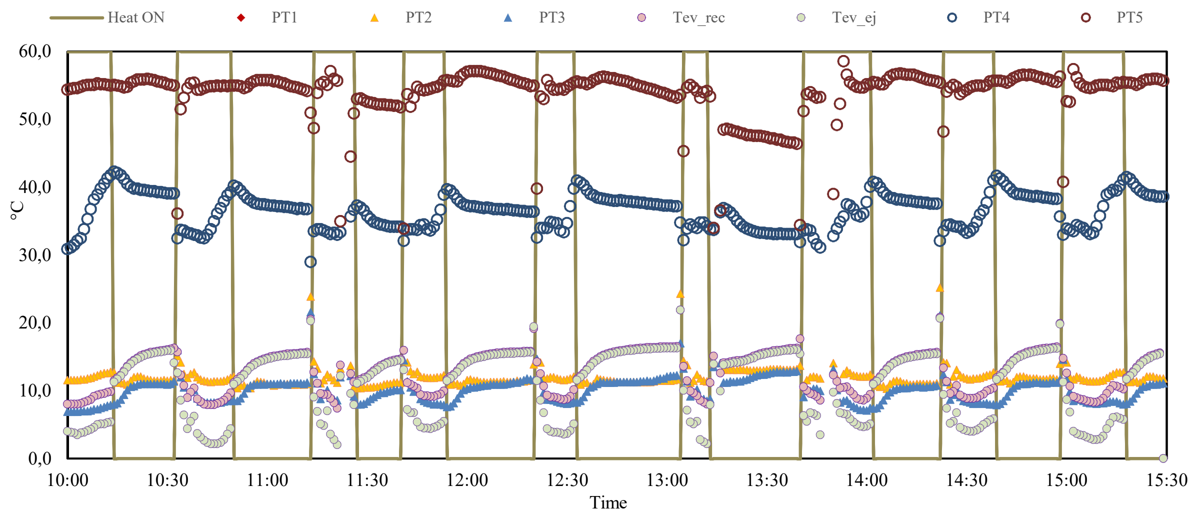


**Figure 10.** Temperature profile on the gravity driven evaporator and ejector driven evaporator during heating working mode**.**

## 5. Conclusions

A $CO_2$ ground water heat pump with an original two evaporator layout implemented, providing heating, cooling and Domestic Hot Water for a hotel was installed and first data from the field are now available for being analysed. So far, only data in heating and DHW working mode conditions are available, in the near future also the cooling working mode will be investigated. First results show that in DHW mode the efficiency of the unit is good; experience in the field demonstrates that it is crucial that all the water system is properly designed to promote stratification and drive cold water to the heat pump. In heating mode, data at partial load are currently available. By acting on the controlling logic of the compressor it will be possible to improve performances, reducing the start and stops when the heating load of the building doesn't match the one provided by the unit.
The peculiar design of the unit, including ejectors and a two-evaporator layout, seems to be the right solution for a $CO_2$ based system serving a hotel. Further experimental and numerical investigations are planned to fully explain the two evaporator's mutual influences and the individual contribution at design and partial load.

## 6. Nomenclature

| | | | |
|---|---|---|---|
| *BPHE* | Brazed Plate Heat Exchanger | | *subscripts* |
| *COP* | Coefficient of Performance [-] | *av* | average |
| *Cp* | Specific heat [kJ $kg^{-1}$ $K^{-1}$] | *d* | demand |
| DHW | Domestic Hot Water | DHW | Domestic Hot Water |
| *E* | Energy [kWh] | *ej* | ejector |
| HVAC | Heating, Ventilation , Air Conditioning | *ev* | evaporator |
| M | Mass flow [kg $s^{-1}$] | *fl* | flooded |
| $P_w$ | Electrical power [kW] | *H* | heating |
| *P* | Pressure [bar] | *in* | inlet |
| *Q* | Thermal power [kW] | *it* | instantaneous |
| T | Temperature [°C] | *q* | produced |
| | | *out* | outlet |

## 7. References

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## 8. Aknowledgements

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