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DESIGN AND CONTROL STRATEGY OF AN INNOVATIVE CO_2 RESIDENTIAL MONOBLOCK HEAT PUMP FOR DOMESTIC HOT WATER

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ABSTRACT

Carbon dioxide (CO_2) as a natural refrigerant (R744) proves its potentiality and suitability for residential heat pumps (HP) producing domestic hot water (DHW). Despite the comparable performances of a CO_2 HP to units adopting other natural and/or synthetic refrigerants, the components cost represents the main barrier to CO_2's wider market share, mainly for higher working pressure and low mass production ratios.

This work presents the thermodynamical and control strategy design of a monoblock HP for DHW with CO_2. The product's uniqueness is being the first with a heating capacity lower than 4 kW on the residential heating market. In particular, the study will focus on the thermodynamic transcritical cycle, component dimensioning and control logic development.

The design took into consideration different technologies and solutions for both gas cooler and water storage tank, improving the heat transfer, coefficient of performance, heating-up time and stratification inside the water tank. On the control side, CO_2 involves new algorithms regulating two different expansion valves aimed to adjust: high-side pressure level, superheating the compressor inlet condition and oil return at the same time.

Keywords: Heat pump, Carbon Dioxide, Domestic Hot Water, Residential, Monoblock

1. INTRODUCTION

The recent review of the European F-gas Regulation (European Commission, REGULATION 2024/573) bans plug-in rooms, monoblock air conditioning and other self-contained heat pump equipment with a maximum rated capacity ≤ 12 kW that contain fluorinated greenhouse gases with a Global Warming Potential (GWP) higher than 150 from 2027. More restrictive bans will arise in 2032, at which time only non-fluorinated gases will be allowed. Since domestic hot water (DHW) heat pumps belong to this category, manufacturers are forced even more towards the transition to natural refrigerants.

In this innovative heat pump for domestic hot water production, the goal is to demonstrate the potential of CO_2 based units to deliver significant higher supply water temperature compared to conventional condensing cycles. This research project will focus on the development of a CO_2 heat pump for domestic hot water production, from initial target data to system dimensioning to control logic development. Firstly, in the component selection process, different technologies are investigated, while secondly, the control of each component is studied. Since the presented unit is a first prototype of its kind in Clivet, this work represents the first step toward further optimization in the design and possible cost reduction to reduce the time to market.

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2. MAIN SECTION

2.1. Vapour compression transcritical cycle and circuit layout

The Lorentzen cycle is the theoretical reference vapour compression cycle for transcritical systems. It consists of isentropic compression, a heat rejection process at constant pressure and gliding temperature, an isentropic expansion and a heat absorption process at a constant temperature, as in Figure 1.

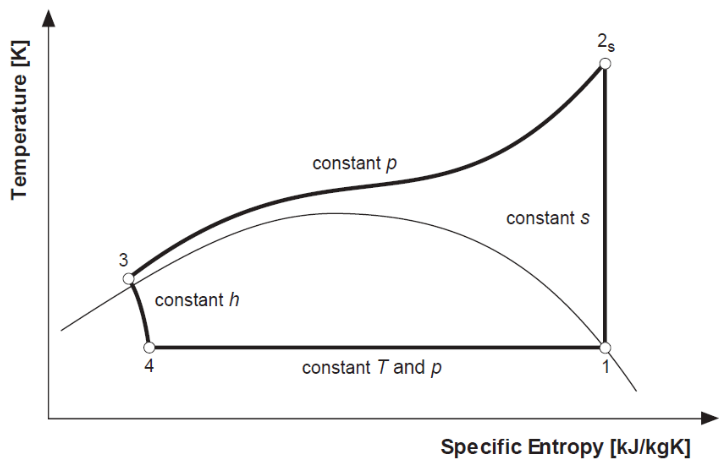

Figure 1: Lorentzen cycle (Klöcker, Schmidt, & Flacke, 1998).

The heat pump prototype was assembled and tested at the Clivet laboratories in Feltre, Italy. Since this was the first prototype of a CO_2 unit. A simplified system, as reported in Figure 2, is adopted in order to create the proper baseline for future optimizations and complexity enhancement.

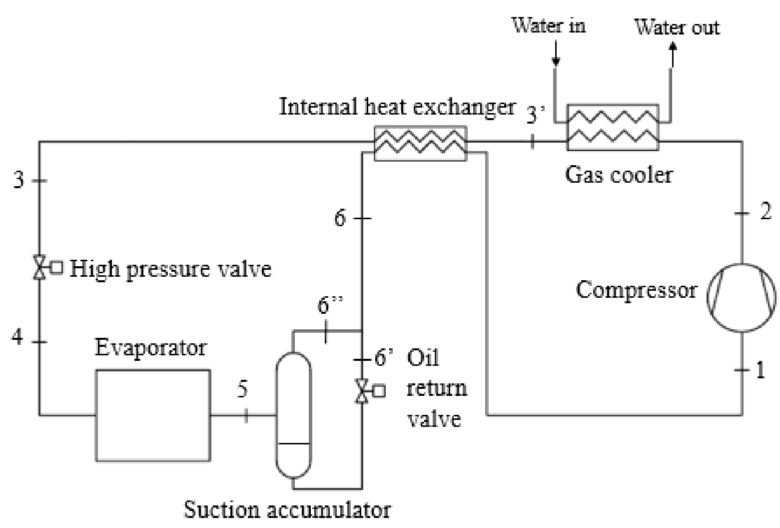

Figure 2: Baseline system layout. 1-2: compression at the compressor, 2-3': heat recovery at gas cooler (water heating), 3'-3: sub-cooling at internal heat exchanger, 3-4: expansion to evaporating pressure by expansion valve, 4-5: evaporation at the evaporator, 5-6" separation and suction of vapour refrigerant, 5-6' oil and liquid refrigerant return and superheating control by oil return valve, 6-1 superheating of suction gas at internal heat exchanger.

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2.2. Component design

For the design phase, alternatives were investigated as far as the components, but for the sake of brevity we will not go into detail on the selection process. Only the ultimate selection will be described.

The design of components is based on CO_2 properties and target performance data, which arise from competitors' benchmark, standard requirement of EN 16147 and ecodesign requirements. Relevant parameters for water heaters are therefore:

- Heating capacity: average heating capacity of the compressor to fulfil the heating up time requirements
- Heating up time: the time required to heat up DHW from 10 °C till the setpoint (target supply hot water)
- Supply hot water: set point of the heat pump
- COP: average COP during heating up cycle
- Tapping cycle profile: typical water usage model determined for a typical household. Five water tapping cycles, ranging from S – XXL, are defined by standard. These cycles vary in terms of the minimum volume of tap water use and the daily profile of its tapping cycle. The minimum duration of such a cycle is equal to 24 hours.
- V40: amount of withdrawn mixed water at 40 °C tapped from the water tank after heating up cycle.
- Efficiency class: water heating energy efficiency is calculated in accordance with ErP (European Commission, REGULATION 812/2013) according to the heat energy of tapped water and electrical energy consumption, determined after tapping cycle tests.

Table2. Target performance data

Parameter	Heating capacity [kW]	Heating up time [h]	Supply hot water [°C]	COP [-]	Profile [-]	V40 [L]	Efficiency Class [-]
Target value	1.5-2	6h30’	85	3	L	235	A+

Tanks and pump

System-wise, when operating in transcritical mode, the COP depends strongly on CO_2 gas cooler pressure level and the required outlet temperature and so water temperature level of the water to be heated, as highlighted in Figure 3. The target is, therefore to maintain and keep inlet water temperature as low as possible at all time. On the other hand, CO_2 HP units can supply high DHW temperature, due to the temperature profile in transcritical mode, enabling high-temperature water storage - meaning a higher V40 is achieved. For those reasons, in this application, two 90 litre tanks, hydraulically connected in series, were adopted to increase stratification, reaching the highest possible temperature gradient between the top of the hot water tank, where supply water is taken for users, and the bottom feeding with cold water, where water is circulated towards the gas cooler. The water can circulate by means of a micropump, typically used in other applications such as drink machines.

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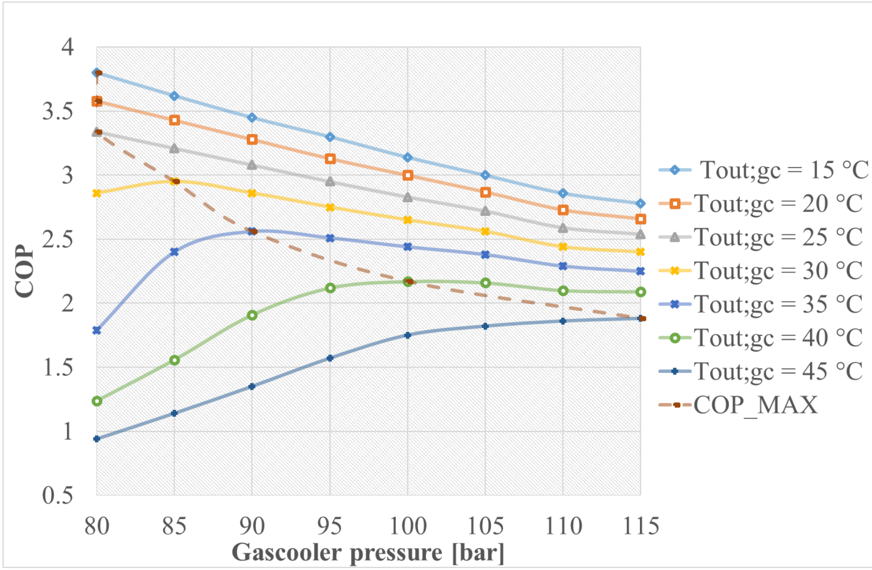

Figure 3: Trend of COP depending on gas cooler outlet temperature and pressure. Initial result of the experimental campaign and optimal COP is represented by the dashed line.

Gas cooler

The gas cooler (GC) is the component which rejects heat from the refrigerant to another medium. During the feasibility analysis, many gas cooler typologies were considered. A multipass plate heat exchanger was selected, where CO_2 directly exchanges heat with DHW. The heat exchanger has more passes in order to reach higher pressure drops and higher shear stresses, which is beneficial to improve heat exchange thanks to higher water velocity, which also enhances the cleaning of the heat exchanger channels from possible limestone formation. Three passages HX lead to higher CO_2 and hot water temperature using fewer plates since the temperature profile of both water and CO_2 is well match each other. The effect of the water outlet temperature was numerically analysed according to the variation of gas cooler pressure, water mass flow rates and inlet water temperature as shown in Figure 4.

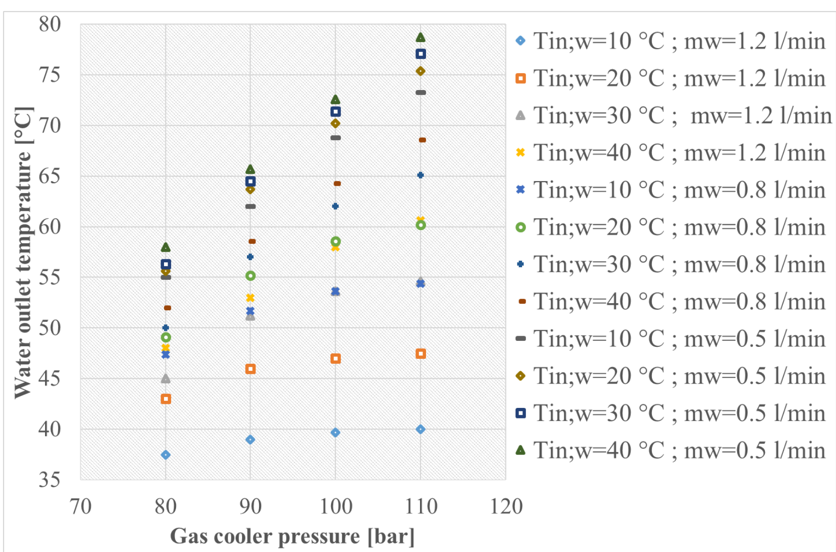

Figure 4: Trend of water outlet temperature depending on pressure and water inlet temperature with an intermediate mass flow rate.

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As a result, an intermediate mass flow rate of 0.5 l/min allows reaching the desired heating power and heating up time with the highest water outlet temperature without surpassing 90 °C to avoid vapour formation. Potential formation of limescale due to high water temperature and fouling on the water side of the gas cooler, has been limited by sizing it with a higher shear stress.

Internal heat exchanger

The useful effects of the internal heat exchanger (IHX) are to cool down CO_2 after the gas cooler and superheat CO_2 at the compressor suction. The IHX is practically a tube-in-tube heat exchanger, where hot high-pressure CO_2 flows internally and cold low-pressure CO_2 externally. From the colder side, It is also advantageous to perform the superheating by the IHX so that the evaporator can work at outlet quality lower than 1, maintaining a high heat transfer coefficient at the evaporator. The copper tube-in-tube typology results in the most suitable solution because it has a simple design with high reliability, availability on the market, possibility of customization and low cost.

Suction accumulator

The aim of an accumulator before the compressor is to separate liquid and gas phase CO_2 to ensure unit reliability. The designed accumulator has an inlet but two outlets: one at the top for gas suction and the other at the bottom for oil return. An additional demister is placed inside the accumulator close to the top outlet to avoid refrigerant droplets being sucked. Ideally, a mesh on the bottom would be required to avoid oil vortexes. Regarding the dimensions, the accumulator cross-section and height are properly chosen in combination with the accumulator refrigerant charge in order to get a standstill pressure at worse ambient conditions, i.e. lower than the design accumulator pressure. Moreover, it is fundamental to calculate the liquid level at different operating conditions to be sure that 30-40% of the refrigerant inside the accumulator is in liquid form.

Evaporator and fan

For the evaporator design, finned tube heat exchanger is selected. Simulations are extremely useful for finding the optimal coil dimension, fin spacing, vertical and horizontal tube spacing, and number of rows and circuits. As an outcome, the finned coil with 7 mm and 1 circuit allows a good compromise between heat load and temperature difference between evaporating temperature and outlet refrigerant temperature. Pressure drop in the heat exchangers working with CO_2 is generally negligible since a pressure drop of 1 bar leads to about a temperature decrease of 1-2 K. From the aeraulic point of view, an airflow rate of about 430 m^3/h is required that can be generated by a centrifugal fan with a 200 mm impeller diameter.

Compressor

The compressor is typically chosen according to its capacity, COP and operating range. The rolling piston double-stage compressor performs a heating capacity between 1.5 kW and 2 kW at following boundary conditions: 0 °C evaporating temperature, 90 bar gas cooler operating pressure, 32 °C gas cooler outlet temperature, 32 °C suction temperature.. The operating frequency is managed by the inverter. Considering 32 °C as fixed suction and gas cooler outlet temperatures, compressor COP varies from 4 (at 90 bar and intermediate rps) to 3 (at 110 bar and low rps) according to the manufacturer compressor performance table. This is sufficient to heat up water from 10 °C to 55 °C in less than 6h 30’. However, most CO_2 compressors with such small

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capacity come from the refrigeration market, and the main issue when adopting in heat pump application is the maximum evaporation temperature, causing an upper limit in operating air temperature. Also, the component cost is a challenging point since similar compressor technology with another natural refrigerant, such as propane, costs 4 times less. Another important lowlight in CO_2 compressors comes out: the lack of models available for heat pumps with a heating capacity of lower than 4 kW on the market.

2.3. Control logic development

High-Pressure valve control

The high-pressure valve is used to keep, by means of a PID control, a constant temperature approach of 2-4 K between CO_2 temperature at the gas cooler outlet and inlet water temperatures. This enhances the movement of the pinch point from the middle of the gas cooler to the outlet, improving heat transfer and so reducing the CO_2 outlet temperature with a beneficial effect on throttling losses and COP. By acting on the valve, it is possible to increase the pressure maintaining the design approach at the gas cooler end during the heating-up cycle, as shown in Figure 5.

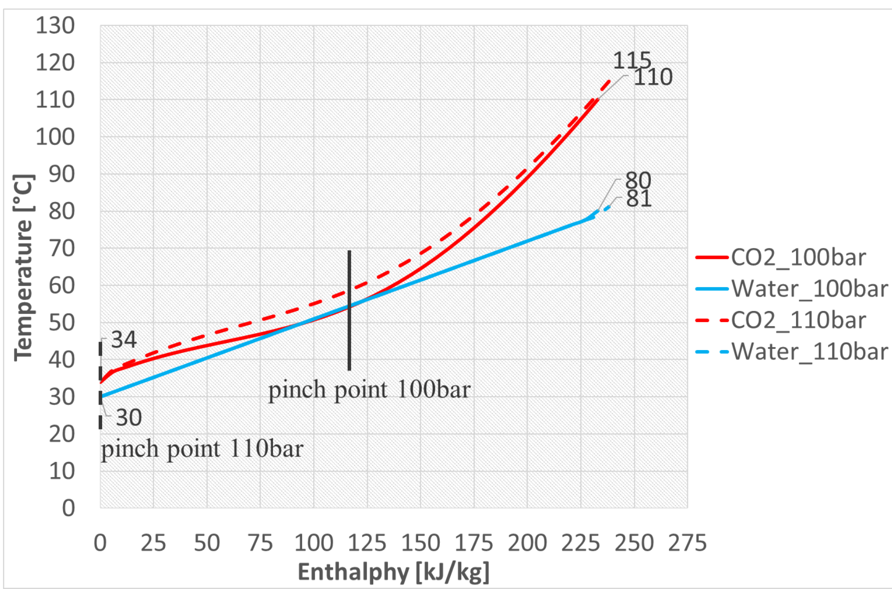

Figure 5: Shiftment of pitch point at the end of the gas cooler by increasing pressure from 100 to 110 bar when inlet water temperature is equal to 30 °C.

The unit stops running when the cold tank water temperature probe measures 40 °C. In this condition, according to the literature value, heat capacity reduces below 60% of the initial heat capacity at 10 °C as water inlet temperature. As a consequence, COP would decrease significantly.

However, from the performance point of view, it is not always convenient to operate according to the approach control for the entire cycle. Referring to previous Figure 5, it is possible to state that:

- By rising water inlet temperature, operating pressure must increase in order to both raise water outlet temperature and get the optimal COP as in Figure 3.

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- By increasing pressure, it comes to a point where outlet water temperature does not change significantly, meaning that a further pressure increase does not lead to a clear increase in water outlet temperature, and it would penalise COP.

These phenomena are linked to the non-linear profile of CO_2 temperature: on a T-s diagram, isobaric curves are steeper in first desuperheating and flatter near the critical point due to a line flex point (see Figure 1).

With this in mind, the following innovative control solutions are developed:

- A first control is applied in the beginning phase of the heating-up, and it consists of measuring the approach at the outlet of the GC, the GC pressure and outlet temperature and consequently acting on the high-pressure valve for the maximisation of COP in reaching a higher water temperature. This control is valid up until the inlet water temperature reaches a certain value (threshold temperature), and then a second control strategy is applied. Compressor rotational speed and water mass flow rate are fixed.
- After reaching the threshold temperature, a second control is applied to further enhance the outlet water temperature in as little time as possible, and it consists of applying the COP optimization function for the control of the working pressure. This function, reported as Eq. (1), derives from test measures by also keeping, in this case, compressor rotational speed and water mass flow rate, as shown in Figure 6.

$$p_{GC} = -0.0007(T_{w\_in})^3 + 0.0905(T_{w\_in})^2 - 2.1728(T_{w\_in}) + 93.69 \qquad \text{Eq. (1)}$$

Oil return and superheat valve control

While in the traditional cycle, the high-pressure expansion valve is used in superheat control, in the current design, the superheating is managed by the second stepper valve placed on the oil return. This valve can also be activated in case of oil return issues. Typically oils used for CO_2 are PAG and POE. Characteristics speaking, POE has a better oil return property and better miscibility but worse lubricant property. However, both accumulate at the bottom of the suction accumulator since above -20 °C, the density of PAG is greater than CO_2; in the case of POE, the limit is at -10 °C.

Compressor, pump and fan

According to the type of control, these components are managed as fixed speed reported in Figure 6.

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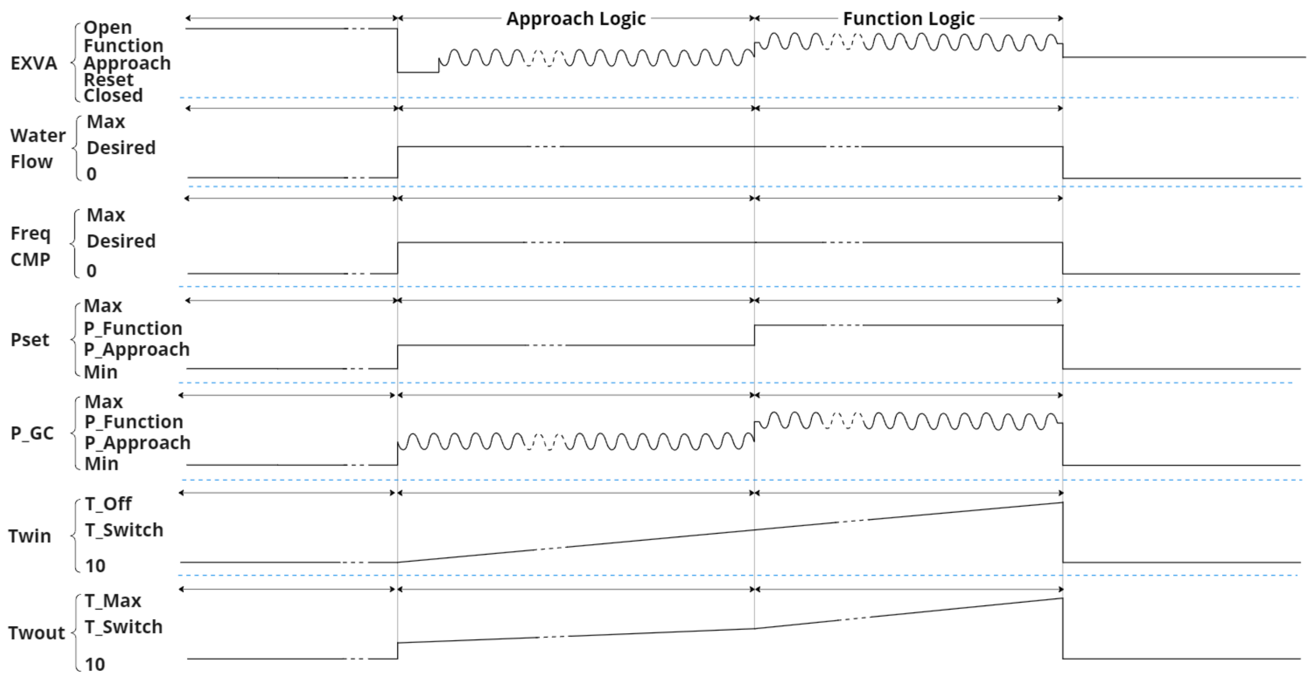

Figure 6: Innovative high-pressure control depending on water inlet temperature.

3. CONCLUSIONS

This study presents the design of a CO_2 heat pump for DHW. Thanks to the transcritical cycle, it is demonstrated that this system can supply up to 70 °C. A higher supply temperature can be reached at a higher inlet water temperature. The COP is shown to be greater than 3 at low pressure, low gas cooler outlet temperature and intermediate compressor frequency. However, due to the dynamic boundary condition of the system, i.e. progressively heating up of water tanks, the COP continuously decreases, and in order to reach the optimal COP, the pressure should increase.
According to this physical phenomenon, the design of control logic becomes extremely important in order to realise the most efficient unit. A two steps control of the high pressure valve is proposed: first according to the GC temperature approach, aimed to maximise the COP in reaching a reasonable water outlet temperature; then, after a defined water outlet temperature threshold, aimed to reach even higher water outlet temperature, while optimising the COP with a experimental control function.

Future studies will also investigate potential improvements in system design. As an example, a still valid technology as an alternative to a plate heat exchanger could be the wrapped D-shape tubes around the tank. This solution could also be advantageous from the maintenance and cost point of view since no risk of limescale and low component price. Another still interesting solution is the tube-in-tube gas cooler, which could lead to a higher uniform energy factor (UEF), as stated in (Kashif, Bo, Ahmed, & Van, 2017), thus more energy efficient and less operating costs. Advanced technology, such as ejectors will also be a topic for the next research activities. Finally, alternative solutions could be investigated on the hydraulics side. For instance, in order to improve stratification, a solution could foresee 4 tanks having a lower ratio between cross-section over height and in series connected; another one would be the addition of intermediate buffles in existing tanks. However, these solutions would cause higher unit costs, which can be counterbalanced by increasing market unit sales price and marginality.

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NOMENCLATURE

CMP	Compressor
COP	Coefficient of Performance
CO_2	Carbon dioxide
DHW	Domestic Hot Water
ErP	Energy related product
GC	Gas Cooler
GWP	Global warming potential
HP	Heat pump
IHX	Internal Heat Exchanger
UEF	Unified Energy Factor

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