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Vapour compression and caloric refrigeration in combination for a new hybrid refrigeration system

Laura NEBOT-ANDRÉS*[(a)], Manuel Gesù DEL DUCA[(b)], Ciro APREA[(b)], Jaka TUŠEK[(c)], Andrej ŽEROVNIK[(c)], ANGELO MAIORINO[(b)], Rodrigo LLOPIS[(a)]

[(a)] Jaume I University. Thermal Engineering Group. Mechanical Engineering and Construction Department.
Castellón de la Plana, 12071, Spain
[(b)] Università degli Studi di Salerno
Fisciano, 84084, Italy
[(c)] University of Ljubljana, Faculty of Mechanical Engineering
Ljubljana, 1000, Slovenia

*Corresponding author: lnebot@uji.es

ABSTRACT

Although caloric refrigeration systems, such as magnetic and elastocaloric ones have shown promising potential in the last decade, their application in stand-alone applications remains scarce. Namely, they can provide high COP values and sufficient cooling power but only at a low temperature span between the heat source and the heat sink. However, the performance of the caloric refrigeration systems drops significantly at higher temperature spans (above 30 K), which are usually required in most common refrigeration applications. This work introduces a new concept or future application that matches perfectly with the current state of development of caloric refrigeration technologies, which is their hybridization with vapour compression systems (VCS) through subcooling. In subcooling applications, the required temperature span is low, therefore, the caloric systems can be used to boost the operation of vapour compression refrigeration systems toward higher efficiencies. Based on experimentally obtained data of different prototypes (magnetic refrigeration system, elastocaloric refrigeration system and CO_2 transcritical refrigeration plant) the energy results and optimum operating conditions of the hybrid caloric-VCS systems will be presented.

Keywords: Hybrid cooling technology, Carbon Dioxide, Elastocaloric refrigeration, Magnetic refrigeration, Modelling, Subcooling

1. INTRODUCTION

Refrigeration, which is crucial for modern society, accounts for around 20% of world's energy consumption and was responsible of 7.8% of the greenhouse gas emissions in 2018 (Refrigeration, 2019)so great efforts are needed to reduce this negative impact of vapour-compression system (VCS). The agreements initiated in Europe (European Commission, 2014) and later extended to the rest of the World (Nations, 2016) want to reduce the usage of hydrofluorocarbon (HFC) substances as refrigerants with high or very-high global warming potential (GWP). The goal is to prevent 80 billion tonnes of CO_2 equivalent by 2050 and limit the global temperature rise below 2 K. To achieve these objectives, natural fluids (hydrocarbons (HC) and CO_2) are being used to reduce the direct effect of GWP but they have other shortcomings. HCs are flammable and their charge is limited, and CO_2 has low efficiency in warm and hot climates. To improve the drawbacks of CO_2, new components have been developed, such as ejectors, and new cycle configurations as the parallel compression system or integrated and dedicated mechanical subcooling systems.

Caloric solid-state refrigeration (CSSR) is a cutting-edge technology, in the process of development, based on the caloric effect of ferric materials. It is characterized by an isothermal entropy change or an adiabatic temperature change, which occurs in a solid-state caloric material upon being subjected to an external stimulus as uniaxial stress, an external field or pressure (IIR, 2022). Depending on the stimulus, the caloric cooling effects are classified in: magnetocaloric (MC), electrocaloric (EC), elastocaloric (eC), and barocaloric (BC). Research performed in caloric cooling technologies showed the high efficiency these systems can offer, quite higher than the one of vapour compression refrigeration (VCR) systems. However, the best performance of CSSR is obtained for low temperature spans, when the temperature difference between the focus is small (IIR, 2022). The main disadvantage of this technology is the small temperature span that is achieved and because of this the applications of this technology are very limited. Nevertheless, it is a very promising technology and efforts should be made to reduce its drawbacks in the future. In conclusion, the development of caloric technologies must follow a continuous effort to be made in parallel with the improvement of VCR systems.

The objective of this work is to show and evaluate the possibilities of merging CO_2 VCR technology with caloric refrigeration systems: magnetocaloric and elastocaloric technologies. We present the possibility of hybrid cooling cycles, where a basic CO_2 cycle is combined to a caloric refrigeration device to perform subcooling. Combining both technologies into a hybrid cooling system allows to transform the disadvantages of each technology into advantages, which leads to a more efficient and environmentally friendlier cooling system. The main objective of this work is to present initial theoretical investigation of the presented hybrid cooling systems and show the potential of improvement.

2. STATE OF CALORIC COOLING TECHNOLOGIES

Caloric cooling are solid-state technologies that do not use refrigerants so are considered to be zero GWP refrigeration technologies. Among them, MC is the most developed one with several prototypes developed and eC is the one with greater caloric effect. In addition to the environmental benefit, these technologies have great COP, close to the Carnot limit, above all for low temperature spans but the difference between the efficiency of these systems and the potential efficiency of caloric materials brings out the need for further research efforts to reach their full potential.

In this section both the state-of-the-art of both technologies is presented as well as their possible application as subcooling system for VCRs.

2.1. The eC technology

Elastocaloric cooling utilises the elastocaloric effect that occurs due to the generation of latent heat during direct and inverse stress-induced martensitic transformation in superelastic shape-memory materials (IIR, 2022). Shape memory materials have the special characteristic of being able to recover their original shape after deformation. Its uniqueness lies in the fact that when the force that causes the deformation disappears, its deformed shape is maintained, and it is necessary to raise the temperature of the material to return it to its original state.

The reason for these properties is the martensitic transformation that can be either temperature-induced (shape memory effect) or stress-induced (superelasticity). The recovery of the original shape with temperature is due to the change in the internal structure of the material. At low temperatures, these materials usually have a laminar and fibrillar arrangement, which allows easy deformation, as some layers can move relative to others; this is called the martensitic state (Figure 1). When heated, the material itself changes to a cubic arrangement, much more rigid, which no longer allows the material to move and therefore no longer allows deformations; this is called the austenitic state.

This transition from the lamellar structure to the cubic structure is strong enough to return the material to its original shape and undo all the deformations it has undergone. Once the temperature is lowered again, the material is once again arranged in a lamellar form and can be deformed again.

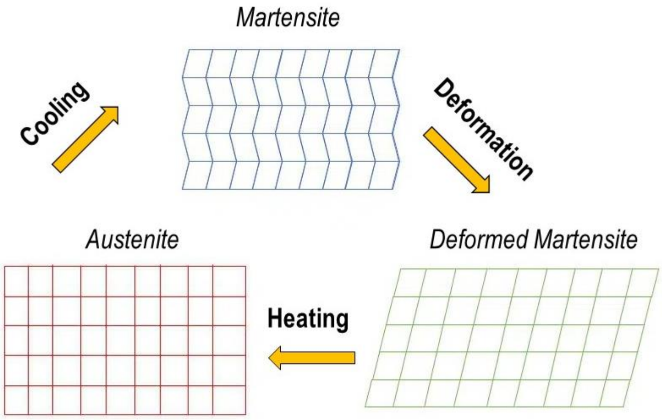


Figure 1. Phases of elastocaloric materials.

In the case of uniaxial compressive loading, as shown in Figure 2, superelastic/elastocaloric cycle is performed. When the external stress is applied, the material is subjected to elastic loading, initializing the martensitic transformation, observed as a change in the slope of the stress-strain curve. The transformation to martensite is an exothermic process leading to the generation of latent heat. In the second step of the cycle, this heated material transfers heat to the environment while compressing stress remains constant and so it starts to cool down. When the initial temperature is reached, the stress in the material decreases. When the compressive stress is removed, it is transformed back into austenite (endothermic process) that is cold and can absorb heat from the environment. The material returns to the initial state, so a continuous cyclic operation is allowed.

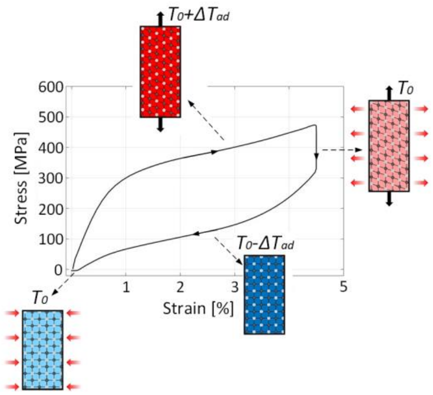


Figure 2. A schematic presentation of superelastic/elastocaloric cycle (Kabirifar et al., 2019).

The maximum temperature of the material is called the adiabatic temperature change (ΔT_{ad}^{+}) and the lower the negative adiabatic temperature change ($-\Delta T_{ad}^{-}$).

Figure 3 presents the prototype of an active elastocaloric regenerator (Tušek et al., 2016). It is a porous structure made of elastocaloric materials through which a heat transfer fluid in pumped in a counter flow direction between a heat sink and heat source. Figure 3 (left) shows a schematic representation of four basic operational steps of active eC regenerator, which are loading, fluid flow towards heat sink (hot external heat exchanger – HHEX), unloading and fluid flow towards heat source (cold external heat exchanger – CHEX). When those four operational steps are continuously repeated a temperature profile is getting established along the length of the regenerator as shown in Figure 3 (right).

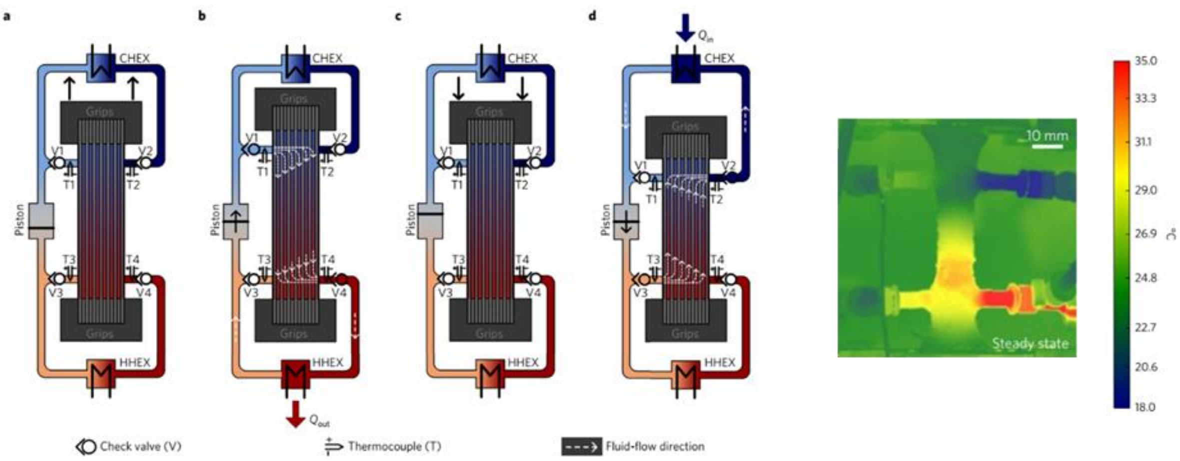


Figure 3. A schematic presentation of four basic operation steps of an active elastocaloric regenerator (left) and an IR image of the elastocaloric regenerator (right) (Tušek et al., 2016).

Based on the elastocaloric regenerator of Figure 3, the linear COP- q_c relations were constructed based on two characteristics points, where one is at zero specific cooling power, which corresponds to zero COP values, while the other was interpolated from the experimentally measured specific heating power – COP characteristics of the eC prototype (Tušek et al., 2016). Figure 4 shows the linear ΔT_{span} - q_c and COP- q_c relation for two operating frequencies. The operationg frequency is the . As it can be seen, when higher is the temperature span (Δt_{span}), lower is the specific cooling capacity and thus lower is the COP. To obtain high COP, only a short Δt_{span} can be provided. It is also seen that the COP is higher when higher is the frequency, but in this case the temperature span is lower.

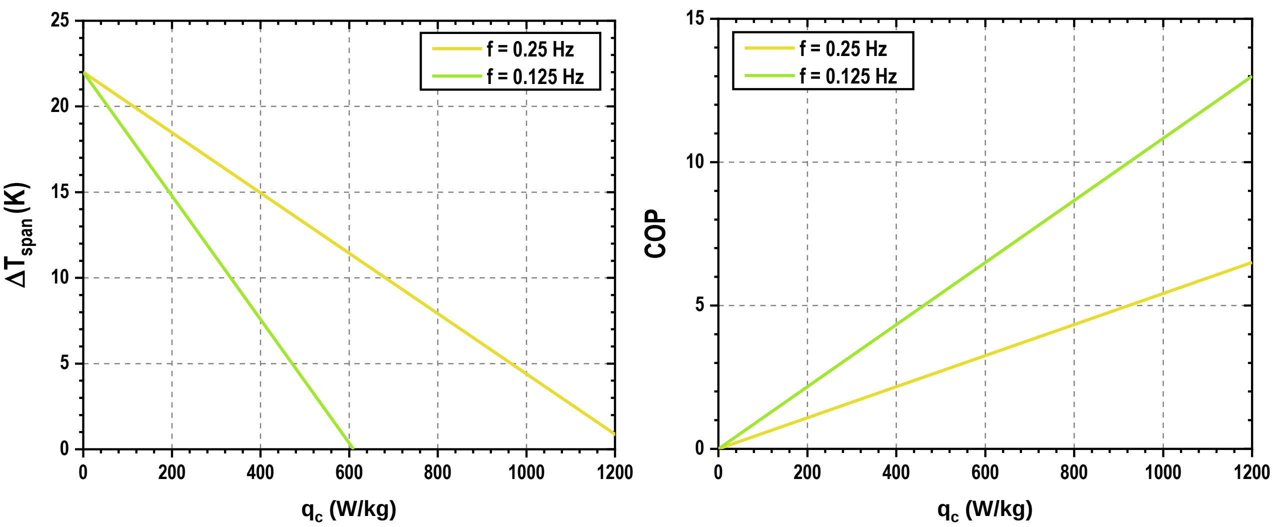


Figure 4. The linear dependence of the temperature span and the COP on the specific cooling power for two different operating frequencies.

2.2. The MC technology

Magnetocaloric was the most investigated caloric cooling technique until a few years ago when the other solid-state technologies became to be considered. Its principle obeys to the effect caused by a magnetic field on the materials that have the property of varying their magnetic entropy when the applied magnetic field is varied. This variation also produces a variation in the temperature of the material. It is then the thermal response when the material is subjected to a magnetic field change, that it's an intrinsic property of magnetic materials.

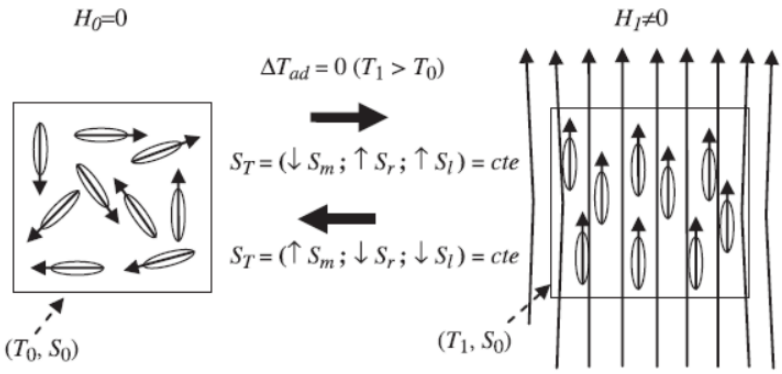


Figure 5. Arrangement of magnetic spin system of an adiabatic sample before and after applying a magnetic field.

An active magnetic refrigerator (AMR) cycle is based on four operational steps: an adiabatic magnetisation, an isofield cooling, an adiabatic demagnetisation and an isofield heating. In the first step, the magnetocaloric material in the AMR is exposed to the external magnetic field, which raises its temperature due to the MC effect. Then, while the magnetic field is maintained at a constant value, a fluid flows through the material (heat regenerating fluid), absorbing heat from it, which is dissipated in the hot heat exchanger. In the adiabatic demagnetisation step, the external magnetic field is removed and the magnetocaloric material cools down. In the final stage of the cooling cycle, without an external magnetic field, the fluid flows through the material in a counter current direction, transferring heat to the material. The fluid then absorbs heat in the cold heat exchanger. At this point, the process is repeated. To date, around 100 prototypes of magnetic chillers have been developed around the world (Greco et al., 2019). A MC heat pump prototype example was developed using NdFeB magnet (Dall'Olio et al., 2021) and could deliver 950 W heating power at 5.6 K temperature span. More recently, a MC water chiller prototype featured 900 W cooling at 18.3 K temperature span (Lionte et al., 2021). These performances are already sufficient for some applications.

Figure 6 shows the COP of a single-layer AMR working with Gadolinium as magnetic material (Del Duca, 2020). The COP is presented as a function of the temperature span, that is the difference between hot and cold sinks. Different COPs can be obtained depending on the fluid flow displacement ratio (V^*) of the regenerating fluid (heat transfer fluid).

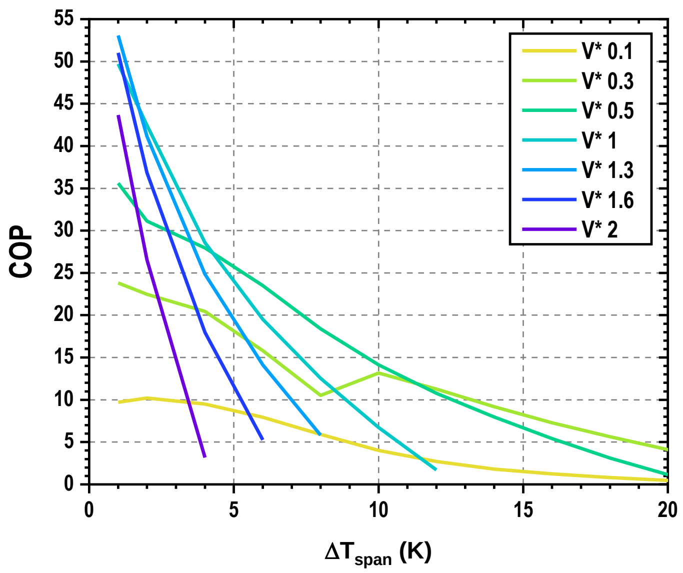


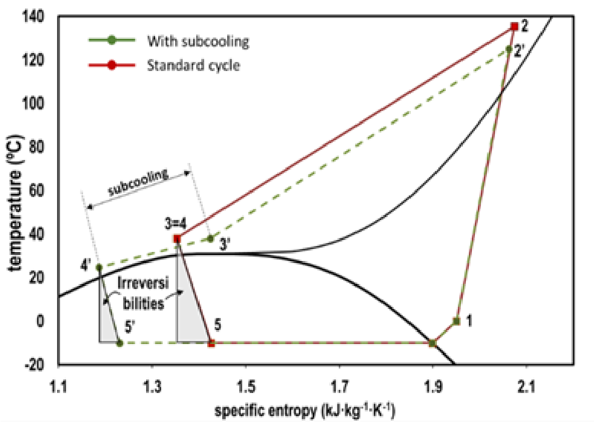
Figure 6. COP of the single-layer Gadolinium AMR as a function of the temperature span for different fluid flow displacement ratios (Del Duca, 2020).

As it can be seen, COP decreases when the temperature span increases and increasing the fluid flow displacement ratio leads to increase the maximum COP but reduces the achievable temperature span. This means that if we want to take advantage of the high COP values of this technology, we have to work with small temperature differences. This is why up to now MC refrigeration has almost only been used for small room temperature refrigerators.

2.3. Subcooling in CO_2 refrigeration system and its potential improvements

In general, the CO_2 VCR has low efficiency when the heat rejection occurs near the critical point, it is when the environment temperature is around 25ºC or more. Special features of the CO_2 force VCR systems be designed as more complex architectures in order to improve the efficiency of the systems.

As shown in Figure 7, the CO_2 refrigeration cycle through evaporation of the refrigerant absorbs energy (Q_o) from a cold source (t_c), then the compressor compresses the refrigerant to high pressure/temperature level, and further in the gas-cooler removes the heat (Q_{gs}) to the heat sink (t_h).



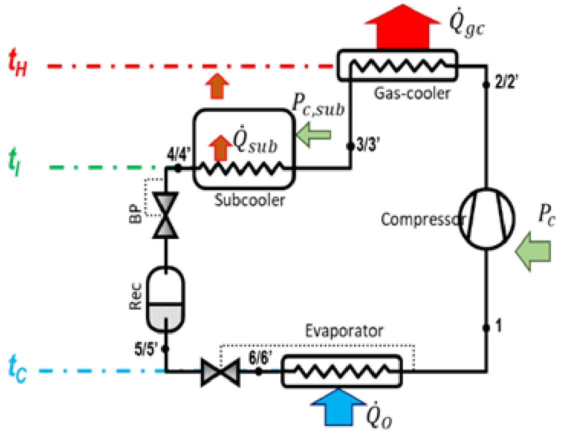


Figure 7. The CO_2 refrigeration cycle with subcooling.

One of the strategies being pursued today to improve CO_2 systems is CO_2 subcooling: cooling the CO_2 at the exit of the gas cooler (point 3 in Figure 7). Sub-cooling has been shown to provide greater benefits at higher ambient temperatures. Its main effects are to increase the cooling capacity and to reduce the optimum heat rejection pressure, the specific compression work and the vapour quality at the evaporator inlet, thus improving the overall energy performance of the CO_2 cycle (Llopis et al., 2018).

Subcooling can be performed easily using an internal heat exchanger (IHX) as presented by Aprea and Maiorino (2008), reaching COP 10% higher. It can also be performed by a dedicated mechanical subcooling, which is a secondary VCR that performs the subcooling and provides increments in COP up to 30.3% (Llopis et al., 2016). An integrated subcooling has been studied too, using the own CO_2 to subcool itself. Nebot-Andrés et al. (2022a) has measured improvements up to 9.5% with respect to the parallel compression cycle. Subcooling methods based on VCR can operate with elevated COP values and large temperature spans between heat sink and cold sources. However, the VCR technology is not designed to the working conditions required for the subcooling process. To obtain the optimum conditions of the subcooling, frequently the auxiliary VCR works out of the operating range of the compressor or operates at compression ratios below 1.5, which is also out of the compressor's manufacturer recommendations (Nebot-Andrés et al., 2021b).

Other auxiliary systems can be potentially applied as subcooling methods for CO_2 refrigeration cycles. For example, solid-state refrigeration technologies can be considered as highly promising candidates for the subcooling auxiliary refrigeration system. They are particularly interesting due to their high efficiency and because they are zero GWP so have negligible overall environmental impact. Thermoelectric (TE) cooling has already been tested as subcooling device achieving relatively high COP values when working at low temperature spans. For example, Astrain et al. (2019) applied an optimized model using which they have predicted that the COP of the TE-VCR hybrid system can be improved for up to 20% compared to VCR only. Furthermore, optimisation analysis on transcritical refrigeration cycle coupled with TE subcooler has shown an improvement in COP and second law efficiency. The good results obtained for this technology leave the door open to testing other solid-state technologies such as magnetocaloric and elastocaloric cooling that can be potentially applied as the subcooling auxiliary refrigeration systems.

3. PROPOSED HYBRIDIZED REFRIGERATION TECHNOLOGY

Both caloric technologies have the inconvenient of the small temperature span they can offer. Therefore, instead of trying to extend this temperature span (penalising performance), we should look for new applications. It has been seen that some very specific applications exist, but this work identifies the possibility of a new application: subcooling the CO_2 in refrigeration plants. It has been shown that subcooling of CO_2 within the VCR cycle enhances performance of CO_2. Since the subcooling requires relatively low temperature spans, applying another VCR system, which are generally not designed to work efficiently at low temperature spans, as a subcooling method is not preferable. The working conditions of the caloric systems fit perfectly with the conditions necessary to carry out this subcooling. That is:

- It has been identified that there is an optimum subcooling degree for transcritical CO_2, which for temperatures from 25ºC to 40ºC is between 10 K and 20 K (Nebot-Andrés et al., 2021a), so it can be affordable for caloric refrigerators.
- In a dedicated subcooling, the temperature between the sinks is lower than the temperature difference where the refrigeration cycle is working. The subcooling system works between the environment temperature and a temperature near to the gas-cooler exit temperature (where the subcooling is performed).
- To ensure that the subcooling system enhances the overall COP, the COP of the auxiliary system (the caloric one in this case) should be larger than the COP of the CO_2 without subcooling (Llopis et al., 2018).

Both need and additional heat transfer fluid, that can be water, to exchange the caloric thermal effect, because caloric materials cannot be pumped. This fluid can be water which is also harmfulness. The heat transfer fluid will be the agent that connects thermally both cycles (the CO_2 and the solid-state one) through a heat exchanger called subcooler. Figure 8 shows the schematic view of the eC or MC subcooling cycle with the main temperatures that must be considered. The regenerating fluid performs heat rejection to the environment and cools down the CO_2 through the subcooler (points 3 to 4).

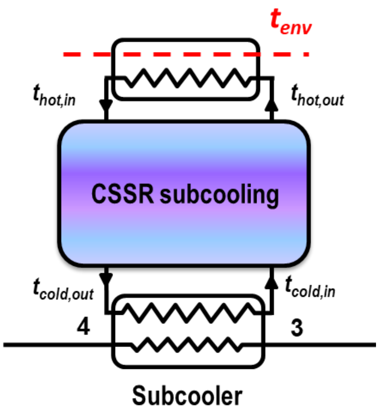


Figure 8. Schematic representation of the temperatures in the solid-state subcooling system.

The hot inlet temperature ($t_{hot,in}$) to the CSSR device is the environment temperature plus an approach (Eq. (1)). The $t_{cold,out}$ has to be calculated to ensure that the CO_2 can be subcooled until t_4, so an approach is also considered (Eq. (2)). The temperature span (Δt_{span}) of the solid-state technologies is defined as the difference between the hot side outlet temperature and the cold side outlet.

$$t_{hot,\mathrm{in}} = t_{env} + \Delta t_{app,eC} \qquad \text{Eq. (1)}$$

$$t_{cold,\mathrm{out}} = t_4 - \Delta t_{app,eC} \qquad \text{Eq. (2)}$$

$$\Delta t_{span} = t_{hot,\mathrm{out}} - t_{cold,\mathrm{out}} \qquad \text{Eq. (3)}$$

The subcooling degree (SUB, Eq. (4)) is then directly related with the environment temperature and the Δt_{span}.

$$SUB = t_3 - t_4 \qquad \text{Eq. (4)}$$

The solid-state refrigerator can be applied to a transcritical CO_2 refrigeration cycle, as shown in Figure 9. The auxiliar system is considered to perform the subcooling. The CO_2 system performs the expansion process in two stages with a liquid receiver in between both (points 4 to 7). The first expansion device is a back-pressure valve (BP, points 4 to 5) and controls the discharge pressure, while the second expansion valve (EXV, points 6 to 7) controls the evaporating process. The liquid receiver is placed between both valves (points 5 to 6). Subcooling is performed in the subcooler (points 3 to 4) placed after the gas-cooler (2 to 3), which thermally connects the CO_2 cycle and the heat transfer fluid of the caloric device (both the MC and the eC).

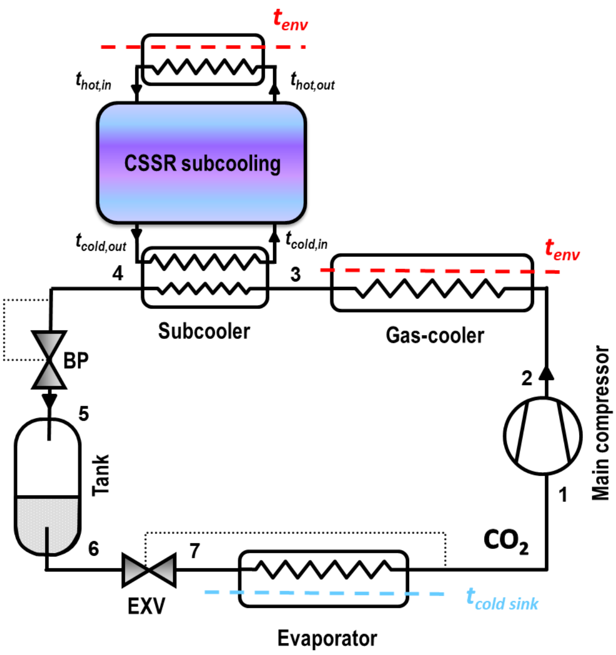


Figure 9. Schematic presentation of a hybrid CSSR-VCR cooling system.

Considering the system of Figure 9, the cooling capacity of the overall system is calculated as Eq. (5) considering isenthalpic expansion and the overall COP (Eq. (6)) is calculated taking into account the power consumption of the CO_2 compressor and the power consumption of the MC device (Eq. (7)).

$$\dot{Q}_0 = \dot{m}_{CO_2} \cdot (h_1 - h_7) = \dot{m}_{CO_2} \cdot (h_1 - h_4) \qquad \text{Eq. (5)}$$

$$\mathrm{COP} = \frac{\dot{Q}_0}{Pc_{CO_2} + Pc_{MC}} \qquad \text{Eq. (6)}$$

$$Pc_{MC} = \frac{\dot{Q}_{SUB}}{COP_{MC}} \quad \text{Eq. (7)}$$

Where $\dot{Q}_{SUB}$ are the cooling needs in the subcooler, calculated as:

$$\dot{Q}_{\mathrm{SUB}} = \dot{\mathrm{m}}_{\mathrm{CO_2}} \cdot (h_3 - h_4) \quad \text{Eq. (8)}$$

4. POTENTIAL IMPROVEMENTS

An hybrid system has been considered based on numerical models validated by experimental data of a transcritical CO_2 plant (Nebot-Andrés et al., 2020) and a magnetic refrigerator prototype, named 8MAG (Aprea et al., 2014), considering the structure presented in Figure 9. The considered device is a rotary permanent magnet magnetic refrigerator where the magnetocaloric material is stationary, and the magnet is rotating. It is characterized by an octagonal shape of the magnetic system and a total number of 8 packed-bed AMRs, supported by an aluminium structure (or magnetocaloric wheel) with 45° spacing.

The overall system is evaluated through an artificial neural network model (ANN) simulating the behaviour of the coupling searching the MC maximum performance ($COP_{MC,max}$). The simulation is run optimizing the main operational parameters:

- Gas-cooler pressure of the CO_2 refrigeration cycle must be optimized to obtain the highest possible overall COP.
- Subcooling degree should also be optimized. It is directly related to the increment in cooling capacity of the CO_2 but also to the Δt_{span} which is directly related to the COP_{MC}.

The overall COP is evaluated for environment temperatures between 25 ºC and 35ºC and several benefits were observed. Regarding the optimum pressure, reductions up to 2 bar were obtained (Table 1). COP of the overall system is always higher than the COP of the pure CO_2 cycle as it can be observed in Figure 10 left, where COP of each device and the hybrid one is presented. Comparing both COP (Figure 10 right) an improvement between 7.5-9% is obtained for all the evaluated conditions. The improvement is more important at temperatures above 28 ºC, where CO_2 is more penalized due to the transcritical operation.

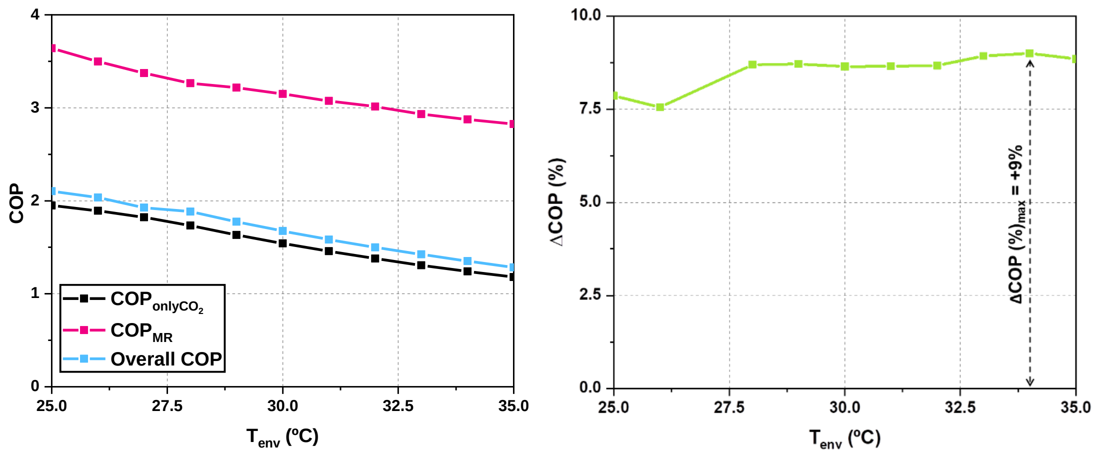


Figure 10. Increments in COP with respect to base system (Nebot-Andrés et al., 2022b).

Table 1. Pressure reduction.

T_{env} (ºC)	25	26	27	28	29	30	31	32	33	34	35	36
ΔP (bar)	0.0	0.0	0.0	0.0	0.8	1.2	1.2	1.4	1.4	1.2	2.4	2.2

5. CONCLUSIONS

CO_2 cooling technologies have been extensively developed in recent years due to regulatory mandates to use more environmentally friendly fluids and also with the design of better energy performance devices. Even so, VCS have reached a point where improvements are becoming increasingly costly to achieve. One of the most promising methods to improve CO_2 systems is subcooling. This can be realized with an additional VCR system reaching important increments in COP and cooling capacity.

On the other hand, CSSR technologies, which are still underdeveloped, present ideal characteristics for the energy transition as they do not use harmful fluids. The problem with these technologies is that in their current state, they are not able to offer large enough temperature spans to be used in the most common applications.

However, this paper proposes a solution that hybridises both technologies, to take advantage of the weaknesses of each and turn them into positive features. CSSR systems can be used as an external method for CO_2 subcooling to improve the energy performance of the refrigeration system. This hybridisation is possible because the subcooling system must work between two temperatures much closer (environment temperature and gas-cooler outlet temperature), where the CSSR system can offer larger COPs. The possibility of combining elastocaloric and magnetocaloric systems, of which some prototypes have already been developed, has been presented.

Some initial calculations of a transcritical CO_2 system thermally connected to an MC refrigerator show the potential of this hybridisation, improving the COP of the overall system by up to 9%.

This study shows the need for further improvement of CSSR systems and proposes a very interesting possible application for them. In addition, in general, transcritical CO_2 VCR introduces high energy irreversibilities in the expansion system, so recovering the energy from the expansion process to activate the CSSR system may also be a possible line of future research.

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NOMENCLATURE

ad	adiabatic
ANN	artificial neural network
AMR	active magnetic refrigerator
app	approach
BC	barocaloric
BP	back-pressure valve
cold	cold side of the heat transfer fluid
COP	coefficient of performance
CSSR	caloric solid-state refrigerator
eC	elastocaloric
EC	electrocaloric
env	environment
EXV	electronic expansion valve
f	frequency (Hz)
GWP	global warming potential
h	enthalpy (kJ/kg)
HC	hydrocarbons
HFC	hydrofluorocarbon
hot	hot side of the heat transfer fluid
$\dot{m}$	mass flow (kg/s)
MC	magnetocaloric
Pc	power consumption (kW)
$\dot{Q}$	cooling capacity (kW)
$\dot{q}_c$	specific cooling capacity (W/kg)
SUB	Subcooling degree (K)
t	temperature (ºC)
TE	thermoelectric
T_{span}	temperature difference between sinks (K)
$V*$	fluid flow displacement ratio
VCR	vapour compression refrigeration
VCS	vapour compression systems
Δ	increment/difference

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