

Article

Multicaloric Coupling Strategies in Ferroic Materials: Sequential vs. Simultaneous Activation for Solid-State Heat Pumping

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Abstract

Solid-state heat pumps based on caloric effects are emerging as a promising alternative to conventional vapor compression systems owing to their use of solid refrigerants with zero global warming potential. However, single-effect caloric technologies are intrinsically limited by the temperature-dependent nature of the caloric response, which typically exhibits a peak adiabatic temperature change within a narrow temperature range. In this context, multicaloric approaches offer a promising pathway to enhance thermal performance by combining multiple external fields. This work focuses on the comparison between simultaneous and sequential (cascade) multicaloric operation, with particular attention to the interaction between field application and the temperature-dependent caloric behavior of the material. A finite element model is developed to investigate a multicaloric solid-state heat pump operating in the air conditioning temperature range. A representative material is considered: $\text{Mn}_{0.6}\text{Ni}_{0.6}\text{Fe}_{0.4}\text{Co}_{0.4}\text{Si}_{0.95}\text{Ga}_{0.05}$, characterized by distinct magnetocaloric and barocaloric responses occurring at different temperature ranges. The analysis explores different field application strategies, including both simultaneous and sequential configurations. The preliminary results suggest that simultaneous multicaloric operation enables a more effective exploitation of the caloric response by aligning the field activation with temperature regions closer to the corresponding peaks. In this framework, cascade strategies appear to offer additional flexibility in tuning system performance under realistic operating conditions. The proposed approach provides a new perspective for the design of multicaloric heat pumps, highlighting the potential role of thermodynamic matching between field activation and material response. Ongoing work is focused on further quantifying these effects and identifying optimal operating conditions.



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Keywords: multicaloric effect; numerical simulation

1. Introduction

The effort to reduce carbon emissions in the heating, ventilation, and air conditioning (HVAC) industry has shifted from a long-range environmental aspiration to a pressing geopolitical and economic requirement [1–5]. Presently, heating and cooling activities make up about 20% of worldwide electricity use, and the dependence on conventional vapor compression (VC) systems creates a dual challenge [6]. While the demand for climate control is on the rise, the artificial refrigerants needed for VC, like HFCs [7,8] and HFOs [9,10],

face significant regulatory pressure [11,12] due to their remaining global warming potential (GWP) and risks to the ozone layer [13], as established by the Montreal Protocol and the following Kigali Amendment [14,15]. In this context, solid-state caloric technologies characterize a significant transformation [16]. These systems employ solid-state refrigerants that have a global warming potential (GWP) of zero [17], effectively separating the cooling-process-related carbon dioxide emissions from the ones due to harmful gas. In addition to their environmental benefits, caloric heat pumps could theoretically achieve efficiencies that are 50–60% greater than traditional vapor compression (VC) systems, leading to a notable reduction in the indirect carbon footprint linked to electricity generation [6,18–20]. The fundamental principle underlying caloric refrigeration is the caloric effect, which is a physical phenomenon observed in certain solid-state materials [21]. This effect is characterized by a change in entropy ΔS_T (during isothermal transformation) or a change in temperature ΔT_{ad} (during adiabatic transformation) resulting from a variation in the intensity of an external field applied to these materials. Nevertheless, despite the development of more than 150 single-effect prototypes [22]—spanning magnetocaloric [23,24], electrocaloric [25,26], and elastocaloric [27–29] technologies—the pathway to market adoption is hindered by inherent material constraints [30–32]. The majority of caloric materials demonstrate a “giant” effect solely within a very limited temperature range, predominantly around a first-order phase transition. To manifest a substantial caloric effect or cooling capacity, these systems often necessitate excessively high intensity on the external forcing field—such as magnetic fields exceeding 2 T or mechanical stresses over 700 MPa—which adversely affects both their cost efficiency and structural integrity [33,34].

To address the limitations of single-effect systems, the scientific community is increasingly shifting its focus to multicaloric refrigeration [35]. This technique utilizes the multiple ferroic or structural orders [36] present in a single material to permit the simultaneous or successive application of different external fields, such as magnetic and barocaloric [37–40]. The potential of multicaloric systems is not merely due to the combined effects of these phenomena but also because they provide the ability to adjust the thermal response [41]. By altering multiple field parameters, it becomes feasible to modify the effective temperature range of a material or to reduce the strength of any individual field needed, thus prolonging the operational life of the regenerator [42,43]. By utilizing a multicaloric system [44], one can take advantage of several benefits: (i) lowering the strength of the individual fields required, thereby saving costs and enhancing energy efficiency [35]; (ii) merging various forward caloric effects to broaden the temperature range and increase useful thermal power; (iii) implementing simultaneous external fields of varying types to produce both forward and inverse caloric effects, which enables, through the multicaloric effect, a shift in the temperature ranges where the material exhibits a significant adiabatic temperature change [45].

Despite the growing interest in multicaloric refrigeration, the literature still lacks application-oriented studies addressing multicaloric systems at the heat pump level. Most of the currently available contributions mainly focus on materials and caloric characterization, while the system-level implications of multicaloric operation remain largely unexplored. In particular, no comprehensive numerical investigations have clarified how the mode of field application affects the performance of a multicaloric solid-state heat pump, especially when comparing sequential and simultaneous activation of the coupled caloric effects.

Within this framework, the present work, within the purposes of the MULTICAL project, aims to analyze the influence of multicaloric effects applied either sequentially or simultaneously in an active multicaloric regenerative cycle. To this end, a finite element model is developed to investigate the thermal behavior and energy performance of a

multicaloric solid-state heat pump operating in the air conditioning temperature range. This study is intended to provide a first application-oriented assessment of how different multicaloric activation strategies affect temperature span, cooling performance and overall system operation.

2. Multicaloric Effect and Multicaloric Material

The multicaloric effect (mCE) is generated by the simultaneous or sequential application of two or more fields, which appears to exhibit better performance than the single caloric effect [46]. Multicaloric materials are defined as materials that possess at least one type of ferroic order and exhibit a caloric effect in response to multiple types of external fields [47]. The literature reports many types of multicaloric cooling and multicaloric composites in which interfacial coupling via stress transduction is used to pump heat. Multicaloric cooling processes have been classified into four main categories [46], depending on whether a material is single-phase or composite and whether a single or multiple fields are applied. Careful design is necessary to achieve additive rather than subtractive effects, and, furthermore, the magnitude of the caloric effect varies with the intensity of the applied field.

The MULTICAL project centers on MC materials that exhibit ferroic orders close to a first-order transition when influenced by various external fields. In particular, this study of the MULTICAL heat pump presented in this paper focuses on multiple fields, acting on single-phase material. The multicaloric system operates within the active multicaloric regenerative refrigeration (AmR) cycle, which consists of four stages:

- (1) Increase in the fields from a minimum to a maximum value in adiabatic conditions, which determines an increase in the temperature of the multicaloric material due to the caloric effect.
- (2) Heat transfer fluid flowing from cold heat exchanger (CHEX) to hot heat exchanger (HHEX) through the use of a pump. The fluid heats up as it passes through the material and releases the heat to the HHEX, whereas the caloric material cools down.
- (3) Adiabatic removal of the fields, further lowering the temperature of the caloric material due to the caloric effect.
- (4) Passage of the fluid from HHEX and CHEX, which, by exchanging heat with the caloric regenerator, cools down and absorbs heat from CHEX, thus realizing the desired cooling effect of the cycle if the heat pump works in cooling mode.

In this arrangement, fields are applied in two scenarios, either sequentially or simultaneously, to maximize energy efficiency for cooling capacity and temperature changes on the cold side of the regenerator. The material under investigation was $\text{Mn}_{0.6}\text{Ni}_{0.6}\text{Fe}_{0.4}\text{Co}_{0.4}\text{Si}_{0.95}\text{Ga}_{0.05}$, which exhibits both the magnetocaloric effect and the inverse barocaloric effect. The magnetocaloric effect occurs in the range from 295 K to 315 K, with a peak of 10 K at a magnetic field of 2 T. In contrast, the barocaloric effect occurs in the range from 270 K to 300 K, with a peak of 17 K at a hydrostatic pressure of 2.5 kbar [48].

By leveraging these different application ranges, it is possible to use the multicaloric effect to operate the heat pump over a 15 K temperature difference, which would be impractical if we relied on the individual effects alone.

3. Geometry of the Multicaloric Regenerator and Validation

The regenerator consisted of a stack of 54 multicaloric plates, encased in $H \times L$ packing measuring $20 \times 45 \text{ mm}^2$, and included channels for fluid flow. The thickness of each channel was indicated by the 0.125 mm gap between two solid plates, each with a thickness of 0.25 mm. A parallel-plate geometry was selected to ensure compatibility with

mechanocaloric and electrocaloric applications, while recognizing that magnetocaloric systems have fewer constraints. Utilizing a parallel-plate geometry facilitated the connection of electrodes necessary for applying a mechanical field through uniaxial tension and generating an electric field in an actual setup.

Moreover, the dimensions of the regenerator were determined based on those of an experimental prototype, which was utilized to validate the numerical model, and the number of plates was chosen to ensure the same volume ratio of the prototype [49,50]. The validation of the 2D numerical model was accomplished by replicating a single regenerator from the 8Mag prototype and comparing it with experimental data obtained from the 8Mag systems during zero-load and fixed-load tests [50]. Although the proposed model generally overestimates the results, it effectively mirrors performance trends, with an average error of 16% observed in fixed-load tests. This discrepancy is attributed to unaccounted-for experimental phenomena such as eddy currents, heat losses, and inactive volumes, as detailed in the model [49].

4. The Numerical Model

The investigation was conducted numerically through the development of a numerical model and solved based on the finite element method. The supporting mathematical models is represented by Navier–Stokes for the fluid and energy equations for the fluid and the solid media:

$$\left\{ \begin{array}{l} \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \\ \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho_f} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \\ \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho_f} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \\ \frac{\partial T_f}{\partial t} + u \frac{\partial T_f}{\partial x} + v \frac{\partial T_f}{\partial y} = \frac{k_f}{(\rho_f C_{f,p})} \left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} \right) \\ (\rho_s C_{s,p}) \frac{\partial T_s}{\partial t} = k_s \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} \right) + \dot{Q}'''_{MULTI} \end{array} \right. \quad (1)$$

The terms u and v represent the values along x and y of the velocity vector of the fluid, ρ_f and ρ_s are respectively the density of the fluid and the caloric solid, p is the pressure of the fluid, ν is the kinematic viscosity of the fluid, $C_{f,p}$ and $C_{s,p}$ are the specific heat of the fluid and the solid at constant pressure, k_f and k_s are the thermal conductivity of the fluid and the solid, T_s and T_f are the temperatures of the solid and the fluid.

In the system in Equation (1), in order, the equation of mass continuity, the two equations of momentum, and the equations of energy for fluid and solid are listed. In addition, the laminar fluid hypothesis is exploited, given the low flow rates, and therefore it is not necessary to use the equations for turbulent flows. Finally, the fluid is not in phase transition, and the surface tension is zero; in this situation, the solid is always be surrounded by the auxiliary fluid. The AmR cycle is simulated as a repeating sequence of 4 stages, and different boundary conditions are applied to each phase of the cycle, as analyzed below:

- The outer boundary of the regenerator is modeled as adiabatic:

$$-\mathbf{n} \cdot (k \nabla T) = 0 \quad (2)$$

- Fluid flow from cold to hot heat exchanger with velocity vector with positive sign. At the fluid-domain boundaries in contact with the cold heat exchanger, an inlet condition is imposed at fixed velocity and temperature $T = T_{cold}$, whereas the remaining regenerator walls are adiabatic.

- Fluid flow from hot to cold heat exchanger with velocity vector with negative sign. At the fluid-domain boundaries in contact with the hot heat exchanger, an inlet condition is imposed at fixed velocity and temperature $T = T_{hot}$, whereas the remaining regenerator walls are adiabatic.

The volumetric multicaloric heat source term, $\dot{Q}'''_{MULTI}$, implemented in the solid energy equation, represents the power density generated or absorbed during the forcing field variations. Depending on the specific field activation strategy explored in the simulation cycle, this generic term is replaced as follows:

$$\dot{Q}'''_{MULTI}(x, y, t) = \dot{Q}'''_{MULTI, sum}(x, y, t) = \dot{Q}'''_{field, 1} + \dot{Q}'''_{field, 2} = \left| \dot{Q}'''_{baro}(T(x, y)) \right| + \left| \dot{Q}'''_{magn}(T(x, y)) \right| \quad (3)$$

$$\dot{Q}'''_{MULTI}(x, y, t) = \dot{Q}'''_{MULTI, seq}(x, y, t) = \begin{cases} \dot{Q}'''_{baro}(T(x, y)) & \text{for } 0 \leq t \leq \frac{t_{field}}{2} \\ \dot{Q}'''_{magn}(T(x, y)) & \text{for } \frac{t_{field}}{2} < t \leq t_{field} \end{cases} \quad (4)$$

$$\dot{Q}'''_{field, i} = \frac{\rho_s C_{s, p} \Delta T_{ad, i}}{\Delta t_{field}} \quad (5)$$

As visible in Equations (1)–(5), multicaloric effects are accounted for as power density balanced in the energy equation of the solid as heat sources that are positive or negative depending on if the field variations determine increasing or decreasing temperature.

The simulations were performed under two scenarios:

- The multicaloric effect is determined by the sum of the effects, as shown in Equation (3);
- The multicaloric effect is determined by the sequential application of the two fields, where the first (barocaloric) is applied in the time interval $[0, \frac{t_{field}}{2}]$, while the second (magnetocaloric) is applied in the remaining time interval $[\frac{t_{field}}{2}, t_{field}]$ (Equation (4)).

The model was implemented through COMSOL 6.4 Multiphysics software and was solved using the finite element method (FEM). The FEM involves the discretization of the domain, which was performed using a customized mesh composed of quadrilateral elements. Additionally, the fluid domain was partitioned into specific layers to ensure an accurate characterization of the boundary layer.

In the simulations, the four steps of the active multicaloric regenerative cycles were repeated cyclically according to the above-introduced mathematical model until meeting the cyclicity criterion:

$$\delta = \max \left\{ T(x, y, 0 + nt_{cycle}) - T(x, y, t_{cycle} + nt_{cycle}) \right\} < \bar{\epsilon} \quad (6)$$

To ensure that the numerical solution was independent of the spatial discretization, a comprehensive grid independence study was performed. Simulations were carried out across different mesh configurations while keeping all initial, operational, and boundary conditions strictly identical. The impact of mesh density on the model's accuracy was evaluated alongside the convergence threshold to determine the optimal number of elements that balanced high precision with reasonable computational cost. Three distinct structured mesh configurations were tested: a coarse grid featuring 10,200 elements, a medium grid with 16,600 elements, and a fine grid comprising 40,000 volumetric elements. Table 1 reports the stabilized cold-side temperature at the end of the multicaloric cycle (sequential mode) for each mesh configuration, evaluated at a fluid velocity of 0.2 m s^{-1} and an operating frequency of 0.5 Hz. The comparison revealed that the temperature divergence between the medium and the finest mesh was extremely marginal, with a maximum deviation of just 0.01 K. Therefore, to optimize the computational efficiency and significantly reduce processing times without sacrificing numerical accuracy, a grid resolution of approximately 20,000 elements was selected for all subsequent simulation campaigns.

Table 1. Discretization settings and grid convergence analysis of model domain.

Mesh Type	Number of Volume Elements	Velocity [m s ^{−1}]	Frequency [Hz]	T _{cold,water} [K]	Maximum Divergence [K]
Coarse	10,200	0.2	0.5	291.5	0.02
Normal	16,600	0.2	0.5	291.18	0.01
Dense	40,000	0.2	0.5	291	-

Furthermore, the baseline adiabatic temperature changes used in this study were derived via Maxwell relations starting from the experimental isothermal entropy data. To account for the potential overestimations or tracking artifacts that the Maxwell formalisms can introduce near sharp first-order transitions, the simulation was intentionally performed on a virtual, idealized material framework. Consequently, the reported metrics represent an upper-bound potential of the system configuration, providing a fundamental baseline for screening multicaloric field-coupling mechanisms.

5. Working Conditions

As described in the preceding paragraphs, the multicaloric material was tested in a parallel-plate configuration in an AmR cycle, with a set temperature of 305 K at the hot side of the heat exchanger and 290 K at the cold side. In the first phase of the cycle, the material was subjected to a multicaloric field consisting of 2 T and 2.5 kbar, which, when applied simultaneously, resulted in a waveform of the effect, as shown in Figure 1. In the sequential field application case, however, it was decided to apply the 2.5 kbar field first and then the 2 T field, which exhibited the waveform shown in Figure 2. The frequency was set to 0.50 Hz, while the heat transfer fluid velocities were selected to fully explore the heat exchanger's thermal profile, yielding the following energy performance metrics:

$$\Delta T_{cold} = T_{cold} - \frac{1}{t_{cycle}} \int_{3t_{step}}^{4t_{step}} T_f(0, y, t) dt \quad (7)$$

$$\dot{Q}_{cold} = \frac{1}{t_{cycle}} \int_{3t_{step}}^{4t_{step}} \dot{m}_f C_f (T_{Cold} - T_f(0, y, t)) dt \quad (8)$$

$$SCP = \frac{\dot{Q}_{cold}}{\dot{m}_s} \quad (9)$$

These performances were evaluated as a function of the volume ratio, calculated as follows:

$$V^* = \frac{t_{fluid} v_{inlet}}{L} \quad (10)$$

The scenario involving the single effect is not reported, since the operating range was in the descending region for the barocaloric effect, and, in the case of the magnetocaloric effect, the effect was further limited, as shown in Figure 2. To achieve a useful cooling effect within the AmR cycle stages without destructive thermal cancellation, the driving fields were synchronized in specific phases. Therefore, the cumulative source term $\dot{Q}'''_{MULTI,sum}$ represents the localized absolute thermal contribution managed by the bed.

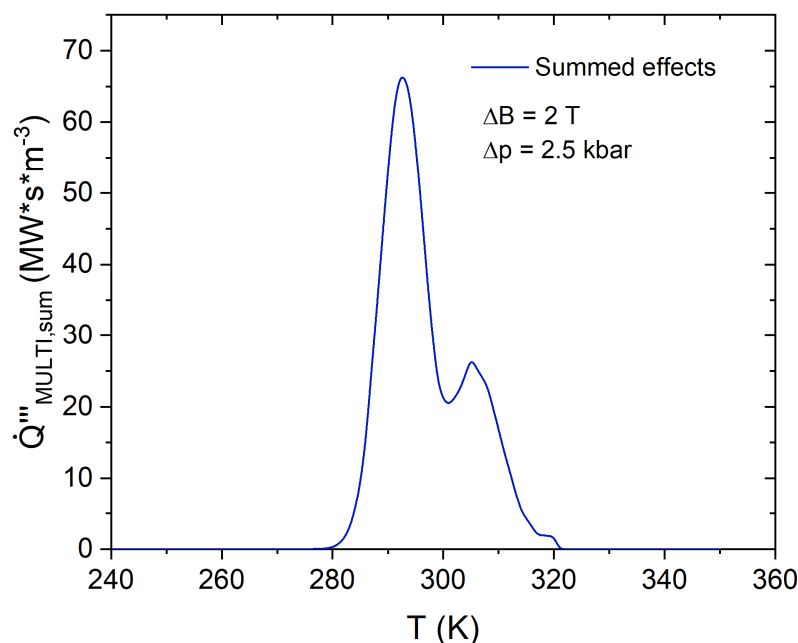


Figure 1. Power density vs. temperature for the summed effects.

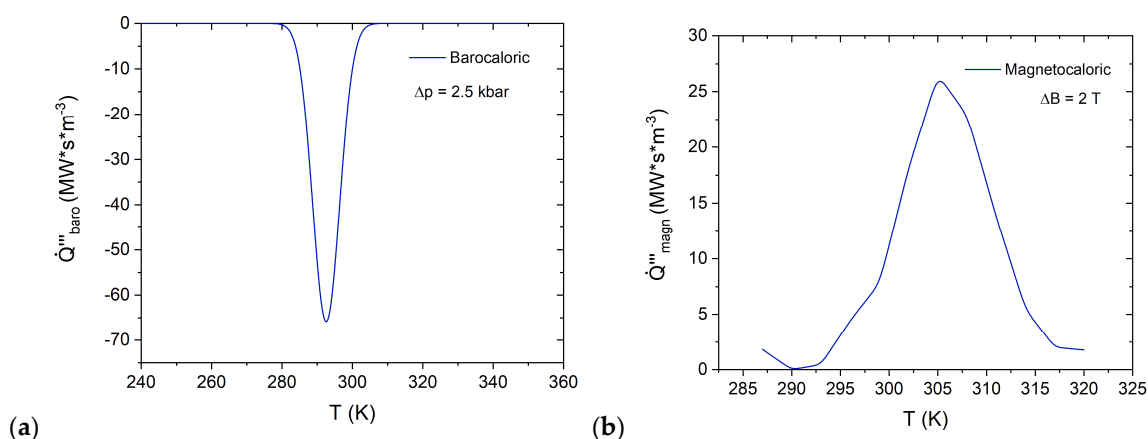


Figure 2. Power density vs. temperature for the: (a) barocaloric effect and (b) the magnetocaloric effect.

6. Results

The numerical results concerning the thermal performance of the multicaloric regenerator under different field activation strategies are summarized in the following plots. Figure 3 illustrates the temperature variation on the cold side of the application (ΔT_{cold}) as a function of the volume ratio (V^*), comparing the simultaneous application of both fields (summed effects, red curve) against the sequential application mode (black curve).

The analysis of the regenerator's performance shows that sequentially applied caloric effects are less advantageous than simultaneous application. This behavior is fundamentally governed by a combination of spatial thermodynamic matching along the regenerator's temperature gradient and the transient thermal paths developed during the cycle stages. The active multicaloric regenerator bed establishes a continuous spatial temperature gradient from the cold boundary (290 K) to the hot boundary (305 K). As illustrated by the individual caloric curves in Figure 2, within this specific temperature snapshot, the inverse barocaloric effect is in its descending region above its 285 K peak (Figure 2a), while the magnetocaloric effect is still limited below its 305 K peak (Figure 2b). When the fields are applied simultaneously, the system achieves an optimal spatial–thermodynamic matching across the coordinate space. The localized sections near the cold end effectively exploit

the baseline of the barocaloric curve, while the sections near the hot end simultaneously leverage the peak of the magnetocaloric curve, operating along the optimized cumulative caloric envelope reported in Figure 1. This continuous cooperative distribution ensures that the entire volume of the regenerator actively contributes to the cooling capacity throughout the loading phase. Conversely, in the sequential mode, the system is highly penalized by a time limitation ($\frac{t_{field}}{2}$) and a dynamic thermal mismatch.

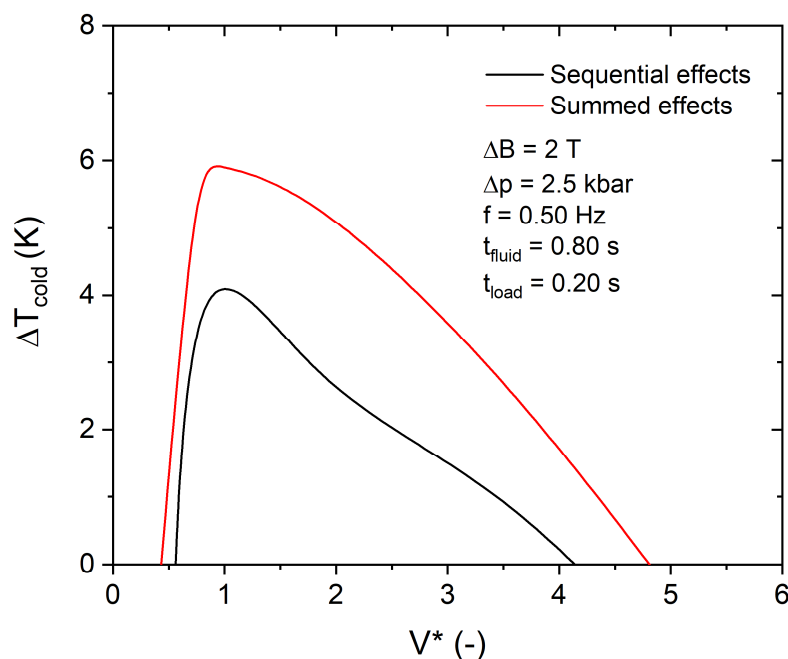


Figure 3. ΔT_{cold} vs. V^* in sequential (black) and simultaneous (red) field application.

In the sequential mode, the activation of the first field causes a local temperature variation; as a result, during the subsequent interval ($\frac{t_{field}}{2}, t_{field}$), the material is forced to operate at a shifted temperature baseline where the second caloric effect is significantly reduced, falling into the descending tail of its ΔT_{ad} curve. This dynamic thermal mismatch severely degrades the cumulative power density. On the contrary, simultaneous activation ensures that the material continuously works at the optimal multicaloric envelope throughout the entire loading duration, maximizing the useful thermal power. In fact, Figure 3 shows the temperature variation on the cold side of the regenerator, where the water is cooled by $\Delta T_{cold} = 6^\circ\text{C}$ as peak, reaching a temperature of approximately 11°C in the simultaneous case. In the sequential application, the peak is only 4 degrees, but this is explainable given that, based on the operating temperature, there is a reduction in the effect when switching from the barocaloric to the magnetocaloric effect. In contrast, in the application of both fields, there is a continuous trend in power density (Figure 1).

Figures 4 and 5 show the cooling capacity and the SCP, both of which follow the same trend. The trend does not match the one shown by ΔT_{cold} , since the cooling capacity is proportional to the fluid mass flow rate, which results in a peak at higher V^* values. Nevertheless, as seen in the previous trend, the application of cumulative effects here also results in high power. The peak in both scenarios occurs at $V^* = 2$, an increase of approximately +124% over the peak of the simultaneous (or summed) effect, and then decreases as the speed increases. The SCP trend mirrors the cooling power, and, for the same material mass, the performance gain is significantly higher when the fields are applied together, reaching peaks of 1292 W kg^{-1} .

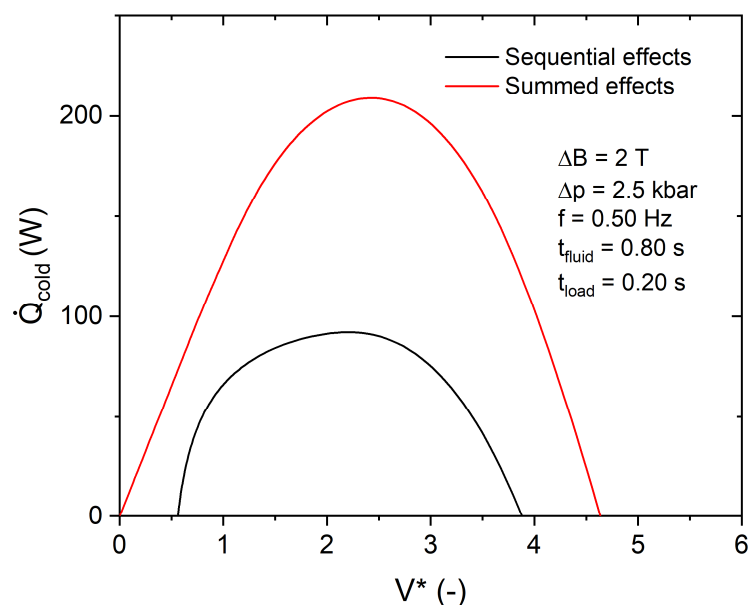


Figure 4. $\dot{Q}_{\text{cold}}$ vs. V^* in the sequential (black) and simultaneous (red) fields application.

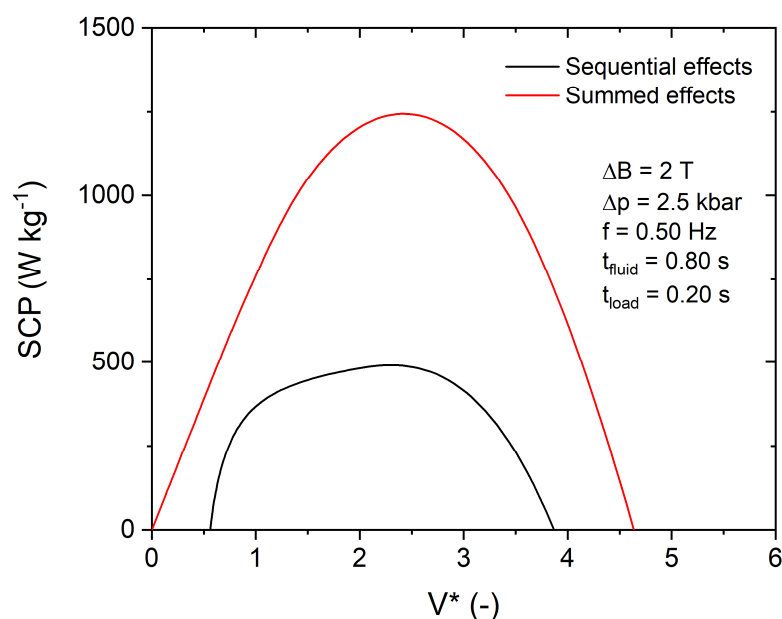


Figure 5. SCP vs. V^* in the sequential (black) and simultaneous (red) field application.

7. Conclusions

The investigation introduced in this paper is based on a new approach to applying multicaloric effect, examining its effects depending on how the fields are applied. In the two scenarios—simultaneous and sequential application—the material's thermal performance at room temperature [290; 305] K is analyzed, and the following conclusions are drawn:

- The simultaneous application of the magnetic field and hydrostatic pressure outperforms the individual effects applied sequentially.
- In cooling mode, the water temperature drops by 6 °C in the best-case scenario, whereas, in the sequential case, peak performance decreases by −33%.
- The cooling capacity is significantly higher in the combined effect, with values exceeding 200 W, and this trend is reflected in the SCP, with a +137% increase compared to the worst-case scenario.

In conclusion, the results highlight the relevance of the field application strategy in multicaloric systems and identify simultaneous activation as the most promising configuration for enhancing the performance of room temperature solid-state heat pumps.

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Nomenclature

Roman symbols

B	Magnetic field induction, T
C	Specific heat capacity, J kg ^{−1} K ^{−1}
H	Height of the regenerator, mm
HTF	Heat transfer fluid
k	Thermal conductivity, W m ^{−1} K ^{−1}
L	Regenerator length, mm
M	Martensite phase temperature, K
$\dot{m}$	Mass flow rate, kg s ^{−1}
m	Mass, kg
n	Number of times
p	Pressure, Pa
$\dot{Q}$	Thermal power, W
$\dot{Q}'''$	Power density, W m ^{−3}
S	Entropy, J kg ^{−1} K ^{−1}
SCP	Specific cooling power, W kg ^{−1}
SMA	Shape memory alloy
T	Temperature, K
t	Time, s
u	Longitudinal velocity component, m s ^{−1}
V*	Volume ratio, -
v	Orthogonal velocity component, m s ^{−1}
$\dot{W}$	Mechanical power
x	Longitudinal spatial coordinate, m
Y	Forcing field in the caloric effect
y	Orthogonal spatial coordinate, m

Greek symbols

Δ	Finite difference
∇	Gradient operator
δ	Infinitesimal difference

ε	Applied deformation, %
$\bar{\varepsilon}$	Infinitesimal quantity
ρ	Density, kg m ^{−3}
σ	Uniaxial stress, MPa
Subscripts	
ad	Adiabatic
cold	Cold
cycle	Cycle
f	Fluid
field	Field
fluid	Fluid flow step of the cycle
hot	Hot
MULTI	Multicaloric effect
p	Constant pressure
pump	Pump
s	Solid, caloric material
seq	Sequential effect application
SMA	Shape memory alloy
span	Span
step	Step
sum	Simultaneously effect application
T	Constant temperature

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