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Highlights

- The performance of a unique TSC embedded in an MC device was investigated numerically.
- A cooling-power density of $2025 \text{ Wkg}_m^{-1} \cdot \text{C}_0^{-1}$ was achieved at 20 Hz and zero temperature span.
- The highest rectification ratio and temperature span achieved were 3.8 and 2.8 K.
- The potential use of the proposed device was demonstrated on a battery system.

Petelin et al., iScience 25, 105517
December 22, 2022 © 2022
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https://doi.org/10.1016/j.isci.2022.105517
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SUMMARY
Compact, solid thermal control devices offer a new way to control the intensity and direction of heat flow between the components of a system, which is crucial for both optimized performance and safety. In this work we study a thin, silicon thermal switch capacitor (TSC) used for heat transport in a magnetocaloric cooling system. A numerical model was developed to quantify the effects of various operating conditions and design parameters on the performance of a magnetocaloric device with an embedded TSC. Based on realistic material properties, a maximum cooling-power density of 4000 Wm⁻² (2025 Wkg⁻¹m⁻¹) was obtained for a zero temperature span and an operating frequency of 20 Hz. The use of the presented device was demonstrated on a battery system, motivating further experimental studies to develop a new, compact cooling device that can be directly attached to a heat reservoir, making it desirable for a variety of applications.

INTRODUCTION
Thermal management is a key challenge for modern technology. It affects a wide range of applications from electronics to medicine, where controlled and efficient heat transfer are important for improved device performance. Therefore, in recent years, researchers have begun to look beyond the state of the art, by exploiting different thermal control devices (TCDs), i.e., thermal switches, thermal diodes, thermal regulators, or thermal capacitors. These types of device provide passive or active control of the heat in an analogous way to how their electrical counterparts control electric current. A thermal switch relies on a non-thermal control parameter such as an electric field, magnetic field, or pressure to change the thermal conductance of the device. A figure of merit for a thermal switch is the switching ratio between the highest thermal conductance achieved (the on state), to the smallest achievable thermal conductance (the off state). Thermal switches can be divided into static and motional thermal switches. Motional thermal switches change their thermal conductance by changing their position and making and breaking thermal contacts, whereas static thermal switches are always in physical contact with adjacent interfaces. On the other hand, the thermal capacitor is a single-pole element, because the other side of this element is always “grounded”, i.e., connected to a heat reservoir. It acts as a temporary heat-storage device, similar to the “self-capacitance” of an insulated electrical conductor in the electrical analogy. The operation of a thermal switch (TS) and a thermal capacitor (TC) can also be combined into a so-called a thermal switch capacitor (TSC) (which is evaluated here), where the TSC first stores the energy and then releases it by means of transporting the heat by moving the material.

Thermal switches have received special attention because of their unique possibilities in solid-state cooling and heating, especially in caloric-effect-based refrigeration and heat pumping. These solid-state technologies represent a serious alternative to existing vapor-compression devices, because of their potential for higher energy efficiency and the use of environmentally friendly, solid-state refrigerants. However, most existing caloric devices use active caloric regeneration, in which the working fluid oscillates through the matrix of the caloric material. This is associated with irreversible viscous and heat-transfer losses and limits the operating frequency and related power density of the device. The implementation of TCDs could improve the performance of caloric devices during high-frequency operation (the number of thermodynamic cycles per unit of time), which is crucial for an increase in the power density (compactness). The Faculty of Mechanical Engineering, University of Ljubljana, Askerceva 6, 1000 Ljubljana, Slovenia

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https://doi.org/10.1016/j.isci.2022.105517

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concept allows the use of static magnetic field sources, like those presented by Klinar et al. and Tomc et al., and represent an energy-efficient route to high-frequency operation. Several mechanisms for the mechanical actuation of solid TSs in caloric technologies have been reported, including electrical actuation, magnetic actuation and mechanical actuation. In most contact-based thermal control devices, heat transfer from the caloric element to the heat exchanger and vice versa can be achieved either by moving the caloric material between a hot and a cold heat exchanger (direct contact) or by using an element that stores energy and transfers it from the caloric material to the heat exchanger (indirect contact), as in our case. Direct contact assured by moving the MCM alternately between the hot and cold heat exchangers, should theoretically result in better thermal performance compared to indirect contact because the heat can be transferred directly. However, when magnetic forces are used to move the MCM, establishing a controlled thermodynamic cycle is challenging because the time required for adiabatic magnetization and demagnetization in the case of the magnetocaloric Brayton cycle must be the same as the time required to move the MCM between the hot and cold heat exchangers. Some authors report that they have found discrepancies between the experimentally determined working cycle and the Brayton cycle targeted by the original design, because of the difficulties mentioned in the case of a direct contact. Although it is true that the self-oscillating system, where no external actuation is required to move the MCM, offers an important advantage in terms of more energy-efficient operation, some authors reported that the MCM no longer achieves contact with the hot and cold heat exchangers for larger air gaps and shows a damped oscillation above and between the equilibrium points. If we were to use a nonmagnetic actuator to move the MCM, the MCM would have to be rigidly clamped during the magnetization, otherwise it would move freely toward the magnetic source. This principle of operation would not be very practical and would also lead to additional heat losses through the rigid structure housing. The main advantages of the indirect approach presented in this work are the controlled working cycle that is independent of the magnetic forces, the lower power consumption for actuation compared to the direct-contact approach with external actuation, and the better reliability of the device under different operating conditions (operation independent of the magnetization/demagnetization time, no need to fix the TSC during magnetization).

The state of the art for theoretically and experimentally evaluated movable solid TSs in caloric technologies is summarized in Table 1, and various concepts of motional solid TSs are presented in Figure 1.

| Actuation | Caloric effect | Switching ratio | Air gap | Analysis type | Reference |
|-----------|----------------|-----------------|---------|---------------|-----------|
| Mechanical | EC             | 29.2            | 230     | Experimentally evaluated | Wang et al., 19 |
| Mechanical | EC             | 27              | 230     | Experimentally evaluated | Wang et al., 20 |
| Mechanical | EC             | 27              | –       | Numerically evaluated | Smullin et al., 23 |
| Mechanical | MC             | 22.5            | 1.1     | Experimentally evaluated | Tsukamoto et al., 24,25 |
| Mechanical | eCE            | –               | –       | Experimentally evaluated | Schmidt et al., 26 |
| Mechanical | eCE            | –               | 1000    | Experimentally evaluated | Ossmer et al., 27,28 |
| Mechanical | EC             | –               | –       | Experimentally evaluated | Defay et al., 29 |
| Mechanical | EC             | 100             | 4000    | Experimentally evaluated | Jia et al., 30 |
| Electric   | EC             | –               | 3000    | Experimentally evaluated | Meng et al., 31 |
| Electric   | EC             | –               | 3000    | Experimentally evaluated | Bo et al., 32 |
| Electric   | EC             | 1000            | –       | Numerically evaluated | Ju et al., 33 |
| Electric   | EC             | 1000            | 975     | Numerically evaluated | Almanza et al., 34 |
| Electric   | EC             | –               | 100     | Experimentally evaluated | Bradesko et al., 35 |
| Magnetic   | MC             | –               | 1.5     | Numerically evaluated | Ahmim et al., 36 |

*EC – electrocaloric, MC – magnetocaloric, eCE – elastocaloric.
We can see from Table 1 that magnetocaloric technologies with solid thermal switches were so far poorly exploited. Moreover, we did not detect research activities on TS–TC integration in any of the caloric technologies. On the other hand, their easy control, rapid operation, compactness and lack of complexity might offer important advantages over other TCDs embedded in caloric technologies.

Therefore, the main objective of our research was to design, model and parametrically evaluate the new concept of a solid TSC, integrated into a magnetocaloric cooling device. For the purposes of this study, realistic material properties and the geometry of each component were considered.

Figure 2 shows the conceptual design of the TSC that is embedded in the magnetocaloric device. The system core consists of two silicon TSCs that modulate the heat flow to and from the magnetocaloric material in a refrigeration cycle that includes the following four steps. (1) In the first step, the magnetic field change causes a temperature change in the magnetocaloric material (MCM) under adiabatic conditions. During this step, both TSCs are not in contact with the MCM and no changes occur in the thermal transport across the interfaces. This defines the TSC’s off state. (2) The heat-transfer process follows in two stages, with the TSC on the hot side being turned on and the TSC on the cold being turned off. First, the hot-side TSC makes thermal contact with the MCM, and during this stage the heat is transferred from the MCM to the TSC. In the second stage, the TSC is in contact with the heat sink, and heat conduction through the thermal interface between the TSC and the heat sink takes place. (3) When both TSCs are switched to the off state, the magnetic field changes to zero or a low magnetic field, leading to a temperature decrease in the MCM. No heat transfer occurs during this stage. (4) Again, heat transfer occurs in two stages, with the hot side of the TSC being in the off state and the cold side of the TSC being in the on state. First, the TSC releases heat while in contact with the MCM. Then, in the second stage, the TSC is moved toward the heat source, where it absorbs heat while in contact with the heat source. As mentioned, the proposed design suggests using electrostatic forces for the TSC’s actuation because electrostatic actuation is compact, has a short switching time in the range of milliseconds, provides a high contact pressure in the range of kPa, and consumes a small amount of power. The heat sink, heat source, and MCM act as an electrode for the electrostatic actuation and should be covered by a dielectric layer (DL) to prevent any short circuit.

Our concept proposes the use of a static electro-permanent magnet to apply and remove the magnetic field, as shown in Figure 2A. The TSC assembly with magnetocaloric material is positioned in the air gap between the two windings. The movement of the Si TSC is caused by the electrostatic forces, whereas...
the heat exchangers and the MCM are separated by thin spacers positioned only at the edges of the device (the purple element in Figure 2A). The Si TSC is freestanding and not attached to the spacers to avoid a thermal bridge between the hot and cold sides (Figure 2A). The spacers should be made of a material with low conductivity to reduce the heat losses. Those heat losses because of the conduction of heat through the spacer are not considered in the numerical model.

For the purposes of the study, a one-dimensional numerical model based on transient heat transfer was developed. Using this tool, we performed numerical analyses for different operating conditions and the thermal and geometrical characteristics of the TSC in a magnetocaloric device. We systematically analyzed the performance of a TSC-based, solid-state magnetic refrigerator, considering different geometrical designs and material properties as well as different operating conditions. For each parameter combination, the temperature span between the heat sink and the heat source was registered. This analysis allowed us to find the best performance of the device and to investigate how the different parameters impact on the performance of the device in terms of heat transfer.

RESULTS
Modeling approach
We investigated the time-dependent behavior of the device with finite-element modeling using a one-dimensional transient heat-transfer model in combination with the magnetocaloric effect. For the purpose
of the study, we evaluated the single-stage magnetic Brayton cycle without regeneration, which consists of two adiabatic and two isofield processes. The adiabatic process occurs during the magnetization and demagnetization, while the heat flows from, or to the magnetocaloric material during the two isofield processes. In this study, we analyzed the effects of the operating conditions on the cooling performance. We investigated different thermal properties (different materials and thermal contact resistances), geometries (different thicknesses), and operating conditions (different frequencies, different contact times, and different cooling powers) to quantify the effects on the temperature span between the heat source and the heat sink as well as the cooling power density, as these are two of the most important criteria for evaluating the cooling device.

In a realistic 3D device, all the edges should be thermally insulated, so the heat losses to the surroundings would be much smaller than the heat transfer by conduction through the solid-solid contacts between the elements of the device. If we also consider that the edges of the device represent only 5% of the total surface area, then the heat losses to the surroundings can be considered as negligible in the numerical model.

Overall, the system consists of an MCM sandwiched between two freely movable, solid TSCs. For the two isofield processes, the governing equation considers only the conduction of the heat, as the whole system consists of solid elements:

\[ k \frac{\partial^2 T}{\partial x^2} - \rho c_p \frac{\partial T}{\partial t} = 0, \]

(Equation 1)

where \( k \) is the thermal conductivity, \( T \) is the temperature, \( x \) is the position in the \( x \) direction, \( \rho \) is the density, \( c_p \) is the specific heat at constant pressure, and \( t \) is the time. Equation 1 was discretized for space and time for each node using the implicit finite-difference method. More information about the numerical model and the corresponding equations and boundary conditions can be found in STAR Methods. A mesh size analysis is performed to ensure numerical accuracy (see Figure S1). The simulations ran over several thermodynamic cycles until the solution converged. A steady state was reached when the temperature changes in the heat sink and heat source for two consecutive cycles were \( \leq 0.0001 \) K. The flowchart of the calculation procedure is shown in Figure 3.

**Thermal switch capacitor in a magnetocaloric device**

The basic unit of the proposed magnetocaloric system consists of a heat sink and a heat source, the MCM, the dielectric layers and two TSCs, which act as a thermal control device for guiding the heat. Thus, by alternating the TSC’s movement, a unidirectional heat flow occurs between the MCM and the heat sink and heat source.

**Heat sink/heat source (HEX)**

Silicon (Si) with a thickness of 500 µm was used as the material for the heat sink and heat source. The thermal conductivity was 150 Wm\(^{-1}\)K\(^{-1}\), the density was 2329 kgm\(^{-3}\), and the specific heat was 700 Jkg\(^{-1}\)K\(^{-1}\). Silicon is not the most commonly used material for the HEXs, but as the contact surface, it plays a crucial role in device performance because of its ultra-flatness. To evaluate a more realistic system and conditions where the heat transfer between the contacts is dominated by the thermal resistance, it is important that the contact surfaces are very flat to achieve better performance in terms of heat transfer.

**Thermal Switch Capacitor**

For reasons similar to those for the HEXs, we used Si with a thickness of 200 µm for the TSC. The ability of the material to store thermal energy plays an important role in the choice of material for the TSC. This can be defined as the heat capacity of the material (\( \rho c_p \delta \)). The TSC serves as an electrode, so it must be electrically conductive, while not having magnetic properties. Otherwise, the switching between the high and low magnetic fields would affect the performance of the device. The minimum voltage required for actuation is 100 V, which is the voltage at which the electrostatic force overcomes gravity for 200 µm thick Si. Our initial experimental results show that this voltage is 8–10 times higher, but it is important to note that the range of the electric current is significantly lower (~mA). In addition, it was experimentally demonstrated that the voltage recovery rate can be as high as 84% for a similar design.33
Dielectric layer
A dielectric layer of SiO$_2$ with a thickness of 50 $\mu$m, a thermal conductivity of 1.38 Wm$^{-1}$K$^{-1}$, a density of 2202 kgm$^{-3}$ and a specific heat of 703 Jkg$^{-1}$K$^{-1}$ was used to prevent any short circuit between the electrodes.

Thermal contact resistance $R_c$
The thermal contact between the TSC and the dielectric layer on the Si was considered by applying the thermal contact resistance in the conduction equation. Simultaneously with the movement of the TSC, the corresponding thermal contact resistance changed, and the air in the air gap was also replaced by fresh air at the ambient temperature (293 K). To focus on the operation of the device, the thermal contact resistance was set to a fixed value of $10^{-6}$ Km$^2$W$^{-1}$ in all of the case studies (except for one case study where the different thermal contact resistances were simulated). In the literature a wide range of different theoretical thermal contact resistance models can be found; however, there is still a large difference between the theoretically and experimentally determined values. This is mostly because of the large number of parameters, such as surface roughness and surface topography, which play important roles at the surface boundary. We used Antonetti’s$^{42}$ correlation to determine the thermal contact resistance between the Si and SiO$_2$ as the TSC and the dielectric layer. The theoretical model chosen is described in supplemental information S4. The model is based on elastic deformation because plastic models are known to overestimate the thermal contact resistance values of a contact-based TCE. The resulting thermal contact resistance based on Antonetti’s correlation is $2.38 \times 10^{-2}$ Km$^2$W$^{-1}$, which is high compared to values for solid-state contacts in the literature.$^{44,45}$ Chien et al.$^{46}$ analyzed theoretically and experimentally the interface

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**Figure 3. Flow chart of the calculation procedure**

Input parameters
- Start
- Magnetic/electric model (Boltzmann cycle)
- $N = N_{iter}$?
- Steady state conditions?
- Last cycle?

- 1. Application of external magnetic field
- 2. Heat transfer – 1. stage
- 3. Heat transfer – 2. stage
- 4. Removal of external magnetic field
- 5. Heat transfer – 1. stage
- 6. Heat transfer – 2. stage

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Materials properties
- geometric properties
- time step, mesh size
- $f, B_0, B_{ext}, \theta_{ang}, R_c, T_{amb}, N_{iter}$

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Theoretical models for $R_c$ include $10^{-6}$ Km$^2$W$^{-1}$.
between Si (TSC) – SiO$_2$ (DI) and obtained a substantially lower value for the interfacial thermal resistance of $3.7 \times 10^{-9}$ K m$^2$ W$^{-1}$. Following this, we have chosen the middle of the range for the thermal contact resistance in the numerical model, i.e., between $2.38 \times 10^{-9}$ and $3.7 \times 10^{-9}$ K m$^2$ W$^{-1}$ to be $10^{-6}$ K m$^2$ W$^{-1}$.

Caloric effect

The magnetocaloric effect was implemented by an instantaneous temperature change of each MCM node whenever there was a field change between the initial ($B_{\text{on}} = 0$ T) and final ($B_{\text{off}} = 1$ T) magnetic field values (see STAR Methods). For the aforementioned magnetic field change, the maximum adiabatic temperature change ($\Delta T_{\text{ad}}$) of gadolinium (Gd) that can be achieved is 3.3 K. In this work, a 250-μm-thick gadolinium (Gd) was used for the MCM, with a density of 7900 kg m$^{-3}$ and a thermal conductivity of 10.5 W m$^{-1}$ K$^{-1}$. The magnetocaloric properties of the Gd were taken from ref. 10, where the mean-field theory was used to obtain the specific total entropy values as a function of the temperature and the internal magnetic field. In all simulations, the magnetization and demagnetization time was set to 5 ms. This value was numerically and experimentally demonstrated by Tomc et al., who developed a static and hybrid electro-permanent magnet with the magnetic energy recovery, which enabled efficient operation at frequencies of about 50 Hz. A longer time would result in a reduced cooling performance, because these two processes should occur under adiabatic conditions.

Boundary conditions

The heat source was isolated (except for one case study where the different cooling-power densities were simulated), whereas the heat sink’s surface was subjected to convective boundary conditions. The ambient temperature was set at 293 K and the convective heat-transfer coefficient $(h_w)$ was 10,000 W m$^{-2}$ K$^{-1}$, which is equivalent to an external water-cooling system with microchannels or a heat-pipe system.

Operating conditions

We performed various case studies where (1) different materials (Si, Ag, Au) were considered for the TSC, (2) the TSC thickness was varied from 50 μm to 200 μm, (3) different materials (Si, Cu, Al) were considered for the heat source and for the heat pipe or the microchannel heat exchanger that served as a heat sink, (4) the heat-source/heat-sink thickness varied between 250 and 3000 μm, (5) the MCM thickness varied between 50 and 750 μm, (6) the contact time varied between 2.5 and 500 ms, (7) the

| Table 2. Parameters used for the simulation |
|-------------------------------------------|
| Operating parameters                      |
| $B_{\text{on}}$ | 1 T |
| $B_{\text{off}}$ | 0 T |
| $F_{\text{on}}$ | 0.1–15 Hz |
| $\Delta T_{\text{ad}}$ (Gd) | 3.3 K |
| $h_w$ | 10,000 W m$^{-2}$ K$^{-1}$ |
| $T_{\text{amb}}$ | 293 K |
| $F_{\text{mag/demag}}$ | 0.005 s |
| $R_{\text{c}}$ | $10^{-6}$ K m$^2$ W$^{-1}$ |
| $T_{\text{amb}}$ | 293 K |
| $h_w$ | 10,000 W m$^{-2}$ K$^{-1}$ |
| $f_{\text{mag/demag}}$ | 0.005 s |
| $R_{\text{c}}$ | $10^{-6}$ K m$^2$ W$^{-1}$ |
| $T_{\text{amb}}$ | 293 K |
| $h_w$ | 10,000 W m$^{-2}$ K$^{-1}$ |
| $f_{\text{mag/demag}}$ | 0.005 s |
| $R_{\text{c}}$ | $10^{-6}$ K m$^2$ W$^{-1}$ |

| Geometric properties |
|----------------------|
| $\delta_{\text{HEX}}$ | 500 μm |
| $\delta_{\text{TSC}}$ | 200 μm |
| $\delta_{\text{MCM}}$ | 250 μm |
| $\delta_{\text{DI}}$ | 50 μm |
| $\delta_{\text{MCM}}$ | 100 μm |

| Material properties |
|---------------------|
| Material c$_p$ (J kg$^{-1}$ K$^{-1}$) | $\rho$ (kg m$^{-3}$) | k (W m$^{-1}$ K$^{-1}$) |
| TSC, heat sink/heat source |
| Silicon (Si) | 700 | 2329 | 150 |
| Magnetocaloric material |
| Gadolinium (MCM) | function of specific total entropy and temperature | 7900 | 10.5 |
| Dielectric layer |
| Silicon dioxide (SiO$_2$) | 703 | 2202 | 1.38 |
cooling-power density varied between 0 and 5000 W m\(^{-2}\), and the thermal contact resistance varied between 3.7 \times 10^{-9} \text{Km}^{-1} and 2.38 \times 10^{-2} \text{Km}^{-1}. The operating frequency varied between 0.1 and 15 Hz in all the case studies. All the parameters and material properties used in the simulations are listed in Table 2.

**Influence of TSC and HEX thickness and material properties on the temperature span at zero cooling power**

In the first scenario studied, three different materials (Si, Ag, Au) were investigated for the TSC and their effects on the temperature difference between the heat sink and the heat source. To further quantify the effects on the cooling performance as a function of the operating frequency, different thicknesses (50, 100, 200 \(\mu m\)) were considered as presented in Table 3. It can be observed from Figure 4A that the optimum operating frequency decreases as the material thickness increases. This is related to the Fourier number (\(Fo = k_{t}/(r c_{p} d^{2})\)). If the material properties of the TSC do not change a great deal over time or during the thermodynamic processes, then the thickness of the TSC is the only quantity that affects the heat diffusion time. At an operating frequency of 15 Hz, the temperature span for a TSC of thickness \(d \geq 100 \mu m\) begins to decrease because the time available for heat diffusion (~28 ms at 15 Hz) is too short for the MCM to perform a complete heat-absorption or rejection step. Therefore, the cooling capacity of the MCM is not fully utilized. This is evident in the temperature profile of the device as a temperature jump at the thermal contact between the MCM and the TSC (see Figure S2). If we use a thinner MCM, e.g., 50 \(\mu m\), the device can be operated at a frequency higher than 15 Hz, and all the heat from the MCM is dissipated during one heat-transfer cycle. However, the cooling capacity decreases when the thickness of the magnetocaloric material is reduced (this case is shown in Figure 5). The conditions under which the maximum temperature span of 2.54 K between the heat source and the heat sink were achieved for all simulated cases correspond to an Au-based TSC with a thickness of 200 \(\mu m\) and an operating frequency of 10 Hz at zero cooling power. Evidently, there is almost no difference in the achieved temperature span for the Au- or Ag-based TSCs, because the volumetric heat capacity is very similar for both materials. For the same thickness of 200 \(\mu m\), the almost-two-times-lower heat capacity of the Si-based TSC results in a smaller temperature span between the heat source and heat sink. If the temperature span does not change with an increase in the frequency from 5 Hz to 10 Hz (e.g., Au TSC, \(d = 200 \mu m\)), this indicates that the heat has diffused through the entire thickness and any additional time for the heat transfer only increases the cycle time without contributing to the heat transfer or the cooling. We can either use a thinner TSC operating at higher frequencies or a thicker TSC operating at lower frequencies to achieve a similar temperature span. However, at lower frequencies, the time required to reach a steady state would increase significantly (see Figure S3). On the other hand, the material properties \(r c_{p}\) and the thickness of the HEX have an opposite effect on the device’s performance, as shown in Figure 4B. A thinner HEX and a low heat capacity lead to larger temperature differences between the heat source and the heat sink at low frequencies. This is because the heat diffuses faster when both the thickness and the volumetric heat capacity of the HEX material are smaller. For example, for a Si-based HEX, which turns out to be the best of the considered materials, the temperature span is higher by a factor of 4.5 for a 250-\(\mu m\)-thick HEX than for a HEX with a thickness of 3000 \(\mu m\), at an operating frequency of 0.1 Hz for both cases. As the frequency increases, the influence of the thickness of the HEX on the cooling performance decreases, but the same device with a thicker HEX takes a

### Table 3. Geometrical and material properties of the TSC and HEX used in the evaluation of the temperature span between the heat source and the heat sink

| Geometric properties | 1. case: TSC | 2. case: HEX |
|----------------------|--------------|--------------|
| \(\delta_{\text{TSC}}\) (Si, Au, Ag) | 50–200 \(\mu m\) | \(\delta_{\text{HEX}}\) (Si, Cu, Al) | 250–3000 \(\mu m\) |
| \(\delta_{\text{HEX}}\) (Si) | 500 \(\mu m\) | \(\delta_{\text{TSC}}\) (Si) | 200 \(\mu m\) |

| Material | \(c_{p}\) (Jkg\(^{-1}\)K\(^{-1}\)) | \(\rho\) (kgm\(^{-3}\)) | \(k\) (Wm\(^{-1}\)K\(^{-1}\)) |
|----------|--------------------------|----------------|----------------|
| Silicon (Si) | 700 | 2329 | 150 |
| Aluminum (Al) | 933 | 2645 | 236 |
| Copper (Cu) | 385 | 8950 | 410 |
| Gold (Au) | 129 | 19200 | 311 |
| Silver (Ag) | 238 | 10640 | 435 |
much longer time to reach the steady state, which should be considered when selecting a heat pipe or a micro-channel heat exchanger.

Influence of MCM thickness on the temperature span at zero cooling power

The influence on the performance and the operation of the MC refrigerator was also tested using different MCM thicknesses. From Figure 5 we can see that by decreasing the MCM’s thickness, the temperature span between the heat source and the heat sink decreases as well. This indicates that there is a minimum thickness of MCM required to achieve a minimum temperature span greater than 0 K between the heat sink and the heat source, and that there is a maximum thickness of MCM at which further increases in thickness no longer affect the cooling performance. For a 150-μm-, 250-μm- and 300-μm-thick piece of Gd the optimum frequency is 10 Hz, for 750 μm it is 5 Hz, and for 50-μm-thick Gd it is out of the analyzed range, so higher than 15 Hz at zero cooling power.

Figure 5. Temperature span as a function of frequency for five different MCM thicknesses at the stationary state and with zero cooling power

Other parameters used in the simulations can be found in Table 2.
Influence of contact time on the temperature span and the switching ratio at zero cooling power

The contact time $t_{\text{contact}}$ between the TSC and the MCM, and between the TSC and the heat sink or heat source, plays an important role in the device’s cooling performance. Figure 6 shows the dependence of the temperature span and the switching ratio $r$ on the contact time $t_{\text{contact}}$, with the magnetization/demagnetization time fixed at 5 ms in all the simulations. The total heat-transfer time for one magnetocaloric cycle is defined as $t_{\text{ht}} = 4 t_{\text{contact}}$ and the switching ratio was calculated as:

$$r = \frac{G_{\text{off}}}{G_{\text{on}}} \quad \text{(Equation 2)}$$

Here, $G$ is the thermal conductance in the off and on states. The temperature span first increases with $t_{\text{contact}}$ until it reaches a maximum value for a given $t_{\text{contact, cut off}}$ and then decreases with a further increase in $t_{\text{contact}}$. In this setup, the maximum value of the temperature span occurs at $t_{\text{contact}} = 27.5$ ms. Similar conclusions can be drawn for the switching ratio, which was defined under stationary-state conditions for both the heating and cooling isofield processes in a single magnetocaloric cycle.

For the device described in the section thermal switch capacitor in a magnetocaloric device, the maximum switching ratio is $r = 3.8$ (cooling isofield process) and $r = 3.5$ (heating isofield process). The two switching ratios differ because the $G_{\text{on}}$ is higher for the heating isofield process during which time the heat is transferred from the MCM to the heat sink. From the switching ratios given in Table 1, it can be seen that the achieved switching ratio is lower than that reported in the literature. Here, it is important to point out that so far the literature lacks a clear definition of the conditions under which the switching ratios were determined. It is not clear whether the switching ratio refers to steady-state or transient conditions, how the on and off states were defined, and at what temperature difference it was achieved. So the comparison is often not appropriate. The results in Figure 6 clearly indicate that a too short $t_{\text{contact}}$ implies a smaller temperature span as there is not enough time to transfer the heat between the elements in the contact. However, when $t_{\text{contact}}$ is too long, the heat also conducts in an unfavorable direction (toward the TSC in the off state), which leads to a reduction in the total temperature span. The contact time not only influences the temperature span; it also influences the time required to reach the steady state and the amplitude of the HEX temperature fluctuations during steady-state operation.

The steady state at the $t_{\text{contact}}$ of 2.5 ms is reached already after approximately 8 s. And at the $t_{\text{contact}}$ of 500 ms, the cooling device reaches steady-state operation after 66 s, with larger temperature fluctuations of the HEX compared to operation with shorter $t_{\text{contact}}$ (Figure S5).

**Figure 6. Results of the parametric analysis for different contact times $t_{\text{contact}}$ between TSC and MCM and between TSC and HEX at zero cooling power**

The black line corresponds to the maximum temperature span for a given contact time $t_{\text{contact}}$ for stationary state and blue lines represent switching ratios $r$, also under stationary-state conditions. The parameters used in the simulations are presented in Table 2.
Influence of cooling-power density on thermal performance

In contrast to the simulations in the previous sections, a constant heat flux on the heat-source side was adopted as the boundary condition to simulate the heat transfer between the MCM and the heat sink/heat source. The cooling power density $q_c$ was varied between 1 and 2000 Wm$^{-2}$ for operating frequencies in the range from 0.1 Hz to 10 Hz and between 0.1 and 5000 Wm$^{-2}$ for an operating frequency of 15, 20 and 25 Hz (Figure 7). Other parameters and geometrical properties used in the simulations are presented in Table 2. As shown in Figure 7, the temperature difference between the ambient and the heat source decreases linearly with an increasing cooling-power density. The decrease in the temperature difference is significantly greater at lower operating frequencies. We can see a special case in Figure 7A, where the device at low operating frequencies operates above the ambient temperature, e.g., at an operating frequency of 0.1 Hz and cooling-power densities of $q_c > 55$ Wm$^{-2}$. In this case the device cannot provide the desired cooling performance because it is not able to remove all the heat from the heat sink, and the heat source starts to heat up, which cancels out the cooling effect. The maximum obtained cooling-power density was 4000 Wm$^{-2}$ at an operating frequency of 20 Hz and a zero temperature span (Figure 7C). This value corresponds to a specific cooling power $q$ of 2025 Wkg$^{-1}$m$^{-1}$m$^{-2}$ for 250-μm-thick Gd, which is more than seven times higher than the achieved cooling power at the zero temperature span of most Gd-based magnetocaloric devices with active magnetocaloric regeneration studied so far.

Influence of thermal contact resistance on the temperature span at zero cooling power

The main idea of this case study is to evaluate a realistic and experimentally achievable design of TSC embedded in a magnetocaloric cooling device with actual properties and geometry. Therefore, we

![Figure 7](image_url)

**Figure 7. Influence of cooling-power density on thermal performance**

Cooling-power density dependence of temperature difference between ambient temperature and heat-source temperature for (A) lower and (B) higher values of $q_c$, during stationary state for operating frequencies in the range 0.1 Hz–15 Hz. (C) Cooling-power density dependence of the temperature difference between ambient temperature and heat-source temperature for operating frequencies of 15, 20 and 25 Hz. Other parameters used in the simulations can be found in Table 2.
evaluated the performance by varying the values of the thermal contact resistance $R_c$ between the TSC and the thermally connected interface, in our case a dielectric layer. As described in the modeling approach section, we assume that the experimental value of the thermal contact resistance will be between $2.38 \times 10^{-6}$ to $3.7 \times 10^{-5}$ $\text{Km}^2/\text{W}$, which are the lower and upper limits of the range of analyzed thermal contact resistances, as shown in Figure 8.

With an increase in the thermal contact resistance, the temperature span becomes smaller at higher frequencies. As an example, if the thermal contact resistance is two orders higher from $2.38 \times 10^{-6}$ to $10^{-5}$ $\text{Km}^2/\text{W}$, the temperature span decreases by 371% at 15 Hz. On the other hand, a decrease in $R_c$ to a certain value ($2.38 \times 10^{-6}$ $\text{Km}^2/\text{W}$), the temperature span is not dominated by the $R_c$ anymore, as a magnitude change in the temperature span is negligible for $R_c$ lower than $10^{-6}$ $\text{Km}^2/\text{W}$, as shown in Figure 8.

Especially interesting is that the temperature span increases for higher values of $R_c$ at lower frequencies. This is related to an increased heat-transfer time and consequently also an increased heat flux (we approximated this particular part by heat diffusion) through the air gap in an undesirable direction, either toward the heat sink or the heat source, depending on the particular step of the thermodynamic cycle of the device, as shown in Figure S4 in supplemental information.

**Discussion of the potential application of the new TSC design**

In this section we show the relevance of our solid TSC for devices whose energy efficiency is decreasing when operating outside the optimum temperature range. Most electronic devices have an optimum operating temperature where their efficiency is the highest. For example, the maximum efficiency of batteries is at 25°C and decreases by 50% for every 10 K above that temperature. When the battery is exposed to a cold environment, its performance also decreases substantially. Maintaining the battery’s temperature in the optimum range is therefore challenging because of the conflicting requirements of conducting heat when the battery is exposed to a cold environment and facilitating cooling when exposed to a hot environment. Within this context, we theoretically evaluated the thermal conditions of a lithium-ion battery in both hot and cold environments. In the first case, a battery at a temperature of 23°C needed to be heated to reach its optimum operating temperature of 25°C. The battery was thermally insulated on one side to prevent heat from simply leaking away to the ambient, whereas the other side was in contact with a magnetocaloric system with a TSC. The convective boundary condition was defined on the heat-source side. In the second case, the ambient and the battery temperature were set at 27°C and the system needed to be cooled. Similar boundary conditions were defined, except that the heat source served as the heat sink. The operating frequency in both cases was set to 10 Hz. Figure 9 illustrates both scenarios. The system consists of a 5-mm-thick Li-ion battery with a density of

![Figure 8. Temperature span between heat sink and heat source as a function of different thermal contact resistances for Si (TSC) and SiO₂ (DL) interface for zero cooling power](image-url)
1940 kgm^{-3}, a heat capacity of 867 Jkg^{-1}K^{-1}, and a thermal conductivity of 4.44 Wm^{-1}K^{-1}. The respective thicknesses of the MCM (Gd), TSC (Si), DL (SiO₂), and heat exchanger (Si) used were 250, 200, 50, and 500 μm. The contact resistance between the TSC and the DL was set to 10^{-6} Km²W^{-1}. The thermal characteristics of all the considered components are listed in Table 2. An additional electrode (Ag, thickness 30 μm) and dielectric layer (SiO₂, thickness 50 μm) were added to electrically isolate the battery system.

We separated the heating and cooling processes because they are not necessarily applied sequentially. Therefore, the first phase corresponds to the heating phase and the second phase to the cooling phase, as shown in Figure 10. When the battery is exposed to an ambient temperature of 23°C, it reaches its optimum temperature in about 104 s, as shown in Figure 10. We would expect that a similar temperature span would occur in the cooling phase. However, at 23°C, the adiabatic temperature change of Gd is 3.1 K, and it is 0.84 K higher than at 27°C, for a magnetic field of 1 T, resulting in better device performance. Thus, the ability to cool or heat depends strongly on the initial temperature of the battery. The maximum adiabatic temperature change for Gd is 3.3 K at 22.1°C and a magnetic field of 1 T. During the cooling phase, the battery system reaches its final state at 25.2°C, which is above the optimum temperature. When the battery is operating in a range closer to the optimum temperature e.g., between 23.5°C and 26.5°C, the battery reaches the optimum temperature in 50.9 s in heating mode. Also, when the device is switched to cooling mode, it reaches the optimum temperature in 72.4 s (See Figure S6 A). We then repeated the same simulations, but changed the initial battery temperature to 20°C (heating) and 30°C (cooling), as shown Figure S6B. In this case the temperature difference is larger than the maximum adiabatic temperature change of the Gd, so to reach the optimum temperature we would need a regeneration process during the magnetocaloric cycles, which will be investigated in our future studies. As a single-stage device without regeneration, this device can be used in applications that require high-precision thermal management with low temperature fluctuations and where rapid switching between the heating and cooling modes is required. Batteries are only one of the potential applications. There are various applications where heat is a by-product, for example, in microprocessor packages, sensors, robots and other power electronic components.

Conclusions
We have numerically simulated the operating characteristics of a magnetocaloric device with two embedded thermal switch capacitors (TSCs). The proposed design represents a new alternative to the hydraulic systems used by most magnetocaloric (MC) devices. In addition to the new approach to guiding the heat flow in an MC device presented here, our main findings and contributions are summarized in the following:

- Our numerical model considers more realistic conditions, which are usually neglected in theoretical studies, such as: the change in thermal contact resistance between adjacent elements because of the movement of the TSC, air exchange in the air gap, realistic values of thermal conductivity in the on and off states (rather than a thermal conductivity of zero in the off state and an infinite value in the on state). Consideration of these parameters should bring the evaluated operation of the device closer to implementation in real applications.
It can be observed that the operating frequency plays the most important role in the temperature span of the device. The use of low frequencies leads to performance degradation because of increased heat-transfer losses in unfavorable directions. However, operating at too high frequencies also reduces the performance of the device. The results show that for the considered device, the best performance in terms of temperature span is achieved at a frequency of 10 Hz for heat capacities of the TSC between $127 \text{ JK}^{-1} \text{m}^{-1}$ and $506 \text{ JK}^{-1} \text{m}^{-1}$. As the frequency or heat capacity of the material continues to increase, the performance decreases because the heat does not diffuse through the entire thickness of the material during each heat-transfer period. If the device is operated at frequencies greater than 10 Hz, the heat capacity should be less than $127 \text{ JK}^{-1} \text{m}^{-1}$ to enhance the performance.

For a device with a fixed MCM thickness of 250 $\mu$m, described in the section thermal switch capacitor in a magnetocaloric device, the maximum temperature difference between the Si heat sink and the Si heat source was 2.5 K at an operating frequency of 10 Hz and operation without regeneration under no-load conditions. The ratio of the temperature span versus the adiabatic temperature change of gadolinium (Gd) under these conditions was 0.8. The maximum temperature span achieved in all the case studies was 2.8 K at an operating frequency of 5 Hz and an MCM thickness that is three times larger, i.e., 750 $\mu$m (Figure 5).

The considered magnetocaloric device with a TSC can operate at higher frequencies than most state-of-the-art active magnetocaloric regenerators (i.e., at frequencies higher than 5 Hz). The time to reach the steady state increases dramatically at lower operating frequencies. The system takes about 66.3 s to reach the steady state at an operating frequency of 0.5 Hz and about 7.8 s at 50 Hz, as shown in Figure S5. At lower operating frequencies, the device exhibits greater undulations compared to operation at higher frequencies.

To achieve the highest switching ratio, the contact time between the TSC and the MCM, and between the TSC and the HEX, must be optimized. The device described in the section thermal switch capacitor in a single-stage magnetocaloric device achieves a switching ratio of $r = 3.8$ with a contact time of 27.5 ms (Figure 6).

Most of the numerical models developed so far involve static solid-state thermal control devices that change their thermal conductance under the influence of an external field. Although these studies show promising results, they usually include unrealistic material properties. The realistic material properties and geometry of the TSC used in our numerical model allow a more realistic evaluation of a potential device. The implementation of a TSC would increase the compactness and reduce the complexity of MC devices, which would allow for a much higher operating frequency and
consequently a higher cooling-power density. For cooling powers above 500 Wm\(^{-2}\) (253 Wkg\(^{-1}\)mcm\(^{-1}\)), the operating frequency must be higher than 1 Hz, otherwise the device under these conditions will not perform cooling. The maximum cooling-power density for a zero temperature span is 4000 Wm\(^{-2}\) at 20 Hz, which corresponds to a specific cooling power of 2025 Wkg\(^{-1}\)mcm\(^{-1}\) for 250-\(\mu\)m-thick Gd. This value is more than seven times higher than the specific cooling power of most of the state-of-the-art Gd-based MC devices with active magnetocaloric regeneration, where specific cooling powers vary between 80 Wkg\(^{-1}\)mcm\(^{-1}\) and 275 Wkg\(^{-1}\)mcm\(^{-1}\) at zero temperature span.\(^{50}\)

**Limitations of the study**

The first limitation of the study is that the numerical results have not been experimentally confirmed, even though we considered realistic properties for the analyzed device. The most important parameter that would need to be evaluated experimentally is the thermal contact resistance. This will be addressed in our future studies. Second, heat losses to the surroundings are considered negligible, because the edges of the device should be thermally insulated and, in any case, represent only 5% of the total surface area of the device. We have further assumed that there is no Joule heating because of electrostatic actuation or increased convection associated with the motion of the TSC. If the Joule heating affects the performance of the device, the heat generation in the corresponding elements representing the electrodes will be included in the numerical model. Heat conduction through spacers separating MCM and heat exchangers is not considered in the numerical model. This has yet to be experimentally proven for our case and will be investigated in our future studies. The power consumption for actuating the TSC was not considered because realistic values are difficult to predict, and therefore the coefficient of performance for the cooling device was not determined. This will also be realized in future experimental work.

**Glossary**

**Nomenclature and units**

|  |  |
|---|---|
| B | Magnetic field (B) |
| Cp | Specific heat (J kg\(^{-1}\) K\(^{-1}\)) |
| E | Electric field (V/m) |
| F | Operating frequency (Hz) |
| G | Thermal conductance (Wm\(^{-2}\)K\(^{-1}\)) |
| H | Surface microhardness (Pa) |
| K | Thermal conductivity (W m\(^{-1}\) K\(^{-1}\)) |
| N | Number of magnetocaloric cycles (/) |
| P | Contact pressure (Pa) |
| \(\dot{q}_c\) | Cooling power density (Wm\(^{-2}\)) |
| \(\dot{q}\) | Specific cooling power (Wkg\(^{-1}\)) |
| R | Switching ratio (/) |
| Rc | Thermal contact resistance (Km\(^2\)W\(^{-1}\)) |
| T | Time (s) |
| T | Temperature (K, °C) |
| X | Position in x direction (m) |

**Greek letters**

|  |  |
|---|---|
| \(\Delta\) | Thickness (m) |
| \(\varepsilon\) | Dielectric constant (/) |
| \(\rho\) | Density (kgm\(^{-3}\)) |
| \(\chi\) | Time (s) |

**Subscripts, superscripts**

|  |  |
|---|---|
| Ht | Heat transfer |
| On | On |
| Off | Off |

(Continued on next page)
Continued

| Abbreviation | Description               |
|--------------|---------------------------|
| Mag          | Magnetization             |
| Demag        | Demagnetization           |
| Amb          | Ambient                   |
| Min          | Minimal                   |
| Ad           | Adiabatic                 |

**STAR★METHODS**

Detailed methods are provided in the online version of this paper and include the following:

- **KEY RESOURCES TABLE**
- **RESOURCE AVAILABILITY**
  - Lead contact
  - Materials availability
  - Data and code availability
- **METHOD DETAILS**

**SUPPLEMENTAL INFORMATION**

Supplemental information can be found online at https://doi.org/10.1016/j.isci.2022.105517.

**ACKNOWLEDGMENTS**

This work was financially supported by the Slovenian Research Agency as part of the Young Researcher PhD program. The authors also acknowledge the financial support of the Slovenian Research Agency for the research project funding no. ARRSJ2-1738 and for the research project no. J7-3148, and the core research program no. P2-0223.

**AUTHOR CONTRIBUTIONS**

Conceptualization, N.P. and A.K.; Methodology, N.P. and K.V.; Software, K.V., N.P., and K.K.; Writing – Original Draft, N.P. and A.K.; Writing – Review and Editing, N.P. and A.K.; Funding Acquisition, A.K.; Visualization, N.P.; Supervision, A.K.; Funding Acquisition, A.K.

**DECLARATION OF INTERESTS**

The authors declare no competing interests.

Received: August 22, 2022
Revised: October 13, 2022
Accepted: November 3, 2022
Published: December 22, 2022

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STAR METHODS

KEY RESOURCES TABLE

| REAGENT or RESOURCE | SOURCE | IDENTIFIER |
|---------------------|--------|------------|
| Software and algorithms | Python Software Foundation | https://www.python.org/ |
| MATLAB R2019a | MathWorks Incorporated | https://www.mathworks.com/ |

RESOURCE AVAILABILITY

Lead contact
Further information and requests for resources should be directed to, and will be fulfilled by, the lead contact, Prof. Dr. Andrej Kitanovski (andrej.kitanovski@fs.uni-lj.si).

Materials availability
This study did not generate new unique materials.

Data and code availability
- All data reported in this article will be shared by the lead contact upon request.
- The code with instructions reported in this article will be shared by the lead contact upon request.
- Any additional information required to analyze the data reported in this study is available from the lead contact upon request.

METHOD DETAILS

The governing equation for heat conduction (convection and adiabatic wall were included as a boundary condition) between the solid elements:

$$\frac{\partial^2 T}{\partial x^2} - \frac{\partial T}{\partial t} = 0 \quad \text{(Equation 3)}$$

Equation 3 was discretized for space (index n) and time (index i) for each node using the finite-difference implicit method. Depending on certain boundary conditions, temperatures were calculated for each discrete time ($t_i$) and space ($x$) with the following set of equations:

- Heat transfer between interior nodes $n$ and adjacent $n + 1$ and $n - 1$

$$T_{n}^{i+1}(2k_{\Delta t} + \rho c_{p}\Delta x^2) - T_{n-1}^{i+1}(k_{\Delta t}) - T_{n+1}^{i+1}(k_{\Delta t}) = T_{n}^{i}(\rho c_{p}\Delta x^2) \quad \text{(Equation 4)}$$

- Heat transfer at the interface between material A (node $n$) and material B (node $n+1$)

$$T_{n}^{i+1}(k_{\Delta t}\Delta x_{a} + \rho c_{p}\Delta x_{a}^2 + k_{a}\Delta t) - T_{n-1}^{i+1}(k_{\Delta t}) - T_{n+1}^{i+1}(k_{\Delta t}\Delta x_{a}) = T_{n}^{i}(\rho c_{p}\Delta x_{a}^2), \quad \text{(Equation 5)}$$

where the total thermal resistance between two dissimilar materials is expressed as:

$$k_{e} = \frac{2k_{a}k_{b}}{\Delta x_{a}k_{b} + 2R_{c}k_{a}k_{b} + \Delta x_{b}k_{a}} \quad \text{(Equation 6)}$$

- Boundary condition: convective heat transfer
\[ T_{n+1}^{i+1} \left( h_x \Delta x \Delta t + k \Delta t + \frac{\rho c_p \Delta x^2}{2} \right) = T_{n+1}^{i+1} (k \Delta t) = T_{\text{amb}} h_x \Delta x \Delta t + T_n \left( \rho c_p \frac{\Delta x^2}{2} \right) \]  
(Equation 7)

- Boundary condition: constant heat flow \( \dot{q}_c \)

\[ T_{n+1}^{i+1} (k \Delta t + \rho c_p \Delta x^2) - T_{n+1}^{i+1} (k \Delta t) = \dot{q}_c \Delta x \Delta t + T_n \left( \rho c_p \Delta x^2 \right) \]  
(Equation 8)

- Boundary condition: adiabatic wall

\[ T_{n+1}^{i+1} \left( k \Delta t + \rho c_p \frac{\Delta x^2}{2} \right) - T_{n+1}^{i+1} (k \Delta t) = T_n \left( \rho c_p \frac{\Delta x^2}{2} \right) \]  
(Equation 9)

- Magnetocaloric effect

applied magnetic field

\[ T_{n+1}^{i+1} = T_n^{i+1} + \Delta T_{\text{ad}} \left( T_{n}^{i}, B_{\text{on}}, B_{\text{off}} \right) \]  
(Equation 10)

removed magnetic field

\[ T_{n+1}^{i+1} = T_n^{i+1} - \Delta T_{\text{ad}} \left( T_{n}^{i}, B_{\text{on}}, B_{\text{off}} \right), \]  
(Equation 11)

where \( \Delta T_{\text{ad}} \) corresponds to the adiabatic temperature change when applying or removing the magnetic field, \( B \) is the magnetic field and \( \text{on} \) and \( \text{off} \) stand for the initial and final values.

- Specific heat of magnetocaloric material:

\[ c_H = \left( \frac{\partial q}{\partial T} \right)_H = T \left( \frac{\partial s}{\partial T} \right)_H \]  
(Equation 12)

The individual equations for each node are then written in matrix form and solved with Thomas’ algorithm used for tridiagonal systems of equations.