Ferrofluidic thermal switch in a magnetocaloric device

Ferrofluidic thermal switch

On state: high thermal conductivity

Off state: low thermal conductivity

Ferrofluidic thermal switches in a magnetocaloric device

off off

off on

on off

off off

Highlights

- A ferrofluidic thermal switch was numerically analyzed in a magnetocaloric device.
- The highest temperature span achieved was 1.12 K for a single embodiment.
- A sensitivity analysis was performed to evaluate the effects of all parameters.

Katja Klinar, Katja Vozel, Timm Swoboda, Tom Sojer, Miguel Muñoz Rojo, Andrej Kitanovski

andrej.kitanovski@fs.uni-lj.si

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Katja Klinar, Katja Vozel, Timm Swoboda, Tom Sojer, Miguel Muñoz Rojo, and Andrej Kitanovski

SUMMARY
Thermal switches are advanced heat-management devices that represent a new opportunity to improve the energy efficiency and power density of caloric devices. In this study we have developed a numerical model to analyze the operation and the performance of static thermal switches in caloric refrigeration. The investigation comprises a parametric analysis of a realistic ferrofluidic thermal switch in terms of the maximum temperature span, cooling power, and coefficient of performance. The highest achieved temperature span between the heat source and the heat sink was 1.12 K for a single embodiment, which could be further developed into a regenerative system to increase the temperature span. A sensitivity analysis is conducted to correlate the relationship between the input parameters and the results. We show that thermal switches can be used in caloric devices even when switching ratios are small, which greatly extends the possibilities to implement different types of thermal switches.

INTRODUCTION
Thermal switches are devices that allow (the on state) or forbid (the off state) heat to flow across them, in a similar way to how their electrical counterparts manage electricity. The first studies on thermal switches began in 1949, when Heer and Daunt (1949) investigated the change in the thermal resistance in superconducting and normal states for tin and tantalum at temperatures below 1 K. Since then, different mechanisms have been developed for applications operating at or above room temperature by implementing solid-state, fluidic, and mechanical thermal switches. These are described in a few recent review papers (Klinar et al., 2021; Swoboda et al., 2021; Wehmeyer et al., 2017). The performance of the thermal switch is determined by the switching ratio (the ratio of heat fluxes in the on and off states), the switching time (the time it takes to transition from on to off, and vice-versa), and the energy efficiency. In addition, it is worth distinguishing between static and moving thermal switches. Static thermal switches (evaluated here) do not change position while switching between the on and off states—they remain in physical contact with the neighboring interfaces at all times. However, moving thermal switches change their positions during the on and off states—they break the physical contact with the neighboring interfaces.

The research activities on thermal switches in caloric technologies for room temperature applications have rapidly increased in the last two decades (Klinar and Kitanovski, 2020). The main advantage of thermal switches over the currently widely used active caloric regeneration with an oscillating fluid flow is that they allow for a higher operating frequency (Kitanovski et al., 2015) (i.e., the number of thermodynamic cycles per unit of time). The higher the operating frequency, the higher the cooling/heating power.

In the literature (Klinar and Kitanovski, 2020), different mechanisms for static and moving thermal switches with electric, mechanic, electro-mechanic, and magnetic actuation have been theoretically and experimentally evaluated. The state of the art for fluidic thermal switches is summarized in Table 1. Silva et al. (2019) and Hess et al. (2019) tackled the implementation of thermal switches in caloric technologies more broadly by designing generalized numerical models. Silva et al. (2019) designed the numerical tool Heatrapy (Silva, 2017; Silva et al., 2018), which makes possible to evaluate caloric devices based on static thermal switches as well as on active caloric regeneration. On the other hand, Hess et al. (2019) presented a numerical model for the evaluation of caloric devices with a cascaded arrangement of thermal switches.

To improve the particular components or whole caloric devices with respect to the temperature span, cooling power, costs, and coefficient of performance (COP), different optimization strategies were used (Silva et al., 2019; Hess et al., 2019).
et al., 2021a; 2021b). However, most of these concern caloric devices based on active caloric regeneration and not thermal switches. For thermal switches, Silva et al., (2021a; 2021b) investigated the effect of the thermal-conductivity variation inside a thermal switch on the performance of the caloric device.

Besides the active caloric regeneration and related thermodynamic cycles (Plaznik et al., 2013), a magneto-caloric device can also operate with so-called single-stage thermodynamic cycles (Kitanovski (2020)), whereas the magnetic, non-regenerative Brayton’s thermodynamic cycle represents one of the most investigated thermodynamic cycles in previous studies. Its operation with thermal switches is illustrated in Figure 1B. The application of the external field increases the temperature of the caloric material due to the adiabatic temperature change ($T + \Delta T_{ad}$) as a consequence of the caloric effect. The removal of the field decreases the temperature of the caloric material ($T - \Delta T_{ad}$). In between these two adiabatic processes, two heat transfer processes occur. On the one hand, during the high isofield process, heat is transferred to the heat sink via the thermal switch 2, which is in the on state. The thermal switch 2 is embodied between the caloric material and the heat sink. Simultaneously, the thermal switch 1, which is embodied between the heat source and caloric material, is in the off state, which prevents heat transfer from the caloric material to the heat source. During the low isofield process, the situation reverses: thermal switch 1 is in the on state, whereas thermal switch 2 is in the off state, meaning that heat is transferred from the heat source to the caloric material and heat transfer from the heat sink to the caloric material is prevented. In the ideal case, heat transfer in the thermal switch is completely suppressed during the off state and the thermal switch represents the ideal adiabatic wall. In the real system, any heat transfer through the thermal switch in the off state leads to irreversible losses that affect the thermodynamic cycle.

Most of the state-of-the-art numerical analyses evaluated ideal thermal switches that exhibited zero thermal conductivity during the off state and a very large (infinite) thermal conductivity during the on state. The main goal of our study was to demonstrate the possibility of applying a realistic thermal switch in a single-stage non-regenerative magnetic Brayton cycle (for which it is well known to be energy inefficient) and thus to provide the missing proof that such an approach, even though inefficient, could still lead to a cooling/heat pumping effect. An example of a heat-regeneration arrangement with thermal switches is given in the discussion section. These principles allow an extension of the potential temperature difference between the heat source and heat sink and also a substantial improvement of the energy efficiency of a device. The reader is referred to Kitanovski et al. (2015) for a more detailed explanation.

## RESULTS

### Numerical model

The model evaluates a caloric embodiment comprising a caloric material sandwiched between two thermal switches, embodied between the heat source and heat sink at each end (Figure 1A). Figure 2 illustrates the flowchart of the numerical program. At the beginning, all the properties and operating parameters of the device are imported from a file. Then, the numerical program is divided into several stages, where each stage corresponds to a process in the Brayton thermodynamic cycle (Figure 1B). The solution of one stage is used as the initial solution for the next stage. All four Brayton thermodynamic processes repeat (minimum $N$ Brayton cycles) until the quasi-steady-state condition is met at the heat source (the change of the temperature in two consecutive time steps is smaller than the set tolerance). Then the program finishes and exports the data for subsequent evaluation.

The heat transfer in the model is based on the implicit finite-difference scheme using Fourier’s law of heat conduction (Equation 1) coupled with the caloric effect (Equations 2A and 2B) in a 1D caloric embodiment.
Nielsen et al. (2011), similar to Silva et al. (2019). Equation (1) considers the properties (thermal conductivity $k$, specific heat $c_p$, and density $\rho$) of each component with regard to time $t$, location $x$, and external trigger $F$.

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

(Equation 1)

Equation (1) is solved for all the nodes of the caloric embodiment. Additional information about the numerical model can be found in STAR Methods and in the supplemental information (Figures S1, S2 and S3).

The caloric effect is implemented by a temperature change of each node of the caloric material, as in most caloric numerical models (Nielsen et al., 2011).

\[
T_{fi} = T_{in} + \Delta T_{ad,app}(T_{in}, F_{fi}, F_{in}) ,
\]

(Equation 2A)

\[
T_{fi} = T_{in} - \Delta T_{ad,rem}(T_{in}, F_{fi}, F_{in}) ,
\]

(Equation 2B)

where $F$ stands for trigger type (magnetic field, electric field, force or pressure), “fi” for the final value, “in” for the initial value, “ad, app” for the adiabatic external field application, and “ad, rem” for the adiabatic external field removal. The model allows the use of any caloric material, as long as we provide the required tables of properties: total entropy in relation to the temperature and trigger intensity.

The thermal conductivity of thermal switches is considered to be time dependent, and following the external trigger change; it is referred to as “low” in the off state and “high” in the on state. The convective boundary
condition is defined on the side of the heat sink; the constant-heat-flux boundary condition is applied for the side of the heat source (simulating cooling power), as presented in Figure 1A.

To bring the results of the numerical simulations closer to the potential experiments, the model includes the effects of the thermal contact resistance and the internal heat generation (Figure 1A). We considered the thermal contact resistances for four interfaces between different embodied components of the caloric embodiment (heat source, heat sink, thermal switches, and caloric material, Figure 1A). Internal heat generation is a consequence of effects accompanying the actuation of the thermal switches that transform into heat and heats the thermal switch (Joule heating, eddy currents, friction, etc.). More about these issues can be found in the following references (Klinar et al., 2021; Swoboda et al., 2021; Wehmeyer et al., 2017).

The exported data at the end of the simulation consist of the temperature evolution for each node, the cooling and heating powers, the magnetic work, and the COP. The numerical model is validated with the numerical model heatrapy (Silva et al., 2019). The details on validation are available in STAR Methods and in supplemental information (Figure S4).

**Magnetocaloric thermal switch**

Here we evaluated a magnetocaloric embodiment consisting of a magnetocaloric material embodied between two thermal switches, a heat source and a heat sink. The model is 1D along the thickness of the magnetocaloric embodiment. The height and width (Figure 1A) of the magnetocaloric embodiment are not defined, except for one exemplary case in Table 2.

**Magnetic field sources**

Three unrelated (electro)magnetic field sources are considered in the magnetocaloric embodiment, one for the magnetocaloric material and one for each of the thermal switches. The (electro)magnetic field applied to the magnetocaloric material is changed in a stepwise manner between $B_{\text{min}} = 0$ and $B_{\text{max}} = 1$ T (e.g., Klinar et al., 2019). The magnetization and demagnetization times are considered to be 5 ms, and the magnetic flux density is homogeneous over the magnetocaloric material. The (electro)magnetic fields applied to each of the switches are changed in a stepwise magnetic field function between 0 and 0.05 T. These two (electro)magnets operate alternately, do not interact, and are not affected by the main magnetic field source (that corresponds to the magnetocaloric material).

**Magnetocaloric material**

We chose gadolinium as the magnetocaloric material. The specific heat of gadolinium is calculated from the mean field theory (Kitanovski et al., 2015) in relation to its temperature. Gadolinium’s density is assumed to be 7,900 kgm$^{-3}$ and its thermal conductivity is 10.5 Wm$^{-1}$K$^{-1}$. The magnetocaloric effect depends on the absolute values of $B_{\text{max}}$ and $B_{\text{min}}$ and the relative change $B_{\text{max}} - B_{\text{min}}$. The temperature and magnetic-field dependence of the specific entropy, specific heat, and adiabatic temperature change are provided in Figures S1–S3.
Heat exchangers
Heat source (the thermal load) and heat sink (the heat exchanger to ambient) are considered to be made of non-magnetic stainless steel. The density is assumed to be 7,870 kg m\(^{-3}\), the thermal conductivity, 15 W m\(^{-1}\) K\(^{-1}\), and the specific heat, 450 J kg\(^{-1}\) K\(^{-1}\).

Thermal switch
The main idea is to implement a static thermal switch with realistic properties. We checked the literature to find the most appropriate principle and materials with high switching ratios and short response times. We chose the experimental results from magnetic nanoparticles dispersed in a heat-transfer oil (HTO) by Katiyar et al. (2016). They measured a 2-fold increase of the thermal conductivity as a consequence of the increased magnetic field from 0 to 0.05 T in a 7.0 vol.% Fe particle concentration. Its thermal conductivity increases owing to the reorientation of the magnetic particles into chain-like structures along the direction of the magnetic field (this defines the direction of the magnetic field). The process is reversible—under zero magnetic field, particles reorientate randomly again, which decreases the thermal conductivity. Although the measurements of thermal conductivity in the experiment by Katiyar et al. (2016) took a few minutes, the thermal conductivity change—the formation of chain-like structures inside the ferrofluid—occurs in a few milliseconds (Zhang et al., 2020). This time can be further decreased if the magnetic field is applied in a perpendicular direction. In the model we assumed a reversible and instant (5 ms, the same as the (de)magnetization process) thermal conductivity change with the change of magnetic field. We calculated the density (Pak and Cho (1998)) and specific heat (Jama et al., 2016) using equations for magnetic nanofluids. Table 2 presents the properties of the thermal switches used in the numerical analysis.

Operating parameters
The operating frequency is set to 20 Hz. The convection boundary condition is defined with an ambient temperature of 293 K and a heat transfer coefficient of 10,000 W m\(^{-2}\) K\(^{-1}\), which can correspond to an external water cooling system. Multiple simulations are performed in a parametric sweep: cooling power is varied between 0 and 900 W m\(^{-2}\), thermal contact resistances and internal heat generation are varied between 0 (ideal case) and maximum values that cancel out the cooling effect of the magnetocaloric embodiment. The quasi-steady-state condition is achieved when the average temperature fluctuation in the heat source is less than 10 K between two consecutive cycles.

The thermal performance of the magnetocaloric embodiment can be analyzed in terms of the temperature span between the heat source and heat sink in the quasi-stationary state (Figure 3B) or in terms of the cooling power. Their relation is as follows: the maximum temperature span is achieved at zero cooling power and the maximum cooling power at zero temperature span. We are looking for a considered magnetocaloric embodiment that has the largest temperature span at zero cooling power.

The temperature profile along the considered magnetocaloric embodiment in the quasi-steady state is related to the thicknesses of the heat source/sink, the thermal switches, and the magnetocaloric material. To achieve the largest-possible temperature span, the thermal switches must thermally compensate for the temperature fluctuations of the magnetocaloric material in an effective way. Figure 3 shows the temperature evolution from the beginning until the quasi-steady-state operation and the temperature profile along the magnetocaloric embodiment during the quasi-steady-state operation for two typical situations. Figures 3A and 3B show the first situation where the thermal switches effectively compensate the oscillations, leading to a constant temperature at the heat sink and the heat source. This makes it possible to have directed heat flux from the heat source to the heat sink with no heat flow in an undesired direction. The second situation is the case where the thermal switch is not able to thermally compensate for the oscillation.

### Table 2. Properties of thermal switches

| Density [kg m\(^{-3}\)] | Specific heat [J kg\(^{-1}\) K\(^{-1}\)] | Low thermal cond. (at \(B = 0\) T) [W m\(^{-1}\) K\(^{-1}\)] | High thermal cond. (at \(B = 0.05\) T) [W m\(^{-1}\) K\(^{-1}\)] | Thermal cond. ratio [\(r\)] |
|-------------------------|---------------------------------|---------------------------------|---------------------------------|-------------------------|
| 1,358                   | 237                             | 0.29                            | 0.58                            | 2                       |

Heat exchangers
Heat source (the thermal load) and heat sink (the heat exchanger to ambient) are considered to be made of non-magnetic stainless steel. The density is assumed to be 7,870 kg m\(^{-3}\), the thermal conductivity, 15 W m\(^{-1}\) K\(^{-1}\), and the specific heat, 450 J kg\(^{-1}\) K\(^{-1}\).
of the temperature of the magnetocaloric material, which is evident from the fluctuation of the temperature of the heat source and the heat sink (Figures 3C and 3D). There is a period where the temperature of the heat sink is below the ambient temperature, which leads to a heat flow in an undesired direction, thus decreasing the cooling effect.

Results of the parametric analysis of the magnetocaloric embodiment: optimizing the thickness

First, the simulations are performed for different thicknesses of the thermal switches, magnetocaloric material, and heat sink/source. When evaluating the different thicknesses of the heat sink and heat source, the highest temperature span is obtained when the heat sink and the heat source each have a thickness of 0.2 mm. More interesting are the results of the different combinations of the thermal switch and the magnetocaloric material thicknesses presented in Figure 4. There is a minimum thickness of magnetocaloric material (0.3 mm in this case) required to achieve the maximum temperature span, but increasing the thickness further under the same conditions no longer affects the maximum temperature span. The thickness of the thermal switch also affects the temperature span. With thinner thermal switches, the temperature fluctuations of the heat sink (Figures 3A and 3C) are too significant, whereas thicker thermal switches tend to accumulate too much heat, which leads to a reduction in the total temperature span between the heat source and the heat sink. The highest temperature span between the heat source and the heat sink in all the conducted simulations is 1.15 K, corresponding to a magnetocaloric material and a thermal switch thickness of 0.3 and 0.1 mm, respectively. However, for this case the temperature fluctuations inside the
heat sink and the heat source are too significant (this case is shown in Figures 3C and 3D). The case without fluctuations of the temperatures in the heat sink and heat source leads to a maximum temperature span of 1.12 K, which corresponds to a magnetocaloric material and thermal switch thickness of 0.3 and 0.25 mm, respectively (this case is shown in Figures 3A and 3B). The thickness of the heat sink and the heat source for all cases is considered to be 0.2 mm. All subsequent analyses will be based on these considered thicknesses for the components of the magnetocaloric embodiment system.

Results of the parametric analysis for the magnetocaloric embodiment: thermal performance

In the following text we present the results for the thermal performance of the considered magnetocaloric embodiment that shows the highest temperature span between the heat sink and the heat source without temperature fluctuations. The influence on the performance can be determined based on the reduction in the maximum temperature span between the heat source and the heat sink and on the values of the following parameters: cooling power, COP, contact thermal resistance between components, and internal heat generation inside the thermal switches.

The cooling power is varied between 0 and 900 Wm\(^{-2}\), and it is simulated via a constant-heat-flux boundary condition from the heat source (heat load to the magnetocaloric embodiment). In our case the temperature of the heat sink is slightly above the ambient temperature. Figure 5 presents both temperature spans: heat source to heat sink and heat source to ambient. The results presented in Figure 5 show that the maximum temperature span between the heat source and the heat sink at zero cooling power is 1.12 K, whereas it is 1.1 K between the heat source and ambient. The maximum cooling power of 850 Wm\(^{-2}\) is achieved for a near-zero temperature span between the heat sink and heat source, which is the case where the temperature of the heat source is still below the ambient temperature \((T_{\text{heat source}}-T_{\text{ambient}} = 0.03 \text{ K}, T_{\text{heat source}}-T_{\text{ambient}} = 0.13 \text{ K})\). At a cooling power of 900 Wm\(^{-2}\), the temperature of the heat source increases above the ambient temperature, thus canceling out the cooling effect. Assuming a 1-mm-high and 1-mm-wide magnetocaloric embodiment, the total mass is 6.18 mg and the specific cooling power of gadolinium is 0.37 Wg\(^{-1}\).

The COP is calculated with Equations (3A) and (3B) using the cooling (thermal load to heat source) and heating (from the heat sink to the ambient) heat fluxes:

\[
\text{COP} = \frac{q_{\text{cooling}}}{q_{\text{heating}} - q_{\text{cooling}}} \quad (\text{Equation 3A})
\]

![Figure 4. Results: optimizing the thickness](image-url)
The COP increases with the cooling power to a maximum COP = 8.5, or COP_{Carnot} = 1,513 for a cooling power of 850 Wm\(^{-2}\). It is important to note, however, that the COP of the considered embodiment can only serve as the performance criterion for the selection of the best configuration of thermal switches and other components. As denoted before, in order to be energy efficient, the real magnetocaloric embodiment also must involve the regenerative process. This can be done by the serial integration of multiple embodiments, and coupling them with the external counter flow of the working fluid, which connects the heat source and the heat sink. Such a configuration is shown in Figure 8.

We also analyzed the effects of the thermal contact resistance and the internal heat generation. Both led to a reduction in the temperature span compared with the case where the two aforementioned effects are neglected. It is difficult to predict their exact values; therefore, we searched for the limiting value that cancels out the cooling effect of the magnetocaloric embodiment. In both cases the worst scenario is evaluated. Following that, the potential prototype requires lower values; otherwise, the cooling device will not work. The results in Figure 6 show that, for the optimal concept of the considered magnetocaloric embodiment, the limiting contact thermal resistance (considered to have the same value for each contact between the components of a magnetocaloric embodiment at all times) in the magnetocaloric embodiment is \( R_{\text{con}} = 0.006 \, \text{Km}^2\text{W}^{-1} \). The value is in accordance with the experimental thermal resistances reported in literature (Cengel (2002)). The limiting value for the constant (at all times) internal heat generation is \( q_{\text{gen}} = 50 \, \text{Wm}^{-2} \) for each thermal switch.

The results of thermal performance section are summarized in Table 3.

**Sensitivity analysis**

A sensitivity analysis was performed to evaluate the multi-parametric effect on the performance of the considered magnetocaloric embodiment. Our model is evaluated with the one-at-a-time (OAT) method, which is the simplest and most common method for a sensitivity analysis (Singiresu 2020). Using OAT, first the nominal case is calculated using the nominal (average) parameters for the conducted simulations. The nominal parameters for our case are presented in Table 4. Then, multiple simulations are performed for a ±50% change of one parameter while keeping the others at their nominal values to define the interval of possible values for each parameter. Then, a tornado chart is plotted, with the parameters having the largest impact displayed on top and the parameters with the smallest impacts shown on the bottom, as illustrated in Figure 7. We decided not to change all the parameters of the model; we fixed the chosen...
magnetocaloric material gadolinium with its temperature and field-dependent properties and an ambient temperature of 293 K. The results for the temperature span between the heat sink and heat source during the quasi-steady state and the COP of the considered magnetocaloric embodiment are presented in Figure 7. The largest effect on the temperature span is observed for the thermal conductivities of the thermal switch, specifically the \( k_{\text{off}} \). This is expected as the thermal conductivity during the off state of the thermal switch is directly related to the undesired heat transfer (from the magnetocaloric material to the heat source during the high isofield heat transfer process and the heat transfer from the heat sink to the magnetocaloric material during the low isofield heat transfer process).

The reason why the result of varying the thickness of the thermal switch in Figure 7A does not touch the nominal vertical line and is presented with discrete points is that the relation is not linear (Figure 4). The optimal thickness is chosen as nominal—all the other thicknesses result in a smaller temperature span.

**Figure 6. Results: thermal contact resistance and internal heat generation**

Evaluation of the thermal performance of the considered magnetocaloric embodiment using the ferrofluidic thermal switch chosen in section of thickness optimization (A) Effect of the thermal contact resistance between the components inside the magnetocaloric embodiment and (B) effect of the internal heat generation inside the thermal switches. Both cases are evaluated for zero cooling power.

Table 3. Operating parameters and summary of the results for the considered magnetocaloric embodiment using a ferrofluidic thermal switch

| Operating parameters | 1 T | Ambient temperature | 293 K |
|-----------------------|-----|---------------------|-------|
| Number of thermal switches in the embodiment | 2 | Thickness of gadolinium | 0.3 mm |
| Magnetocaloric material | Gadolinium | Thickness of thermal switch | 0.25 mm |
| Heat sink/source material | Stainless steel | Thickness of heat source/sink | 0.2 mm |
| Thermal switch material | HTO with Fe nanoparticles | Operating frequency | 20 Hz |
| Total thickness of the embodiment | | | 1.2 mm |

| Results | 1.12 K | Max contact thermal resistance (zero cooling power) | 0.006 Km²W⁻¹ |
|---------|--------|-----------------------------------------------|----------------|
| Max cooling power | 850 W m⁻² | Max internal heat generation (zero cooling power) | 50 Wm⁻² |
| Max COP | 8.5 | | |

Example: assuming 1-mm-high and 1-mm-wide magnetocaloric embodiment (Figure 1A)

| Mass of two thermal switches | 0.68 mg |
| Mass of both, heat source and heat sink | 3.2 mg |
| Mass of magnetocaloric material | 2.3 mg |
| Max specific cooling power | 0.37 W g⁻¹ gadolinium |
On the other hand, $B_{\text{max}}$ has the largest effect on the COP, since it affects the magnetocaloric effect and the cooling power. Namely, the large magnetocaloric effect concerns the large adiabatic temperature change, which further decreases the share of the irreversibility related to the heat transfer (Kitanovski et al., 2015). Both graphs in Figure 7 show the small contribution to the temperature span and the COP by the convective heat transfer coefficient and the thermal conductivity of the heat source and heat sink.

The OAT sensitivity analysis for the chosen nominal case revealed the parameters that influence the temperature span and the COP the most. The three most important parameters for the temperature span are the thermal conductivities of the thermal switches during the off and on states and the maximum magnetic field $B_{\text{max}}$, whereas the most important parameters for the COP are the maximum magnetic field $B_{\text{max}}$, the cooling power, and the operating frequency. However, the sequence of parameters and the size of their interval in the tornado chart could be different for a different nominal case.

**DISCUSSION AND FUTURE WORK**

A numerical model has been developed to evaluate the static thermal switches in their embodiment with the caloric material together with the heat source and heat sink. The presented model represents the most comprehensive evaluation tool in the literature, because it also includes effects that are usually neglected (e.g., thermal mass, thermal contact resistance between components, internal heat generation in the thermal switch). In this way the results are expected to be very close to those measured with experimental setups, which is crucial for the future development of thermal switches for application in caloric technologies.

We wanted to consider a realistic thermal switch; however, the literature on appropriate ferrofluids that exhibit a fast and sufficiently large change in thermal conductivity is limited. All the properties of the considered ferrofluidic thermal switch are taken from Katiyar et al. (2016), except the response time, which was not provided in the reference. We considered a response time of 5 ms, which is the same as the (de)magnetization process. The highest temperature span for the considered gadolinium magnetocaloric device at 20 Hz is 1.12 K. The maximum cooling power at a zero temperature span is 850 Wm$^{-2}$, whereas the specific cooling power at the near zero temperature span is 0.37 W g$^{-1}$ for gadolinium (assuming a total height of the embodiment of 1 mm and a width of 1 mm). The value is comparable with gadolinium’s specific cooling power when an active magnetocaloric regenerator is used instead of thermal switches. The analysis of the limiting thermal contact resistance (that cancels out the cooling effect) between each component ($R_{\text{con}} = 0.006 \text{ K m W}^{-1}$) and the limiting internal heat generation (that cancels out the cooling effect) of $q_{\text{gen}} = 50 \text{ W m}^{-2}$ confirm the feasibility of building the prototype device.

The simplest figures of merit for a particular embodiment are its COP, maximum cooling power, and the ratio of the temperature span between the heat source and heat sink versus the adiabatic temperature change of the magnetocaloric material. These figures of merit are rather low for the considered embodiment; however, the temperature span and, consequently, the cooling power and the COP can be further increased with, e.g., thermal regeneration (Hess et al., 2019; Kitanovski et al., 2015). This will also require multiple embodiments consisting of a plural number of “layered” and different magnetocaloric materials according to their Curie temperature. Figure 8 shows how the thermodynamic cycle of operation for the exemplary case of potential implementation of four embodiments should look like. The embodiments

| Table 4. Parameters chosen for the nominal simulation |
|---------------------------------|---------------------------------|---------------------------------|
| Thermal switch thickness 0.25 mm | Therm. cond. gadolinium 10.5 Wm$^{-1}$K$^{-1}$ |
| Heat source/sink thickness 0.2 mm | Therm. cond. heat source/sink 15 W m$^{-1}$K$^{-1}$ |
| Gadolinium thickness 0.3 mm | Cooling power 100 W |
| $\rho$ Thermal switch 1358 kg m$^{-3}$ | Convection coefficient 10,000 W m$^{-2}$K$^{-1}$ |
| $c_p$ Thermal switch 237 J kg$^{-1}$K$^{-1}$ | Internal heat gain 1 W m$^{-2}$ |
| $k_{\text{on}}$ 0.58 W m$^{-1}$K$^{-1}$ | Thermal contact resistance 0.0001 m$^2$K W$^{-1}$ |
| $k_{\text{off}}$ 0.29 W m$^{-1}$K$^{-1}$ | $B_{\text{min}}$ (magnetocaloric material) 0.1 T |
| $\rho$ Heat sink/source 7870 kg m$^{-3}$ | $B_{\text{max}}$ (magnetocaloric material) 1 T |
| $c_p$ Heat source/sink 450 J kg$^{-1}$K$^{-1}$ | $t_{\text{mag/demag}}$ 0.005 s |
| $\rho$ Gadolinium 7900 kg m$^{-3}$ | Frequency 10 Hz |

| Thermal switch thickness 0.25 mm | Therm. cond. gadolinium 10.5 Wm$^{-1}$K$^{-1}$ |
| Heat source/sink thickness 0.2 mm | Therm. cond. heat source/sink 15 W m$^{-1}$K$^{-1}$ |
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| $c_p$ Heat source/sink 450 J kg$^{-1}$K$^{-1}$ | $t_{\text{mag/demag}}$ 0.005 s |
| $\rho$ Gadolinium 7900 kg m$^{-3}$ | Frequency 10 Hz |
are de/magnetized alternatively, i.e., I. and III., and II. and IV. at the same time. The heat transfer between the embodiments is enabled by the external, unidirectional, and continuous counter fluid flow, which also allows for heat regeneration between both isofield processes. The heat transfer fluid also couples the heat-sink and the heat-source heat exchanger with the high- and low-field regions of the layered embodiments.

Future work should include an extension of the numerical model to moving thermal switches (those that move between the different positions) and the extension into multiple embodiments that form a realistic (magneto)caloric device. Moreover, because of the very large number of influential and temperature- and time-dependent parameters, optimization methods are required (including the possible use of neural networks) that will reduce the computation time and serve for the validation of the numerical model with future experiments.

Limitations of the study
The main limitation is the experimental proof of the promising numerical results. As described earlier in the discussion, we analyzed a thermal switch for which we considered experimentally measured properties except for the response time. We were not able to find experimental evidence that the response time for such a thermal switch could be 5 ms. This problem remains open and will be realized in future work.

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

Figure 7. Sensitivity analysis tornado charts for the results of the OAT sensitivity analysis
(A) The effect on the temperature span between the heat source and heat sink,
(B) The effect on the COP of the considered magnetocaloric embodiment. Dotted vertical line represents the nominal case, described in Table 4.
Figure 8. Arrangement for heat regeneration

T-s diagram of the layered embodiment of thermal switches (as it would most likely look), where heat regeneration between isofield heat transfer processes is enabled by the specific arrangement of the device parts. This is a proposed composition that has not yet been realized in magnetocalorics. The dotted line shows the flow direction and continuous counter fluid flow of the heat transfer fluid.
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STAR★METHODS

KEY RESOURCES TABLE

| RESOURCE                | SOURCE                                      | IDENTIFIER               |
|-------------------------|---------------------------------------------|--------------------------|
| Software and algorithms | Python 3                                    | Python Software Foundatio| https://www.python.org/ |
|                         | Heatrapy                                    | Python library for simula| https://github.com/djsilva99/heatrapy |
|                         | Origin 2021b                                | OriginLab Corporation    | https://www.originlab.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.
- Code with instructions reported in this article will be shared by the lead contact upon request.
- Any additional information required to reanalyse the data reported in this study is available from the lead contact upon request.

METHOD DETAILS

Implicit finite-difference numerical model for evaluation of heat transfer was designed in Python programming language. The main equation

$$\frac{\partial T}{\partial t} + \frac{k(T)}{c_p(T)\rho} \frac{\partial^2 T}{\partial x^2} = 0$$  \hspace{1cm} (Equation 1)

was discretized in time (index \(i\)) and space (index \(m\)) for different nodes. The equations were rewritten in a trigonal matrix using the coefficients \(a, b, c, z\) in the following order:

$$\begin{bmatrix}
b_0 & c_0 & 0 & 0 & 0 & 0 \\
a_1 & b_1 & c_1 & 0 & 0 & 0 \\
\vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\
\vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\
0 & 0 & 0 & a_{m-1} & b_{m-1} & c_{m-1} \\
0 & 0 & 0 & 0 & a_m & b_m \\
\end{bmatrix} \begin{bmatrix}
T_{i+1}^0 \\
T_{i+1}^1 \\
\vdots \\
\vdots \\
T_{i+1}^{m-1} \\
T_{i+1}^m \\
\end{bmatrix} = \begin{bmatrix}
z_0 \\
z_1 \\
\vdots \\
\vdots \\
z_{m-1} \\
z_m \\
\end{bmatrix}$$  \hspace{1cm} (Equation 2)

where, for example, the coefficients

- for the calculation of \(T_{i+1}^0\) of the node 0 (the left border – heat/flux boundary condition) are

$$a_0 = 0$$  \hspace{1cm} (Equation 3)

$$b_0 = k_{\text{hex}} dt + \rho_{\text{hex}} c_{p,\text{hex}} dx^2 / 2$$  \hspace{1cm} (Equation 4)

$$c_0 = -k_{\text{hex}} dt$$  \hspace{1cm} (Equation 5)

$$z_0 = T_0^0 \rho_{\text{hex}} c_{p,\text{hex}} dx^2 / 2$$  \hspace{1cm} (Equation 6)
• node m, right convection boundary condition for $T_m^{i+1}$

$$a_m = -k_{hax} \text{dt}$$  \hspace{1cm} (Equation 7)

$$b_m = (h \text{dx} + k_{hax}) \text{dt} + \rho_{hax} c_{p,hax} \text{dx}^2 / 2$$  \hspace{1cm} (Equation 8)

$$c_m = 0$$  \hspace{1cm} (Equation 9)

$$z_m = hT_{\text{air}} \text{dt} \text{dx} + T_m \rho_{hax} c_{p,hax} \text{dx}^2 / 2$$  \hspace{1cm} (Equation 10)

The unknown temperatures were solved with Thomas’ algorithm and the heat fluxes for each time step at the interface of the heat source and heat sink were calculated. Multi-parametric analysis was carried out to find the parameters for optimal cooling performance. Gadolinium properties are provided in Figures S1–S3.

The numerical model presented in this work was validated with the heatrapy numerical model by Silva et al. (2019). We set the same parameters for the magnetocaloric device and the gadolinium properties. We then run the simulations (parameters are written in section 3 in supplemental information) and compared the results in Figure S4 in supplemental information.

QUANTIFICATION AND STATISTICAL ANALYSIS

Sensitivity analysis (one-at-a-time) was carried out to see the effects of parameters in the numerical model.