Numerical optimization of the operation of a magnetocaloric heat pump

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Abstract

Magnetocaloric refrigeration is a very attractive alternative to traditional vapour compression devices for heating and cooling systems. It is potentially an environmental friendly and efficient technology.

Magnetocaloric devices are complex systems, where not only the individual parts have to be developed stand-alone, but also investigated as part of a system. The design of a novel magnetocaloric heat pump at the Technical University of Denmark gives, in particular, the chance of analyzing the interplay of flow system and magnetic field. Numerical simulations are performed to find the optimum working conditions. The method of analysis, used within this dissertation, leads to results applicable to the particular prototype but of general interest.

The necessity of considering the width of the regenerators in the identification of the best working conditions gives rise to the idea of a new 2D approach. Furthermore, the usage of such a heat pump in a real context faces possible deviations of the working parameters from the design values. The realization of partial load conditions is not trivial, and is definitely different from vapour compression systems.

Eventually, experimental measurements of the resistances through the active magnetic regenerators give the chance to validate the pressure drop correlation used in the model. The regenerators are found to have different resistances compared to each other. A possible balancing of the circuit is therefore investigated.
La refrigerazione magnetocalorica è un’interessante alternativa ai tradizionali sistemi di riscaldamento e raffreddamento a compressione di vapore. Rappresenta potenzialmente un’efficiente tecnologia a basso impatto ambientale.

Gli apparecchi magnetocalorici sono sistemi complessi, dove non solo bisogna sviluppare i singoli componenti, ma è fondamentale studiarli anche in relazione agli altri elementi, come parte di un sistema. La progettazione di una nuova pompa di calore magnetocalorica, costruita all’Università tecnica della Danimarca, fornisce la possibilità, in particolare, di analizzare l’interazione reciproca tra campo magnetico e sistema di flusso. Simulazioni numeriche consentono lo studio delle condizioni di lavoro di ottimo. Il metodo di analisi seguito conduce a risultati riferiti al singolo prototipo ma di interesse generale.

La necessità di considerare l’ampiezza dei rigeneratori porta, poi, allo sviluppo di una approssimazione 2-dimensionale. Quindi, si analizza l’introduzione di un apparecchio del genere in un sistema reale. Possibili deviazioni dai parametri di progetto e i loro effetti sono prese in considerazione. Viene studiata la non banale e fondamentale realizzazione delle condizioni di carico parziale.

Infine, misure sperimentali della resistenza al flusso attraverso i rigeneratori danno la possibilità di convalidare la correlazione per la caduta di pressione usata nel modello. Viene, inoltre, evidenziato che i rigeneratori hanno resistenze diverse tra loro. Si studia, quindi, un possibile bilanciamento del sistema.
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Introduction

Magnetocaloric refrigeration is an emerging technology, that aims at competing with traditional vapour compression systems. It is based on the exploitation of the magnetocaloric effect (MCE), that is the reversible adiabatic temperature change that magnetocaloric materials (MCM) undergo at magnetization.

As the adiabatic temperature change of a single material is of a few degrees even near its Curie temperature, a single sample cannot simply be used in heating and cooling devices. The widespread solution is the realization of an active magnetic regenerator (AMR), in which the MCM is also a thermal regenerator.

In a typical cycle, the AMR is magnetized, and its temperature increases. A fluid flow from the cold side, at $T_C$, to the hot side, at $T_h$, ensures the warming up of the working fluid. The AMR is, then, demagnetized, and its temperature decreases. Finally, the fluid flow in the hot-to-cold direction makes the fluid cool down. The state of the magnetic field during the period defines the magnetic field profile. The fraction of the period when there is fluid flowing through the regenerator (blow fraction) and there is none (pause fraction) defines the flow profile.

This thesis takes its place within the ENOVHEAT project, with the final goal of building a magnetocaloric heat pump for the residential sector. This device has thirteen regenerators filled with first order magnetocaloric materials, and a permanent magnet that rotates over the regenerators.

The choice of the magnetocaloric materials and the design of a variable magnetic field are fundamental, but the single components of the machine have also to be considered together as an overall system, with the purpose of maximizing the performances of the heat pump.

The study is carried out numerically. Results may be, then, easily put into practice in the ENOVHEAT prototype, thanks to the use of two solenoid valves and an independent control system for each regenerator. Indeed, the opening and closing time of each single valve, the volume flow rate, and the motor speed are controlled via a FPGA integrated circuit.
The common thermodynamic cycles for magnetocaloric devices are the Brayton one, that is adiabatic magnetization and demagnetization, and isofield fluid flows; and the Ericsson cycle, where magnetization and demagnetization are isothermal [24],[62].

Within this dissertation, it is shown that a well-defined thermodynamic cycle may be detrimental for the machine's performances. A broader investigation of the interplay between magnetic field profile and flow profile leads to better results.

Benedict et al. [60] show that it is not convenient to transfer heat during magnetization and demagnetization. Premature heat transfer during magnetization or demagnetization does not let the MCM reach low and high temperatures.

Teyber et al. [61] point out that displacing the fluid when the magnetic field is near the maximum and minimum values, increases the average magnetic field change during the blow periods. A late blowing limits the amount of heat that can be exchanged, and the higher temperature difference between the regenerator and the ambient causes higher thermal losses.

Bjørk and Engelbrecht [10] examine numerically the non-synchronization of magnetic field and flow profiles. They show significant performance drops for early ramps, no changes for slightly late ramps, and performance drops for very late ramps.

Existing prototypes [27], [30], [31],[32] do not typically utilize first order materials. Indeed, first order materials show hysteretical behaviours, and have a significant MCE only really close to their Curie temperature. Hysteresis is minimized in the LaFeSi compound of the ENOVHEAT machine with the introduction of Mn [16]. About the inelasticity of such a device, numerical simulations represent possible realistic operating conditions. Moreover, numerical results suggest the importance of partial load conditions to achieve high efficiency.

Eriksen et al. [66] point out that the balancing of the circuit within a magnetocaloric device can be critical. The independent control system of each regenerator gives new chances of dealing with this issue.
**Numerical implementation**

Simulations are carried out via the solution of one-dimensional energy equations for the solid AMR (1) and the liquid working fluid (2), which gives the temperature fields of the two.

\[
\begin{align*}
\frac{\partial}{\partial x} \left( k_{eff} A_c \frac{\partial T_s}{\partial x} \right) + \frac{Nu k_f}{d_n} a_s A_c (T_f - T_s) &= \rho_s (1 - \epsilon) \rho_s \left[ c_H \frac{\partial T_s}{\partial t} + T_s \left( \frac{\partial s_h}{\partial H} \right) \frac{\partial H}{\partial t} - T_s \frac{\dot{S}_g}{m_s} \right] \\
A_c \epsilon \rho_f c_f \frac{\partial T_f}{\partial t} &= \frac{\partial}{\partial x} \left( k_{disp} A_c \frac{\partial T_f}{\partial x} \right) - \frac{Nu k_f}{d_h} a_s A_c (T_f - T_s) - \dot{m}_f \left( \frac{\partial h_f}{\partial x} \right) + \frac{\partial p}{\partial x} \frac{\dot{m}_f}{\rho_f}
\end{align*}
\]

(1)

(2)

The solid equation has a term related to the irreversibilities. It refers to the magnetic hysteresis as the area swept by the hysteresis curve [50]. To account for the trapezoidal regenerators and have a consistent problem, the non-uniform shape of the regenerators must be included in the model [49]. H, the internal field in the regenerators include, not only the applied external field, but also the demagnetizing field. To take it into consideration, but not to make the problem too complex, the Aharoni equation for a rectangular prism is used [58].

Because of the high operating frequencies, the considerable heat transfer between solid regenerator and fluid, and the insulation of the regenerators, heat losses should not represent a substantial percentage of the total heat. Therefore, heat losses are neglected, to have a solution system as easy as possible.

Proper correlations are chosen to describe the convective heat transfer, the fluid dispersion, and the pressure drop. The selection of the Macdonald rough correlation [69] is supported by experimental measurements of the regenerators' resistances (figure 1). This gives also the chance of underlining the considerable error, that could arise from the use of other correlations common in the sector literature (the Ergun [67] and the Macdonald smooth [69]), and of the total porosity, instead of the open one. It is, then, necessary to highlight that the presence of epoxy in the matrix could play an important role.

The initial conditions are a linear temperature profile from \(T_c\) to \(T_h\) both in the fluid and in solid. The solid’s walls are adiabatic, hence there is not heat exchanged at \(x=0\) and \(x=L\). About the boundary conditions of the fluid equation, they depend on the direction. For flows in the hot-to-cold direction at \(x=0\) there is an enthalpy flux entering the control volume at \(T_h\); at \(x=L\)
there is no variation of heat exchanged. The opposite happens for flows in the cold-to-hot direction.

From the temperature profiles, the heating and cooling capacity, and the COP are derived. The work done by the system is the sum of magnetic work and pump work but the efficiencies of both the pump and motor are not taken into consideration. It is, of course, an overestimation of the performances of the machine but it allows analysing the system itself, without focusing on particular motors or pumps, that is not of our interest.

**Fig. 1**: Experimental pressure drop versus volume flow rate for each regenerator, average resistance of the regenerators and analytical results using different correlations

Although the bed, its width, characteristics and their influences on 1D variables are contemplated within the solution equations, they are only solved along the flow direction. To choose the right optimum working conditions, the width of the bed should also be considered. A 2D simulation would be required, but this implies high computational cost. Note though that, what affects most results happens in the axial direction, along which the fluid flows. This is why the 1D model gives good results. In order to take into account the width of the regenerator, this could be divided into various partitions, and within each partition, a simulation could be performed. Different portions of the simulated regenerator undergo different magnetizations, and they are exposed to the same magnetic field but shifted in time.
Results

Figure 2 shows some among the studied flow profiles in relation to the average magnetic field profile (left hand side). They are represented with the same utilization. On the right hand side of figure 2 are results obtained from the simulations in the COP, heating capacity graph, for AMR frequencies equal to 0.5 Hz, 1 Hz, 2 Hz.

Names of flow profiles refer to the overall blow fractions (tb). For example 0.95tb means that there is fluid flowing through the regenerator for 0.95 of the period.

Fig. 2: On the left, the magnetic field profile and the analysed flow profiles (with the same utilization). On the right, simulations outcomes in the COP, heating capacity graph for various AMR frequencies.

Low AMR frequency, involves long heat exchanging, and hence high COP; whereas high frequency, involves high mass flow, and hence high heating capacity. Low blow fraction causes both low COP and low heating capacity. In this case, when the fluid flows, the mass flow rate is very high. This involves high losses and low NTU, and consequently low effectiveness and high pressure drop. Too high blow fraction could cause a reduction of the temperature span during de-/magnetization, but still 0.95tb has high heating capacity and COP. This derives from the use of a magnet, which produces a magnetic field with a large maximum plateau and a relative short ramp. And this is of general interest, because such a magnet approximates the ideal magnetic field profile with a high magnetic flux region and a sharp passage to the zero field one.

Focusing on reaching the highest heating capacity with the greatest COP, 0.95tb is the best flow profile.
All the previously studied flow profiles are synchronized with the magnetic field. Taking the best flow profile, 0.95tb, the chance of non-synchronizing is considered. In figure 3, the shifting step is expressed in term of fraction of the period, with a sign to indicate the direction: minus corresponds to an earlier ramp, plus corresponds to a later ramp.

For example, the so called `-0.02 (period)` is the 0.95tb with an earlier ramp, translated 2/100 of the period, compared to the synchronized profile. `-0.02(period)` is, therefore, the degree of the non-synchronization.

![Fig. 3: On the left, the magnetic field profile, the best flow profile, and the interval of shifting, when simulating non-synchronized conditions. On the right, results are expressed as normalized heating capacity as a function of the non-synchronization.](image)

Fig. 3: On the left, the magnetic field profile, the best flow profile, and the interval of shifting, when simulating non-synchronized conditions. On the right, results are expressed as normalized heating capacity as a function of the non-synchronization.

Results show great improvements with non-synchronized flow profiles. At 2 Hz the maximum heating capacity increases of 13% if the ramp is slightly delayed. Similar improvements are reached with shifting between 0.02(period) and 0.06(period). There is a big immediate decrease of performances for early ramps and performances decrease anyway for very late ramps. Note that, the best flow profile (0.95tb and non-synchronized) does not correspond to any standard thermodynamic cycles, and so, this atypical analysis, i.e. considering not only defined thermodynamic cycles, seems to be advantageous.

For a better comprehension of the outcome, it is convenient to compare the temperature profiles of regenerator and fluid with the synchronized flow profile and the non-synchronized one. Figure 4 represents the analysed flow profiles and the spatial averaged temperature profiles over the AMR period with the mass flow rate at which the highest heating capacity is achieved.

Note that, in the non-synchronized case, there is hot fluid blowing through the regenerator during magnetization, and cold fluid blowing during demagnetization. This allows the regenerator to reach higher temperatures at high magnetic field, that is when the regenerator
rejects heat. Accordingly, lower temperatures are achieved at low field, when the regenerator absorbs heat. Therefore, during the ramps the fluid contributes to warm up the regenerator (magnetization) and cool it down (demagnetization). And this represents a loss. On the other hand, it might be that this contributes to realize a greater magnetocaloric effect. As the fluid reaches higher temperature span when the field is constant, the heating capacity increases. It might be that Bjørk and Engelbrecht found the synchronization not profitable because of their magnetic field, flow profile, or simulated magnetocaloric materials. The losses for extra heating and cooling the regenerator during (de)-magnetization could have been greater than the increase of heating and cooling capacities.

Simulations are run to study realistic applications of the heat pump. The first test consists in increasing and decreasing the temperature of the hot and cold reservoirs, while keeping constant the temperature span. The left hand side of figure 5 represents the heating capacity at 2 Hz of the simulation with the original parameters and with an increase and decrease of $T_h$ (and consequently $T_c$) of 1°C. With just one degree of displacement, there is a tremendous drop of performances. This demonstrates the absolute inelasticity of the machine, and therefore, the great difficulties of using it in practice. To mitigate the effects of different working temperatures than the designed ones, two reservoirs by each heat exchanger, one upstream and one downstream, may be used. This would increase the inertia of the system, and would allow more stable running conditions of the machine.

![Fig. 4: On the left, synchronized and non-synchronized flow profiles. On the right, spatial averaged temperature profiles of solid and fluid in the two cases.](image)
The second test shows the heating capacity if the machine is used with a cold reservoir at a different temperature that the design one. The $T_h$ is always 310 K, the $T_c$ varies and so the span accordingly. The right hand side of figure 5 represents the heating capacity in the various conditions. Note that, with higher temperature span the heating capacity falls rapidly. The layers with lower Curie temperature are not exploited. With lower temperature span, the heating capacity raises much less markedly. This demonstrates the necessity of choosing a cold reservoir with constant temperature (like a bore hole). Moreover, the design $T_c$ has to be the coldest temperature that can be reached by the reservoir. Indeed, a little decrease would make the heat pump useless.

**Fig. 5:** Effects on the heating capacity with a slight change in the reservoir temperatures, while keeping the same temperature span (left hand side); effects on the heating capacity with a change in the temperature span, while keeping the hot reservoir temperature constant (right hand side)

As shown in figure 2, the lower the AMR frequency, the higher is the COP. For this reason, it is convenient, if possible, to realize partial load conditions. Moreover, since such a magnetocaloric machine has long transient responses, an on/off cycle would be detrimental for the machine’s operation. Figure 6 shows some possible modulating strategies. For the realization of high heating capacity, there is no appreciable difference between modulating just the mass flow rate and both the mass flow rate and the frequency. Instead, the difference becomes significant at low heating capacities. It is never convenient to regulate just the frequency, and, as we said before, on/off cycles, other than inefficient, would be difficult to realize.

For practical applications, it would be, therefore, advisable to select some combinations of mass flow rates and frequencies, corresponding to various heating capacities.
Eventually, the measurement of regenerators’ resistances introduces the issue, whether it is convenient to balance the circuit. This would imply having the same amount of fluid flowing through each bed, and so the realization of the specific best working conditions. On the other hand, the pump work would increase. A quantitative analysis demonstrates that, in the present case, the balancing would cause a higher increase of work than the gain in heating capacity. This outcome is not applicable to general situations, but shows the a straightforward balancing may not be positive.

An independent control system for each regenerator may be useful to have different flow profiles for various regenerators. It is not advisable to use different flow profiles, just to have the same utilization, because the performances would decrease.

Nevertheless, internal unbalances, that are different resistances in the same bed in the hot-to-cold direction and the cold-to-hot one (usually caused by an asymmetrical hydraulic circuit), would lead to steady state conditions different than the designed ones. As shown in figure 5, even small deviations from the design parameters lead to very inefficient working conditions. Asymmetrical flow profiles could be used to overcome the issue.
Conclusions

Different flow profiles, that is when the working fluid blows through the AMR with respect to magnetic field profile, are considered. Usually only some defined thermodynamic cycles are investigated, but, as it is shown, this is not sufficient to define the best flow profile. This is non-synchronized with the magnetic field profile and without any pause fraction. The novel magnetic field with short ramps and large maximum high field regions, could justify the absence of pause fractions in the best flow profile. An extra heating and cooling of regenerators during, respectively, magnetization and demagnetization, seems to benefit the magnetocaloric effect and allows the system to realize a larger fluid temperature span. Experiments on the heat pump are, nevertheless, needed to confirm the numerical results.

Working conditions with different input parameters than the designed one, that is dissimilar hot and cold reservoir temperatures and temperature span, are examined. The use of first-order materials imposes a strict respect of the designed parameters, and therefore, a difficult use of the heat pump in real applications. Even small deviations cause big drops of performances.

The realization of partial load conditions through a control strategy that delivers high efficiency, is fundamental, especially for magnetocaloric devices. The optimum control strategy is the regulation of both machine frequency and working fluid mass flow rate.

Finally, having beds with different resistances, a study whether it is worth balancing the circuit is performed. This demonstrates that it is useless, or even counterproductive, to balance the circuit. The independent control system could be useful to solve dangerous internal unbalances, caused by an asymmetrical hydraulic system.
Introduction

Magnetocaloric refrigeration is an emerging technology, which aims at competing with traditional vapour compression systems. It is based on the exploitation of the magnetocaloric effect (MCE), that is the reversible adiabatic temperature change that magnetocaloric materials (MCM) undergo at magnetization. Heating or cooling devices can be consequently built. The study of magnetocaloric materials and the design of the variable magnetic field are fundamental, but the engineering of a device requires a wider effort. Single components have to be considered together, and analyzed as part of a system.

This thesis deals with the construction of the novel magnetocaloric heat pump at DTU and its numerical optimization. The best working conditions, in order to achieve the highest possible efficiency, are investigated.

This dissertation is structured in five chapters.

The first chapter is a theoretical background on the magnetocaloric effect, materials and technologies utilized for this application. A few existing prototypes are briefly presented.

Chapter 2 gives an overview of the ENOVHEAT heat pump with the description of its components and the control system.

Chapter 3 is the presentation of the numerical code used for the simulations within this investigation. It is based on an existing code, which has been, then, improved and modified to implement the prototype.

Chapter 4 gives the outcome of the numerical optimization. Various combinations of the magnetic field profile and the flow profiles are analyzed. Results are compared to the literature and experimental measurements on another prototype at DTU. Following this is an examination of possible behaviours of the heat pump in a real context, where working parameters are not exactly the designed ones. Finally, it is stressed the importance of partial load conditions for magnetocaloric devices and the way they are achieved.

Eventually, measurements of the regenerators resistances are given in chapter 5. This brings some useful considerations for the numerical implementation of the pressure drop term, and opens a discussion whether it is convenient to balance the machine’s hydraulic circuit.
1. **Background**

The electrical consumption for heating and cooling purposes represents one of the greatest items in the energy of the world. New efficient solutions could represent a big step forward in the reduction of the energy requirements. Among all the possible alternatives to traditional vapour compression systems, magnetocaloric refrigeration is one of the most promising. Moreover, today’s cooling/heating systems use environmentally detrimental gases. Several countries still use refrigerant gases with high Ozone Depletion Potential (ODP), and even most of the gases used in the West have high Global Warming Potential (GDP).

As magnetocaloric refrigeration does not utilize any kind of gases and shows theoretically high efficiency, it could be a competitive and environmental friendly solution.

1.1. **The magnetocaloric effect**

The magnetocaloric effect can be described as a reversible temperature change of a magnetic material as it undergoes an adiabatic magnetization or demagnetization.

It was first observed by Weiss and Picard in 1918 and already in the 30s was used for refrigeration at very low temperatures [1]. It was used in this range of temperatures, where the specific heat of the cooling material is very small, and, therefore, even a modest magnetocaloric effect can produce a significant temperature drop of the substance. Applying the same technology at higher temperatures with the knowledge of the time would have brought to unsatisfying results.

In 1935 Urban, Weiss, and Trombe discovered the magnetocaloric effect of Gadolinium, but the discovery did not arouse much interest [2].

Only in 1976, Brown proposed to use Gadolinium for near-room-temperature magnetic refrigeration [3].

Barclay and Steyert, then in 1982, gave birth to an active magnetic regenerator (AMR). It was a fundamental step forward, which led to new perspectives in the development of this technology [4].
Since that time, the research in this field has been spreading more and more, leading to the discovery of promising materials and to the construction of various prototypes [5].

A clear image of the magnetocaloric effect can be given considering the variations of entropy during such a transformation. Considering a magnetic material, the total entropy can be divided into three components: the entropy related to the magnetic moments, $s_m$, to the lattice, $s_{lat}$, and to the conduction electrons, $s_{el}$. For the sake of this basic explanation, it could be assumed that the magnetic entropy depends on temperature and magnetic field, and the other two components only on the temperature. That is:

$$s(T, H) = s_m(T, H) + s_{lat}(T) + s_{el}(T)\quad (1.1)$$

During magnetization, the magnetic entropy decreases and this can be physically seen as an alignment of the spins. If the process is adiabatic, the total entropy remains constant. This involves that the lattice and electronic entropies increase, i.e. the temperature of the material raises.

Likewise, if the material is adiabatically demagnetized, the magnetic entropy increases, and consequentially the lattice and electronic entropies lower, i.e. the temperature of the material decreases.

Up till now the magnetocaloric effect has been described as an adiabatic temperature change, but it can be equivalently seen as an isothermal entropy change. If the magnetization process is isothermal, instead of adiabatic, the material experiences an entropy change, which corresponds to a heat transfer to the environment.

The above-named quantities: the isothermal entropy change, $\Delta s_{iso}$, and the adiabatic temperature change, $\Delta T_{ad}$, are both fundamental for the characterization of the magnetocaloric effect.

Figure 1.1 shows these quantities in the T,S diagram. Isofield lines are plotted for the total entropy and its components.
Chapter 1. Background

Fig. 1.1: $\Delta s_{iso}$ and $\Delta T_{ad}$ are shown in the T,S diagram. Isofield lines are plotted for the total entropy and its components. (From Pecharsky and Gschneidner, [6])

Various thermodynamic cycles can be realized within cooling/heating machines, thanks to materials, which exhibit considerable magnetocaloric effects (MCE), called magnetocaloric materials (MCM). This theme will be further analyzed in section 1.8.

Note that $\Delta s_{iso}$ and $\Delta T_{ad}$ are independent of each other. In order to evaluate the magnetocaloric effect of one material, they have to be both examined. If a material has high $\Delta T_{ad}$ but low $\Delta s_{iso}$, it could transfer only a small quantity of heat. On the other hand, if a material has high $\Delta s_{iso}$ but low $\Delta T_{ad}$, it would have high heat to exchange but the low temperature difference makes the heat transfer difficult. The ideal magnetocaloric material should have both high $\Delta s_{iso}$ and $\Delta T_{ad}$ [7].

1.2. Characterization of magnetocaloric materials

All magnetic materials show a magnetocaloric effect, even though it is usually rather small. Notwithstanding, a significant magnetocaloric effect is usually experienced by ferromagnets during their magnetic transition. The most remarkable one is the transition from ferromagnet to paramagnet, which occurs at the Curie temperature of the substance. At this temperature the lattice thermal vibration overcomes the magnetic alignment of the spins and the material loses his spontaneous magnetization.
Magnetocaloric materials may be classified based on the kind of their transition. In first order materials the magnetization changes discontinuously, whereas in second order materials there is a continuous change of magnetization (figure 1.2). This influences the properties of the substance at transition; in particular, the different behaviour of the isofield specific heat, $c_H$, is emblematic.

Figure 1.3 shows the features of generic first and second order materials. For first order substances, the specific heat has a high peak, which remains like that also for high field curves. The presence of the high field stabilizes the ferromagnetic phase and the transition is shifted at higher temperatures. This explains also the behavior of the isothermal entropy change: the height is not very sensitive to the magnetic field but the peak gets wider. Note that the shifting of the $c_H$ peak with increasing magnetic fields can be seen as the magnetocaloric effect itself for first-order materials.

For second-order materials, though the specific heat has a sharp peak at zero field, it gets flatter for high fields. The $\Delta s_{iso}$ and $\Delta T_{ad}$ curves increase in height and width for high fields, but the position of the peak does not change.

$\Delta s_{iso}$ and $\Delta T_{ad}$ for first order materials are generally higher but they are experienced only in proximity of the Curie temperature. A small deviation from it implies a great reduction of the resulting effect. This will also be shown in section 4.7 for the device analyzed in the present project.

First order transitions often imply structure transformations and volume discontinuities; and this may lead to mechanical stresses [9].

\[ \text{Fig. 1.2: Magnetization in first order materials (left) and second order materials (right). (From Eriksen,[8])} \]
Another interesting characteristic of the magnetocaloric effect is that its dependency on the magnetic field is to a power lower than one, around 2/3 for second order materials [10] (figure 1.4).

This implies that, in order to maximize the magnetocaloric effect, it is more important to ensure to start at a field as close as possible to 0, rather than slightly increasing the magnetic field [9].
First-order materials may show hysteretical effects in both temperature and magnetization. Hysteresis implies that, the transition from zero field to high field and the transition from high field to zero field do not occur at the same temperature. If such a magnetocaloric material is used in a device, only the reversible effect is taken into account, and thus, the $\Delta T_{ad}$ may get much smaller [12]. Figure 1.5 shows the magnetization and entropy curves around the Curie temperature for a first-order material without and with hysteresis.
Fig. 1.5: Magnetization and entropy curves function of temperature for a first-order material without hysteresis (top) and with hysteresis (bottom). *(From Gutfleisch et al., [12])*

So far, we have been talking about $\Delta s_{iso}$ and $\Delta T_{ad}$ of magnetocaloric materials. Moreover, it is desirable to avoid hysteresis. Nevertheless, other properties have to be taken into account.

First of all, the Curie temperature of the substance has to be close to the working point. About the thermal conductivity, from the thermal point of view, the optimal MCM should be highly anisotropic. On one hand, it should have thermal conductivity as high as possible, so that a proper heat exchange between the MCM and the working fluid is ensured. On the other hand, magnetic devices are usually built up in regenerators (section 1.7) to obtain a high temperature gradient. High conductivity would imply high axial conduction, and consequently a decrease in the regenerator performances. The ideal magnetocaloric regenerator would have high “normal/tangential” conductivity and low axial conductivity [13]. The MCM should have, then, high surface area, so that the heat transfer with the working fluid is maximized. At the same time, the pressure drop, that the fluid undergoes flowing through the MCM, should be as low as possible.
A good MCM should also have good mechanical properties and, therefore, be easily manufactured. It should be stable and have good resistance to corrosion. It does not have to be toxic, and of course, its environmental footprint has to be as low as possible.

After this general characterization, an overview on the most common magnetocaloric materials is given.

### 1.3. Materials

The first and probably most investigated material is the Gadolinium. It has a second-order transition with a \( \Delta s_{iso} \approx 3 \frac{J}{kgK} \) and \( \Delta T_{ad} \approx 2.5 \) K for a change in magnetic field of \( \Delta(\mu_0 H) = 1 \) T.

Many Gd based alloys, containing other rare earths have been studied. They show excellent qualities but the prohibitive cost of such materials makes them unusable.

\( Gd_5Ge_2Si_2 \) shows, what was called at its discovery, a “giant magnetocaloric effect”. Its MC effect is, indeed, greater than Gadolinium’s one, but it is worth using it if only high magnetic fields are applied. It is a first-order material and shows significant hysteresis, but it can be reduced with some additive metals.

A class of promising materials is \( La(FeSi)_{13} \) compounds. They have good magnetocaloric properties and relative low cost. They undergo a first-order phase transition. Substituting metals are needed to regulate the Curie temperature and control the hysteresis. One possibility is the insertion of Cobalt: it shifts the \( T_C \), but it decrease \( \Delta s_{iso} \) and \( \Delta T_{ad} \). Another way is to add hydrogen. The hydrogen atoms, occupying interstitial positions, work as expanders among the iron atoms and stabilize the ferromagnetic state. To properly regulate the \( T_C \), a partial hydrogenation would be required but it is not easy to perform, and the resulting compound would not be stable. What is currently done is a full hydrogenation with the subsequent addition of Manganese to adjust the \( T_C \). With Mn, the transition is also shifted from first to second order (Figure 1.6). A considerable drawback of the hydrogenation process is that the substance gets brittle. It is therefore no longer possible to manufacture solid structures, but a powder can still be used with adhesive-bonding techniques.

Indeed, \( La(FeMnSi)_{13}H_x \) is the chosen magnetocaloric material for the ENOVHEAT prototype. Its properties are \( \Delta s_{iso} \approx 10 \frac{J}{kgK} \) and \( \Delta T_{ad} \approx 3.5 \) K under \( \Delta(\mu_0 H) = 1 \) T.
Chapter 1. Background

Among other interesting alternatives are the Heusler alloys. They show really high magnetocaloric properties but also large hysteresis. MnAs compounds exhibit also good properties but the presence of the toxic As makes them less attractive. Perovskite manganites have good qualities but not so high magnetocaloric effect. FeRh alloys, though being quite expensive and presenting high thermal hysteresis, are of great interest. [9], [12], [14], [15].

![Fig. 1.6: Δs_{iso} (top) and ΔT_{ad} (bottom) of two La(FeMnSi)_{13} H\textsubscript{z} compounds. The Mn content increases from right to left. Gd properties are also plotted for the sake of comparison. (From Krauts et al.,[16])](image-url)
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1.4. Essential thermodynamics

This is an overview of the basic thermodynamics that is used to describe the magnetocaloric effect. It will be then useful in following chapters for the description of the measurement techniques of the MCE and the presentation of the code, implemented for the analysis of magnetocaloric systems.

As said before, the entropy is only considered as a function of temperature and magnetic field. That is:

\[ ds(T, H) = \left( \frac{\partial s}{\partial T} \right)_H dT + \left( \frac{\partial s}{\partial H} \right)_T dH \]  

(1.2)

For an adiabatic process, the entropy remains constant, i.e.:

\[ \left( \frac{\partial s}{\partial T} \right)_H dT + \left( \frac{\partial s}{\partial H} \right)_T dH = 0 \]  

(1.3)

It is now useful to introduce the isofield specific heat:

\[ c_H(T, H) = \frac{1}{T} \left( \frac{\partial s}{\partial T} \right)_H \]  

(1.4)

So that the isofield variation of entropy with temperature can be rewritten in term of the isofield specific heat. Hence, the differential temperature change for an adiabatic process is:

\[ dT = -\frac{T}{c_H(T, H)} \left( \frac{\partial s}{\partial H} \right)_T dH \]  

(1.5)

The adiabatic temperature change can be derived. Mind that the \( \Delta T_{ad} \) is not only a function of the initial and final field but also on the starting conditions. The integral is path (\( \gamma \)) dependent. It yields:

\[ \Delta T_{ad}(T_i, H_i, H_f) = -\int_{\gamma} \frac{T}{c_H(T, H)} \left( \frac{\partial s}{\partial H} \right)_T dH \]  

(1.6)

The isothermal entropy change can be easily derived from equation (1.4):

\[ \Delta s_{iso}(T, H_i, H_f) = \int_0^T \frac{c_H(T, H_f)}{T} dT - \int_0^T \frac{c_H(T, H_i)}{T} dT \]  

(1.7)

\( \Delta s_{iso} \) and \( \Delta T_{ad} \) can be also expressed as functions of the magnetization.
Starting from the differential expression of the free energy and using the Schwarz theorem, the Maxwell relation is obtained:

$$\left( \frac{\partial S}{\partial H} \right)_T = \mu_0 \left( \frac{\partial m}{\partial T} \right)_H$$  \hspace{1cm} (1.8)

(1.6) and (1.7) as functions of the magnetization, using the Maxwell relation, become:

$$\Delta T_{ad}(T_i, H_i, H_f) = -\int_{c_H(T, H)} T \mu_0 \left( \frac{\partial m}{\partial T} \right)_H dH$$  \hspace{1cm} (1.9)

$$\Delta s_{iso}(T_i, H_i, H_f) = \int_{H_i}^{H_f} \mu_0 \left( \frac{\partial m}{\partial T} \right)_H dH$$  \hspace{1cm} (1.10)

The Maxwell relation is derived for equilibrium conditions, and it assumes that the magnetization is differentiable with temperature. It might, therefore, not be valid if the material has an hysteretical behaviour, that is not in equilibrium. Moreover, for first-order materials the magnetization shows a discontinuity during the phase transition (see figure 1.2). J.R. Sun et al. show that the Maxwell relation may, however, be applicable for first order materials using a step function [17]. Concerning the hysteresis, there are various theoretical approaches for considering it [9], but this is not the proper place to go through them. For the purpose of this thesis, the Maxwell relation can still be regarded as valid (the MCMs utilized in the heat pump have low hysteresis, because of the addition of Mn).

1.5. Measurement of the magnetocaloric effect

There are several ways in which the magnetocaloric effect can be measured [9], [15]. The most straightforward procedure is to measure directly the adiabatic temperature change. After having isolated the sample, the temperature is measured as different magnetic fields are applied. Varying the magnetic field while the sample is kept well insulated is not trivial but there are more than one possibility. A common solution is to move the sample in and out a Halbach cylinder (chapter 1.6). The temperature can be simply measured by means of thermocouples.

Another method for characterizing a MCM is to measure the magnetization of the material as a function of magnetic field and temperature. $\Delta s_{iso}$ is, then, indirectly calculated using equation (1.10). The determination of the magnetization of the sample arises from...
measurements of the induced voltage in pick-up coils when the sample is made vibrating and the magnetic field is kept constant.

The last method that we present is the detection of the specific heat using a calorimeter. A widespread device of this kind is the differential scanning calorimeter (DSC). Here two Peltier elements are used to measure the heat flux through two surfaces. One is kept empty as reference; on the other one the sample is placed. A Peltier cell allows to get accurate measurements. A heat flux through it gives rise to a voltage difference.

The sample and the Peltier elements are in a chamber, where the temperature is varied and accurately measured. Vacuum is kept in the chamber and an infrared shield surrounds it, in order to avoid thermal dispersions and have a measurement as precise as possible.

The Peltier elements in the DSC may also be used actively. Their voltage can be controlled to provide a precise heat flux and keep the sample at constant temperature while the field is varied. In this way, instead of mapping the specific heat, the isothermal entropy change is measured directly. Indeed, the heat required to keep isothermal the sample is equal to the isothermal entropy change times the sample temperature.

Figures 1.7, 1.8 show the DSC at DTU
Fig. 1.7: DSC (differential scanning calorimeter) at DTU. (1) is a magnet, that can be lifted up and down, and can be made rotating; (2) is the chamber with the Peltier elements (figure); (3) is the vacuum pump.

Fig. 1.8: DSC at DTU. Magnification of the chamber (2 in figure 1.7) where the two Peltier cells are. Note the sample on top on one of them.
1.6. Magnetic field

After the magnetocaloric material, the other essential component in a magnetocaloric device is the magnetic field source. It represents the most expensive part of the system. Therefore great attempts have been done to design efficient but as cheap as possible solutions [18]. There are three ways a magnetic field can be generated: through electromagnets, superconducting magnets, and permanent magnets.

Electromagnets are made of a soft iron core surrounded by coils. The circulation of electric current in the coils generates the magnetic field. The soft magnetic material gets also magnetized in turn, and the total magnetic flux density is the sum of these two contributions. With the use of electromagnets, high magnetic fields can be obtained and the device itself is easy and cheap. The main drawback, which prevents these devices from the utilization in magnetocaloric applications, is the operative cost. Besides the continuous use of electricity, it is necessary to cool the coil, because it heats up for Joule effect. The ratio between magnetizing energy and the lost Joule energy is, furthermore, very small.

This issue could be solved with superconducting magnets. Superconductors possess extraordinary properties that are zero electrical resistance and the Meissner effect and, hence, zero magnetic flux density inside the superconductor. Very high fields can be generated but usually very low temperatures are required. Recently, superconductors with relative high critical temperatures have been discovered [19], but still the use of a superconducting magnet would require an additional expensive cryogenic system. Nevertheless, it might make sense to install such a device for big magnetocaloric systems.

Finally, the most common solution is permanent magnets. They are very expensive, and, furthermore, require high designing and manufacturing skills, but they do not require expense of energy during the operation of the machine.

Several characteristics of permanent magnets have to be taken into account [20], [21], [22]. The remanence is the magnetization left behind in a ferromagnetic material after an external field is removed. The coercivity is the ability of a ferromagnet to withstand an external magnetic field without becoming demagnetized. The maximum energy product is a measurement of the energy that can be stored in a magnet. Figuratively, it is the greatest rectangular that is contained in the hysteresis curve. The permeability is the relation of the magnetic field to the magnetic field density.

Except for magnetic properties, also the operating temperature range and the mechanical characteristics are fundamental for the realization of a magnet. The latter includes corrosion
resistance and manufacturability. Eventually, the electric resistivity of the material refers to the possible appearance of eddy currents during operative conditions.

There are many kinds of permanent magnets. For magnetic heating/cooling systems only Nd-Fe-B magnets have been used [18]. They have high energy product, high remanence and coercivity, but are very expensive.

As said before, it is more relevant to assure the coexistence of both regions of high magnetic field and low magnetic field, rather than just having a high magnetic field. In the low field regions, the field has to be as close as possible to 0. Moreover, as the magnet is very expensive, the high field region has to be exploited as much as possible. This has led to deep investigations on possible magnet assemblies.

The most utilized geometry derives from the Halbach array. It is a special arrangement with high field on one side of the array and almost zero field on the other side. Figure 1.9 shows a Halbach cylinder, which creates a high field region in the inside of the cylinder and ideally 0 zero field outside.

In order to compare different magnet assemblies a proper figure of merit is needed. Bjørk [10] proposes the $\Lambda_{\text{cool}}$, defined as follows:

$$\Lambda_{\text{cool}} = \left( \frac{B_{\text{mag}}^2}{V_{\text{mag}}} - \frac{B_{\text{out}}^2}{V_{\text{field}}} \right) \frac{V_{\text{field}}}{V_{\text{mag}}} P_{\text{field}} $$

(1.11)

where $B$ is the average flux density in the high flux density region, $B_{\text{out}}$ is the average flux density in the low flux density region, $V_{\text{field}}$ is the volume of the high field region, $V_{\text{mag}}$ is the volume of the magnet and $P_{\text{field}}$ is the fraction of the AMR cycle when the MCM is in the high flux density region.

$\Lambda_{\text{cool}}$ allows considering both the difference between high and low magnetic regions. The volume of the magnet refers indirectly to the cost of it. The possible utilization of magnetocaloric material depends on the volume of the high field region and the $P_{\text{field}}$. 


Fig. 1.9: Halbach cylinder with the representation of the quantities utilized in the definition of $\Lambda_{cool}$. (From Bjørk et al., [11])

1.7. **AMR**

Even the MCM with the greatest magnetocaloric effect undergoes an adiabatic temperature difference of some degrees, too little to build a cooling/heating device. The widespread solution to this issue is the use of an active magnetic regenerator (AMR)[4]. Here the MCM is also a thermal regenerator, hence the name. In this way a much higher temperature gradients can be built.

The AMR is usually made of different layers with various Curie temperatures. During magnetization and demagnetization all the temperature gradient moves upwards and downwards. The working fluid flows first from the cold-to-hot direction and then in the hot-to-cold direction and its temperature follows the gradient across the regenerator.
Regenerators have different geometries. The most common ones are the packed sphere bed, wires-like, perforated-plates and parallel-plate AMRs.

The aim of the regenerator’s design is to maximize the heat transfer between the solid matrix and the fluid, while having the lowest pressure drop as possible.

The packed bed, for example, has high heat transfer coefficient but high viscous dissipation as well. The parallel-plate has low pressure drop but also lower heat transfer coefficient.

Figure 1.11 shows a comparison among various geometries.
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Fig. 1.11: Different regenerator geometries: (a) packed sphere bed; (b) parallel-plate matrix; (c) circular micro-channel matrix; (d) rectangular micro-channel matrix; (e) packed screen bed. The dark regions are the solid matrix, the white regions flow channels. *(From Lei et al., [23])*

From the study of Lei et al. [23], it emerges that the parallel-plate, the micro-channel matrix and the packed screen bed have the highest performances. Figure 1.12 shows the trend of the friction factor, $f_F$, the Nusselt number, $Nu$, and the ratio between the two, $Nu/f_F$, as functions of the Reynolds number. The pressure drop is proportional to the friction factor; the heat transfer is function of the Nusselt number. The parallel-plate, the micro-channel matrix and the packed screen bed show the highest $Nu/f_F$ ratios.

The packed sphere bed is, however, the most common configuration, because of the technological difficulties in fabricating the others.

Indeed, the production process usually limits the development of high-efficiency regenerators.
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Fig. 1.12: Friction factor, \( f_f \) (a), Nusselt number, \( \text{Nu} \) (b), and the ratio between the two, \( \text{Nu} / f_f \) (c), as functions of the Reynold number, \( Re_h \), for different geometries. Data are relative to porosity, \( \epsilon = 0.36 \), and hydraulic diameter, \( D_h = 0.20 \text{ mm} \). In brackets the reference of the data. (From Lei et al., [23])

Among the characteristic parameters for the description of the AMR is the number of transfer unit, NTU. It is the ratio between the convective heat transfer and the fluid thermal capacity. I.e.:

\[
NTU = \frac{hA}{m_f c_f}
\]  

(1.12)

where \( h \) is the heat transfer coefficient, \( A \) is the total exchange surface, \( m_f \) the fluid mass flow rate, \( c_f \) the fluid specific heat.

High NTU involves that high heat is transferred.
Another fundamental figure is the utilization:

\[ u = \frac{m_f c_f}{2f m_s c_s} \]  \hspace{1cm} (1.13)

where \( f \) is the AMR frequency, \( m_s \) the solid refrigerant mass, \( c_s \) the solid specific heat.

The utilization is the ratio between the fluid thermal capacity and the solid thermal capacity. It is used to correctly compare different working conditions. Small \( u \) indicates that less mass of fluid blows through the regenerator.

The effectiveness, \( \eta \), of a regenerator is the ratio of the amount of heat that is transferred during a blow process to the maximum possible heat transfer. The effectiveness is a function of the utilization and the NTU, i.e. \( \eta = f(u, NTU) \). Generally, as the NTU increases the regenerator effectiveness gets higher.

### 1.8. Thermodynamic cycles

It is possible to exploit the magnetocaloric effect to create a cooling/heating device. When the MCM is magnetized, its temperature increases. When the MCM is demagnetized, its temperature decreases. Between the two transformations, a working fluid exchanges heat with the solid refrigerant. The whole cycle can be realized in different ways. The most common ones are the Brayton and the Ericson cycles (figure 1.13).

The Brayton cycle consists in two adiabatic and two isofield transformations. The MCM is magnetized adiabatically (1-2), the working fluid blows through the regenerator in the cold-to-hot direction, while the field remains constant (2-3). The MCM is then demagnetized adiabatically (3-4). Finally, the working fluid blows through the regenerator while the field is still equal to 0, but this time in the hot-to-cold direction (4-1). In the Ericsson cycle, the magnetization (1-2) and demagnetization (3-4) processes are isothermal, instead of adiabatic.

There is, therefore, fluid blowing through the regenerator even during these two transformations. In real applications, the Ericsson is difficult to realize. The heat flux during magnetization and demagnetization is not constant along the regenerator. To achieve an isothermal transformation, the fluid flow should vary in time and space along the AMR. Therefore, practically only quasi-isothermal temperature changes could be achieved.
Fig. 1.13: T,S diagrams for the Brayton cycle (on the left) and the Ericsson (on the right). *(From Gomez et al., [24])*  

The reference cycle is, as usual the Carnot (figure 1.14). It is made of two adiabatic (1-2, 3-4) and two isothermal (2-3, 4-1) transformations. There is, indeed, no region at constant magnetic field. The heat exchange is isothermal and, therefore, inefficiencies are minimized. The Carnot cycle is not used by practical devices though. The temperature interval between hot and cold reservoirs is restricted by the adiabatic temperature change. Moreover, the Carnot cycle requires a varying magnetic field that is not practicable for standard cooling/heating devices. Basically, it could be objected that, in an AMR single particles could performance Carnot cycles, but the whole AMR would, however, perform an hybrid cycle (combination of adiabatic and isothermal processes). The Carnot cycle is still used for refrigeration at very low temperatures where a magnetocaloric single step can be well performed, and it is worth using superconducting magnets to create variable fields. Most of the existing devices operate under the Brayton cycle [5].
1.9. Existing prototypes

Several prototypes have been built in the last decades. Among all of them, two distinctions can be done: linear/reciprocating or rotating prototypes, and moving magnet or moving regenerators.

Usually performances are expressed in terms of zero-temperature-span cooling/heating load, that is the maximum one, and no-load maximum temperature span.

A brief review of some distinguishable devices follows.

The first magnetocaloric refrigerator operating near room temperature was built by NASA in 1976. It was a reciprocating device, following the Stirling cycle, provided with an electromagnet. It was able to achieve a no-load temperature span of 47 K [3].


These devices showed good performances, but the power needed to run a cryocooler jeopardized future developments of superconducting magnets for magnetocaloric devices.

With the new century, new devices using permanent magnets appeared.
Zimm and the Astronautics presented a small device with six rotating regenerators, a stationary permanent magnet and rotary valves [27] (figure 1.15). They tried the machine with different Gd-based alloys and LaFeSiH, obtaining results such as a no-load temperature span of 25 K and a cooling load with no temperature span of 44 W.

In 2002 Rowe and Barclay, from the University of Victoria, published the first results of their reciprocating prototype [28]. In 2006 new results on the same machine, such as a no-load temperature span of 44 K with a 1.5 T magnetic flux, showed the promising potentialities of magnetic refrigeration [29].

In 2006 the Tokyo Institute of Technology presented a device with stationary Gd-based regenerators and rotary permanent magnet [30]. It was capable of producing a cooling power of 540 W with no temperature span.

A few prototypes, both reciprocating and rotating, have been developed at DTU [31],[32]. It makes sense to describe Eriksen’s device [33], which showed good performances, and because it will be considered again within this dissertation.

Fig. 15: Photograph and scheme of Astronautics 2001’s prototype. (From Zimm et al., [27])
The device’s AMR is a hollow cylinder made of 11 beds of Gd-Y spherical particles. Each bed is vertically oriented, and has four layers of MCMs with different Tc. A Halbach cylinder rotates around the AMR and an inner iron core. The two-pole magnet is N50 and it generates a magnetic field up to 1,37 T. Each bed is connected with the cold and the hot reservoirs. At the bottom of the AMR there are the outlet and inlet cold manifolds, connected in turn with the cold heat exchanger. An electrical heater generates the thermal load. On top of the AMR, two tubes for each bed connect the bed with the inlet and outlet hot manifolds. These are connected to the hot exchanger, where the fluid is cooled down by means of a chiller. Poppet valves and check valves are used to impose the reciprocating movement of the fluid, from the hot to the cold side and from the cold to the hot side. A properly designed cam ring, attached to the rotating cylinder, make open and close the poppet valves (figure 1.18). Check valves ensure that the fluid moves only in the appropriate direction. Adjustment valves are, eventually, needed to balance the circuit.

**Fig. 1.16:** Photo of Eriksen’s device. The main body contains the AMR and the Halbach cylinder. In the upper part the tubes connect the regenerators to the manifolds and the pump.
Fig. 1.17: Cutaway section and scheme of Eriksen’s prototype. The highlighted components are: (1) rotating cam rings; (2) rotating magnet; (3) regenerator bed; (4) hot inlet manifold; (5) hot outlet manifold; (6) adjustment valve; (7) poppet valve; (8) check valve; (9) cold inlet manifold; (10) cold outlet manifold; (11) heat exchanger with hot reservoir; (12) pump; (13) heat exchanger with cold reservoir. (From Eriksen,[8])

Fig. 1.18: Magnification of Eriksen’s prototype. (1) is a poppet valve; (2) the cam ring
2. The ENOVHEAT heat pump

The work of this thesis is part of a wider project, ENOVHEAT. The aim of the project is a broad theoretical and technological study for the realization of a magnetocaloric heat pump. It started in 2012, and it involves several institutes and people around the world.

The first years of the project were dedicated to initial investigations on magnet assemblies [21] and magnetocaloric materials [34]. The research has moved to the optimization of the components of a magnetocaloric heat pump, such as regenerators and flow system. Practical aspects like the integration of the machine in a building and the control strategies have been included in the optimization process. All these studies aim to find a practical concretization with the realization of a prototype, which will be described in the next paragraphs.

The heat pump has thirteen regenerators filled with first order magnetocaloric materials that undergo a continuous magnetizing and demagnetizing cycle. This is possible thanks to a magnet, that rotates over the regenerators. The magnet is surrounded by iron and the regenerators are mounted on an iron structure. The latter leads the lines of the magnetic field across the MCM. The regenerators are apart from each other. Even though the distance between them increases the thermal losses and a portion of the high field region is not used, the teeth of the iron structure, upon which the regenerators are mounted, direct the magnetic field lines. This creates a higher magnetic field across the regenerators.

In order to exchange heat with the MCMs, a proper flow system allows having the cold and hot blow through the regenerators at the right time, according to the periodic variation of the magnetic field. The flow across the regenerators is reciprocating. Several measurements of temperature, pressure and torque are performed, both to control and characterize the machine.
2.1. Magnetocaloric materials and regenerators

The regenerators are made of ten layers of first order magnetocaloric materials with different Curie temperatures. In particular, $\text{La(FeMnSi)}_{13} \text{H}_z$ are used [34]. Using such an alloy allows obtaining similar materials with different Curie temperatures, slightly changing the amount of the chemical components. See chapter 1.3 for a description of the material.

There are thirteen regenerators in the machine and this choice is not random. The number of regenerators has to be odd to avoid the presence of magnetically balanced angular positions. An even number of regenerators would imply a certain amount of stable equilibrium conditions during the machine cycle, due to the symmetry of the system regenerators-magnetic field. This would result in a significant cogging torque and, hence, an unbalanced work input.

Among the possible regenerator geometries, the packed sphere bed is chosen. The goal of this choice is to maximize the surface area of the magnetocaloric material, and hence the heat
transfer efficiency between the MCM and the fluid. For this purpose, spheres should be as small as possible. In spite of this, the entropy change of LaFeSi compounds strongly decrease as the particle size is reduced below $160 \mu m$ [35]. According to Lyubina [36], this could be explained by the removal of constraints (due to the magnetovolume effect) imposed by grain boundaries in the porous matrix (on the other hand the hysteresis gets narrower). Alternatively, another explanation is given by Skokov [37]: it could be that the $T_c$ of individual fragments scatters around the $T_c$ of the bulk material. This could lead to a broadening and lowering of the MCE. A middle values is therefore chosen. The particles of MCM in the regenerators have a diameter between 400 and 550 $\mu m$.

The drawback of using such a regenerator is the high pressure drop that the fluid experiences (see chapter 1.7).

The spheres are bound together by epoxy. The presence of an external substance lowers, of course, the magnetocaloric effect of the regenerator, but this addition is inevitable, since the hydrogenation causes the decrepitation of the material [38]. The epoxy allows having a solid and compact regenerator, that can stand mechanical stresses, caused by the MCM phase transition and the fluid blowing.

Practically La(FeMnSi)$_{13}$H$_x$ and epoxy powders are mixed together. The composite undergoes a compaction process. During this, the pressure does not have to be too high, otherwise a further decrease of particle size will occurs [38]. The compound is then brought to high temperatures to let the epoxy melt. An homogeneous porous matrix is so realized.

Figure 2.2 shows a real picture of a regenerator and figure 2.3 shows the different layers within the regenerators. The lengths of the layers are not equal. They are designed according to the magnetic field distribution throughout the bed, so that each bed possesses the same magnetic energy.
Chapter 2. The ENOVHEAT heat pump

Fig. 2.2: Picture of a regenerator in the housing

Fig. 2.3: Cutaway view of a regenerator with the representation of the ten different layers

The housing of the regenerator is 3D printed. It is designed to resist to mechanical stresses and to reduce as much as possible thermal losses to the environment. Nielsen et al. [39] show how sensible performances are to the characterization of the housing. There is a strong dependence on the fluid characteristics, the diffusivity and thermal mass of the wall, and the
thermal utilization. Their model predicts a reduction in regenerator effectiveness up to 18% compared to the ideal case of a regenerator with no walls.

In the ENOVHEAT regenerators an embossment is applied to the bottom surface to minimize the contact with the iron ring and decrease conductive losses. To reduce the convective loss between the housing and the ambient a rubber layer is added between the regenerator and the lid on the top of the MCM.

Note that, the regenerator and, so, the housing have not parallel walls but tapered. This is done to increase the specific heating/cooling power (per unit volume) of the device and to follow the architecture of the machine: since the magnet is the very expensive part of the machine, the magnetized volume has to be exploited as much as possible. Moreover, the tapering gives another advantage. At the cold side the fluid has higher viscosity, hence the pressure drop is generally higher for cold fluid. If the cold side of the regenerator is placed at the wider end, the velocity there decreases and the overall pressure drop is lower [40].

The housing has proper void volumes between the entrances and the regenerator. This is done to give the fluid enough space to distribute itself in all the section of the bed.

### 2.2. Magnet

The magnet is designed via an optimization method based on the desired magnetic field [41]. Assuming a linear relation between the magnetic flux density, $B$, the magnetic field, $H$, and the remanent magnetization or remanence, $B_r$, and, thus, the validity of the relation $B = \mu H + B_r$, the reciprocity theorem can be used. Considering two magnetic systems, 1 and 2, the theorem is:

$$\int B_{r,1} \cdot H_2 \, dV = \int B_{r,2} \cdot H_1 \, dV$$ (2.1)

where $V$ is the volume of the system.

The equation expresses the equivalence between the magnetic energy, in terms of $B_{r,1}$, in system 1, under the influence of the magnetic field generated by system 2, $H_2$, and the magnetic energy in system 2, $B_{r,2}$, under the influence of the magnetic field generated by system 1, $H_1$. Moreover, if $B_{r,1}$ and $H_2$ are parallel everywhere, and likewise $B_{r,2}$ and $H_1$ do, the scalar products in the equation can be simply handled.
The theorem can be useful in the realization of a magnet and therefore the determination of its remanence, $B_{r,1}$, for a desired magnetic field, $H_1$. A virtual system, 2, is considered. If the remanence of the virtual system, $B_{r,2}$ is aligned to $H_1$ in every point, the remanence of system 1 has to be aligned to the virtual magnetic field, $H_2$.

After determining the required remanence for the realization of the desired magnetic field, the optimal magnet shape is chosen. Magnets with complex remanence profiles are usually made of different smaller magnets, called segments. The procedure that allows identifying the best segmentation is described in [42].

The optimal magnet structure is then simplified, taking into account the production system. In the present case, the chosen practical segmentation does not affect magnet’s performances significantly. Figure 2.4 shows the resulting magnet.

Two identical halves compose the magnet. Each half is surrounded by iron on all sides, except for the bottom surface. In total the magnet is composed by 56 segments.

The magnet segments are all made of Sintered Neodymium. The majority of them are N50, the others are N50M. Segments of the second type have higher coercivity, that is the ability of the material to withstand demagnetizing fields. N50M segments are used where high demagnetization is experienced [21].

Figure 2.5 represents the applied magnetic field profile to each regenerator during one machine period. Note that it is not constant along the length of the bed. Since the magnet has two regions of high field, one period of the machine corresponds to two AMR cycles (magnetization of the MCM, fluid flow, demagnetization of the MCM, fluid flow). Further in the text, the magnetic field profile will be represented as its spatial average, and only one AMR cycle will be considered (like figure 2.6). $x$ indicates the bed length fraction, $\tau$ the machine period fraction and $\phi$ the AMR period fraction.
Fig. 2.4: Magnet: magnet segments and iron yoke

Fig. 2.5: Magnetic field as a function of the length fraction (x) and the machine period (τ). Note that the machine period corresponds to two AMR periods
2.3. Flow system

In order to exchange heat with the regenerator a mixture of water and anticorrosion additive is used. The additive is required to avoid the oxidation of the iron-based MCM. Two solenoids valves and two check valves for each regenerator are needed to control the flow. When the MCM is magnetized and it warms up, there is cold fluid flowing through the regenerator in the cold-to-hot direction. Vice versa, when the MCM demagnetizes and cools down, there is warm fluid flowing through the regenerator in the hot-to-cold direction. The presence of solenoid valves let control and easily vary the flow profile, i.e. the fraction of cycle when the fluid blows through a regenerator in one direction, in the other one, and when there is no fluid blowing. In the ENOVHEAT machine, the valves can also be regulated independently. Therefore, it could be possible to have different flow profiles for each regenerator. The choice of using solenoid valves is because of the difficulties experienced with the previous prototype developed at DTU, which has mechanical valves. Mechanical valves do not allow fast modifications of the flow circuit, while solenoid valves make the system more flexible. This is of great importance especially concerning the issue of coupling regenerators with different resistances or having different resistances through the same bed but in the two opposite direction.

Fig. 2.6: Spatial average of the magnetic field profile “seen” by each bed during one AMR cycle
The possibility of using solenoid valves has also been considered by Cardoso and Hoffmann [43]–[45]. They underline the potential of using electro valves to improve the efficiency of the magnetic device and to reduce the power consumption. They also stress the noise reduction in such a machine, compared to a device with mechanical valves. Each branch of the circuit is provided with a check valve to ensure that the flow flows always in one direction. The working fluid going to the regenerators or coming from them is collected/distributed in/from manifolds. A centrifugal variable speed pump is used to circulate the fluid. To work in steady state conditions, after the fluid has heated up by passing through the magnetized AMR, the fluid has to be cooled down to its initial temperature. For this purpose a thermal bath together with a heat exchanger are used at the hot side of the machine. A heater is installed at the cold side to simulate the cooling load. Figure 2.7 is a schematic representation of the flow system, considering only one bed for the sake of simplicity.

Fig. 2.7: Schematization of the flow system, considering just one regenerator. Components: 1 = chiller; 2 = heat exchanger; 3 = pump; 4 = outlet solenoid valve; 5 = inlet solenoid valve; 6 = regenerator; 7 = check valves; 8 = heater.
2.4. Control system

The heat pump is provided of several electronic measurement devices for the control and analyses of the working conditions via LabVIEW.

An encoder fastened to the shaft, supplies continuous measurements of the angular position of the magnet. According to this reading, the valves of the regenerators are open or close. The LabVIEW interface gives the possibility to decide at which angle each valve should open and how much it should stay open.

As a certain precision and accuracy is needed, an integrated circuit FPGA is used. Especially at high frequencies, the system has to be as precise and reactive as possible, because even delays of fractions of second can have a significant impact on the performances of the machine. The IC FPGA is an external independent device that can memorize the LabVIEW program and easily execute several operations in parallel.

A torque meter is installed on the shaft. This measurement is used to calculate the shaft work, which could be seen as an easy prediction of the magnetic work. The volume flow rate elaborated by the pump is adjustable and the volume flow rate is measured with a flow meter. Pressure sensors are installed in the manifolds. Volume flow rate and pressure drop measurements allow to calculate the pump work.

The motor has variable speed and this can also be chosen via LabVIEW. Varying the speed of the motor corresponds to varying the AMR frequency.

Several temperature measurements are performed in the manifolds and the heat exchangers. In one regenerator the temperature is also measured in each layer. Thermocouples are mainly preferred because they are cheap, small and have a fast response. RTD are used when higher accuracy is needed.

The control system is designed to guarantee a fixed temperature at the hot side and an adjustable heat load at the cold side. This would like to resemble practical working conditions, where the heat pump has the cold side at fixed temperature and the user may want to vary the heat load at the hot side, to regulate the temperature of the house. According to this logic the heating power of the heater is adjusted as a function of the outlet temperature. The heating power exchanged with the chiller is calculated knowing the mass flow rate, the inlet and the outlet temperatures of the heat exchanger. The temperature and the mass flow rate of the water from the chiller are, then, regulated accordingly.
The LabVIEW routine calculates the Carnot COP with the reservoir temperatures and the COP considering the heating power, the pump work and the shaft work.

Figure 2.8 shows a picture of the ENOHEAT heat pump.

Fig. 2.8: Picture of the ENOVHEAT heat pump
3. The AMR model

The analysis, and hence, the optimization of the machine’s working conditions are carried out via Matlab simulations.

The work is based on a one-dimensional numerical model, developed at University of Wisconsin [46], but substantially modified.

The model aims to determine the temperature field of a regenerator. From that, the results are extended to the overall system made of several regenerators. The code, then, calculates other quantities, like heating and cooling capacity based on the temperature profiles. The model assumes that the fluid and solid temperature profiles are functions only of the flow direction. Coupled one-dimensional partial differential equations in space and time, describing the temperature of the regenerator and the fluid, are solved.

The regenerator ends are assumed to be adiabatic. During the blow period the fluid enters the regenerator with the set up temperature of the hot or cold reservoirs, respectively from the cold side and the hot one of the regenerator.

Many other models, even two or three-dimensional, can be found in the literature [47]. Nevertheless, for the purpose of this analysis, the 1D model seems to be sufficient, exhibiting advantages such as lower computational time, high efficiency and relatively high accuracy. While 2- or 3D models can solve the coupled fluid flow and heat transfer equations directly, the 1D model requires correlations to calculate heat transfer and pressure drop inside the regenerator.

A closer description is presented below.
3.1. 1D AMR model

The first law of thermodynamics is considered locally, therefore differentiating to the time only. Small letters refer to quantities per unit of mass. The heat exchanged is written in terms of entropy for an irreversible process. Concerning the work, no volume work is done by/to the system but the magnetic work has to be included. This yields to:

\[
\rho_s \frac{\partial u_s}{\partial t} = \rho_s T_s \frac{\partial s_s}{\partial t} + \mu_0 \frac{\partial m}{\partial t} - \rho_s T_s \frac{\dot{S}_g}{m_s}
\]  

(3.1)

where \( \rho_s \) is the density of the solid, \( u_s \) the internal energy of the solid, \( t \) the time, \( T_s \) the solid temperature, \( s_s \) the specific entropy of the solid, \( \mu_0 \) the vacuum permeability, \( H \) the magnetic field strength, \( m \) the magnetization of the material, \( \dot{S}_g \) the irreversible entropy production rate, and \( m_s \) the solid mass.

The magnetic work is derived from the energy generated by an electric field acting on a charge current and then modified with the Gauss’s theorem and Faraday’s law [48]. The last term, that takes into account the irreversibilities, contains also the magnetic hysteresis [49]. Note that the entropy generation term is not present in the Engelbrecht’s original version. A deeper description will be given in section 3.3.

Using equation (1.8), (3.1) becomes:

\[
\rho_s \frac{\partial u_s}{\partial t} = \mu_0 H \frac{\partial m}{\partial t} + \rho_s \left[ T_s \left( \frac{\partial s_s}{\partial T_s} \right)_H \frac{\partial T_s}{\partial t} + T_s \left( \frac{\partial s_s}{\partial H} \right)_{T_s} \frac{\partial H}{\partial t} - T_s \frac{\dot{S}_g}{m_s} \right]
\]  

(3.2)

That is, the variation of the internal energy is given by the magnetic work (the first term after the equal sign) and the heat transfer (in brackets).

Referring to a regenerator with a cross sectional area, \( A_c \), the heat transfer with the solid is due to mainly two mechanism, heat conduction and convection:

\[
\delta q = \frac{\partial}{\partial x} \left( k_{eff} A_c \frac{\partial T_s}{\partial x} \right) + \frac{Nu k_f}{d_h} a_s A_c (T_f - T_s)
\]  

(3.3)
where \( x \) is the axial position, \( k_{eff} \) the effective thermal conductivity, \( T_f \) the temperature of the fluid, \( \text{Nu} \) the Nusselt number, \( k_f \) the thermal conductivity of the fluid, \( d_h \) the hydraulic diameter, and \( a_s \) the specific surface area.

The regenerator is a porous medium through which the fluid flows. There is, therefore, a fraction of the total volume of the regenerator, occupied by the liquid, \( \epsilon \), and a fraction occupied by the solid, \( (1 - \epsilon) \).

Considering eq. (3.3) for the porous regenerator and combining it with the heat transfer component of eq. (3.2), it yields to the governing equation for the solid:

\[
\frac{\partial}{\partial x} \left( k_{eff} A_c \frac{\partial T_s}{\partial x} \right) + \frac{\text{Nu} k_f}{d_h} a_s A_c (T_f - T_s) = \rho_s (1 - \epsilon) \rho_s \left[ C_H \frac{\partial T_s}{\partial t} + T_s \left( \frac{\partial s_s}{\partial H} \right) \frac{\partial H}{\partial t} - T_s \frac{\dot{g}}{m_s} \right]
\]  

(3.4)

To calculate the energy equation for the fluid, an element of length \( dx \) is considered. The figure below (figure 3.1) shows all the contributions to the energy balance.

\[ \dot{W}_{p,1} = p_1 A_c u \]
\[ \dot{Q}_{k,1} = -k_{disp} A_c \frac{\partial T_f}{\partial x} \]
\[ \dot{H}_1 = \dot{m}_f h_{f,1} \]

\[ \dot{W}_{p,2} = p_2 A_c u = (p_1 - \frac{\partial p}{\partial x} dx) A_c u \]
\[ \dot{Q}_{k,2} = \dot{Q}_{k,1} - \frac{\partial}{\partial x} (k_{disp} A_c \frac{\partial T_f}{\partial x}) dx \]
\[ \dot{H}_2 = \dot{m}_f (h_{f,1} + \frac{\partial h}{\partial x} dx) \]

\[ \dot{Q}_c = \frac{\text{Nu} k_f}{d_h} a_s A_c (T_f - T_s) \]

\[ dx \]

Fig. 3.1: An element of fluid
\( W_p \) is the pressure work in the unit of time;
\( \dot{Q}_k \) is the heat exchanged between different elements of the fluid by dispersion;
\( \dot{H} \) is the enthalpy flow;
\( \dot{Q}_c \) is the heating power exchanged with the solid by convection.
The terms at the outlet of the element are approximated with the Taylor expansion (see the pressure term)

The corresponding energy equation for the fluid can be found with the energy balance. That is the variation of the energy in the control volume is equal to the outflow of energy minus the inflow. It yields to:

\[
A_c \epsilon \rho_f c_f \frac{\partial T_f}{\partial t} = \frac{\partial}{\partial x} \left( k_{disp} A_c \frac{\partial T_f}{\partial x} \right) - \frac{Nuk_f}{d_h} \alpha_s A_c (T_f - T_s) - \dot{m}_f \left( \frac{\partial h_f}{\partial x} \right) + \frac{\partial p}{\partial x} \dot{m}_f \rho_f
\]

where \( k_{disp} \) is the thermal conductivity due to fluid dispersion, \( \dot{m}_f \) the mass flow rate, \( h_f \) the fluid specific enthalpy, \( \partial p/\partial x \) the pressure drop, \( \rho_f \) the fluid density and \( c_f \) the fluid specific heat. Mind that, the product \( A_c \epsilon u \) in the pressure term can be equivalently written as \( \dot{m}_f / \rho_f \).

In the above equation, the fluid is assumed incompressible, and consequently the continuity equation is eliminated. The terms from the left hand side are: heat storage, heat conduction/dispersion, heat convection, enthalpy flow and viscous dissipation.
The same result can be achieved considering the general energy equation for a fluid:

\[
\rho \frac{DE}{Dt} = -div(p\bar{u}) + div(\bar{t} \cdot \bar{u}) + div(k \nabla T) + S_E
\]

where \( DE/Dt \) is the material derivative of the energy, \( \bar{u} \) the velocity vector, \( \bar{t} \) the stress tensor, \( S_E \) the source term. Equation (3.5) can be obtained from (3.6) applying continuity and regarding convection as the source term.

In the AMR a reciprocating flow is realized. There is fluid blowing from the hot to the cold side, and, vice versa, from the cold to the hot side. We assume that the hot reservoir (\( T_H \)) is at \( x=0 \), and the cold reservoir (\( T_C \)) is at \( x=L \). The mass flow is positive if it goes in the hot-to-cold direction, it is negative if it goes in the cold-to-hot direction.
Chapter 3. The AMR model

The initial conditions, at t=0, are a linear temperature profile from $T_H$ to $T_C$ both in the fluid and in the solid.

The solid’s walls are adiabatic, hence there is not heat exchanged at $x=0$ and $x=L$.

About the boundary conditions of the fluid equation, they depend on the direction. For positive flow at $x=0$ there is an enthalpy flux entering the control volume at $T_H$, and there is no dispersion with the wall; at $x=L$ there is no variation of the enthalpy flux and no dispersion with the wall. The opposite happens for negative flows.

The energy equation (of the solid or of the fluid) at $(i,j)$ can be simply written in such a form:

$$a_{i-1,j}T_{i-1,j} + a_{i,j}T_{i,j} + a_{i+1,j}T_{i+1,j} = b_{i,j}$$

(3.15)

where $a$ is the coefficient of the relative temperature and $b$ the known term.

Analogously, considering both the fluid and solid energy equations, it yields to the solving system. For the temporal instant $j$, the linear system can be written as:

$$[A]\{T\} = \{B\}$$

(3.16)

Where $[A]$ is the matrix of the coefficients of the temperatures, $\{T\}$ is the vector of the temperatures, $\{B\}$ is the vector of the known terms.

The system is solved using the backslash operator of MATLAB. $A$ is square and banded, therefore the banded linear solver is implemented.

The obtained temperature fields are compared with the temperature fields at the previous iteration. If the difference is lower than a specified tolerance, the solution is found.

Eventually, about the number of spatial and temporal steps, the Courant number is defined:

$$CFL = \frac{u_p}{dt}$$

(3.17)

where $dt$ is the temporal step, $dx$ is the spatial step, $u_p$ is the velocity inside the regenerator, the pore velocity (see section 5.2)

In order to have easy convergence, the condition to impose is $CFL<<1$. 

3.2. Correlations for heat transfer and pressure drop

The correlations for the heat transfer and the pressure drop refer to the following quantities. The hydraulic diameter, \( d_h \), for packed spheres is:

\[
d_h = \frac{2}{3 \left( 1 - \varepsilon \right)} \cdot d_p
\]  

(3.7)

where \( d_p \) is the particle diameter [50].

The Reynolds number can be based on the hydraulic diameter, or on the particle diameter:

\[
Re_f = \frac{d_h |\dot{m}|}{A_c \mu_f}
\]  

(3.8)

\[
Re_p = \frac{d_p |\dot{m}|}{A_c \mu_f}
\]  

(3.9)

with \(|\dot{m}|\) the absolute value of the mass flow rate.

The Prandtl number is:

\[
Pr = \frac{c_f \mu_f}{k_f}
\]  

(3.10)

The specific surface area, \( a_s \) is [50]:

\[
a_s = 6 \frac{1 - \varepsilon}{d_p}
\]  

(3.11)

The correlation for the heat transfer, used in the model, is the Wakao and Kaguei [51] for packed sphere beds. It derives from fitting experimental data and it is expressed in the form

\[
Nu = f (Re^n, Pr^m) : \quad Nu = \frac{\alpha d_h}{k_f} = 2 + 1.1 Re_p^{0.6} Pr^{1/3}
\]  

(3.12)

where \( \alpha \) is the convective heat transfer coefficient.

A modified Ergun correlation is used to predict the pressure drop. Since there was the possibility of measuring the pressure drop through the regenerators, this part will be analysed more in detail in chapter 5.

Finally, about the different conductivities, \( k_{eff} \) and \( k_{disp} \), it is not sufficient to consider the intrinsic conductivity of the regenerator and the fluid. The fluid, flowing through the
regenerator, mixes itself and this phenomenon, called dispersion, may be seen as an axial conduction term, \( k_{\text{disp}} \). The effective heat conductivity, \( k_{\text{eff}} \), contained in the solid equation, is related to both the fluid and solid static conductivities through a suitable correlation [39].

### 3.3. Changes of the original code

Lei [52] suggests various modifications and improvements in the code. He develops the energy equation for the housing wall, in order to take into account for thermal losses. He proposes the use of MC limiters for the space discretization. In this way the spatial discretization of one cell, depends on the neighbours. He extends the above-mentioned energy equations for regenerators with non-uniform shape.

Brey et al. [53] give indications about how to consider the irreversible term. The rate of entropy generation, due to the magnetic hysteresis, is related to the area within the hysteresis curve.

\[
T_s \frac{\dot{S}_g}{\dot{m}_s} = \nu M_{\text{irr}} \left| \frac{d\mu_0 H}{dt} \right| \tag{3.13}
\]

where \( \nu \) is the specific volume of the magnetic material, \( M_{\text{irr}} \) the irreversible magnetization per unit volume, i.e.:

\[
M_{\text{irr}} = \frac{|M_{\text{neg}}(\mu_0 H, T) - M_{\text{pos}}(\mu_0 H, T)|}{2} \tag{3.14}
\]

where \( M_{\text{neg}} \) is the volume magnetization when the change in field is negative, \( M_{\text{pos}} \) the volume magnetization when the change in field is positive. Indeed, \( M_{\text{irr}} \) is defined to recover the area swept by the hysteresis curve. The entropy generation term in the present code is modelled in the same way.

Although Lei suggestions, losses with the ambient are neglected. Because of the high operating frequencies, the considerable heat transfer between solid regenerator and fluid, and the insulation of the regenerators, heat losses should not represent a substantial percentage of the total heat. Results do not differ remarkably considering or not the heat losses, and without the energy equations for the housing wall the solution system becomes much easier.
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The energy equations are discretized with the Crank Nicolson temporal scheme. It is unconditionally stable for all values of the time step [54], but the approximate solution can still contain spurious oscillations. This is avoided in the present case, choosing the right values of $dx$ and $dt$ [55], so that:

$$dt < \rho c \frac{dx^2}{k} \tag{3.15}$$

The advantage of the Crank Nicolson scheme over the fully implicit is that it has second order accuracy. To discretize the equations in the spatial domain care must be taken. The discretization scheme has to possess conservativeness, boundedness and transportiveness. The central differencing scheme is conservative and its Taylor series truncation error is second-order. Therefore, it is not very accurate but it is an easy scheme. Problems could arise at high Peclet numbers. The Peclet number, $Pe$, is the ratio of the advective transport rate to the diffusive one. I.e. in the present case:

$$Pe = \frac{\text{advection}}{\text{diffusion}} = \frac{\rho_f u_c f}{k_{\text{disp}} A_c/dx} \tag{3.16}$$

where $dx$ is the length of the spatial discretization element.

According to the Scarbourough criterion [56] if the $Pe$ is higher than 2, the central differencing does not satisfy the boundedness conditions. Moreover, it does not recognize the direction of the fluid and so, it does not possess transportiveness as well. Indeed, for highly advective problems the central differencing produces wiggles, that is the solution oscillates around the exact one, and this could lead to physically impossible solutions. For high $Pe$, therefore, a method like the upwind would be more appropriate.

The Peclet number in the analyzed problem, for the studied velocity range, is always much higher than 2. Simulations have been run with both the central differencing and the upwind. Still, the central differencing scheme produces satisfying results. This is probably because of the strong coupling between solid and fluid. The temperature profiles of the two are very similar and close to each other and therefore the properties of the fluid are strongly related to the position. MC limiters may be used, as suggested by Lei. The simulation would be more accurate for sure, but, since substantial differences are not seen, and the use of MC limiters would increase the computational cost, the central differencing scheme is chosen.

Another difference with the original code is the discretization of the enthalpy term. In the present case, the value in a point is calculated by the values of the neighbours using the
central difference, instead of just the value itself. This provides higher accuracy to its estimation.

Just as an example, the discretization of the enthalpy term with the central differencing scheme and the Crank-Nicolson method is reported beneath.

Considering the point \((i,j)\) (that means at the spatial step \(i\) and at the time instant \(j\)) and its west and east neighbours, respectively \(i-1\) and \(i+1\), the integration over the control volume and time step \(dt\) yields to:

\[
\int_{j-1}^{j} \left( \int_{CV} \frac{\partial h_{f,i,j}}{\partial x} dV \right) dt = \frac{1}{2} \left( \frac{h_{f,i+1,j-1} + h_{f,i+1,j}}{2} - \frac{h_{f,i-1,j-1} + h_{f,i-1,j}}{2} \right) dt
\]

Eventually, it is not precise to consider only the external magnetic field. If a magnetic field is applied to a magnetic material, this becomes magnetized. The magnetization, \(m\), is the description of the state of the magnetic polarization, i.e. the magnetic dipole moment per unit volume. The dipolar force acting between individual magnetic moments produces, in turn, a magnetic field opposed to the magnetization. This is called demagnetizing field inside the material and stray field outside.

The demagnetizing field plays a big role in magnetocaloric systems. It is strongly dependent on the geometry and the orientation of the MCM, the direction and magnitude of the applied magnetic field, and other quantities like the temperature of the material. The importance of considering the demagnetization field is shown in various publications. Bahl and Nielsen [57] investigate the change of MCE as a function of the orientation of the samples, and find that the effect may be significant (the adiabatic temperature change varies of 30% depending on the orientation of their Gadolinium samples). Smith et al. [58] examine the effects on the demagnetization by different Curie temperature and a linear temperature profile in a magnetocaloric regenerator. In a case like this, that is also similar to the present one, the internal field becomes spatially asymmetric and great differences are seen depending on the direction of the field. Christensen et al. [59] consider this issue in stacked rectangular prisms, to approximate an AMR made of different layers. They show that the demagnetizing field has a great influence on the total internal field, and thus, it is important to consider the full geometry of the body.

It is, so, convenient to take into consideration the demagnetization also in the present case.
The internal magnetic field, $H$, is:

$$H = H_{\text{app}} - \mathbb{N} \cdot m$$  \hspace{1cm} (3.18)

where $H_{\text{app}}$ is the external applied field, $m$ is the magnetization, and $\mathbb{N}$ is the demagnetizing tensor. $\mathbb{N}$ is a tensor because the internal field varies spatially inside the sample. However, for ellipsoids, when both the applied field and the magnetization are along a principal axis the demagnetization factor becomes a scalar. The same may be performed for other geometries as well, interpreting the demagnetization factor as an average.

Nielsen presents a 3D model for the calculation of the demagnetizing field and hence the internal field [60]. The magnetization of the sample at a point is dependent on the internal field, which in turn depends on the magnetization. An iterative procedure is, therefore, followed.

In the present case the problem has to be simplified because of the one-dimensional nature of the model. Instead of considering the demagnetization tensor, an average demagnetization factor can be assumed. The Aharoni equation is chosen [61]. It is a geometrical correlation valid for a rectangular prism, which extends over the volume $-a \leq x \leq a, -b \leq y \leq b, -c \leq z \leq c$. Figure 3.2 shows the demagnetizing factor in the $z$-direction, $D_z$.

Mind that this is a simplified assumption mainly for two reasons.

The regenerators in the ENOVHEAT prototype are trapezoidal (figure 2.2) and Aharoni demagnetizing factor is for a rectangular prism. Moreover, Aharoni assumes the magnetization completely parallel to the applied field, but this may not be true, considering the impact of the corners.

Secondly, Aharoni expression does not take into account of the different magnetocaloric materials and the temperature gradient across the regenerator. Indeed, during operations it could happen that there are a few layers in the ferromagnetic state, and others in the paramagnetic one. Using Aharoni equation, this cannot be simulated.

Within this study, because of the attention that will be given to the magnetic field profile and the flow profile, these two are represented with great accuracy in the Matlab code. In the original code, they are described by simple functions. In the current simulations the specific values of flow profile and magnetic field profile are specified for every time step and spatial step.
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Fig. 3.2: Aharoni equation. $D_z$ is the demagnetizing factor in the $z$ direction; $a$, $b$, $c$ are the half dimensions of the prism.

3.4. Simulation of the ENOVHEAT heat pump

The ENOVHEAT prototype has trapezoidal regenerators (see figure 2.2). It is, therefore, necessary to consider the variation of the cross sectional area in the code, as Lei suggested. The model is extended to the heat pump, multiplying the results by the number of beds; for the ENOVHEAT device. It is possible to simulate various working frequencies modifying the time of a cycle. The heating capacity and the cooling capacity are calculated as the energy exchanged with, respectively, the hot and cold reservoir divided by the cycle period and multiplied by the number of beds. That is:

$$
\dot{Q}_c = \sum_j -m_{f,j} \int (h_{f,x=L,j} - h_{f,T_c}) dt / \tau * nbed \tag{3.19}
$$

$$
\dot{Q}_h = \sum_j m_{f,j} \int (h_{f,x=0,j} - h_{f,T_h}) dt / \tau * nbed \tag{3.20}
$$

where $\tau$ is the cycle period and $nbed$ is the number of beds.

The COP is:

$$
COP = \frac{\dot{Q}_h}{W} = \frac{\dot{Q}_h}{\dot{Q}_h - \dot{Q}_c} \tag{3.21}
$$

The work done by the system is the sum of magnetic work and pump work but the efficiencies of both the pump and motor are not taken into consideration.

It is, of course, an overestimation of the performances of the machine but it allows analysing the system itself, without focusing on particular motors or pumps, that is not of our interest.

The calculation of the Carnot COP:
\[ \text{COP}_{\text{Carnot}} = \frac{T_h}{T_h - T_c}, \]  

(3.22)

Let's calculate the second law efficiency as:

\[ \eta_{II} = \frac{\text{COP}}{\text{COP}_{\text{Carnot}}}. \]  

(3.23)

The properties of the magnetocaloric materials are given as entropy and magnetization curves as functions of temperature for different magnetic field. The entropy curves are generated from experimental measurements of the specific heat of the materials.

### 3.5. Summary

The code designed for the modelling of a regenerator and then especially modified for the implementation of the ENOVHEAT heat pump has been presented. It uses the finite volume method to solve the energy equations of the solid and the fluid. Moreover, it should be more accurate than the original one because of the Crank Nicolson temporal discretization scheme. The central differencing scheme is, then, chosen, neglecting the great advective characteristic of the problem. This is done because of the strict coupling between fluid and solid temperatures. Other relevant features of the utilized code are the presence of the magnetic hysteresis and the demagnetizing effect, which are mandatory for a precise description of the phenomenon.

The outputs of the model are the temperature profiles of solid and fluid. From this, the heating and cooling capacities, the COP, and the entropy production rate are derived. The trends of these quantities allow us to compare different working conditions of the heat pump.
The magnet and the active magnetic regenerator (AMR) are the heart of the heat pump. Their design is fundamental to achieve the goal of the ENOVHEAT project. Besides, the single components of the machine have to be considered together as an overall system, with the aim of maximizing the performances of the heat pump.

The useful effect of the heat pump, that is the heating capacity, arises from the heat exchange between the working fluid and the magnetocaloric material (MCM). As the MCM is magnetized, an increase of temperature occurs, the heat transfer fluid is pumped through the porous AMR bed in the cold-to-hot direction, and heat is transferred from the MCM to the working fluid. Vice versa, as the MCM is demagnetized, its temperature decreases, and the heat transfer fluid is pumped through the AMR in the hot-to-cold direction, in order to cool down the fluid. The way the fluid blows through the AMR, the flow profile, and its coupling with the magnetic field deeply influences the heating capacity and the COP trends. Note that, talking about different flow profiles with respect to the magnetic field may correspond to the definition of different thermodynamic cycles.

The issue of properly combining magnetic field and flow profile has already been studied in recent years, but it was usually referred to some particular cases or general thermodynamic cycles. On the contrary, this analysis is a wide investigation that is not stuck in the possibility of realizing just some defined thermodynamic cycles. Moreover, this is also easy to realize in practice with the ENOVHEAT heap pump, which is provided of a proper control system. This brings us to results of general interest.

In the following section, the extension of the results in a novel 1D+1D model extension, to account for the regenerator width, is presented.

Further in this chapter, some deviations to the original design parameters are investigated. This aims at considering the introduction of the heat pump in a real context, where theoretical conditions are always difficult to perform.
Finally, how to realize partial load conditions is a relevant matter in magnetocaloric devices. The reason why will be later explained. Besides, there are more alternatives to realize them.

### 4.1. Flow profiles

Each magnetocaloric regenerator is subjected to a periodic magnetic field. During this period, the flow profile defines when there is fluid blowing in one direction, when in the other one (blow fractions), and when there is no fluid blowing (pause fraction).

Analysing the problem in steady state, the blow fractions in the two directions have to be the same, so that the flow is balanced. This means that the amount of fluid per cycle flowing from the hot to the cold side, and the fluid flowing from the cold to the hot side, must be equal.

In this analysis, first flow profiles synchronized with the magnetic field are considered; later on, also non-synchronized profiles will be studied.

The aim of this work is to find the flow profile that maximizes the heating capacity, with the possible highest COP.

In order to have a general overview of the matter, different working conditions are examined. Low mass flow rate through the regenerator implies low velocity and therefore low heat transfer within the regenerator. High mass flow rate reduces the NTU and makes the regenerator less effective. Practically, high mass flow rate causes the flattening of the temperature gradient across the regenerator, and, for this reason, it has to be avoided.

Moreover, high mass flow rate causes an increase of pressure drop. Thus, there is a maximum heat corresponding to an intermediate range of mass flow rates. Since the heat transfer coefficient is proportional to the velocity raised to a coefficient that is smaller than one (according to Wakao and Kaguei [51]) and the relation of the pressure drop to the velocity is almost quadratic in the range of velocities considered, the COP decreases for increasing mass flow rate.

Different working frequencies are contemplated. Low frequencies correspond to high COPs because the fluid is given enough time to exchange heat with the MCM. High frequencies correspond, on one side, to high heat transfer because the total mass flow is higher but, on the other side, to low COPs. Working with very high frequencies does not bring any advantage though, because it is not given enough time to the heat exchange to happen[8].

Further on frequencies between 0.5 Hz and 2 Hz will be considered.
In the current case, it is not trivial to define an optimization function to find the right flow profile because of the different variables the problem depends on, and the parameters that are aimed. The computational cost would, then, be rather high. It is easier to define some distinguishing cases and simulate the operation of the machine.

In figure 4.1 some among the studied flow profiles are presented, in relation to the average magnetic field profile. They are represented with the same utilization. Beneath are the results (figure 4.2) obtained from the simulations, that are COP and heating capacity, as functions of the volume flow rate, for frequencies equal to 0.5 Hz, 1 Hz, 2 Hz. Names of flow profiles refer to the overall blow fractions (tb). For example, 0.95tb means that there is fluid flowing through the regenerator for 0.95 of the period. All flow profiles have the same slope of the transition between the blow fraction and the pause fraction, which will be further on called ramp. In practice, electro valves with an opening and closing time of fractions of second are used. It is, then, difficult to calculate the influence of the partial open valve in the circuit but it should be rather small, since the pressure drop through the regenerator is much greater. Therefore, zero ramps could be used but, to avoid discontinuities in the flow profile, a value has been chosen.

**Fig. 4.1:** Mean magnetic field profile and different flow profiles with the same utilization. Names refer to the overall blow fraction (tb)
Fig. 4.2: Heating capacity vs volume flow rate, COP vs volume flow rate for frequencies equal to \(0.5\), \(1\), \(2\) Hz (from top to the bottom) and various flow profiles.
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From the results, it is clear that low blow fraction causes both low COP and low heating capacity, because there is not enough time for the heat exchange to occur. 0.95tb, 0.77tb and 0.66tb have similar heating capacity but 0.95tb has higher COP especially at high frequency. Focusing on reaching the highest heating capacity with the greatest efficiency, 0.95tb is the best flow profile.

A common way to represent these results is by means of the COP vs Heating capacity diagram. It allows a fast fair comparison among the flow profiles (figure 4.3). Note that, there are three lines for each flow profile, for the three frequencies: 2 Hz is the one with the highest heating capacity; 0.5 Hz is the one with the highest COP; 1 Hz is the middle one. Each line corresponds to increasing mass flow rate. It reaches almost immediately the maximum COP. The maximum heating capacity is at medium mass flow rates.

![COP vs Heating capacity diagram](image)

**Fig. 4.3:** Performances of the various flow profiles in the COP, Heating capacity diagram. The results at 0.5, 1 and 2 Hz are plotted. Lower heating capacities and higher COPs correspond to low frequencies.

### 4.2. Entropy production rate

Another way of comparing the flow profiles and confirm the results is to evaluate the entropy production rate of the cycle with the chosen different flow profiles [49], [52]. The greater the entropy production is, the greater irreversibilities the transformation has. Therefore, this is a measurement of the efficiency of the physical process; and the research of the best flow profile should correspond to the minimization of the entropy production.
The total entropy production rate is given by the sum of three terms:

\[ \dot{S}_{ht} \], the entropy production due to the heat transfer between the solid and the fluid and the heat transfer between the fluid and the hot and cold reservoirs;

\[ \dot{S}_{vd} \], the entropy production rate due to the viscous dissipation while the fluid flows through the regenerator;

\[ \dot{S}_{ac} \], the entropy production rate due to the axial conduction in the solid and the axial conduction in the fluid (or previously called fluid dispersion).

\[
\dot{S}_{ht} = \frac{1}{\tau} \int_{0}^{\tau} \int_{0}^{L} h_{as} A_{c} \frac{(T_f - T_s)^2}{T_f T_s} \, dx \, dt \\
+ \frac{1}{\tau} \int_{0}^{\tau} \int_{0}^{L} |\dot{m}_f| c_f \left( \ln \frac{T_C}{T_{f,x=L}} + \frac{T_{f,x=L} - T_C}{T_C} + \ln \frac{T_H}{T_{f,x=0}} + \frac{T_{f,x=0} - T_H}{T_H} \right) \, dx \, dt
\]  

\[ (4.1) \]

\[
\dot{S}_{vd} = \frac{1}{\tau} \int_{0}^{\tau} \int_{0}^{L} \left| \dot{m}_f \right| \frac{\partial p}{\rho_f T_f} \, dx \, dt
\]  

\[ (4.2) \]

\[
\dot{S}_{ac} = \frac{1}{\tau} \int_{0}^{\tau} \int_{0}^{L} \left[ k_{stat} A_{c} \frac{1}{T_s^2} \left( \frac{dT_s}{dx} \right)^2 + k_{disp} A_{c} \frac{1}{T_f^2} \left( \frac{dT_f}{dx} \right)^2 \right] \, dx \, dt
\]  

\[ (4.3) \]

\[
\dot{S}_{tot} = \dot{S}_{ht} + \dot{S}_{vd} + \dot{S}_{ac}
\]  

\[ (4.4) \]

where \( \tau \) is the period of one cycle, \( L \) the length of the regenerator, \( x \) the axial coordinate, in the range from 0 to \( L \), \( T_C \) the temperature of the cold reservoir, \( T_H \) the temperature of the hot reservoir.

In figure 4.4, the entropy production rates of the cycles with the same flow profiles as before, are shown.

Also from this investigation 0,95tb appears to be the best flow profile.
4.3. Non-synchronized profiles

All the flow profiles, previously analyzed, are always synchronized with the magnetic field. They are synchronized if the middle point of the maximum of the magnetic field corresponds to the middle point of the blow fraction. In this chapter results of the investigation of non-synchronized flow profiles are presented. Figure 4.5 represents the magnetic field profile, the original 0,95tb flow profile, which was found out to be the best one, and the shifting interval. The 0,95tb profile is shifted several steps from the earliest ramp to the latest ramp.
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Fig. 4.5: Magnetic field profile, the best flow profile, the 0.95tb, and the interval of shifting, when simulating non-synchronized conditions

The results of some non-synchronized flow profiles are shown below (figure 4.6). In the figures, the shifting step is expressed in term of fraction of the period, with a sign to indicate the direction: minus corresponds to an earlier ramp, plus corresponds to a later ramp. For example, the so called “-0.02 (period)” is the 0.95tb with an earlier ramp, translated 2/100 of the period, compared to the synchronized profile. -0.02(period) is, therefore, the degree of the non-synchronization.

The graphs are COP and heating capacity as functions of volume flow rate, relative to frequency 0.5Hz and 2Hz. This is to show that different results are achieved at low or high frequencies but, in any case, the best flow profile is always another one but the synchronized one.

Among these flow profiles and results, the flow profile with the highest heating capacity and relative high COP should be preferred. Considering results at low frequencies and, hence, lower heating capacity is still important though, as it will be shown later on.
Fig. 4.6: Heating capacity vs volume flow rate, COP vs volume flow rate for frequencies equal to 0.5, 2 Hz of various non-synchronized 0.95tb flow profiles

Note that the COP increases for later flow profiles. Even the +0.10(period), which has lower heating capacity, shows the highest COP peak. This is probably due to the drop of magnetic work that follows the shifting.

Figure 4.7 shows heating capacity results in a clearer way. It plots the normalized maximum heating capacity, that is the ratio of the maximum heating capacity to the maximum heating capacity of the synchronized flow profile, 0.95tb, as a function of the shifting of the flow profile.
Results show great improvements with non-synchronized flow profiles. At 2 Hz the maximum heating capacity increases of 13% if the ramp is slightly delayed. Similar improvements are reached with shifting between 0,02(period) and 0,06(period). There is a big immediate decrease of performances for early ramps and performances decrease anyway for very late ramps.

Figure 4.8 shows, in the end, the best flow profile. Note that, it does not correspond to any standard thermodynamic cycles, and so, this atypical analysis, i.e. considering not only defined thermodynamic cycles, seems to be advantageous.
4.4. Comparison of the results

Because of various delays in the ENOVHEAT project, it was not possible to run the prototype and, in this way, to validate experimentally the outcome of the simulations.

What obtained is very interesting and there are two aspects of the results, which were unforeseen, but reasonable, as it will be shown. These are the best flow profile without pause fractions and an improvement of performances for non-synchronized profiles. Waiting for the possibility of running some experiments, it is worth deeply analyzing the results, comparing them with information found in the literature, and performing some qualitative experiments with the other prototype at DTU [33].

Firstly, only the absence of any pause fraction in the best profile will be considered. Benedict et al. [62] show that it is not convenient to transfer heat during magnetization and demagnetization. Premature heat transfer during magnetization or demagnetization does not let the MCM reach low and high temperatures.

Teyber et al. [63] point out that displacing the fluid when the magnetic field is near the maximum and minimum values, increase the average magnetic field change during the blow period. On the contrary, a late blowing limits the amount of heat that can be exchanged, and the higher temperature difference between regenerator and ambient causes higher thermal...
losses. For this reason, performances may increase with a pause if the magnetic field varies greatly, but it does not have to be too large. They underline, then, that if you compare flow profiles with the same utilization, in a flow profile with a great pause, the volume flow rate, during the blow fraction, is higher. This implies higher axial dispersion and viscous dissipation.

Plaznik et al. [64] compare different performances of the Brayton cycle (adiabatic de-/magnetization), the Ericsson (isothermal de-/magnetization), and hybrid Brayton-Ericsson (where the de-/magnetization process is a combination of an adiabatic and an isothermal process). See section 1.8 for a deeper explanation.

Figure 4.9 shows the thermodynamic cycles analyzed by Plaznik in a similar way than how we previously did.

![Diagram of magnetic field profile and flow profile](image)

**Fig. 4.9:** Magnetic field profile and flow profile studied by Plaznik et al. The flow profiles correspond to the Brayton, the Ericsson and the Hybrid cycles. *(From Plaznik et al., [64])*

Their investigation shows that the Ericsson has higher COP than the Brayton cycle, and the latter one has higher cooling capacity than the Ericsson. Anyway the Hybrid cycle shows the
best performances: cooling capacity similar to the Brayton and COP similar to the Ericsson cycle.

In brief, the coupling of flow profile and magnetic field profile is a delicate matter. A proper increase of temperature has to be guaranteed during magnetization and, likewise, a proper decrease of temperature during demagnetization. This ensures high heat transfer. High fluid flow rate implies high losses, and it risks flattening the temperature gradient across the bed. Therefore high fluid flow rate, that is realized for large pause fractions and, hence, small blow fractions, involves lower COP.

Coming to the present case, analogously, the flow profiles with little pause fractions show the highest heating capacities. The flow profile with no pause fraction, the so called “0,95tb”, has the highest COP. The last one has been chosen as the best flow profile, because it also has a heating capacity as high as other flow profiles with small pause fractions. This could derive from the use of a magnet, which produces a magnetic field with a large maximum plateau and a relative short ramp. Indeed, in all the studied papers the magnetic field profile has long ramps and smaller plateaus.

If, experimentally confirmed, this outcome would be of great interest. The magnetic field profile in the ENOVHEAT’s machine aims to approximate the ideal case of a cycle with the maximum field in half of the period, an immediate ramp and zero field in the other half of the period. Future prototypes could therefore have similar magnetic field as the ENOVHEAT one.

The chance of confirming once more the numerical model and these results comes from the experimental tests performed on Eriksen’s machine (see chapter 1.9) [8], [7], another prototype at DTU.

For a fairer comparison, simulations with the magnetic field profile of Eriksen’s machine and two flow profiles, respectively with long and short pause fraction, are performed (figure 4.10). Note that, these simulations are done with the same model implemented for the ENOVHEAT machine. The only thing that changes is the simulated magnetic field. Therefore, their outcomes cannot be straight applied to Eriksen’s machine, but they could still be used for a qualitative comparison.
Fig. 4.10: Eriksen’s prototype magnetic field profile and the two examined flow profiles

Fig. 4.11: COP, Heating capacity for the two flow profiles
Figure 4.11 provides the performances of the two cases (frequency is equal to 1 Hz). It is shown that, with the flow profile 0,50tb, a considerably higher heating capacity may be reached, and therefore 0,50tb would be the preferred flow profile, even if it has lower COP. 0,95tb, on the contrary, shows higher COP.

Similar experiments are realized with Eriksen’s prototype. Here it is difficult to realize different flow profiles, because they are determined by a special cam ring, which make the poppet valves open and close (see chapter 1.9). The machine is run with two cam rings, one corresponding to the short pause fraction and the other one to the long pause fraction. In this prototype, the cooling capacity is fixed and the temperature span is measured. With a cooling capacity of 110 W and a volume flow rate of 3,1 l/min, the highest temperature span is given by the low blow fraction flow profile (12 K against 8,5 K), and the highest COP is shown by high blow fraction flow profiles (6,6 against 4,9). Moreover, the highest second law efficiency is achieved for the flow profile with the highest blow fraction and so shortest pause fraction. This agrees with the outcome of the simulations, and it could be assume that the results for the ENOVHEAT heat pump might be realistic. Experimental measurements on the new prototype will validate them once and for all.

It is, anyway, clear that, the magnetic field has great importance. Eriksen’s machine’s magnetic field has a longer ramp and a shorter maximum. With such a magnetic field, the magnetocaloric material magnetizes more slowly. It is therefore more convenient to wait until a certain magnetocaloric effect is realized, before letting the fluid flowing through the bed. And this can be seen from the model as well.

It is worth repeating that, within these simulations only the magnetic field of Eriksen’s refrigerator is considered, all the other parameters are the ones of the analyzed heat pump. This outcome is, therefore, not applicable to the prototype, but it is only useful for the qualitative validation of the results.

About non-synchronized magnetic field profile and flow profile, it is shown that with a slightly delay of the flow profile, performances increase appreciably.

The non-synchronization between magnetic field and the flow profile has been already considered from the scientific community. According to the simulations in [10] no improvements are expected in this case. Bjørk and Engelbrecht [10] show significant performance drops for early ramps, no changes for slightly late ramps, and performance drops
for very late ramps. Even if in the present case great performance drops are experienced for early ramps (see -2/100 (period)), and performance drops are seen for very late ramps, a performance increase is observed for slightly late ramps.

A deeper analysis of the case is therefore performed. For this aim, it is reasonable to compare the temperature profiles of regenerator and fluid with the synchronized flow profile and the non-synchronized (non-sync) one. For an easier comprehension, figure 4.12 reports the analysed flow profiles. Figure 4.13 represents the spatial averaged temperature profiles over the AMR period with the mass flow rate at which the highest heating capacity is realized.

Note that, in the non-sync case there is hot fluid blowing through the regenerator during the magnetization, and cold fluid blows during the demagnetization. According to the temperature field (figure 4.13), this allows to reach higher temperatures in the regenerator at high magnetic field, when the regenerator rejects heat. Accordingly, lower temperatures are achieved at low field, when the regenerator absorbs heat. So during the ramps the fluid contributes to warm up the regenerator (magnetization) and cool it down (demagnetization). And this represents a loss. On the other hand, it might be that this contributes to realize a greater magnetocaloric effect. As the fluid reaches higher temperature span when the field is constant, the heating capacity increases.

There is, indeed, a balance between positive and negative contributions given by the non-synchronization.

It might be that Bjørk and Engelbrecht found the synchronization not profitable because of their magnetic field, flow profile, or simulated magnetocaloric materials. The losses for extra heating and cooling the regenerator during (de)-magnetization could have been greater than the increase of heating and cooling capacities.
Another way to prove the goodness of these outcomes is to compare the entropy production rate (table 4.1).
Table 4.1: Entropy production rate

<table>
<thead>
<tr>
<th>Entropy production rate</th>
<th>Synchronized profile</th>
<th>Non-synchronized profile</th>
</tr>
</thead>
<tbody>
<tr>
<td>[J/K]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat transfer between fluid and solid</td>
<td>0.5</td>
<td>0.418</td>
</tr>
<tr>
<td>Viscous dissipation</td>
<td>1.61 * 10^{-2}</td>
<td>1.61 * 10^{-2}</td>
</tr>
<tr>
<td>Flow dispersion</td>
<td>8.25 * 10^{-2}</td>
<td>8.42 * 10^{-2}</td>
</tr>
<tr>
<td>Axial conduction</td>
<td>7.0 * 10^{-3}</td>
<td>7.0 * 10^{-3}</td>
</tr>
<tr>
<td>Heat transfer between fluid and hot reservoir</td>
<td>4.3 * 10^{-3}</td>
<td>5.4 * 10^{-3}</td>
</tr>
<tr>
<td>Heat transfer between fluid and cold reservoir</td>
<td>2.9 * 10^{-3}</td>
<td>3.6 * 10^{-3}</td>
</tr>
<tr>
<td>Total EPR</td>
<td>0.612</td>
<td>0.535</td>
</tr>
</tbody>
</table>

The total entropy production is lower in the non-sync case. Although the entropy production rate due to the heat transfer between the reservoirs is slightly higher, the component referring to the heat exchange between fluid and solid is significantly lower. Figure 4.14 shows the spatial averaged temperature difference between solid and fluid in the two cases. The temperature difference for the non-sync profile is indeed lower. This implies lower entropy production rate.

Fig. 4.14: Spatial averaged temperature difference between solid and fluid in the synchronized and non-synchronized case
Results may, therefore, be reasonable. An extra heating and cooling of the regenerators during de-/magnetization by the fluid could increase the performance of the heat pump. Experimental data will confirm or disagree with these outcomes.

4.5. **Width extension 1D+1D approach**

Possible operating conditions have been considered, so far, with a 1D model. Although the bed, its width, characteristics and their influences on 1D variables are contemplated within the solution equations, they are only solved along the flow direction. To choose the right optimum working conditions, the width of the bed should also be considered. A 2D simulation would be required, but this implies high computational cost. Note though that, what affects most the results happens in the axial direction, along which the fluid flows. This is why the 1D model gives good results. In order to take into account the width of the regenerator, this could be divided into various partitions, and within each partition, a simulation could be performed. Different portions of the simulated regenerator undergo different magnetizations, and they are exposed to the same magnetic field but shifted in time.

It is not trivial to define how the bed should be divided and where simulations should be performed. It is advisable to divide the bed into equal partitions, and for each one perform the respective simulation along the middle line (figure 4.15). In this way, every simulation accounts for the same amount of material. This is not a 2D simulation, instead it is a reconstruction of a 2D case using multiple 1D model runs. The solving equations are still solved along a line and do not refer to the surface, nor do they consider directly transversal effects. This approach, however, allows us to consider the bed’s width.
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Fig. 4.15: Example of partitions (3) of the regenerator and middle lines, along which the simulations are performed

It is not easy to determine how many partitions should be made. A sensitivity analysis may be performed. This would lead to coherent results if the analysed flow profiles were all synchronized, but performances can vary in sensitivity, to the shifting of the period. In some cases, one partition is sufficient; in other cases more partitions give different results. This is also suggested by figure 4.7. For that case and considering the heating capacity, the effect of the 2D averaging is less relevant around the maximum plateau.

The results shown below are related to three partitions (figure 4.16).

About names, for example, “2D +0,02(period)” refers to the average of three simulations: one with the +0,02(period) flow profile itself, that is the middle line, and two simulations, which correspond to the middle lines of the right hand partition and of the left hand partition.

At 2Hz, 2D +0,06(period)” still shows one of the highest heating capacities and a high COP. This is why +0,06(period) is chosen as the best flow profile.

This analysis could seem useless because the 0,95tb+0,06(period) had already been identified as the best flow profile. Nevertheless, a closer examination would emphasize the relevance of this approach. For example, the difference of heating capacity of +0,02(period) is significant. Note in figure 4.6 that +0,02(period) has the highest heating capacity. Considering the width of the regenerator, it leads to a considerable drop in its heating capacity curve (figure 4.16). At the peak the difference is about 8% of the value.
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4.6. Comparison with traditional VCS

The second law efficiency may be calculated according to the results with the best flow profile. Working with low volume flow rates and low frequencies would imply high COP and, therefore, high efficiency, but the most interesting working condition is the one with the highest heating capacity.

The highest heating capacity, that is $\dot{Q} = 1100 \text{ W}$, is realized at 2 Hz with a volume flow rate of 2460 l/h. It corresponds to a COP=2.6. At the working temperatures $COP_{\text{Carnot}} = 12.4$. It yields that the second efficiency is $\eta_{II} = \frac{\text{COP}}{COP_{\text{Carnot}}} = 0.21$.

Of course, if the choice of the operating condition would be done in order to maximize the COP, a COP of 5 may be reached but at very low heating capacities.
We may compare the result with the performances of a borehole heat exchanger / conventional heat pump system. In such a complex system, performances vary significantly according to the design parameters. For a general analysis, considering the cold side temperature around 10°C and the hot side temperature at least 35°C, a heating capacity of around 4 kW, a COP = 4 could be assumed [65].

The traditional system is, therefore, much more efficient than the magnetocaloric heat pump. With a much higher heating capacity, the COP is definitely greater. Moreover, note that this data are calculated without taking into account the efficiency of motor and pump in the magnetocaloric heat pump.

This highlights that, though magnetocaloric devices are environmental friendly and theoretically very efficient, still a good deal of research is needed, before these systems might compare with traditional VCSs.

4.7. Deviation from design parameters

First order magnetocaloric materials show high isothermal entropy changes with magnetization and demagnetization around their Curie temperature but the $\Delta s$ curve is very sharp. They are, therefore, on one hand very attractive for their performances but, on the other hand, it is important to operate very close to the Curie temperature of the materials. This is the case of the analyzed heat pump. During real operations, it is not straightforward to realize the right optimum conditions with the design parameters.

Various simulations are, so, performed to study how much performances vary when $T_c$, $T_h$, and $T_{span}$ are changed.

Figure 4.17 represents the magnetocaloric effect, as isothermal entropy change, of ten layers of first-order materials with the same Curie temperatures of the materials in the ENOVHEAT machine’s regenerators. The figure is just illustrative: the represented values do not correspond to the real data, nor to the data used in the code. The machine is designed for working between a cold reservoir, $T_c$, at 285 K and a hot reservoir, $T_h$, at 310 K, with a total temperature span of 25 K.
Fig. 4.17: Illustrative representation of the magnetocaloric effect of the layers. The variation of entropy is considerable only close the relative Curie temperature of the material. The materials and their relative Curie temperatures are chosen to work between $T_c$ and $T_h$.

The first test consists in increasing and decreasing the temperature of the hot reservoir and consequently of the cold reservoir, while keeping constant the temperature span. Figure 4.18 represents the heating capacity at 2 Hz of the simulation with the original parameters and with an increase and decrease of $T_h$ (and consequently $T_c$) of 1°C. It is clear that, with just one degree of displacement, there is a tremendous drop of performances. This shows that the working conditions of the ENOVHEAT heat pump have to be precisely the same as the designed ones. It demonstrates the absolute inelasticity of the machine, and therefore, the great difficulties of using it in practice.

To mitigate the effects of different working temperatures than the designed ones, two reservoirs by each heat exchanger, one upstream and one downstream, could be used. That would increase the inertia of the system and would allow more stable running conditions of the machine.
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Fig. 4.18: The heating capacity varies sensibly, just slightly changing the reservoirs temperatures. The temperature span is kept constant

The second test shows the heating capacity if the machine is used with a cold reservoir at a different temperature that the design one. The hot temperature is always 310 K, the cold temperature varies and so the span accordingly. Figure 4.19 represents the heating capacity in the various conditions. Note that, with higher temperature span the heating capacity falls rapidly. The layers with lower Curie temperature are not exploited. With lower temperature span, the heating capacity raises much less markedly. It is curious to notice that, for temperature span smaller than 25K, the maximum appears at higher flow rates. As said before, the heating capacity reaches a maximum because it is influenced by two phenomena. The higher the volume flow rate is, the higher the heat transfer coefficient is. The temperature gradient along the regenerator flattens itself with too high volume flow rates though. With a smaller temperature span the regenerator is less sensible to the latter phenomenon. This demonstrates the necessity of choosing a cold reservoir with constant temperature (like a bore hole). Moreover, the design $T_c$ has to be the coldest temperature that can be reached by the reservoir. Indeed, a little decrease would make the heat pump useless.
After studying the optimum working conditions, it is worth investigating how to operate in practice.

The simplest way to run the heat pump would be through on-off cycles. This is the control logic for old vapour compression systems, but for magnetocaloric devices it is not the proper control strategy. As it can be seen from the previous graphs, it is not very efficient to work at high heating capacities, where the COP is rather low. Moreover, a magnetocaloric heat pump has very long transient responses. When the system is shut off and then restarted, it requires long time to reach steady state again. During this interval the pump and the motor work without providing any useful effect.

If the maximum heating capacity is not demanded, it is better to work at lower frequencies and mass flow rates, thus higher COP. It is, therefore, necessary to use a variable speed pump and motor.

The plot COP vs heating capacity (figure 4.20) gives an immediate representation of the better working conditions at every heating capacity.

**Fig. 4.19**: Change of heating capacity at different Tspan. The hot temperature is kept constant at 310 K

**4.8. Partial load conditions**
Figure 4.21 compares other alternatives, which are to regulate the heating capacity, modulating the mass flow rate, the frequency and both. It is clear that the last strategy is the most efficient.

**Fig. 4.20:** Performances of the best flow profile at various frequencies.

**Fig. 4.21:** Possible modulation strategies to achieve partial load conditions. From the worse to the best solution: modulating only the frequency; on/off cycle; modulating only the mass flow rate; modulating both the frequency and the mass flow rate.
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For the realization of high heating capacities, there is no appreciable difference between modulating just the mass flow rate and both the mass flow rate and the frequency. Instead, the difference becomes significant at low heating capacities. It is never convenient to regulate just the frequency, and, as we said before, on/off cycles, other than inefficient (figure 4.21), would be difficult to realize.

For practical applications, it would be, therefore, advisable to select some combinations of mass flow rates and frequencies, corresponding to various heating capacities.

Another way of realizing partial load conditions is to decrease the hot temperature and so the temperature span. If the heat pump provides the system with working fluid at lower temperature, the heat transfer between the fluid and the ambient will be smaller. Figure 4.22 shows in the (COP, heating capacity) plot the results of the working conditions with the original parameters (Thot 310; Tspan 25) and with lower hot reservoir temperature (Thot 308; Tspan 23 and Thot 306; Tspan 21) at frequencies equal to 0.5 Hz, 1 Hz and 2 Hz.

The simulation with the original parameters gives always better performances: modulating both the frequency and the mass flow rate is still the best option.

Since working at low heating capacity implies high COP, installing a thermal storage would be beneficial for the system. It would cost and occupy space but it would guarantee more efficient working conditions.
**Fig. 4.22:** Decreasing the hot temperature and temperature span as another possible partial load strategy. Plot of performances for frequencies equal to 0.5, 1, 2 Hz (lower heating capacities and higher COPs correspond to low frequencies).

The original parameters let realize higher performances: modulating mass flow rate and frequency remains the best control strategy.
4.9. Summary

Different flow profiles, that is when the working fluid blows through the AMR with respect to magnetic field profile, are considered. Such a study was typically neglected in the literature. Often only some defined thermodynamic cycles are investigated, but, as it is shown, this is not sufficient to define the best flow profile. This is non-synchronized with the magnetic field profile and without any pause fraction. Further data analysis and an experimental validation aim at proving the reasonability of the outcome of the simulations. The novel magnetic field with short ramps and large maximum high field regions, could justify the absence of pause fractions in the best flow profile. An extra heating and cooling of regenerators during, respectively, magnetization and demagnetization, seems to benefit the magnetocaloric effect and allows the system to realize a larger fluid temperature span.

Experiments on the heat pump are, however, needed to confirm the numerical results.

A width extension, 1D+1D approach is presented. It aims at a better implementation of the interplay between magnetic field and flow profile, considering the width of the regenerators, without resorting to 2D models. Significant deviations are not experienced in the particular case of the best flow profile, but its relevance is shown. It is, anyway, a fair reflection, that could become fundamental for wider (respect to the magnetic field periodic variations) regenerators.

Working conditions with different input parameters than the designed one, that is dissimilar hot and cold reservoir temperatures and temperature span, are examined. The use of first-order materials imposes a strict respect of the designed parameters, and therefore, a difficult use of the heat pump in real applications. Even small deviations cause big drops of performances. Possible solutions, like the introduction of reservoirs to mitigate these effects, are suggested.

Eventually, the practical modulation of the heat pump is investigated. Under this point of view, the realization of partial load conditions through a control strategy that delivers high efficiency, is fundamental, especially for magnetocaloric devices. The optimum control strategy is the regulation of both machine frequency and working fluid mass flow rate.
5. **Flow balancing**

From previous experiences it is known that regenerators, which are designed to be identical, can differ in practice [8]. This is due to the production system. The arrangement of magnetocaloric spheres and so the pores in the regenerators cannot be the same. Similarly, the flow passages leading to the individual regenerator beds will always have some variations, especially for low volume production and prototyping. This influences the behaviour of each bed and therefore of the machine, and can be significantly detrimental for its performances [66]. In the ENOVHEAT prototype, an extra chance of solving any unbalance is given by the independent flow control system for each bed.

A clear evidence of the difference among the regenerators is the unequal flow resistance among them. The pressure drop through each regenerator has been measured for various volume flow rates. This allows to compare the regenerators and to study mutual influences during normal working conditions. Moreover, the measurements let validate the use of the chosen correlation for the pressure drop, and highlight the inaccuracy of the classic Ergun equation, and the effects of some common mistakes.

A brief explanation of the theoretical calculation of the pressure drop through a porous media precedes the presentation of the experimental data, in order to justify the choice of the currently used pressure drop correlation.

### 5.1. Pressure drop through a porous medium

The pillar relation for the calculation of the pressure drop trough a porous media is the Darcy law [50]. It considers a cylindrical pipe with a porous bed made of uniform-size particles, randomly and loosely packed. The flow of a Newtonian fluid through the bed is 1-dimensional and the problem is solved in steady state. The Darcy law refers to the filter velocity, that is

\[ u_D = \frac{4\dot{m}}{\rho_f \pi D^2} \]  

(5.1)
where $\dot{m}$ is the mass flow rate, $\rho_f$ the density of the fluid and $D$ the diameter of the pipe. 

Darcy realizes that the pressure drop through the porous bed is a function of the filter velocity, the viscosity of the fluid and the permeability of the bed. The last quantity includes geometric and hydrodynamic characteristics of the porous media. The Darcy equation is:

$$-\frac{dp}{dx} = \frac{\mu}{K} u_p$$  \hspace{1cm} (5.2)

Where $-dp/dx$ is the pressure drop, $\mu$ the viscosity of the fluid and $K$ the permeability of the solid matrix. This model, and hence, the linear relation between pressure drop and filter velocity works good for Stokes flows. That is liquid flowing at low velocities, where viscous effects dominate over inertial effects. It is, indeed, easy to note the similarity between the Darcy law and the Hagen Poiseuille equation, which describes the laminar flow of a Newtonian fluid.

There are various models and correlations, based on the Darcy equation, that aim to predict the permeability of the solid matrix. Examples are the capillary model or the hydraulic radius model, better known as Carman-Kozeny. The latter one modifies the Hagen Poiseuille, considering the tortuosity of the bed and a shape parameter. Another similar correlation, used for packed sphere beds, is the Rumpf and Gupte [67].

The Darcy law is not suitable for high velocities, as inertial forces become more and more relevant. In this interval, the flow resistance is given by the sum of the viscous contribution and the inertial one. At very high velocities the first one becomes negligible and inertial forces become dominant.

These considerations bring Ergun [68] to formulate a correlation that could take into account both viscous and inertial effects:

$$-\frac{dp}{dx} = 150 \left(1 - \frac{1}{\varepsilon} \right)^2 \frac{\mu}{d_p} u_p + 1,75 \frac{1}{\varepsilon^3} \frac{1 - \varepsilon}{d_p^3} \rho_f u_p^2$$ \hspace{1cm} (5.3)

where $\varepsilon$ is the porosity, $d_p$ the particle diameter.

The pressure drop has a parabolic dependency on the velocity: the linear term refers to the viscous contribution and the quadratic one to the inertial ones.
Hicks [69] notes that the Ergun equation may not be applicable for \( \frac{Re_p}{1-\varepsilon} > 500 \), and it is almost not applicable when channeling occurs. Channeling is the reduction of flow resistance due to large pores adjacent to the boundary surfaces.

Since the Ergun correlation was developed (1952), many other relations based on the Ergun one have been presented.

Macdonald et al. [70], presenting their modified Ergun correlation, underline the importance of considering the porous medium. The original Ergun equation takes into account the porosity and the particle diameter; they go even further, including the roughness of the solid matrix. This allows better estimations of the pressure drop on one hand but on the other one, requires attention. The numerical coefficients in (5.3) may change, according to the specific case. Moreover, it has to be kept in mind, that the presence of non spherical particles, consolidated media, and wide particle size distributions has big influences in the results. These aspects could be, with some limitations, be included in a modified \( d_p \), since this quantity should be seen as a measurement of the average channel diameter, rather than the actual particle diameter. Giving an interval of pressure drop dependent on the roughness of porous material, the Macdonald correlation can predict different experimental data quite well. Finally, also the Macdonald equation gives unsatisfactory results for \( \frac{Re_p}{1-\varepsilon} > 500 \).

It is worth including the flow classification of Dybbs and Edwards [71]. It refers to the Reynolds number (different from the previous \( Re_p \)) which, further on, will be here called Dybbs Reynolds number:

\[
Re_d = \frac{\rho_f u_p d_a}{\mu}
\]  

where \( u_p \) is the average pore velocity, and \( d_a \) the average characteristic length scale of the pores.

Their classification is as follows:

- \( Re_d < 1 \): Darcy/Stokes flow;
- \( 1 - 10 < Re_d < 150 \): inertial flow;
- \( 150 < Re_d < 300 \): unsteady laminar flow;
- \( Re_d > 300 \): unsteady flow.
5.2. Comparison with experimental data

Measurements are taken by means of a flow meter and two pressure gauges, before and after the regenerator. The data are collected using LabVIEW. The accuracy of the flow meter is 1% of the full scale for measures <50% of the full scale, 2% of the reading value for measures >50% of the full scale; the accuracy of the pressure transducer is 0.3% of the full scale (4 bar relative) for measurements over 100 mbar. Measurements have been taken between 100 l/h and 550 l/h, every 50 l/h in both directions, corresponding to the hot-to-cold direction and the cold-to-hot one. The temperature of the water is kept constant, to assure the same value of viscosity during the experiments. This is realized with a thermal bath. The water is at 20 °C. Results are shown in figure 5.1. Each point coincides with the average of hundreds of measurements. The interpolations of the data of the highest and lowest resistances are depicted.

![Figure 5.1](image.png)

**Fig. 5.1:** Pressure drop versus volume flow rate for each regenerator. Interpolations of the data of the highest and lowest resistances are shown.

In order to compare the experimental data with various correlations for the prediction of the pressure drop, some assumptions, regarding the characteristics of the regenerators, have to be done. The regenerators have length equal to \( L = 0.059 \text{ m} \). At 20 °C and in the analysed range of pressure, the density of the water is 998 kg/m\(^3\), the dynamic viscosity is 0.0010 Pa·s (NIST). The regenerators have a trapezoidal shape (figure 2.2), so their cross sectional area is not
constant but the middle value can be taken: \( A = 0.001043 \, \text{m}^2 \). About the sphere diameter and the porosity, the matter is more complex. The regenerators are made of magnetocaloric spheres bounded with epoxy. Therefore, the bed is not an unconsolidated media and the epoxy could play a significant role. The spheres are, with good approximation, spherical and their diameter varies between 0.00045 and 0.00055 m. An average value could be assumed: \( d_p = 0.0005 \, \text{m} \). Nevertheless, as said before, the particle diameter is used just as an evaluation of the average pore diameter. Indeed, the presence of epoxy could affect the reliability of this assumption. Concerning the porosity, the real porosity of the regenerator can be easily calculated from volume and mass measurements, but the open porosity is required. The latter refers only to open pores, that the liquid can therefore reach. In order to have an exact value of the open porosity, measurements would be necessary, due to the presence of epoxy. An estimate will be used. According to the model of Haughey and Beveridge [72], for loose random packing, a porosity of 0.4 can be assumed.

The Carman-Kozeny, the Rumpf and Gupte, the Ergun, and the Macdonald correlations are computed consequently for the same range of volume flow rates of the experimental data. Figure 5.2 shows the experimental points (like in figure 5.1) and the analytical results of the above-listed equations.

To identify the flow regimes at the different volume flow rates and so the velocities, the Reynolds number based on the pore velocity, \( u_p \), is needed. To determine the pore velocity the Dupuit-Forchheimer relation is used:

\[
\text{u}_p = \frac{u_p}{\epsilon} \quad (5.5)
\]

The sphere diameter, then, is employed as the average characteristic length scale of pores. It yields that the Dybbs Reynolds number is included between 17, corresponding to 50 l/h, and 183, corresponding to 550 l/h. Such high Reynolds numbers denote the dominance of inertial effects over viscous effects. This is also clear from figure 5.2: the Darcy linear relations, like the Carman-Kozeny and the Rumpf-Gupte, strongly underestimate the experimental data.

The calculation of \( \frac{Re_p}{1-\epsilon} \) that is between 11 and 138 for the various flow rates, assure that the Ergun and Macdonald correlations are in the range of applicability. Furthermore, thanks to the careful assembly of the regenerators, channeling should not occur.
Figure 5.2 shows that the Macdonald rough can predict quite well the average pressure drop of the regenerators. The relative error between the two curves goes from around 30% at low volume flow rates to around 2% at high volume flow rates, with an average of 13% (figure 5.3). The high error at low volume flow rates is probably caused by the higher inaccuracy of the instruments at low flow rates.

Eventually, the Macdonald rough equation may be used in the code. It would be anyway wise to analyze more deeply the effects of the epoxy, in order to have right values of the open porosity and average characteristic length scale of the pores.

**Fig. 5.2:** Experimental pressure drop versus volume flow rate for each regenerator, average resistance of the regenerators and analytical results using different correlations
Fig. 5.3: Parity plot. Comparison between analytical data (Macdonald rough correlation) and the average of the experimental data. +/- 10% lines are depicted.

In models and dissertations, that can be found in the field literature (examples [46] [75] [8]), the Ergun correlation is sometimes used, and the Macdonald smooth frequently utilized. Moreover, it is not given importance to the difference between real and open porosity. Results, instead, show the goodness of the Macdonald rough correlation and the great inaccuracy of the Ergun and Macdonald smooth correlations for purposes like this one (figure 5.2). It is, then, of considerable importance accounting for the open porosity. The use of the total porosity (around 0.50 in the present case) would have brought to general substantial underestimations of the regenerator resistances (figure 5.4)
5.3. Circuit balancing

Although the differences of pressure drop of most regenerators are quite low, there are a few curves, which differ more. From figure 5.1, it is clear that, differences increase at high volume flow rates since, due to the parabolic trend, the ratio between resistances is not constant. The issue of different resistances is therefore significant. The nominal working condition, corresponding to the maximum heating capacity, occurs at rather high volume flow rates and so great differences of pressure drop.

It is, hence, convenient to study how to operate: is it worth balancing the circuit with external resistances, or is it better to leave the circuit as it is? Balancing the circuit would imply to have equal overall resistances in each branch of the system and, therefore, equal volume flow rate of liquid flowing through each bed. In this way, the optimum working conditions can be realized in each bed. On the other hand this would lead to an increment of the pumping work.

Afterwards, the possibility of using different flow profiles for each bed is considered in the perspective of the circuit balancing.

The flow profile indicates when, during the AMR cycle, there is fluid flowing through the bed in the two directions (blow fraction) and when there is no fluid flowing though the bed (pause fraction).
During normal operations, the fluid flowing through each bed follows the chosen flow profile. The fluid flows in the hot-to-cold direction when the regenerator is demagnetized and in the cold-to-hot direction when the regenerator is magnetized. If the 0.95tb profile is used, there are no pause fractions, meaning that, there is always fluid blowing through each regenerator. Since the ENOVHEAT heat pump has 13 beds, there are six or seven open beds in the cold-to-hot direction and seven or six open beds in the hot-to-cold direction. The continuous transition between six and seven open beds will be analyzed later on. Figure 5.5 shows a simplified (only the beds are shown) hydraulic scheme of the machine.

**Fig. 5.5: Simplified hydraulic circuit of the heat pump (only the regenerators are considered)**

Within the discussion whether it is more convenient to add localized pressure drops to balance the circuit or not, it has to be clarified that choosing heating capacity or COP implies fixing a mass flow rate. Therefore, knowing the resistances of the beds and fixing the total volume flow rate, the pressure drop is then univocally determined. This involves, that the problem can be simplified only considering the first six or seven beds in parallel. So, between analysing six or seven beds, for the sake of simplicity, we will further on refer to six beds in parallel. To generalize the problem, it is assumed that the six beds are two with the highest resistance, two with the lowest resistance and two with intermediate resistance.

The first case is the realization of a heating capacity or COP that is not the maximum one (example figure 5.6). Without balancing the circuit, the fluid divides itself unevenly among the
beds. This situation is preferred though, because some beds give higher performances, compared to the beds with intermediate resistance, and the others give lower performances than the intermediate ones. This results in overall performances that are almost equal to the performance of the intermediate beds. If the circuit is balanced, pretty much the same conditions are realized but with additional pressure drops and, hence, higher work.

![Graph showing volume flow rates and corresponding heating capacities for regenerators with different resistances](Image)

**Fig. 5.6:** Example of volume flow rates and corresponding heating capacities for regenerators with different resistances (values are related to an equivalent heat pump)

The second case, that is the realization of the greatest heating capacity, is probably the most interesting one. If the circuit is not balanced, the beds with intermediate resistance will have a volume flow rate that corresponds to the highest heating capacity, but the other beds will have higher and lower volume flow rate, both corresponding to lower heating capacity (see figure 5.7). The maximum heating capacity is realized at a volume flow rate that do not coincide with the maximum COP. Therefore, for the non-balanced circuit, like in the previous case, the overall COP is almost equal to the one corresponding to the ideally balanced circuit. In the balanced circuit, the heating capacity is equal to the nominal maximum one, but localized pressure drops are added, and so the COP is lower. A quantitative study follows to
understand if the drop of heating capacity in the non-balanced circuit is more significant than the drop of COP in the balanced circuit.

**Fig. 5.7:** Example of volume flow rates and corresponding heating capacities for regenerators with different resistances (values are related to an equivalent heat pump)

It is known from the previous chapter (see figure 4.7), that the heat pump generates the greatest heating capacity, $\dot{Q}_{\text{max}}$, equal to 1110 W, with frequency equal to 2 Hz and volume flow rate, $\dot{V}_{\text{max}}$, of 2460 l/h. It corresponds to COP=2.6.

If the system was ideally balanced, the volume flow rate through each regenerator would be equal to $\dot{V}_i = \dot{V}_{\text{max}}/6 = 410$ l/h, for a correspondent heating capacity equal to $\dot{Q}_i = \dot{Q}_{\text{max}}/6 = 185$ W. Calculations are done considering two beds with low resistance (beds number 1 and 2), two beds with intermediate resistance (beds number 3 and 4), and two beds with high resistance (beds number 5 and 6). To find the pressure drop and relative volume flow rate through each regenerator, due to the parabolic trend of the pressure drop, an iterative method has to be used. Solving the system, composed of the six equations, obtained from the interpolation of the experimental data, a pressure drop of 1.03 bar is found. This corresponds to the following volume flow rates: $\dot{V}_1 = 448$ l/h, $\dot{V}_2 = 434$ l/h, $\dot{V}_3 = 420$ l/h, $\dot{V}_4 = 417$ l/h, $\dot{V}_5 = 370$ l/h, $\dot{V}_6 = 370$ l/h. It is possible to use the same curve as before, heating capacity as a function of volume flow rate at constant frequency, equal to 2 Hz, for the chosen best flow
profile, 0.95tb +0.06(period). The curve refers to the whole machine and therefore its values have to be divided by six. In this way the heating capacity of single regenerators can be estimated. It yields that the corresponding heating capacities are: $\dot{Q}_1 = 183 \text{ W}$, $\dot{Q}_2 = 184 \text{ W}$, $\dot{Q}_3 = 185 \text{ W}$, $\dot{Q}_4 = 185 \text{ W}$, $\dot{Q}_5 = 181 \text{ W}$, $\dot{Q}_6 = 181 \text{ W}$, for a total heating capacity of $\dot{Q}_{tot} = 1099 \text{ W}$. This yields to a COP=2.57, that is almost the same as in the ideally balanced circuit.

If localized resistances are added, so that in every branch the same overall resistance is realized, the system becomes balanced. The heating capacity is the maximum one but the COP gets lower. Extra work, called balancing work, $\dot{W}_b$, is added to the original work, $\dot{W}_0$, to win the additional pressure losses. The original work is calculated considering an average resistance, given by the correlation. The balancing work is, therefore, the difference between the highest pressure drop, related to $\dot{V}_i$, and the pressure drop given by the correlation, multiplied by the overall volume flow rate.

$$\dot{W}_b = \dot{V}_{tot} (\Delta p_{max} - \Delta p_{correlation}) = 14 \text{ W}$$

Hence, the COP = $\frac{\dot{Q}_{max}}{\dot{W}_0 + \dot{W}_b} = 2.5$.

Finally, it is clear that it is more convenient not to balance the system. The balancing of the system would give 11 W more of heating capacity, providing extra 14 W of work though. Differences are, anyway, rather small.

Another case is considered, but calculations will not be reported. The realization of a heating capacity close to the maximum would lead to such a condition that, the difference of heating capacity between the bed with the lower resistance and the middle one and the difference of heating capacity between the bed with the higher resistance and the middle one, would differ. This happens because the heating capacity curve changes its slope significantly close to the maximum. (see figure 5.8). Even in this case, calculations show that the balancing of the circuit would lead to higher work than the gain in heating capacity.
Fig. 5.8: Example of volume flow rates and corresponding heating capacities for regenerators with different resistances (values are related to an equivalent heat pump)

To minimize the problem of beds with different pressure drops in parallel, the relative position of the beds in the machine can be significant. It is preferable to have beds with similar pressure drop open in one direction in the same moment. So that the volume flow rate through each open bed is very similar to the others. Since there are two regions of high magnetic field and they are opposite to each other, opposite regenerators will be open at the same time, and therefore, they should have similar pressure drop.

Figure 4.7 shows a possible arrangement of the 13 beds.
It is worth discussing about using different flow profiles within the same machine. It could seem reasonable to use various flow profiles for regenerators with different pressure drop. The chosen best flow profile, 0.95tb, which is without pause fraction, could be used for regenerators with high resistance. Other flow profiles with smaller blow fraction and a certain pause fraction could be employed for regenerators with low resistance. In this way all the regenerators would have the same utilization, but the efficiency of the machine would decrease. Just consider two open beds, for the sake of simplicity, but now assume that they have different flow profiles. When the beds are both open, the system is still unbalanced, so the volume flow rate through each bed is not the optimum one. If, then, the bed with the lowest pressure drop, closes before, the volume flow rate flowing through the other bed will be much higher, and hence, far from the optimum condition.

There are two other situations when using a different flow profile than the 0.95tb+0.6(period) could be useful though.

It could happen that the flow resistance through one bed is different in the two directions. This condition could occur if external components are installed asymmetrically in the circuit. Dissimilar flow resistances imply dissimilar volume flow rates in the two directions. The heat fluxes hot-cold and cold-hot would be different and this would lead the system to an
equilibrium condition that is not the designed one and so inefficient. Indeed, figure 4.18 and 4.19, show that slight changes bring to tremendous performance drops. With an asymmetrical flow profile, that is with different blow fractions in the two directions, this issue could be solved.

As stated before, in the analyzed heat pump there are six or seven beds open in the same moment. The transition between six and seven open beds, and vice versa, could cause spikes in pressure and volume flow rates. Besides, it could affect the efficiency of the machine. If this was problematic, the issue could be managed using a different flow profile: to add the right pause fraction could avoid this continuous transition and let work only six open beds at the same time. To examine the matter, and study whether this continuous transitions is detrimental for the performances of the machine, a simulation with the actual flow profile is run (figure 5.10). The flow continuously changes: when there is the passage from six to seven open beds, the flow decreases, spreading in more beds.

**Fig. 5.10:** Normalized magnetic field profile and actual flow profile. The latter one takes into consideration that the heat pump works with a constant transition between six and seven beds in the same direction
Figure 5.11 shows the results of the simulation, i.e. the comparison between the performances of the ideal constant profile and the actual one. Note that, points with similar heating capacity and COP do not necessarily have the same volume flow rate. However, the total amount of liquid flowing through the beds is equal in the two cases. It is clear that it does not make sense to investigate different solutions since the performances are pretty much equal.

Fig. 5.11: Difference of performances between the ideal best profile (0.95 $t_b$ + 0.6(period)) and the actual flow profile
5.4. **Summary**

A discussion about pressure drop through the regenerators and the overall hydraulic system is presented.

Theoretical backgrounds on pressure drop through a porous medium introduce a comparison among different analytical correlations for the calculation of the pressure drop through a regenerator. This allows us to show the great inaccuracy of some correlations used in the literature, and to underline the importance of using the open porosity. The use of the Macdonald rough correlation is suggested. It fits quite well the experimental measurements of the beds resistance. It is also highlighted that the lack of information regarding the effects of the epoxy in the regenerators, may affect the results. A deeper analysis would be required.

As beds have different resistances, a study whether it is worth balancing the circuit is performed. This demonstrates that it is useless, or even counterproductive, to balance the circuit. It is also shown that placing beds with similar resistances opposite to each other could minimize the issue of different flow rates through them.

The use of unlike flow profiles for different regenerators or other profiles than the $0.95tb +0.6(\text{period})$ is considered. It is not advisable to use different flow profiles just to have the same utilization because performances would decrease. An asymmetrical flow profile could be used to solve internal unbalances, that are different resistances in the same bed in the hot-to-cold direction and the cold-to-hot one.

Eventually, the working conditions with the actual flow profile with the continuous six-seven open beds transition are examined. The simulation result shows that performances do not change appreciably from the ideal case.
6. Conclusions

The new magnetocaloric heat pump, developed within the ENOVHEAT project at DTU, is presented. It has thirteen active magnetic regenerators with first order magnetocaloric materials and a rotating magnet. A proper flow system assures the right exchange between the regenerators and the working fluid. The way the fluid flows through the regenerators with respect to the magnetic field is called flow profile.

A numerical optimization of the prototype's working conditions is performed via a Matlab code. The energy equations of solid regenerator and fluid are simulated using the finite volume method. The characteristics of the device, the magnetocaloric materials, the actual magnetic field, and the working parameters are implemented. The magnetic hysteresis and the demagnetizing field are included in the code. The output quantities are the temperature profiles of solid and fluid during the machine operating cycle. Secondary parameters, such as heating and cooling capacity, COP, and second law efficiency, are consequently determined.

In order to compare different working conditions, previous studies deal mainly just with thermodynamic cycles. Here the interplay between magnetic field profile and flow profile is investigated in a broader way. Cases that do not correspond to precise thermodynamic cycles are analyzed. This lead to results of general interest. In practice, the use of an independent control system for each bed allows to control and vary the flow profiles and, hence, to realize various working conditions. A comparison between the heating capacity and COP curves of different working conditions let choose the best flow profile. This does not have pause fractions, i.e., there is always fluid blowing through the regenerator, and it is non-synchronized with the magnetic field. The outcomes are unforeseen but reasonable. They are, consequently, compared with the literature, and qualitatively experimentally with another prototype at DTU. The absence of pause fraction in the best flow profile may be caused by the use of a magnet producing a
magnetic field with short ramps and large high field regions. If confirmed, this outcome would be of great interest.
The non-synchronization makes hot fluid flow during magnetization and cold fluid flow during demagnetization, and this may contribute to increase the magnetocaloric effect. The temperature of the regenerators is higher in the high-field region, and lower in the zero-field region. This could explain the best performances of a slightly delayed flow profile. Experimental validation is, however, required.

The described model is 1D. To take into consideration the width of the beds, a 2D model would be required, but it would be computationally expensive. An alternative 2D approach, 1D+1D model, is presented. The beds are divided into more partitions, and the 1D model is solved in each of them. The average result is, then, more accurate than the single simulation.

The regenerators of the ENOVHEAT prototype are made of first-order materials. They exhibit a significant magnetocaloric effect, but only in proximity of their Curie temperatures. Simulations with slightly different input parameters than the designed ones, are performed. The results show how disadvantageous it is not to work within the very right range of temperatures, and demonstrate the difficulties in employing such a device in real applications. The introduction of reservoirs may mitigate this disadvantage.

The proper running of the heat pump is fundamental to achieve high performances. How to better realize partial load conditions is investigated. This is particularly relevant for magnetocaloric devices: as shown, the COP changes sensibly. Simulations show that the best way is to vary both motor frequency and mass flow rate.

A different topic, but still connected with the optimization of the heat pump, is analyzed. Measurements of the flow resistance across the various regenerators let validate the use of the pressure drop correlation. Moreover, the inaccuracies of some correlations used in the literature are highlighted.

It is shown that beds' resistances vary among each other. The study of a possible circuit balancing follows. If the regenerators have different resistances, the amount of fluid flowing through them will differ. This would lead to the impossibility of
realizing the same working conditions in each bed. The overall performances of the machine would, therefore, decrease. Adding proper localized resistances would balance the circuit. All the beds would work in the same way, but the pump work would increase. A quantitative study demonstrates that, it is more convenient not to balance the system. In the ENOVHEAT prototype, an extra chance of solving any unbalance is given by the independent flow control system for each bed. This could be useful for internal unbalances, but it is not for external unbalances.

Eventually, considering that the heat pump has thirteen regenerators, there is a continuous transition between six and seven open beds in each direction. The simulation of this issue shows that it does not affect performances.

As already remarked, experimental tests on the machine will be essential to validate the reported numerical results.
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Nomenclature

Roman letters

\( a_s \) Specific surface area \([1/m]\)
\( a_i \) Coefficient of the matrix \( A \)
\( [A] \) Matrix of the coefficients of the temperatures
\( A_c \) Regenerator cross sectional area \([m^2]\)
\( b_i \) Known term of vector \( [B] \)
\( B \) Magnetic flux density \([T]\)
\( [B] \) Vector of known terms
\( c_f \) Fluid specific heat \([J/kg*K]\)
\( c_H \) Isofield specific heat \([J/kg*K]\)
\( d \) Diameter \([m]\)
\( d_h \) Hydraulic diameter \([m]\)
\( d_p \) Particle diameter \([m]\)
\( dt \) Temporal step \([s]\)
\( dx \) Spatial step \([m]\)
\( D \) Material derivative
\( f \) Frequency \([Hz]\)
\( f_f \) Friction factor
\( h \) Specific enthalpy \([J/kg]\)
\( H \) Magnetic field \([A/m]\)
\( H_i \) Initial magnetic field \([A/m]\)
\( H_f \) Final magnetic field \([A/m]\)
\( i \) Spatial node
\( j \) Temporal node
\( k \) Thermal conductivity \([W/(m*K)]\)
\( k_{disp} \) Thermal conductivity due to fluid dispersion \([W/(m*K)]\)
\( k_{eff} \) Effective thermal conductivity \([W/(m*K)]\)
\( K \) Permeability of the solid matrix \([Pa]\)
\( L \) Regenerator length \([m]\)
\( m \) Specific magnetization \([A*m^2/kg]\)
\( \dot{m}_f \) Fluid mass flow rate \([kg/s]\)
\( m_s \) Solid mass \([kg]\)
\( M \) Volume magnetization \([A/m]\)
\( M_{irr} \) Irreversible volume magnetization \([A/m]\)
\( M_{neg} \) Volume magnetization during demagnetization \([A/m]\)
\( M_{pos} \) Volume magnetization during demagnetization \([A/m]\)
\( Nu \) Nusselt number
\( p \) Pressure \([Pa]\)
\( Pe \) Peclet number
\( Pr \) Prandtl number
Nomenclature

\( \dot{q} \)       Heat flux [W/kg]
\( \dot{Q}_c \)   Cooling capacity [W]
\( \dot{Q}_h \)   Heating capacity [W]
\( Re_f \)    Reynolds number based on the hydraulic diameter
\( Re_p \)    Reynolds number based on the particle diameter
\( s \)        Specific entropy [J/(kg*K)]
\( \dot{S}_g \)       Irreversible entropy production rate [W/K]
\( S_{p,ac} \)   Entropy production rate due to axial conduction [W/K]
\( S_{p,ht} \)   Entropy production rate due to insufficient heat transfer [W/K]
\( S_{p,vd} \)   Entropy production rate due to viscous dissipation [W/K]
\( S_{p,tot} \)  Total entropy production rate [W/K]
\( t \)           Time [s]
\( tb \)         Blow fraction
\( T \)         Temperature [K]
\( \{T\} \)       Temperature vector [K]
\( T_C \)    Cold reservoir temperature [K]
\( T_f \)    Fluid temperature [K]
\( T_H \)    Hot reservoir temperature [K]
\( T_s \)    Solid temperature [K]
\( u_D \)    Darcy/filter velocity [m/s]
\( U \)      utilization
\( V_n \)    Normalized volume flow rate
\( \chi \)    bed length fraction

Greek letters

\( \alpha \)    Convective heat transfer coefficient [W/(m²*K)]
\( \gamma \)    Integration path
\( \Delta S_{iso} \)    Isothermal entropy difference
\( \Delta T_{ad} \)    Adiabatic temperature difference
\( \epsilon \)    Porosity
\( \eta \)     Effectiveness
\( \eta_{II} \)    Second law efficiency
\( \mu_0 \)     Vacuum permeability [T*m/A]
\( \mu_f \)    Fluid dynamic viscosity [Pa*m/s]
\( \phi \)      AMR period fraction
\( \rho_f \)    Fluid density [kg/m³]
\( \rho_s \)    Solid density [kg/m³]
\( \tau \)     Machine period fraction

Abbreviations

AMR    Active magnetic regenerator
CFL    Courant number
COP    Coefficient of performance
DSC    Differential scanning calorimeter

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### Nomenclature

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>EPR</td>
<td>Entropy production rate [W/K]</td>
</tr>
<tr>
<td>GWP</td>
<td>Global warming potential</td>
</tr>
<tr>
<td>MCE</td>
<td>Magnetocaloric effect</td>
</tr>
<tr>
<td>MCM</td>
<td>Magnetocaloric material</td>
</tr>
<tr>
<td>nbed</td>
<td>Number of beds</td>
</tr>
<tr>
<td>NTU</td>
<td>Number of transfer unit</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone depletion potential</td>
</tr>
<tr>
<td>RTD</td>
<td>Resistance temperature detector</td>
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### Other symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>ℕ</td>
<td>Demagnetizing field</td>
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Bibliography


