Active magnetic regenerator refrigeration with rotary multi-bed technology

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Publication date: 2016

Document Version
Publisher's PDF, also known as Version of record

Link back to DTU Orbit

Citation (APA):

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Abstract

Magnetic refrigeration is an emerging cooling technology with potential advantages over conventional vapor compression, the most important being higher efficiency. This thesis presents experimental and theoretical research into the possibilities of realizing this potential with actual active magnetic regenerator (AMR) prototypes. The starting point is the design and experiments with a rotary multi-bed prototype at the Technical University of Denmark. Promising results were obtained with this machine in terms of temperature span and cooling power. However, issues limiting the energy efficiency, mainly relating to heat leaks and flow system friction losses, have given rise to new ideas for taking the technology a step further. On this background, a second generation multi-bed prototype was designed, built and used in experimental investigations.

A central feature of the new prototype is a novel system for handling the heat transfer fluid, providing a reciprocating flow inside the AMR beds while ensuring a continuous unidirectional flow in the surrounding flow circuit, communicating with the hot and cold reservoirs. With this system it is possible, via an arrangement of poppet valves and check valves, to control the flow rate versus rotational angle of the magnet system providing a time varying magnetic field in the beds with very minor losses compared to more traditional rotary valve based systems.

Numerical AMR modeling capturing the variations in the azimuthal direction inside the beds has been used to investigate the effect of the shape of this flow profile, which confirms the importance of carefully optimizing it for the desired operating conditions.

Numerical modeling and heat transfer calculations addressing heat leaks through the walls of the regenerator housing has revealed a necessary trade off between the amount of magnetocaloric material and an insulating air gap in the magnetized volume provided by the Halbach-like cylindrical permanent magnet system, when designing for high efficiency rather than maximum cooling power.

The central part of the magnet system is a flux conducting iron core which was laminated for electrical and thermal insulation to minimize heat leaks and eddy current losses. Experimental investigations with different configurations of iron and insulation in the core focusing on the impact on
temperature span and COP were conducted. AMR experiments with the new prototype revealed strong impacts on COP and cooling power by minor adjustments of the individual valves controlling the flow in each bed. This effect, inherent to rotary multi-bed AMRs, is addressed with a numerical modeling approach and confirmed experimentally with the new prototype by carefully evening out the variations by the means of needle valves. An experimental performance analysis of the new prototype was carried out. A breakdown of the losses indicate pressure drop in external components and regenerator losses as the main contributors to entropy generation. While the former may be reduced by simple design improvements, the latter is non-trivial and requires detailed geometrical optimization assisted by numerical modeling and improved manufacturing techniques. Finally, possible applications are discussed and a concept of operating the AMR machine in combination with a thermal storage is introduced and demonstrated experimentally. Furthermore, a novel shunt valve technology, which was developed as a spin-off from the magnetic refrigeration research, is presented.

Et centralelement i den nye prototype er et nyskabende system til håndtering af varmetransport-væsken som sørger for en reciprokerende strømnings i de enkelte AMR beds i kombination med en kontinuerlig strømnings i det omgivende væskesystem, som kommunikerer med det varme hhv. kolde reservoir. Med dette system er det, via et system af knastaktuere ventiler og kontraventiler, muligt at kontrollere volumenstrømmen som funktion af rotationsvinklen af magnetisk felt i de enkelte beds med meget beskedne tab sammenlignet med traditionelle rotationsventil-baserede systemer til følge.

Numerisk AMR modelering, som tager variationerne i den azimuthale retning i de enkelte beds i betragtning, er anvendt til at undersøge effekten af en form for dette volumenstrøms-profil, hvilket bekærer vigtigheden af en omhyggelig optimering i henhold til de ønskede operationsbetingelser. Numerisk modelering og beregninger angående ønsket varmetransmission gennem regenerator-væggene har afsløret nødvendigheden af et kompromis mellem mængden af magnetokalorisk materiale og et isolerende (luftfyldt) mellemrum i det magnetiserede volumen som tilvejebringes af det Halbach-lignende, cyldriske system baseret på permanente magneter, når der designes med høj effektivitet frem for maksimal køleffekt for øje.

Den centrale del af magnetisk system er en flux-ledende jernkerne, som er laminieret for at opnå elektisk og termisk isolering med henblik på at at
minimere tab hidrørende fra uønsket varmeoverførsel og inducerede eddy-strømme. Dette er undersøgt eksperimentelt med fokus på køleeffekt og COP vha. forskellige kombinationer af jern og isoleringsmateriale i kernen. AMR eksperimenter med den nye prototype har afsløret kraftige påvirkninger af COP og køleeffekt hidrørende fra små justeringer af de ventiler som regulerer strømningen i de enkelte beds. Denne effekt, som er generel for multibed AMR-maskiner, adresseres med udgangspunkt i numerisk modellering og bekræftes eksperimentelt med den nye prototype ved omhyggeligt at udjævne variationerne vha. nåleventiler.

Acknowledgements

It all began with my section leader, Nini Pryds, who gave me the opportunity to pursue my ideas emerging during the work with magnetic refrigeration and encouraged me to convert them into a real machine, use it in my research and collect it all in the present PhD thesis. For this I would like to express my sincere gratitude towards you personally and the management at DTU Energy for supporting the idea. Although it has been a long and hard struggle, much harder than I ever expected, I was always surrounded by excellent and helpful colleagues without whom this would never have been possible. I have especially enjoyed the countless hours of enthusiastic mechanical design discussions and making ideas into reality in the lab with Kurt Engelbrecht. I am deeply grateful for the way you, Christian Bahl and Rasmus Bjørk have closely followed my work as supervisors from week to week and each made your own crucial contributions individually and as a team. Also Kaspar Nielsen, who shares my passion for what we (quite exclusively) perceive as intelligent humor, has played a crucial role throughout the project and I dare say I could not have been supported by a finer team of experts within every aspect of magnetic refrigeration. You were always ready to discuss whatever issue that came up and I am truly impressed by your profound skills and knowledge. More people have played important roles along the way. I would like to acknowledge Adnan Bijedic, Jørgen Geyti and Finn Saxild for technical support and excellent craftsmanship. I also owe a special debt of gratitude towards each of my good colleagues, former and present, Tian Lei, Andrea Insinga, Henrique Bez, Anders Smith, Jaime Lozano, Jaka Tusek, Lars von Moos, Jan Haertel, Regina Bulatova, Jesper Jensen, Dennis Christiansen, Kjeld Behm, Pernille Nielsen, Brit Hansen, Inge Biering and Stinus Jeppesen for fruitful discussions, kind help and great inspiration along the way. And thank you to Kristina Navickaité who just moved into my office together with Stefano Dall’Olio, cheering me up with various cakes and tea during these hard times of writing and working. And I really enjoy continuing the work on magnetocaloric prototypes with Stefano - you are doing a great job! Finally, I would like to express my deepest gratitude towards my entire family. Especially my two awesome sons, Thor and Asbjørn, and my beloved wife, Mette. I thank you for being so immensely caring, patient and understanding - you guys are simply the best!
List of publications

(Related to the present thesis only)

Articles in scientific journals (Peer reviewed)

- Eriksen, D., Engelbrecht, K., Bahl, C. R. H., Bjørk, R. (2016). Exploring the efficiency potential for an active magnetic regenerating refrigerator. *STBE Not-in-kind HVAC* Accepted manuscript (referred to as Eriksen et al. (2016b)).


**Articles in proceedings (Peer reviewed)**


**Published patent application**

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Chapter 1

Introduction

Magnetic refrigeration is an emerging cooling technology which is currently a subject of intense research as it may be a beneficial alternative to conventional vapor compression. It relies on the subsequent magnetization and demagnetization of a solid refrigerant, which may be considered analogous to the compression and expansion of a gas in a conventional refrigerator. The solid magnetic refrigerant, often denoted a magnetocaloric material (MCM), heats up when a magnetic field is applied to it and cools down when the field is removed. Removing a field from such a material was first utilized for obtaining temperatures close to absolute zero by Giauque and MacDougall (1933) and it has been used in cryo cooling applications ever since. However, it is only possible to directly obtain a cooling of a few Kelvin by such a “single shot” or adiabatic demagnetization refrigeration (ADR) technique with a reasonable magnetic flux density. With the introduction of the active magnetic regenerator (AMR) by Barclay (1983), in which a porous matrix of MCM is subsequently magnetized and demagnetized while heat is transferred in and out by means of a fluid, it became possible to build up a temperature span between a cold and a hot reservoir many times larger than what is possible with the ADR technology. This initiated a growing interest in developing a technology for refrigeration near room temperature based on this principle. The potential advantages are:

- Absence of hazardous and/or environmentally harmful gaseous refrigerants.
- Absence of a compressor opens the possibility of silent operation.
- Possibility for high efficiency.

While the last point has been the main driver of the development, it is still not well understood how to utilize the potential for high efficiency and how high it can be. Investigating this through designing and testing
actual prototypes is the main scope of the present thesis. But first, the basic principles will be introduced in a little more detail.

\section{Physical background}

\subsection{The magnetocaloric effect}

Magnetic refrigeration is based on the magnetocaloric effect in magnetic materials. This physical effect, discovered by Weiss and Piccard (1918), may be described by considering a block of ferromagnetic material, see fig. 1.1. In the initial state, its spins are randomly oriented. If the material is then magnetized by an applied magnetic field in a certain direction, they will align according to this. Furthermore, if the field is applied under adiabatic conditions, the temperature of the material increases by the amount $\Delta T_{ad}$.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig1.png}
\caption{A ferromagnet with randomly oriented spins is magnetized adiabatically. Due to the magnetocaloric effect, its temperature increases.}
\end{figure}

An explanation may be given in terms of entropy. The total mass specific entropy of the material, $s_{total}$, is composed of three parts: A magnetic part, $s_{mag}$, an electronic part, $s_{ele}$, and a lattice part, $s_{lat}$, i.e.

$$s_{tot} = s_{mag} + s_{ele} + s_{lat}. \quad (1.1)$$

As the magnetization process may be considered reversible (in some materials), the total entropy remains unchanged due to the adiabatic conditions. However, the magnetic entropy decreases as the spins are aligned. Therefore, the electronic and lattice parts must in total increase by the same amount. Hence, the temperature rises. If instead the magnetic field is applied at a fixed temperature, the specific entropy will decrease by the amount $\Delta s_{m}$, which is referred to as the magnetic entropy change.
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1.1.2 Basic refrigeration cycles

The magnetocaloric effect can be utilized to perform thermodynamic refrigeration (or heating) cycles between a cold and a hot heat reservoir. Such a refrigeration cycle may be illustrated by considering a piece of MCM which initially is in an unmagnetized state, A, at the temperature $T_H$ of the hot reservoir, see Fig. 1.2. A magnetic field is then applied adiabatically to the MCM which reaches a magnetized state, B, with an increased temperature. While the field is kept constant, heat is rejected to the hot reservoir, until thermal equilibrium is achieved. From this state, C, the magnetic field is removed adiabatically. The MCM thereby reaches state D, at which its temperature is lower than that of the cold reservoir. This allows the acceptance of a heat load from the cold reservoir while going from state D to D’ on the way towards a closing of the cycle by once again reaching the temperature of the hot reservoir at state A. In the same figure, an ideal (Carnot) cycle ($A' \rightarrow B' \rightarrow C' \rightarrow D'$) working between the same hot and cold reservoirs is indicated.

Figure 1.2: Brayton-like magnetocaloric cycle and Carnot cycle working between hot and cold reservoirs at temperatures $T_C$ and $T_H$ respectively.

1.2 Magnetocaloric materials

This section is meant to briefly provide a basic understanding of some important characteristics of MCMs for use in actual magnetocaloric devices. More detailed descriptions can be found in the literature, see e.g. Smith et al. (2012) (applications oriented) or Tishin (2007) (materials oriented). The magnetocaloric effect can be found in any ferromagnet. However, it is most pronounced in the vicinity of the so-called Curie temperature, above which the ferromagnet has undergone a phase transition to become a para-
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magnet. Thus, when utilizing MCMs for refrigeration, matching their Curie temperatures with the temperatures of the application is essential. Magnetocaloric materials are categorized according to the nature of the magnetic phase transition, which may be of 1st or 2nd order, see Figure 1.3. The basic understanding of these different material categories is important for the choice between candidates for refrigeration applications.

![Figure 1.3: Qualitative illustration of the second order (left) and first order (right) magnetization behavior of ferromagnetic materials around the Curie temperature.](image)

In general, the magnetization, $M$, of a ferromagnet decreases as a function of temperature, $T$. In the absence of an applied external field, $H$, the magnetization reaches zero at the Curie temperature, $T_C$. For a first order material, this is associated with a discontinuity in the magnetization corresponding to a latent heat during the phase transition. For a second order material, there is no latent heat. In this case, however, the first derivative of the magnetization curve is discontinuous at $T_C$ (with no applied field). If an external field is applied, the magnetization curves shift as indicated in Figure 1.3.

Pure gadolinium, which has a typical second order phase transition, is the only element with a Curie temperature near room temperature. It can be considered the best performing MCM available, with the significant drawback of being expensive (Smith et al. (2012)). It is much used in magnetocaloric devices around the world and may as such be considered a benchmark material. The interest in first order materials for magnetic refrigeration was boosted by the discovery of the so-called “giant magnetocaloric effect” in Gd$_5$(Si$_2$Ge$_2$), see Pecharsky and Gschneidner Jr. (1997). On one hand, the latent heat of first order materials, which may be associated with microstructural changes, can enhance the magnetic entropy change and hence the magnetocaloric effect. On the other hand, such materials
can exhibit a significant hysteresis, both thermally and magnetically. Such irreversibilities inevitably imply losses when the material is utilized in a refrigeration cycle. Therefore it is crucial to understand hysteresis in MCMs and the impact on refrigeration cycle performance, see e.g. von Moos (2014).

When considering and comparing MCMs for magnetic refrigeration, their adiabatic temperature changes and magnetic entropy changes, as described above, are essential quantities. To enable a meaningful comparison, the initial value, \( H_0 \), and final value, \( H_1 \), of the applied magnetic field must be specified. Both \( \Delta T_{\text{ad}} \) and \( \Delta S \) depend on the heat capacity, \( C \) (see e.g. Smith et al. (2012)):

\[
\Delta T_{\text{ad}}(T_0; H_1, H_0) = -\int_\gamma \frac{T}{C_H(T, H)} \cdot \left( \frac{\partial S}{\partial H} \right)_T dH,
\]

where \( \gamma \) is an isentrope from \( H_0 \) to \( H_1 \), and

\[
\Delta S(T, H_0, H_1) = \int_0^T \frac{C_H(T, H_1) - C_H(T, H_2)}{T} dT.
\]

Thus, the heat capacity as a function of magnetic field and temperature, which may be experimentally evaluated, is an essential characteristic of an MCM. Furthermore, the Maxwell relation

\[
\left( \frac{\partial S}{\partial H} \right)_T = \mu_0 \left( \frac{\partial M}{\partial T} \right)_H,
\]

where \( \mu_0 \) is the vacuum permeability, reveals the possibility of determining the isothermal entropy change through integration of magnetization data, which may be obtained experimentally, see e.g. Hansen (2010).

Apart from the quantities directly related to the magnetocaloric effect, thermal conductivity, \( \kappa \), is essential. It should be high enough to ensure a sufficient heat transfer in and out of the MCM during a cycle. On the other hand, a high conductivity can also hurt the performance of a refrigerator based on MCM when the material is arranged in a regenerator as described below.

### 1.3 Active magnetic regenerators

The near room temperature magnetic refrigeration prototypes reported in literature during the recent years almost exclusively use permanent magnets as magnetic field sources, since the losses associated with electromagnets or even superconductors are considered prohibitively large, see e.g. Kitanovski et al. (2015).

With permanent magnets, \( \Delta T_{\text{ad}} \) is only up to a few K for the known MCMs relevant for near room temperature refrigeration. Therefore, MCMs
are in real applications arranged in an AMR consisting of a porous matrix of MCM and a heat transfer fluid capable of exchanging heat with the solid and absorb/reject heat from/to the cold and hot reservoirs respectively. Such an arrangement is illustrated in Figure 1.4. By using it to perform what is known as the AMR cycle, a temperature gradient much larger than the $\Delta T_{ad}$ of the refrigerant can be obtained. The cycle consist of four steps:

1. The MCM is magnetized, e.g. by the application of a field from a permanent magnet. The MCM heats up throughout the matrix.
2. The heat transfer fluid is pushed in the direction from the cold to the hot reservoir and excess heat, $Q_H$, is rejected.
3. The MCM is demagnetized and its temperature drops consequently.
4. Finally, the fluid is pushed back towards the cold end and a heat load, $Q_H$, is absorbed.

Figure 1.4: Schematic illustration of the active magnetic regenerator cycle.

1.3.1 Numerical modeling of the AMR

Understanding the AMR theoretically is a multiphysics problem involving the magnetocaloric effect itself in combination with fluid flow and different heat transfer and dissipation mechanisms. Existing numerical models solving this problem exist and are used in different ways in the present work.
The starting point is a model resolving the regenerator in 1D, namely the
direction of the fluid flow and temperature gradient, see Engelbrecht (2008).
The model solves coupled differential equations describing energy balances
in the fluid phase and the porous solid regenerator matrix respectively.

Figure 1.5: Modelled regenerator bed with length, \( L \), and cross sectional
area, \( A_c \), consisting of a porous solid MCM matrix with a fluid flowing
through it at a mass flow rate of \( \dot{m} \).

For an infinitesimal control volume of length \( dx \) with cross sectional area,
\( A_c \), see Figure 1.5, the governing equation considering volume specific energy
balance for the fluid can be expressed as:

\[
\frac{\dot{m} c_f}{A_c} \frac{\partial T_f}{\partial x} + h a_s (T_f - T_s) + \rho_f c_f \frac{\partial T_f}{\partial t} - k_{\text{disp}} \frac{\partial^2 T_f}{\partial x^2} = \frac{\dot{m}}{\rho_f A_c} \frac{\partial p}{\partial x}. \tag{1.5}
\]

The terms represent, from left to right, enthalpy change of the fluid,
convective heat transfer from fluid to solid, energy storage, axial dispersion
and viscous dissipation. \( c_f \) is the mass specific heat capacity of the fluid,
\( T_f \) and \( T_s \) are the temperatures of the fluid and solid respectively, \( h \) is
the convective heat transfer coefficient from fluid to solid, \( a_s \) is the volume
specific surface area of the MCM, \( \rho_f \) is the fluid density, \( \epsilon \) is the porosity,
\( t \) is time, \( k_{\text{disp}} \) is the thermal conductivity of the fluid due to axial dispersion
and \( p \) is the pressure.

The corresponding governing equation for the solid can be written as:

\[
ha_s(T_f - T_s) + k_{\text{eff}} \frac{\partial^2 T_s}{\partial x^2} = (1 - \epsilon) \rho_s c_{H,s} \frac{\partial T_s}{\partial t}. \tag{1.6}
\]

Here the terms, from left to right, represent convective heat transfer
from fluid to solid, axial conduction, transfer of magnetic work and energy
storage. \( k_{\text{eff}} \) is the thermal conductivity of the combined fluid/solid matrix,
\( H \) is the magnetic field, \( \rho_s \) is the density of the MCM and \( c_{H,s} \) is the mass
specific heat capacity of the MCM for a fixed magnetic field.

Such models are in the present work used both as tools for development/optimization and investigation of actual AMR devices.

1.3.2 AMR devices

An increasing number of AMR test devices and prototypes have been reported during the recent years. A detailed review can be found in Kitanovski et al. (2015). However, to establish a reference framework against which the results of the present work can be discussed and evaluated, a few important devices will briefly be described here. Firstly, AMR devices may be categorized in different ways based upon their conceptual configuration, see e.g. Sharpa et al. (2012). In the present work, a simplified categorization will be used, based upon the way the MCM is magnetized and demagnetized, see Figure 1.6.

![Basic categorization of magnetocaloric machines](image)

Figure 1.6: Basic categorization of magnetocaloric machines.

The linear reciprocating machines, in which the magnet and regenerator are displaced linearly with respect to each other, are generally simple to construct and operate. They are mainly used for testing different regenerators and operating conditions. However, their operating frequencies are quite limited due to the reciprocating motion of large masses and they are not considered feasible for highly efficient operation.

In rotary devices the MCM is magnetized by magnet arrangements rotating relatively to the regenerator(s). Here, a distinction between “single bed per magnet” and “rotary multi-bed” machines is made. Machines of both types have demonstrated promising performance.

An example of a well documented state of the art prototype within the “single bed per magnet” category is the prototype at the University of Victoria. With this device, a temperature span of 33 K was recently reported, see Arnold et al. (2014). In this type of devices, the central bore in a nested Halbach array is utilized for the regenerator. By rotating the concentric magnet arrays relatively to each other, the resulting field can be alternated
between zero and a maximum value, see Figure 1.7 a. This gives the inherent advantage of a well controlled homogeneous magnetic field in the regenerator. The design and construction of regenerators is simple due to the circular cross section of the bore. However, in order to achieve efficient operation, it is necessary to combine a number of such systems mechanically and fluid flow-wise. The main argument against this type of devices however, is the fact that the magnet is only utilized for half of the AMR cycle, since the demagnetization is done by letting the field from the concentric magnet arrays cancel each other out. This is not desirable from an economical standpoint, since the magnet is the single most expensive component in
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an AMR machine, and should therefore be used for magnetizing MCM as much as possible, see e.g. Bjørk et al. (2010).

In rotary multi-bed systems, a number of compartments comprising the MCM are arranged annularly and the alternating magnetic field is provided by a magnet arrangement rotating relatively to the beds. One layout of this concept is sketched in Figure 1.7 b.

Compared to the single bed per magnet concept, a rotary multi-bed system could in principle utilize the magnetic field at all times if the beds are directly adjacent to each other. A group at the Tokyo Institute of Technology and Chubu Electric Power Co. presented a rotary four bed device in 2005, see Okamura et al. (2005). Further developments by this group has more recently resulted in a prototype with a reported COP of 2.5 at a temperature span of 5 K with gadolinium as MCM, see Okamura and Hirano (2013). At the Astronautics Corporation of America, another group is developing rotary multi-bed prototypes with the first results published in 2005, see Zimm et al. (2005). Recently, this group has reported a rotary device working at COPs above 2 at a temperature span of 10 K using a regenerator with six layers of the first order MCM La(FeSi)$_{13}$H with different $T_C$’s, see Jacobs et al. (2014). The progress made by the mentioned groups and others during the recent years emphasizes the importance of pushing the development and understanding needed to possibly reach the goal of commercially relevant AMR refrigeration.

A schematic representation of a rotary multi-bed system is shown in Figure 1.8. As indicated in the figure, the regenerator beds have to communicate flow-wise with a surrounding circuit, transferring heat in and out at the cold and hot ends, respectively. Therefore, some kind of valve interface is needed. In order to fully exploit the potential benefits of rotary multi-bed concepts, significant scientific and technological challenges affecting mainly the design of regenerators and heat transfer fluid flow system need to be overcome. Facing these challenges with a combined theoretical and experimental approach, aiming at demonstrating devices as close to commercially relevant applications as possible, has been the main driver behind the work described in the present thesis.

1.4 Working hypothesis

It is often said that magnetic refrigeration can provide high efficiency due to the fact that the magnetocaloric effect is reversible (in second order materials), but so is compression and expansion of an (ideal) gas. From an AMR modeling standpoint it seems possible to utilize MCMs for refrigerators with high COPs, see eg. Engelbrecht et al. (2006). Already in 1998, remarkably high COPs (above 6) were demonstrated experimentally with an AMR device by Zimm et al. (1998). This was, however, obtained with
1.4. WORKING HYPOTHESIS

Figure 1.8: Rotary multi-bed system including external flow circuit

a 5 T superconducting magnet and neither the energy consumption of this, nor the seal friction was included in the COP. How to actually achieve the goal of high COP in a real system based on permanent magnets is not well understood and, so far, only a few groups have even reported COPs of actual prototypes. It should also be stressed, that a COP in itself is of little relevance if not associated with a useful temperature span. The investigations presented in this thesis take the design and results of the first rotary AMR prototype at DTU as a starting point for looking into the possibilities of actually obtaining higher efficiencies within magnetic refrigeration. This is done by addressing individual issues that are considered crucial to device efficiency both during the novel design of a 2nd generation prototype and by subsequent experimental and theoretical analysis. The main intentions behind the presented work may be summarized as a working hypothesis:
Hypothesis

It is possible to design a rotary multi-bed permanent magnet magnetocaloric refrigeration device working near room temperature, which:

- Delivers a relevant combination of temperature span and cooling power and
- does so at a competitive COP.

1.5 Thesis outline

The work on rotary multi-bed magnetic refrigeration at the Technical University of Denmark began with the MagCool project, which resulted in a prototype presented in Engelbrecht et al. (2012). In this project, the author took part in the development, realization and experimental characterization of the machine, with special focus on the heat transfer fluid flow distribution system and mechanical design. The design and experimental results of this machine are presented in Chapter 2.

While this machine was indeed state of the art considering high cooling powers and high frequency operation, a number of challenges limiting the efficiency were discovered. This work has facilitated a range of ideas for alternative concepts enabling better designs. On this background, the author was given the opportunity to design, build and test a second generation rotary multi-bed prototype, with the primary aim of demonstrating efficiencies at least twice as high as those of the 1st prototype. This work has been documented in the chapters 3 through 6, which may be regarded as the "backbone" of the thesis. Each of these chapters is a peer reviewed publication, followed by a non-published addendum providing related additional information and background. As each of these chapters is per se an article in its own, it may be read individually. However, this also implies that some information is repeated several times throughout the thesis.

Finally, perspectives of the technology are discussed in Chapter 7, primarily based on the new prototype. Here, possible applications are discussed and exemplified with a demonstration setup in which the machine is coupled with an actual refrigeration cabinet through a thermal storage tank.

Furthermore, as an alternative use of MCM/permanent magnet systems, a novel valve technology, which was developed in parallel with the magnetic refrigeration activities, is also briefly presented in Chapter 7. This detour provides an example of a new application emerging as a spin-off from the main activities.
Chapter 2

1st rotary DTU prototype

Figure 2.1: The 1st rotary prototype developed, constructed and tested at the Technical University of Denmark.

The first rotary prototype built at DTU was an important step towards demonstrating the potential for magnetic refrigeration as a promising emerging technology. It was a result of a multidisciplinary effort of scientists, engineers and technicians involving materials research, magnet design, AMR modeling, mechanical design and construction. The design concepts were described by Bahl et al. (2011). Experimental results were published by Bahl et al. (2014) and Engelbrecht et al. (2012). In this chapter, the basic concept and design will be described and discussed from the perspective of the present thesis. Emphasis will be put on the heat transfer fluid flow system, as this was the main effort of the author. The performance of the machine and identified challenges relating to efficiency are discussed as this provides an important basis for the further work towards a more efficient device.
CHAPTER 2. 1ST ROTARY DTU PROTOTYPE

2.1 Design

The prototype is a rotary multi-bed device as described in the introduction. The objective for the design was to achieve a continuous operation and thereby utilization of the magnetized volume, but also to enable a continuous pump driven heat transfer fluid flow circuit communicating with hot and cold reservoirs.

The overall layout of the machine is illustrated in Figure 2.2. It was chosen to have a stationary magnet assembly with a cylindrical gap, see Figure 1.7b, in which a rotating multi-bed regenerator can rotate. The magnet has four high field and low field regions\(^1\) Having four poles rather than two enables a doubling of the AMR cycle frequency per rotational frequency of the regenerator. This is important from an efficiency point of view as the parasitic losses associated with dissipation in the rotating machinery scales with the rotational speed.

\(^1\)The design of the magnet is primarily a result of work carried out by R. Bjark and is described in details in Bjark (2010).
2.1. DESIGN

2.1.1 Regenerator

The relatively long spatial extension of the magnetized volume (250 mm) was meant for parallel plate regenerators as these are characterized by a low pressure drop compared to other geometries, such as packed spheres. The choice of parallel plates is linked with the development of a novel tape-casting technology of the oxide MCM $\text{La}_{0.67}\text{Ca}_{0.33-x}\text{Sr}_x\text{Mn}_{1.05}\text{O}_3$ which took place at DTU Energy simultaneously with the design and construction of the actual prototype. The Curie temperature of this material can be tuned by varying x, see e.g. Dinesen (2004). Bulatova (2014) describes the development of functionally graded plates with five different curie temperatures for the prototype.

Ideally, the air gap in the magnet would be filled with such MCM plates, equally distributed as sketched in Figure 1.7b. They should then be rotated around the center of the magnet while a flow system should distribute a fluid between the plates according to their position in the varying magnetic field sweeping over them. The flow system (described below), however, needed to be compartmentalized by effectively grouping a number of adjacent plates and supply them with the same flow from dedicated channels. Besides this, the need for mechanical support of the plates facilitated a compartmentalized structure with a carrying skeleton filled with cassettes containing the MCM, see Figure 2.2. The compartmentalization furthermore supports the ability to replace single cassettes in case of failure or to easily test different sets of regenerators.

Due to manufacturing difficulties and delays of the tape casted plates, sets of regenerator cassettes filled with gadolinium spheres were manufactured and used for testing the machine, see Engelbrecht et al. (2012) and Bahl et al. (2014). The beds used in the tests described below each contained 117 grams of Gadolinium spheres between 0.25 and 0.8 mm, filling up a length 100 mm.

2.1.2 Flow system

The heat transfer fluid used was a mixture of water and 20% ethylene glycol based automotive antifreeze to avoid corrosion of the gadolinium. The overall flow diagram is illustrated in Figure 2.3. In this realization of the rotary multi-bed concept illustrated in Figure 1.8, the regenerator beds are moving in and out of the stationary high and low magnetic field regions. Sliding valve interfaces are ensuring a reciprocating flow in the beds timed with the variations in the magnetic field. In this way each bed experiences a cold-to-cold blow while the MCM is magnetized and a hot-to cold blow while it is demagnetized. The flow in the circuit surrounding the regenerator on the cold and hot side is unidirectional and continuous. The flow is driven by a gear pump. The hot heat exchanger (HHEX), in which generated heat,
\( \dot{Q}_H \), is rejected, is connected to a chiller (not shown) enabling a controllable temperature of the fluid entering the hot side of the regenerator. To enable removal of heat generated by the pump, it is placed before the HHEX. On the cold side of the regenerator, the flow is passing through an electrical heater providing a controlled heat load, \( \dot{Q}_H \).

A consequence of the stationary magnet/rotary regenerator configuration is the need for dynamic rotary sealing. Figure 2.4 shows the working principle of the valve/rotary flow head system. The flow heads were manufactured in nylon using selective laser sintering (SLS) rapid prototyping. This was done in order to enable a good structural stability, which was needed as the flow heads had to transfer the torque between motor, regenerator and stationary valves. The flow channels exit/enter the regenerator at a relatively large diameter. However, the flow heads narrow down the bundle of flow channels. This reduces the necessary diameter of the sliding valve interface, which greatly reduces valve friction. With required hydraulic diameters of 8 mm, the channels are packed quite closely to reduce the diameter as much as possible without compromising the mechanical stability of the part.

The fluid going in and out of the regenerator through a flow head is split in two separate channels going nearly all the way to the MCM in order to

![Flow system diagram](image-url)
avoid mixing/dead volume. However, a small chamber is present at the very entrance to the MCM matrix to even out the flow profile over the entire cross section, see Figure 2.5.
2.1.3 Performance and challenges

The prototype has been characterized in a large number of experimental investigations, see Engelbrecht et al. (2012), Lozano et al. (2012), Lozano et al. (2013) and Bahl et al. (2014). The machine has been operated at high frequencies of up to 8 Hz. A maximum no load temperature span of 27 K and a no load cooling power of 1010 W were obtained. Some of the best intermediate results obtained are summarized in table 2.1.

<table>
<thead>
<tr>
<th>$\Delta T$ [K]</th>
<th>13.8</th>
<th>18.9</th>
<th>20.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{Q}_C$ [W]</td>
<td>400</td>
<td>200</td>
<td>100</td>
</tr>
</tbody>
</table>

A detailed experimental performance analysis, which is mainly a result of work carried out by Dr J. Lozano (Lozano et al. (2013)), has shown a maximum 2nd-law efficiency of 5.6%. By operating the system in different configurations, e.g. with and without the valves mounted, it has been shown that friction generated by the rotary lip seals and at the sliding valve interfaces between valve rotors and stators are one of the main issues limiting efficiency. Figure 2.6 (left) shows a cut-away view of the rotary valve system. A stainless steel ring with 48 cut-outs corresponding to the flow channels is mounted on the rotating nylon flow head, providing structural and dimensional stability. The static outer part was made of high density polyethylene, enabling relatively low friction. However, due to the difference in coefficients of thermal expansion between the plastic and steel cylinders, which is roughly an order of magnitude, a relatively tight fit was necessary, even though this increases friction. In a later design of the valve stator, a thin plastic ring was mounted inside a steel cylinder. This enabled a better control of the rotor/stator fit at varying temperatures. However, a significant friction is still present in this design. Besides the necessity for an increased work input to drive the rotation, the friction is heating up the fluid at the cold side of the regenerator, which subtracts from the cooling power. Both problems are obviously limiting the efficiency of the system.

Another main issue limiting efficiency, that was pointed out in the performance analysis, was heat leaks through regenerator housing and flow heads. An IR-image of the cold flow head illustrates this, see Figure 2.6 (right). While insulating external flow components may be relatively easy, the heat transfer between rotating regenerator and magnet is more complicated to address, as volume taken up by insulation will "waste" valuable magnetized volume.

As mentioned above, the prototype was originally designed for low pressure drop parallel plate regenerators, but in reality tested with packed spheres. This has greatly increased the pressure drop that had to be handled by the flow system. This has caused a problem of internal flow bypass taking place...
2.1. DESIGN

Figure 2.6: Quarter sectional view (left) and IR-image(right) of flow head at the cold end during operation

at the hot side valve interfaces. The basic mechanism of this is illustrated in Figure 2.7: Due to the pressure difference between inlet and outlet valve interfaces at the hot side, a portion of the fluid can short circuit, avoiding to pass through the high resistance of the regenerator. It has been detected by the means of flow meters mounted on both sides of the regenerators, that the flow rate of this bypassing flow was typically on the order of 50% of that of the flow going through the regenerator. In later initial experiments with parallel plate regenerators (not published), the bypass was reduced to only a few percent. The presence of such a bypass flow inevitably reduces system efficiency due to increased pumping power. Limiting the bypass flow in the present case required a tighter fit between valve rotor and stator, even though this increased the friction, i.e. a trade off had to be made.

Figure 2.7: Internal leak path between hot inlet and outlet in the sliding valve interface

Finally, a major challenge has turned out to be uneven driving torque ("cogging torque") due to the distribution of MCM in the magnetic field. As can be seen from Figure 2.2, the compartmentalized structure of the
regenerator results in a significant amount of non-magnetic space between the beds. In the present case only 37% of the cross section was occupied by MCM. When a bed is at the center of a magnetized region in the azimuthal direction, it is in a stable magnetic equilibrium. Due to symmetry, the combination of four magnetic poles and twenty-four beds causes the entire regenerator to be in stable equilibrium 24 times per revolution. In stead of having a torque which is balanced between beds going in and out of magnetic field regions simultaneously, this gives rise to a significant cogging torque and an unbalanced work input, which causes drivetrain losses hurting the system efficiency.
Chapter 3

Design and experimental tests of a rotary active magnetic regenerator prototype

Abstract:
A rotary active magnetic regenerator (AMR) prototype with efficiency and compact design as focus points has been designed and built. The main objective is to demonstrate improved efficiency for rotary devices by reducing heat leaks from the environment and parasitic mechanical work losses while optimizing the utilization of the magnetized volume. Heat transfer calculations combined with 1D AMR modeling have revealed the necessity for an insulating air gap between magnet and regenerator when designing for high efficiency. 2D finite difference AMR modeling capturing the interplay between heat transfer fluid flow and an inhomogenous time-varying magnetic field in the individual regenerator beds has been used in the design process. For one operating point a COP of 3.1 at a temperature span of 10.2 K and a cooling power of 103 W were measured. Major issues limiting the performance have been identified and improvements are outlined for future work.

3.1 Introduction

Magnetic refrigeration is a promising alternative to conventional vapor compression technology. It is based on the magnetocaloric effect in ferromagnetic materials, hereafter referred to as magnetocaloric materials (MCM). As a consequence of this effect, the temperature of an MCM will, under adiabatic conditions, change as a response to a change in an applied magnetic field, such that the temperature will increase when the field is increased and vice versa. The effect, which is most pronounced in the vicinity of the Curie temperature of the material, can be utilized in heating or cooling devices; see e.g. Smith et al. (2012). In 1982 Barclay demonstrated a cooling device based on a concept where the MCM itself was used to regenerate heat which was transported between a cold and a hot reservoir via a heat transfer fluid; see Barclay (1982). Since then this principle, known as the active magnetic regenerator (AMR) cycle, has been used in an increasing number of devices with the aim of making magnetic refrigeration near room temperature a competitive alternative to conventional vapor compression technology. For a more comprehensive description of the AMR cycle; see Engelbrecht et al. (2012). In general, magnetic refrigeration has the advantage of not using gaseous refrigerants. The absence of a compressor opens the possibility of silent operation. Furthermore, some of the losses associated with vapor compression are avoided, which may lead to a higher efficiency. Already in 1998, COPs above 6 were obtained at cooling powers exceeding 500 W, see Zimm et al. (1998). However, this device used a liquid helium cooled superconducting 5 T magnet and the power used for the magnet itself was not included in the COP calculation. Superconducting magnet AMRs are not economically viable with the present technology. Over the years, permanent magnets have been used in an increasing number of published devices; see e.g. Bjørk et al. (2010) and it seems that devices based on rotary concepts with permanent magnets have a good potential for high performance; see e.g. Yu et al. (2010).

A design with a rotary magnet structure and regenerator with reciprocating flow provided by a diplacer has produced high temperature spans, e.g. a temperature span of 29 K has been achieved by Tura and Rowe (2011). Recently, a temperature span of 33 K was demonstrated for an improved version of the same device; see Arnold et al. (2014). In other devices, a compartmentalized regenerator and a magnet system are mechanically rotated relative to each other. The flow system in these devices is of major importance. By applying valve systems to control the heat transfer fluid flow, it is possible to achieve a continuous flow circuit driven by a pump to transport heat to and from the regenerator while ensuring a reciprocating flow in the regenerator compartments; see e.g. Tusek et al. (2010), Engelbrecht et al. (2012) and Jacobs et al. (2014). By carefully balancing magnetic forces and fluid flow, a smooth and efficient operation may be obtained. However, during each
AMR cycle, the work performed on the regenerator is negative during magnetization and positive during demagnetization. In order to obtain efficient operation which is comparable to numerical AMR model predictions, the work performed by the regenerator during magnetization has to be utilized. In a multi bed rotary AMR configuration, such as the one presented in this paper and the earlier device presented by Engelbrecht et al. (2012), this is done by always having beds moving into the magnetic field, thus contributing to the driving torque, while others are moving out. However, a poorly mechanically and magnetically balanced configuration will result in large torque fluctuations and drivetrain losses. How to achieve an optimum configuration in this regard is not well understood. With the presented device, an improved drivechain is realized by minimizing the thickness of the walls separating the regenerator beds in order to obtain a more even distribution of MCM and hence a more smooth driving torque. Furthermore, an odd number of regenerator beds are combined with a two pole magnet in order to avoid magnetic equilibrium positions during rotation. Although much work has been conducted over the years to improve the efficiency of the devices, there are still significant technical challenges that need to be overcome. Recently, it was shown that care should be taken to reduce a number of parasitic losses associated with the overall COP of an AMR system; see Lozano et al. (2013). In this paper we report a compact new regenerator design, bringing these losses down. In the process, detailed numerical AMR modelling has been used as a design tool to address heat loss issues and the inhomogeneous magnetic field in regenerator compartments.

### 3.2 System design

The AMR device consisting of a regenerator, a magnet and a flow control system is shown in Figure 3.1. For the device presented here, the MCM is confined in a cylindrical regenerator which is divided into eleven compartments. The regenerator is fixed on the outside of an iron core that is a part of the magnet system. The core consists of laminated plates of iron and glass fiber reinforced epoxy (GFRE) in order to minimize losses due to eddy currents and thermal conduction in the axial direction. On the upper and lower sides of the regenerator, valve arrangements ensure reciprocating flow of the heat transfer fluid in the regenerator compartments and a continuous, unidirectional flow in the external flow circuit.

Special care has been taken to reduce losses on the cold lower side of the regenerator. The outer magnet ring, which is the only rotating part of the device, is situated on the outside of the regenerator and mounted concentrically with the iron core. The structure is rotated at an AMR frequency which can be varied between zero to approximately 4 Hz. The rotating outer part not being mechanically connected to a central shaft
enables simple connection between the flow distribution system mounted above the regenerator and the external flow circuit. On the hot top side of the device, hoses supply fluid in and out of 11 regenerator beds timed with the rotation of the magnet. The timing of the flow is ensured by poppet valves in the flow distributor. These valves are actuated by cam rings rotating along with the magnet. A detailed description of the flow system is available in Eriksen et al. (2015a). The excess heat in the hotter outgoing fluid is rejected in the hot heat exchanger (HHEX), before going back into the regenerator. On the cold side, the fluid goes through an electrical heater, simulating a heat load.

### 3.2.1 Magnet system

A cross section of the magnet, iron core and regenerator is shown in Figure 3.2a. The magnet consists of a number of blocks of permanent NdFeB magnets, magnetized in the indicated directions. In combination with flux-conducting iron yokes and an iron core, the structure forms a Halbach-like array with two high field and two low field regions in the gap where the regenerator is situated. These regions will be swept over the regenerator as the magnet rotates.

The 12 pieces of permanent NdFeB magnet, grade N50, have a total volume of 1.5 L. The flux density in the gap between iron core and magnet was measured using a three-axis Hall probe (Arepec s. r. o. AXIS-3) as a function of radius, length and azimuthal angle. There is a good agree-
3.2. SYSTEM DESIGN

The calculations reveal a potential to increase the maximum flux density from the current 1.13 T to 1.4 T. This is considered for future work.

3.2.2 Regenerator

A photograph showing the regenerator filled with spheres of Gd and three different compositions of Gd$_{100-x}$Y$_x$ from Santoku Corp. (Japan) is shown in Figure 3.3. These materials are arranged such that they will operate near their respective Curie temperatures. The diameter of the spheres is between 500 µm and 600 µm for the Gd and between 300µm and 500µm for the Gd$_{100-x}$Y$_x$ compounds. As the magnitude of the magnetic flux density decreases near the ends of the 100 mm long magnet, only the central 90 mm is utilized as AMR. The total mass of MCM is 1.7 kg. The regenerator housing consists of two concentric non-magnetic stainless steel cylinders, grade 304, with a thickness of 0.5 mm. The walls separating the 11 regenerator beds are made from 0.5 mm thick GFRE plates. This material was chosen to avoid eddy currents while ensuring sufficient mechanical stability.

After packing the regenerator with MCM spheres, it was closed at the ends with PVC/PA parts with stainless steel wire mesh screens facing the MCM and holes for flow system connections. The dimensions of the housing material inside the magnet were minimized using the commercially available finite element software Comsol Multiphysics for strength calculations, tak-
3.3 Modeling used as design tool

This section describes the use of numerical modeling as a tool for addressing both heat losses and the effects of having an inhomogeneous magnetic field in the regenerator compartments. The modeling has been used for the design and dimensioning of the presented prototype. However, the issues addressed are inherent to the considered type of rotary devises. The modeling is based on a 1D numerical AMR model solving coupled governing equations based on energy balances for the MCM and heat transfer fluid respectively. The model takes property data for these materials, given fluid flow and applied magnetic field as functions of time during the cycle and a given temperature span and cycle frequency as inputs. The main outputs are the hot and cold heat flows and the input power. The model was originally validated experimentally against an AMR prototype described by Zimm et al. (2006), see Engelbrecht (2008). The model that is used to cap-
ture spatial variations in magnetization in a regenerator bed solves the same governing equations in 2D as described below. This model was based on previous passive 2D porous bed regenerator modelling described in Nielsen et al. (2013).

### 3.3.1 Parasitic heat loss through regenerator housing

![Diagram](image)

Figure 3.4: (a) Regenerator with temperature gradient situated between iron core and rotating outer magnet. (b) Air gap between regenerator and rotating magnet at a given z. Forced convection with linear velocity profile between 0 and U and heat flux, $q''$, into the regenerator.

Special attention has been given to the heat leakage between the regenerator and the magnet/iron core as earlier work has identified this as an important issue limiting the system performance; see Lozano et al. (2013). The situation is illustrated in Figure 3.4. Between the regenerator housing and the rotating outer magnet is an air gap. The magnet, having a large thermal mass and good thermal transport properties, is considered to be at the ambient temperature, $T_\infty$, and the regenerator wall is considered to have a linear temperature gradient between $T_C$ and $T_H$ in the $z$-direction. The heat transfer through the regenerator housing due to both forced and natural convection has been considered. The rotation of the magnet will create a friction driven air flow in the gap, which is treated as a Couette flow; see e.g. Incropera and Dewitt (1996). This simple case of forced convection has an analytical solution revealing that the heat flux in the radial direction consists of only two terms: viscous dissipation in the air and conduction driven by the temperature difference between regenerator and magnet. For the rotational speeds relevant for the present application, it is found that the viscous dissipation may be neglected. Furthermore, the natural convection was modeled with a simple finite element model made in Comsol Multiphysics. Based on this it has been concluded, that for an air gap of up to a few mm, the heat transfer is still dominated by conduction. Fur-
thermore, the natural convection may be limited by physical barriers to the axial flow. Therefore, only the heat transfer by conduction is considered in the following, both for the gap between regenerator and rotating outer magnet and for the equally wide gap between regenerator and iron core. Since the present device design was focused on efficiency, estimates of the influence of how the conduction based loss influences the COP have been made, based on the geometry shown in Figure 3.4, with \( d = 100 \text{ mm} \), \( D = 132 \text{ mm} \) and \( L = 90 \text{ mm} \). It was assumed that the regenerator housing consisted of stainless steel with a thickness of 0.5 mm. For simplicity, it was assumed that the hot end of the regenerator was at the ambient temperature of 30°C and that the cold end was at 0°C. In this situation, all the heat flux is going into the regenerator from the magnet and the iron core, which was not considered to be laminated in this case. Then, the total heat flux into the inside and outside of the regenerator was integrated over its length, giving a total heat loss, \( Q_{\text{Loss}} \). The cooling power, \( Q \), produced by the device and the work input, \( W \), which consists of both magnetic work, \( W_{\text{mag}} \) and pump work, \( W_{\text{pump}} \), i.e.

\[
W = W_{\text{mag}} + W_{\text{pump}},
\]

was calculated using the 1D numerical AMR model described above, see also Engelbrecht and Bahl (2010), assuming a varying flow rate and an AMR operational frequency of 0.75 Hz. The COP was then evaluated as:

\[
COP = \frac{Q}{W}.
\]

Next, as a rough estimate, the calculated loss was simply subtracted from the cooling power when calculating the COP with losses, \( COP_L \):

\[
COP_L = \frac{Q - Q_{\text{Loss}}}{W}.
\]

Results from these calculations are shown in Figure 3.5a, where the flow rate is increased from 4.2 L min\(^{-1}\), corresponding to a utilization of \( \phi = 0.4 \), until the cooling power reaches zero. Here, the utilization is defined as

\[
\phi = \frac{m_f c_f}{m_s c_s},
\]

where \( m_f \) is the mass of fluid flowing through a bed during a blow period, \( m_s \) is the mass of MCM in a bed and \( c_f = 4200 \text{ J kg}^{-1} \text{ K}^{-1} \) and \( c_s = 300 \text{ J kg}^{-1} \text{ K}^{-1} \) are the heat capacities of the fluid and MCM respectively. In each case, the cooling powers plotted in Figure 3.5a reach their maximum values at approximately \( \phi = 0.8 \).

It is clear that, without the loss term included, the regenerators with the smaller air gaps and hence higher masses perform better in most cases, both
in terms of cooling power and COP. However, when the utilizations become low, higher COPs can be achieved with larger air gaps. This is considered to be an effect of smaller air gaps giving larger cross sectional areas, which leads to increased axial heat conduction losses. This is captured directly by the AMR model.

![Figure 3.5: (a) Modeled COP and cooling power with varying flow rate. Different thicknesses of air gap, with and without losses included in the COP calculation. (b) Highest COP from the same data set at fixed cooling powers plotted as a function of air gap.](image)

The figure furthermore shows, that when the heat losses through the regenerator walls are included, the COPs of the smaller air gaps become significantly decreased. Therefore, when designing for high COP, it will be desirable to have a certain air gap, which in this case is larger than 1 mm, even though it slightly decreases the maximum cooling power. In Figure 3.5b the COP is plotted as a function of the air gap for different cooling powers. Based on these considerations, an air gap of 2 mm was chosen in the present case. Such a tradeoff between insulating air gap and the amount of MCM in the magnet is inherent to this type of rotary AMR devices. Having a clearance between rotating and stationary components furthermore has the advantage of reducing the demands for manufacturing tolerances.

### 3.3.2 Inhomogeneous magnetic field and 2D AMR modeling

The field in a regenerator bed is inhomogeneous, meaning that the MCM experiences an applied field that varies with the azimuthal direction. This makes it non-trivial to optimize the flow rate of the heat transfer fluid during the cycle because the entire bed experiences the same fluid flow rate. To address this issue, a finite difference AMR model, based on previous work by Nielsen et al. (2013), was developed that resolves the regenerator in both the $z$ and $\phi$ directions indicated in Figure 3.2. This 2D model has been used
as a design tool to iterate between different combinations of magnet design, regenerator design and flow profile (flow rate versus rotation) design. In this process, different magnetic field profiles have been developed and investigated, using the iterative finite element based methodology described in Bjørk et al. (2010) and Bjørk et al. (2011). This has been done in combination with a variety of regenerator and flow profile designs modeled with the 2D AMR model. The investigated flow profiles are trapezoidal, meaning that they consist of full flow periods, no flow periods and linear ramps. This combined modeling approach has been used as a tool to design the present prototype and the methodology will be described in further detail in future works.

Once the inner and outer dimensions of the magnet and regenerator were fixed, the number of beds was varied in the 2D AMR model; see Figure 3.6.

![Figure 3.6: Cooling power and COP versus number of beds and resulting width of the individual regenerator compartments. The utilization is 0.4](image)

It is a clear result that having more beds is desirable, although not from a practical standpoint because many beds require smaller dimensions, more valves and more intricate assembly. The number of beds was chosen to be eleven as a compromise in the present case. This number was also chosen not to be a multiple of the number of magnetic poles to improve mechanical efficiency.

### 3.4 Interfacing

Adjustable operational parameters and instrumental readings are handled by a dedicated LabView PC program, allowing the control of operational frequency and heater power. Data were acquired using a National Instruments
3.5. EXPERIMENTAL RESULTS

AMR experiments have been conducted with the hot side temperature held constant around 18°C. A series of experiments at an AMR frequency of 0.75 Hz and a fluid flow rate of 3 L min$^{-1}$ resulted in the cooling curve shown in Figure 3.7.

![Cooling curve with corresponding COP values. The utilization was 0.3](image)

The indicated COP values are calculated according to Eqs. 1 and 2 with $\dot{Q}$ being the power supplied to the heater providing the cooling load. $\dot{W}_{\text{mag}}$ is taken as the power supplied to the motor and $\dot{W}_{\text{pump}}$ is evaluated as the measured fluid flow rate times the total pressure drop over the regenerator. Thus, the efficiency of the pump itself is the only parameter which was not included in the COP. The current pump is over dimensioned in order to be able to cover a wide range of pressure drops and flow rates for experimental purposes. Demineralized water with 5% ethylene glycol based automotive
antifreeze was used as heat transfer fluid. Measured as a fraction of the Carnot efficiency, the best result is obtained at a temperature span of 10.2 K at a cooling load of 102.8 W. In this case the COP of 3.1 is 11.3% of the Carnot Efficiency.

![Figure 3.8](image)

Figure 3.8: COP versus temperature span showing significant improvement compared to rotary prototype previously presented by DTU.

The highest COP values obtained so far at different temperatures are plotted in Figure 3.8, where a significant improvement compared to the first rotary DTU prototype is also shown, see Lozano et al. (2013). Beside the minimized parasitic losses, the driving torque fluctuations are greatly reduced, hence drive train losses are also reduced as described above. So far the device has only been operated at very modest conditions, especially in terms of flow rate and pressure drop, in order to avoid possible mechanical overloading of the regenerator.

The actual flow rates in some of the regenerator compartments have been measured as a function of magnet rotation angle using the portable flow meter.

The result of such a measurement can be seen in Figure 3.9, where also the expected flow profile is indicated together with the magnet profile. It is clear, that there is a significant discrepancy between the expected and measured flow rates, indicating that adjustments to the flow system are needed. Already, the system has shown strong responses to such adjustments. It is evident that, for the bed measured in this case, the total flow is less in the cold to hot direction than in the hot to cold direction. This is hypothesized by the authors to be due to the way different beds with somewhat different flow resistances are connected during operation. This flow balancing issue might also be an important factor limiting the achieved temperature
Figure 3.9: Normalized magnet and measured flow profiles in one of the eleven regenerator compartments. Red indicates hot to cold and blue indicates cold to hot blow.

spans, which are quite modest compared to what could be expected from the present staggered regenerator using materials with four different Curie temperatures as described in the introduction.

The potential for getting a higher magnetic field with a different iron core as mentioned above combined with adjustments to the flow system indicate a good potential for improved results in the future.

3.6 Conclusion

A rotary AMR prototype has been designed and built. The focus has been on enhanced performance in terms of efficiency and compact design at temperature spans and cooling powers relevant for commercial applications. During the design process, heat transfer calculations combined with 1D AMR modeling have revealed that there will be a trade-off between the amount of MCM and insulating air gap between magnet and regenerator in rotary prototypes when designing for high COP. Furthermore, 2D AMR modelling has proven to be an important tool for the design of AMR device configurations where the magnetic field varies as a function of rotation angle. Experimental results from the presented prototype have shown promising results, including a temperature span of 10.2 K at a cooling load of 103 W and a COP of 3.1. A potential for obtaining a higher magnetic field and the need for improving the balancing of heat transfer fluid flow have been identified and significant improvements of system performance are expected in the future.
3.7 Addendum

3.7.1 Flow system design

The inherent problems with friction losses and potential internal leakages in rotary valve systems, such as the one used in the 1st DTU prototype, can be minimized by different design efforts, see e.g. Lozano (2015). In the present prototype however, the concept of rotary valves is abandoned in favor of a novel flow system based on poppet valves. A detailed description of the flow system can be found in Eriksen et al. (2015a), see Appendix A.1.

![Figure 3.10: Array of poppet valves and rotating cam rings.](image)

The central unit in this system is the "Hot flow distributor" in Figure 3.1. It is based on 22 poppet valves arranged in two annular arrays, see Figure 3.10. The individual valves are actuated by cam rings that are rotating along with the magnet. The use of an array of poppet valves enables a very low friction loss compared to traditional rotary valves. Furthermore, internal leakage is eliminated as the valves have a positive seal when closed. A small work input is needed to open the valves due to the biasing spring force. However, this work is partially returned as the valve is closing. If the number of valves opening and closing is equal at any time, the net work becomes zero, if friction and spring irreversibilities are disregarded. The poppet valves used are of the type MJV-2-MG from Clippard Minimatic. They were chosen partially because they support a reasonably compact ma-
chine design. A number of the valves were tested in series with a flow meter and a centrifugal pump to measure the characteristics. The valves were, to a crude approximation, seen to open linearly with a travel of 0.6 mm. Obtaining a desired trapezoidal flow profile in the AMR machine was then a question about the profile of the cam rings.

The 2D model described in section 3.3.2 was used to investigate the influence on cooling power and COP of varying fractions of pauses, $\tau_p$, ramps, $\tau_r$, and full flow $\tau_f$ of the total AMR cycle. Figure 3.11 shows results from this for low and high ramp fractions of $\tau_r = 0.2$ and $\tau_r = 0.6$, respectively. It can be seen that the cooling power for the short ramp fraction is very sensitive to the pause fraction, while the COP is not. For the long ramp fraction, the cooling power is much less sensitive to the pause fraction. However, in this case the COP does become sensitive to the ramp fraction, but even in the worst cases it does not become significantly lower than what can be obtained with the short ramp fraction.

![Figure 3.11: Modeled COPs and cooling powers for varying flow profiles.](image)
The AMR frequency was 1 Hz and the temperature span was 30 K.

Based on these considerations it was chosen to design for $\tau_r = 0.6$, $\tau_p = 0.2$ and $\tau_f = 0.2$ as this should make it possible to obtain a good combination of cooling power and COP while at the same time accepting minor deviations.
from the desired flow profile. Furthermore, the design allows for a later replacement of the cam rings to test other flow profiles.

![Cam profile lift](image)

Figure 3.12: Cam profile lift (left). The circular profile is plotted to the right (cam profile up-scaled 40 times relative to circle.

The cam profile lift consists of three regions as indicated in Figure 3.12. Between 0° and 9° the valve is closed. There is a 0.5 mm extra travel to ensure full closing. Between 9° and 36° the valve travels 0.6 mm and linear opening is assumed. Between 36° and 45° the valve is given an extra 0.6 mm of travel to ensure full opening. The profile of the cam is symmetric around 45°.

Initially, to test the valve controllability with the actual cam rings in the machine, one of the two valve discs were installed together with a cam ring. Each inlet hose was connected to a manifold in series with a centrifugal pump and a needle valve. Then, for varying rotational angles, the flow rate through one poppet valve at a time was measured. The initial results revealed significant discrepancies between flow profiles generated by each of the two cams. Furthermore the profiles deviated from the expected profile. To investigate this problem, a dial indicator was installed and the actual profiles of the cam rings as well as the supporting aluminum cylinder in which they were mounted, were measured. As a result, it was found that the supporting cylinder was out of round, exceeding the expected tolerances. Attempts to correct this by conventional machining failed, seemingly due to residual stresses in the material. Finally, the cylinder geometry was corrected manually by the means of sanding and application of shim tape, see Figure 3.13. As a result, the rings could now be mounted in a cylinder round and centered within a few hundredths of a mm. Finally the cam profiles were measured again, see Figure 3.13.
3.7. ADDENDUM

Figure 3.13: Measurements of cam rings and supporting cylinder before and after manual corrections. Lift up-scaled 40 times relative to circles.

Figure 3.14: Flow characteristic (left) and flow profiles for a valve actuated by two cams on the same ring.
3.7.2 Measured in situ flow profile
From Figure 3.9 two things are evident about the measured flow profile:

1. The ramp is much shorter than expected

2. The flow profile has "spikes" during the open period

In the early phase of the design process, poppet valves were tested individually with a centrifugal pump to measure flow rate versus travel. The result of such a test can be seen in Figure 3.14 (left). It can be seen, that there is an almost linear regime where most of the opening takes place. As a rough approximation, it is simply assumed that the valve opens linearly over a travel of 0.6 mm. After the corrections of the cam ring cylinder described above, it was possible to obtain measured flow profiles as indicated in Figure 3.14 (right). It is clear, that there is a good agreement between the two flow profiles related to the two cams on the ring. Furthermore, the shape of the flow profile is considered satisfactory, with the described linear approximation of the measured valve characteristic in mind. This result shows, that it was indeed possible to obtain an in situ flow profile which is characterized by long ramps as desired. However, when tested in the finished machine with the regenerators installed, the ramps were almost absent, as mentioned above. This may be explained by the fact that the poppet valve is connected in series with a regenerator bed with a high flow resistance, relative to that of the valve itself. In practice this means, that as soon as the valve just begins to open, the flow resistance of the series connection of the valve and regenerator bed is dominated by the resistance of the bed. As a consequence, the effective ramp becomes very short. Further considerations regarding the flow profile and explanations of the "spikes" mentioned above can be found in Chapter 4.

3.7.3 Magnetocaloric materials
Choice of MCM
As described in Chapter 1, the magnetocaloric effect may be high in first order materials and good performance in AMR devices utilizing such materials has been reported in literature (Jacobs et al. (2014)). However, using first order materials requires layered regenerator beds with small spacing between Curie temperatures of the layers. Furthermore, hysteresis issues can be difficult to deal with and may significantly hurt efficiency. For the current regenerator it was therefore chosen to use gadolinium based materials, which may as mentioned be considered a benchmark material within magnetic refrigeration near room temperature. However, as it was considered important to have a well functioning regenerator at higher temperature spans, some layering was desired. It is possible to lower the Curie temperature of
gadolinium by alloying with different elements, see e.g. Wollan (1967) and Kaji et al. (2010). Here, gadolinium alloys with terbium, erbium, holmium (Tishin and Spichkin (2003a)) and yttrium (Kito et al. (2007)) were considered, see Figure 3.15. Based on raw material prices\(^1\) and linear fits of the data plotted in Figure 3.15, mass specific costs of gadolinium alloys with a Curie temperature of \(T_C = 273\) K were estimated, see Table 3.1.

![Figure 3.15: Curie temperature vs. rare earth element fraction.](image)

**Table 3.1: Raw materials cost**

<table>
<thead>
<tr>
<th>Element, E</th>
<th>Gd</th>
<th>Tb</th>
<th>Er</th>
<th>Y</th>
<th>Ho</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost [USD/kg]</td>
<td>210</td>
<td>2500</td>
<td>275</td>
<td>120</td>
<td>8600</td>
</tr>
<tr>
<td>Cost, (Gd_{1-x}E_x(T_C = 273K)[USD/kg])</td>
<td>1075</td>
<td>221</td>
<td>210</td>
<td>1608</td>
<td></td>
</tr>
</tbody>
</table>

From this, yttrium and erbium were the obvious candidates. Judging from Tishin and Spichkin (2003b) and Kito et al. (2007), the magnetic entropy change is roughly the same in the two cases, applied fields and Curie temperatures being equal. The final choice of \(Gd_{1-x}Y_x\) alloys were based on availability of spheres in the relevant range of diameters rather than an optimization of number of layers and spacing between \(T_{CS}\). Together with pure gadolinium from the first DTU prototype, the alloys used were found to cover a reasonable range of \(T_{CS}\) for a refrigerator application.

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\(^1\)From www.mineralprices.com/default.aspx#Rare and www.chemicool.com (January 2013)
Characterization of MCM

As described in Chapter 1, one of the main characteristics of a magnetocaloric material is its magnetic entropy change, $\Delta s_m$, when a magnetic field is applied at a certain temperature. This may be derived by a Maxwell relation (Equation 1.4) from magnetization data. However, this indirect method may give rise to uncertainties due to numerical differentiation. For this reason, an instrument for direct measurement of $\Delta s_m$ via a technique based on calorimetry was developed at DTU Energy (Nielsen et al. (2015)). In this instrument, two Peltier elements are placed at the top of a cold finger with a controlled temperature in vacuum. On one of the elements an MCM sample can be mounted. As a magnetic field change is imposed on the sample while its temperature is kept constant, a heat flux through the Peltier element can be evaluated from the difference in voltage signal between the two elements. When this signal is integrated over the time it takes to go from one field to another, the corresponding magnetic entropy change may be evaluated. Details are given in Nielsen et al. (2015). The main contribution to the development of the instrument by the author was the mechanical setup shown in Figure 3.16. The purpose of this arrangement is to provide a magnetic field with a time varying flux density. An important requirement was that the field direction had to be fixed in space in order to have a constant demagnetization factor of the sample. Typically, the sample will have an orthorhombic shape in which case the demagnetization factor may be evaluated as described by Aharoni (1998). By the means of two equally sized bevel gears driven by the same pinion (see Figure 3.16) the inner and outer magnets can be rotated in opposite directions with angular velocities of equal magnitude, which ensures the constant field direction.

This instrument was used for characterizing the materials used in the presented AMR prototype. In order to do this, a pellet of each of the four materials was made by spark plasma sintering (SPS) using an in-house SPS unit (Dr Sinter 515s from Syntex Inc., Japan). During this process, the powder was pressed together while subjected to a pulsating high current density causing a sintering of the powder to a dense pellet\(^2\). To obtain samples of approximately $2.5 \times 2.5 \times 1$ mm\(^3\), the pellets were cut on a diamond saw\(^3\). The resulting samples were then used in the $\Delta s_m$-instrument to obtain the results plotted in Figure 3.17. Each of the six plots shows the magnitude of the magnetic entropy change corresponding to a field change between 0 T and the value indicated in the legend. As expected the curves peak near the Curie temperatures indicated in Figure 3.3. It is also evident, that there is a minor decrease in the maximum absolute entropy change in the $Gd_{1-x}Y_x$ alloys compared to the pure Gd.

\(^2\)This sample preparation was done with the assistance of K. K. Nielsen.
\(^3\)This was done by P. H. Nielsen.
3.7.4 Regenerator housing

In order to minimize the heat conduction in the regenerator housing, it is desirable to use a wall which is thin and has a low thermal conductivity. It should, however, be strong enough to withstand the mechanical stresses due to both pressure and temperature variations. In Table 3.2 different non-magnetic candidates for housing materials are listed. \( \sigma_f \) is the stress at which the material will fail due to either yield of fracture. In this case, the maximum allowable stress is set to \( \sigma_{max} = 0.75 \sigma_f \). To get an estimate of the wall thickness needed, the situation is considered equivalent to that of a cylindrical thin walled pressure vessel, in which the hoop stress can be calculated (Beer and Russel Johnston (1992)) as

\[
\sigma_{\text{hoop}} = \frac{P \cdot d}{2 \cdot h},
\]

(3.5)

where \( P \) is the pressure, \( d \) is the diameter of the vessel and \( h \) is the wall thickness. As approximate values resembling the situation for the regenerator outer cylinder, \( P = 10 \) bar, and \( d = 120 \) mm is used. The corresponding wall thicknesses are indicated in Table 3.2. Finally, considering the different thermal conductivities, the axial heat conduction in the wall is calculated, assuming a temperature gradient of \( \Delta T = 35 \) K. It is clear, that even though a small wall thickness can be used with aluminum, there will be a quite significant axial conduction. It is clear, that this conduction loss can be very small with the plastic materials, but this requires quite thick walls. This
Figure 3.17: Entropy changes of the used regenerator materials measured with the instrument described in Nielsen et al. (2015).

will not be feasible due to the radial heat leak considerations discussed in Section 3.3.1, as the space in the radial direction that is needed for insulation should rather be air, which gives a better insulation. The GFRE is an interesting option. However, it was found that a stainless steel cylinder was easier to manufacture in the present case, and a final choice of this material with a thickness of 0.5 mm was chosen. However, GFRE was used for the
walls separating the regenerator compartments as described in Section 3.2.2. This analysis did not include the way the regenerator wall takes part in the regeneration process and reduces the regenerator effectiveness. It can be recommended to use numerical modeling for doing so in future designs, see Nielsen et al. (2013).

<table>
<thead>
<tr>
<th></th>
<th>(\sigma_f)</th>
<th>(\sigma_{\text{max}})</th>
<th>(t)</th>
<th>(\kappa)</th>
<th>(Q_{\text{axial}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stainless steel 304</td>
<td>205</td>
<td>154</td>
<td>0.39</td>
<td>16</td>
<td>1.64</td>
</tr>
<tr>
<td>Aluminum 201-T6</td>
<td>400</td>
<td>300</td>
<td>0.20</td>
<td>205</td>
<td>10.8</td>
</tr>
<tr>
<td>Titanium</td>
<td>940</td>
<td>705</td>
<td>0.09</td>
<td>22</td>
<td>0.49</td>
</tr>
<tr>
<td>PA (Nylon)</td>
<td>45</td>
<td>33.8</td>
<td>1.78</td>
<td>0.28</td>
<td>0.13</td>
</tr>
<tr>
<td>PVC</td>
<td>58</td>
<td>43.5</td>
<td>1.38</td>
<td>0.16</td>
<td>0.06</td>
</tr>
<tr>
<td>PEEK</td>
<td>115</td>
<td>86.3</td>
<td>0.70</td>
<td>0.25</td>
<td>0.05</td>
</tr>
<tr>
<td>Glass fiber (GFRE)</td>
<td>390</td>
<td>293</td>
<td>0.21</td>
<td>25</td>
<td>1.35</td>
</tr>
</tbody>
</table>

Table 3.2: Comparison of material candidates for regenerator housing.

With the chosen housing geometry and materials, simulations of mechanical stress and stain were done using Comsol Multiphysics, see Figure 3.18, assuming an imposed temperature gradient of \(\Delta T = 35\) K and an internal pressure of \(P = 10\) bar, which is considered an extreme case. In this case, the maximum displacement of the wall is seen to be an order of magnitude less than the sphere diameter, which is considered acceptable for ensuring a closely packed sphere bed.
Figure 3.18: Simulations of stress and strain in the regenerator housing. Applied pressure $P = 10$ bar and temperature difference of 40 K.
Chapter 4

Experimental studies with an active magnetic regenerating refrigerator

Abstract:
Experimental results for an active magnetic regenerator (AMR) are presented. The focus is on whether or not it pays off to partly substitute soft magnetic material with non-magnetic insulation in a flux-conducting core in the magnet system. Such a substitution reduces losses due to heat conduction and eddy currents, but also reduces the magnetic field. Two different cores were tested in the AMR system with different cooling loads and it is shown, that in the present case, replacing half of the iron with insulation lead to an average reduction in temperature span of 14%, but also a small decrease in COP, hence the substitution did not pay off. Furthermore, it is shown experimentally, that small imbalances in the heat transfer fluid flow greatly influence the system performance. A reduction of these imbalances through valve adjustments resulted in an increase in the temperature span from approximately 16 K to 27.3 K.

4.1 Introduction

Magnetic refrigeration is a solid state refrigeration technology that is a potential replacement for vapour compression in near-room-temperature applications. Cooling is based on the magnetocaloric effect in the solid refrigerant. The magnetocaloric effect causes the material to heat up when magnetised and cool down when demagnetised when ferromagnetic refrigerants are used. The effect is reversible in many materials and therefore may be used to construct an efficient cooling cycle. A water-based heat transfer fluid is used to transfer heat from the solid refrigerant to the external heat exchangers used to absorb a cooling load and to reject heat to the ambient. Because the fluid is single phase, superheat and throttling losses are avoided. Magnetocaloric materials (MCMs) are an active research topic and several families of materials that are well-suited to air conditioning and refrigeration applications have been identified, see e.g. Pecharsky and Gschneidner Jr. (2006). Because the temperature change of known materials when magnetised in a strong permanent magnet is much lower than the operating temperature span of a practical cooling device, MCMs are implemented in a regenerative cycle such as the active magnetic regenerator (AMR) cycle. In the AMR cycle, a porous MCM regenerator with an entrained heat transfer fluid is put in thermal contact with hot and cold thermal reservoirs. The cycle consists of four processes: first the MCM is magnetised causing its temperature to increase. Then the heat transfer fluid is pumped from the cold reservoir to the hot reservoir and heat is rejected to the ambient while the MCM is cooled. The MCM is then demagnetised and its temperature decreases. Finally, fluid is pumped from the hot reservoir to the cold where a cooling load is absorbed. The MCM is warmed by the fluid flow and the system returns to its original temperature.

Experimental AMRs have been demonstrated in a range of configurations and applications Kitanovski et al. (2015). Some of the first devices used superconducting electromagnets Zimm et al. (1998) but more recent devices have used more practical permanent magnets. Recent work on AMR devices has focused on making more compact devices by increasing the operating frequency of the device Tura and Rowe (2011) and on increasing the efficiency Lozano et al. (2013). An AMR developed for electronics cooling produced a cooling power of 2.5 kW at an 11 K temperature span Jacobs et al. (2014). Our group presented a rotating AMR using Gd as a refrigerant that produced 100 W at a span of 20.5 K Engelbrecht et al. (2012). Analysis performed on this device Lozano et al. (2013) showed that the system suffered reduced performance from heat leaks to the cold reservoir and to the regenerator, drivetrain losses and friction losses in the valve system. This paper presents a newly developed device designed to reduce losses compared to the previously reported device in order to improve efficiency. The effects of heat leaks through parts of the magnet system and eddy current losses are
investigated experimentally by building a composite magnet core with lower thermal and electrical conductivity and comparing results to those with a solid iron magnet core. Finally, the effect of variations in the fluid flow in individual regenerator beds is studied and the effect of adjusting the flow in each bed is quantified at a specific operating condition. The temperature span, cooling power, and COP of the new device are given for a range of operating conditions.

4.2 Description of the AMR device

Figure 4.1: Photo of the AMR device assembled. Tubes at the top of the regenerator are to and from the pump and the gear drive is shown.
CHAPTER 4. EXPERIMENTAL STUDIES...

The AMR device presented here is designed for better regenerator performance, improved thermal isolation, reduced friction and better drivetrain efficiency. The goal is to demonstrate a COP that is significantly higher than our group’s previously presented device while still producing a practical temperature span and cooling power. The AMR consists of 11 independent regenerator beds that are stationary and arranged in an annulus. The regenerator housing is a thin sheet of stainless steel and an air gap at the inner and outer diameters acts as thermal insulation. The magnetic field is provided by 1.5 L of permanent NdFeB magnet in a rotating Halbach-like assembly mounted on the outer diameter of the regenerator and a stationary soft magnetic core mounted on the inner diameter of the regenerator. The magnet assembly consists of two high field regions and two low field regions, meaning that each rotation of the magnet gives two full AMR cycles in each regenerator bed. A summary of the materials used to construct the regenerator is given in Table 4.1. A photo of the assembled device is shown in Fig. 4.1. The magnet is rotated by an electric motor with a gear drive and fluid flow is provided by a single pump operating continuously. The timing and fluid flow direction is controlled by the fluid flow system, which is explained in detail in Section 4.4.

Table 4.1: Summary of regenerator materials

<table>
<thead>
<tr>
<th>Layer</th>
<th>Material</th>
<th>Mass (kg)</th>
<th>Curie Temperature (K)</th>
<th>Sphere diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Gd</td>
<td>0.71</td>
<td>291</td>
<td>0.5-0.6</td>
</tr>
<tr>
<td>2</td>
<td>Gd$<em>{97.5}$Y$</em>{2.5}$</td>
<td>0.20</td>
<td>287</td>
<td>0.3-0.5</td>
</tr>
<tr>
<td>2</td>
<td>Gd$<em>{95}$Y$</em>{5}$</td>
<td>0.39</td>
<td>283</td>
<td>0.3-0.5</td>
</tr>
<tr>
<td>2</td>
<td>Gd$<em>{90}$Y$</em>{10}$</td>
<td>0.41</td>
<td>272</td>
<td>0.3-0.5</td>
</tr>
<tr>
<td></td>
<td>Total</td>
<td>1.70</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

4.3 Magnet core

The magnet core is used to guide magnetic flux from the permanent magnets to reduce the total reluctance of the magnetic circuit and to give the desired magnetic field profile in the gap where the regenerator is mounted. However, the core will also act as a thermal short circuit between the hot and cold reservoirs, causing energy that could be used for cooling to be conducted instead to the hot reservoir. Finally, the fluctuating magnetic field in the magnet core will cause some eddy current heating inside the core, which will increase motor work and decrease COP. Eddy currents scale with the squared size of the material perpendicular to the magnetic field, the fluctuations in the magnetic field and with the electrical conductivity of the material. The effects of eddy current heating, axial thermal conductivity and permeability on AMR performance are studied here by testing the device using two different magnet cores. The composite core was constructed
of alternating 2 mm thick disks of iron and fiberglass and bonding them with epoxy. The composite core has lower electrical and thermal conductivities in the axial direction and a lower average magnetic permeability. It will reduce eddy currents and thermal losses through the magnet core but will also reduce the strength of the magnetic field, which will reduce the magnetocaloric effect in the regenerator. The iron core was constructed of 2 mm thick iron disks bonded with epoxy. The iron core will have a higher magnetic field while generating more eddy currents and having increased heat losses due to thermal conduction.

4.3.1 Magnetic field profiles for both magnet cores

The measured magnetic field in the middle of the gap where the regenerator is mounted is shown in Fig. 4.2. The figure shows that the maximum field is approximately 18% higher for the iron core and the low field region is closer to 0 T. The magnetocaloric effect in the regenerator will be higher when using the iron core, which will result in better heat transfer between the heat transfer fluid and the MCM. Loss mechanisms such as eddy current heating and axial heat losses will be lower for the composite core.

Figure 4.2: Measured magnetic field with the composite and iron core installed in the magnet assembly. Measurement uncertainties are less than 0.5%.
4.3.2 AMR experiments with the two cores

A number of AMR experiments have been carried out with each of the two cores mounted in the system. Figure 4.3 shows the applied cooling power and achieved COP for a series of such experiments. In all cases the system was operated at an AMR frequency of 0.7 Hz and a heat transfer fluid flow rate of approximately 3.2 L/min. The COP was evaluated as

\[
COP = \frac{\dot{Q}_L}{W_{motor} + \Delta p_{reg} \cdot V}
\]  

(4.1)

where \(\dot{Q}_L\) = applied cooling load [W], \(W_{motor}\) = electrical power supplied to motor [W], \(\Delta p_{reg}\) = pressure drop over regenerator [Pa], \(V\) = volumetric flow rate [m\(^3\) s\(^{-1}\)]. Measurement uncertainties for these parameters are given in Table 6.1.

Thus, the efficiency of the pump itself was not included in the calculation since this component was highly over dimensioned in order to be able to cover a broad range of experimental conditions. In the considered series of experiments, three different cooling loads were applied when the composite core was mounted. After replacement with the iron core, the experiments were repeated with the same cooling loads. It is clear from Figure 4.3, that for all three cooling loads the achieved temperature span was higher with the iron core than with the composite core with an average increase of 14%. However, it can also be seen that there is an almost linear relationship between temperature span and COP for both cores. Furthermore, the COP at a given temperature span is consistently higher with the iron core, but the difference is small, which is also evident by the fact that the average 2nd law efficiency only increased by 3.7% by going from the composite to the iron core.

The adiabatic temperature change, \(\Delta T_{ad}\), which is the temperature increase that takes place in a magnetocaloric material when a certain magnetic field is applied to it under adiabatic conditions, is often used to characterize the material. In general, \(\Delta T_{ad}\) increases with increasing field, which also increases heat transfer between the MCM and the heat transfer fluid. In the present case this explains why the temperature span increased when the composite core was replaced with the iron core leading to a higher maximum field. The fact that the COP at a given temperature span, and hence the 2nd law efficiency, increased only slightly, may be attributed to the losses due to increased axial conduction in the core. In order to get an indication of the influence of the eddy current losses, an experiment was performed by rotating the magnet without the regenerator mounted. A thermocouple was mounted in good thermal contact with the core and the motor power was recorded as a function of rotational speed. This was done for both of the cores and resulted in no significant temperature increase or change in...
4.4. FLOW CONTROL

Figure 4.3: Steady state AMR results with the two different cores.

Motor power as would have been expected if significant eddy current losses were present. All in all the results indicate, that in the present case it does not pay off to trade maximum magnetic field for insulating material in the core. It should however be noted, that the investigated system includes an insulating air gap between core and regenerator housing, see Eriksen et al. (2015c) which breaks the heat conduction path, and without which the axial conduction in the core would have been expected to play a more significant role. Finally it should be noted, that the described AMR experiments have revealed significant sources of uncertainty in the AMR results originating from the fluid flow system, which has motivated investigations described below.

<table>
<thead>
<tr>
<th>Measurement uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature span</td>
</tr>
<tr>
<td>0.2 K</td>
</tr>
</tbody>
</table>

4.4 Flow control

The exact adjustment of the flow control system has proven to play a key role to the system performance.

This section presents experiments showing the effect of such adjustments. Each regenerator bed is connected to two poppet valves controlling the flow
in and out of the hot side, respectively. The valves are actuated by a dedicated cam ring for each of the two flow directions. Each ring has two cams corresponding to two AMR cycles per revolution of the two pole magnet with which they rotate. At the cold side of the regenerator, a compact manifold with check valves ensures a unidirectional flow through an electrical heater where a heat load can be applied. For a detailed description of the flow system, see Eriksen et al. (2015a).

4.4.1 Flow balancing experiments

A portable flow meter (Omega Microflow TFB322D) was connected in series with the poppet valve supplying fluid to an arbitrarily chosen bed. Without rotating the magnet/cam rings, the pump was turned on and the flow rate of the fluid entering the chosen bed was measured. Next, the magnet/cam rings were incrementally rotated one degree at a time, and for each position the flow rate was measured. This was repeated until the flow profiles corresponding to the two cams on the ring controlling the hot to cold blows ("In, cam1" and "In, cam 2" in Figure 4.5a) were mapped out. The procedure was repeated with the flow meter connected in series with the poppet valve controlling the flow out of the hot side of the bed and the flow profiles of the cold to hot blows ("Out, cam1" and "Out, cam 2" in Figure 4.5a) were
4.4. FLOW CONTROL

measured.

Figure 4.5: Flow profiles normalized with model predictions measured for both cams on each of the two cam rings controlling flow in and out of a regenerator bed before adjustment (a) and after adjustment (b).

In Figure 4.5 the flow profiles are plotted together with a theoretical flow profile based on the assumption of a constant volumetric flow rate in the entire system, which is divided between the individual beds depending on the degree of opening of all the poppet valves. The figure represents stationary operation of the flow system and the actual flow profile may be slightly different under dynamic operating conditions. The assumption of a constant overall flow rate is based on the use of a gear pump and verified by a flow meter (Omega FMG71) in the main flow circuit. It can be seen, that there is a good agreement between the shape of the measured and the theoretical flow profiles. However, there is a discrepancy between the cold to hot blow and the hot to cold blow of 14.5 % on average, indicating a higher flow resistance in the hot to cold direction. To compensate for this, a simple ball valve in series with the poppet valve controlling the flow in the cold to hot direction in the considered bed was partially closed to even out this difference, resulting in the flow profiles in figure 4.5b. To investigate the effect of this adjustment, an AMR experiment was conducted with a frequency of 0.63 Hz and a flow rate of 2.48 L min$^{-1}$. After reaching steady state at a temperature span of 19.8 K, the adjustment was eliminated by fully reopening the ball valve. This resulted in a rather dramatic decrease in the temperature span which dropped approximately 3.4 K down to 16.4 K, see Figure 4.6.

4.4.2 Future work

The fact that the described adjustment of the flow rate in only one arbitrarily chosen bed had such a significant impact on the system performance
motivates a more thorough investigation of this effect. From an earlier characterization of the eleven beds conducted with a single blow test setup, it is known that some variation in flow resistance from bed to bed is present. For a flow rate of 2.22 L min$^{-1}$ the standard deviation in the flow resistance of the 11 beds was 5.9% of the mean value of 56.7 kPa min L$^{-1}$. A hypothesis of the authors is, that this variation in bed flow resistances is responsible for the direction dependent flow rates observed in the experiment described above due to the way the beds are connected to each other during operation, and that this effect may be minimized by ensuring a more uniform flow resistance of all the beds through valve adjustments. This will be investigated in future work, but at the time of writing such an adjustment has increased the temperature span from approximately 16 K to 27.3 K at a frequency of 1 Hz and a flow rate of 3 L min$^{-1}$. The effect of the adjustments on other system performance metrics such as cooling power and COP will also be quantified.

4.5 Conclusions

An experimental study investigating the effect of replacing soft magnetic material with thermal and electrical insulation in the flux conducting core of the magnet assembly in an AMR device has been carried out. Two
different cores were used, both made by stacking 2 mm layers of iron glued together with electrically insulating epoxy. One of the cores was made as a composite where every second iron layer was replaced with glass fibre acting as insulation. The scope was to investigate the effect of losses due to axial conduction and eddy currents in the core and evaluate the influence of reducing these losses at the expense of a reduction in the magnetic field in the regenerator. The study has shown a significant decrease of the temperature span for fixed cooling loads with the composite core which is attributed to the reduced magnetic field. Even though the difference in COP was small, the iron core also proved to be superior in terms of efficiency. The effect of the eddy current losses has proven to be small and the effect of axial heat conduction in the core is limited by the insulating air gap between core and regenerator. It is concluded, that in the present case, optimizing the magnetic field with the iron core is more important than limiting its thermal and eddy current losses by laminating it with insulating material. Furthermore it is shown through experiments, that small variations in the flow resistances of the beds drastically limit the performance of the system. A first attempt of balancing out these resistances has led to an increase in temperature span from approximately 16 K to 27.3 K for a given set of operating conditions.

4.6 Addendum

4.6.1 Eddy current simulations

To get further insight into the eddy currents that may arise in the iron core as the magnet rotates, a finite element model was set up using the Comsol Multiphysics software. The modeled geometry is evident from Figure 4.7: An iron disc with a diameter similar to that of the actual iron core is situated concentrically inside a magnet assembly with a cross section corresponding to that of the prototype magnet. Corresponding remanent magnetic flux densities are assigned to each of the magnet elements. In the model, the magnet is fixed in space and the iron disc is rotating at a constant angular velocity. In this way, the resulting magnetic and electrical fields will be fixed in space. Therefore, a stationary solution to the equations of current conservation and Amperes law may be found. The equations of the model are:
\( \nabla \cdot \mathbf{J} = 0 \) \hspace{1cm} (4.2)

\( \nabla \times \mathbf{H} = \mathbf{J} \) \hspace{1cm} (4.3)

\( \mathbf{B} = \nabla \times \mathbf{A} \) \hspace{1cm} (4.4)

\( \mathbf{E} = -\nabla V \) \hspace{1cm} (4.5)

\( \mathbf{J} = \sigma \mathbf{E} + \sigma \mathbf{v} \times \mathbf{B}, \) \hspace{1cm} (4.6)

where \( \mathbf{J} \) is the current density, \( \mathbf{H} \) is the magnetic field intensity, \( \mathbf{B} \) is the magnetic flux density, \( \mathbf{E} \) is the electric field intensity, \( V \) is the electric scalar potential, \( \sigma \) is the electrical conductivity and \( \mathbf{v} \) is velocity. In the modeled case, the lorenz term, \( \sigma \mathbf{v} \times \mathbf{B}, \) which is only included in the rotating disc, gives rise to a force providing a torque in the direction of the rotational axis. Hence, a work input is required to drive the rotation. Furthermore, the induced current density causes the disc to heat up due to Joule heating. Energy conservation requires the work input and heat generation to be equal.

Figure 4.7: Simulation of magnetic field (left) and electric potential and current density (right) in a disc with a height of 50 mm rotating at what corresponds to an AMR frequency of 0.25 Hz.

Figure 4.8 (left) shows the results of a variation of the disc height. It is clear that there is a good agreement between the work input and generated Joule heating as expected. A 2nd order polynomial provides a good fit as
a phenomenological model. However, it has not been possible to obtain converged solutions for disc heights less than 15 mm. Furthermore, the fit does not quite go to zero power when the height goes to zero. As can be seen from Figure 4.7 (right) the electrical field gradient gets high at the edges of the disc close to the center of the high magnetic field regions and the largest current densities are found near the top and bottom of the disc. As the disc height gets smaller this ”skin effect” becomes even more dominating and the model fails to capture this, even with very refined meshes. However, the obtained results are used to calculate a total required work input if the iron core was stacked of discs, electrically insulated from each other, with the different simulated heights (”Core total” in Figure 4.8 (left)). The model predicts a maximum total dissipation of 7.4 W at $f_{\text{AMR}} = 0.5$ Hz. For a

![Figure 4.8: Simulation results showing total power dissipation in the core due to eddy currents vs. disc thickness (left) and frequency for a fixed disc thickness of 50 mm (right).](image)

disc height of 50 mm, the frequency was varied, see Figure 4.8 (right). In this case a good fit passing closely by the origin is provided by a 2$^{\text{nd}}$ order polynomial.

From these calculations it is concluded, that for the disk height of 2 mm used in the prototype, the eddy current losses are indeed very minor for the relevant operational frequencies, which supports the conclusions of the described experiments.

### 4.6.2 Flow profiles

To address the issue of the spikes in the flow profiles discussed in Chapter 3, a simple Matlab model was set up to calculate the flow in each bed during operation. As the flow is driven by a gear pump providing a quite constant displacement, it is assumed that there is a constant total flow rate in the circuit. For every rotational angle, the relative position of each bed and the
magnet/cam ring system is calculated. By assuming linear opening of the valves, corresponding to the ramps dictated by the trapezoidal cam profiles, the corresponding degree of opening of each valve is then calculated. For a given angle, the total flow in one direction, say hot-to-cold, is then considered to be distributed between the beds that are open in that direction according to the degree of opening of the respective valves.

As discussed in Section 3.7.2, the actual ramps of the flow profiles were shorter than expected due to the high flow resistance of the beds. From the measured flow rate in a bed shown in Figure 3.9, a trapezoidal flow profile with a full flow of 0.68, a ramp of 0.15 and a pause of 0.17, as fractions of the AMR cycle, is estimated. With these values, the corresponding modeled flow profile is shown in Figure 4.9. A direct comparison is difficult as the experimental flow profile was measured with a $5^\circ$, resolution. However, the presence of spikes is explained by the model.

![Figure 4.9: Modeled flow profile corresponding to initial experiments presented in Chapter 3](image)

**Cam ring replacement**

It was not possible to obtain long ramps in the flow profiles, even though the ramps on the cams were relatively long, as discussed above. Furthermore, these long ramps on the cams made it difficult to control the exact positions at which the valves would actually open and close. For the initial experiments presented in Chapter 3, the machine was calibrated by means of the radial adjustment mechanisms of the poppet valves shown in Figure 3.10. This was done by first setting the radial positions of the valves actuated by the lower cam ring controlling the flow in the cold-to-hot direction. Then
4.6. ADDENDUM

Figure 4.10: Simulated cooling power and COP at varying flow fractions for different utilizations.

an AMR experiment with fixed flow rate and frequency was conducted and the poppet valves actuated by the upper cam ring (hot-to-cold direction) were then adjusted on the fly, to maximize the resulting temperature span. However, it was decided to replace the original cam rings with ones having a very short ramp, corresponding to "on-off" operation in order to obtain a better control of the actual opening/closing positions.

Before the choice was made, a series of 2D simulations with varying flow profiles were done, using the same model as in Chapter 3. In these simulations, the temperature span was set to $\Delta T = 30$ K and the frequency was $f_{AMR} = 1$ Hz. A short ramp fraction of 0.05 was used. The results are shown in Figure 4.10. For modest utilizations, which gives the highest cooling powers and COPs at this relatively high temperature span, it is clear that the COP increases while the cooling power decreases with increasing flow fractions above 0.5. For a utilization of 0.2, high COPs are predicted. However, the cooling powers are relatively low in this case. As the model does not take parasitic losses into account that will in reality subtract from the obtainable cooling power, it was decided to try a cam profile with a flow fraction of 0.5. New cam rings with this profile were manufactured and installed. The experiments presented in the present and the remaining chapters of the thesis were conducted with these rings.

Changing the flow profiles affects the number and shape of the spikes during the flow period as discussed above. In the case of a ramp of 0.05, the modeled flow profiles for different flow fractions are shown in Figure 4.11. In the case of the chosen flow fraction of 0.5, the profile has two relatively high spikes, which is to some extend in agreement with the measurement results shown in Figure 4.5. In this case the second spike was, however, less pronounced. This was later found to be due to small misalignments of the poppet valves and more recent flow profile measurements of other beds have
Figure 4.11: Modeled flow profiles for varying flow fractions.

shown better agreement with the model predictions. Finally, it should be noted that it is possible to minimize the "jaggedness" of the flow profile by altering the flow fraction. In the present case, a value of 0.7 would give a quite smooth profile, see Figure 4.11.
Chapter 5

Effects on flow balancing on active magnetic regenerator performance

Abstract:
Experiments with a recently constructed rotary multi-bed active magnetic regenerator (AMR) prototype have revealed strong impacts on the temperature span from variations in the resistances of the flow channels carrying heat transfer fluid in and out of the regenerator beds. In this paper we show through numerical modeling how unbalanced flow in the beds decreases the cooling power and COP for a dual bed device. Furthermore, it is shown how resistance variations in multi-bed devices give rise to unbalanced flow in the individual beds and how this decreases cooling powers and COPs of the machines by approximately 30% and 50%, respectively.

CHAPTER 5. EFFECTS OF FLOW BALANCING...

5.1 Introduction

As a promising alternative to conventional vapor compression refrigeration with a potential for high efficiency, magnetic cooling based on active magnetic regenerators (AMRs) is moving closer to a possible commercialization. This is reflected in an increasing number of reported prototypes with improving performance over the recent years, see e.g. Kitanovski et al. (2015).

An AMR consists of a porous matrix of magnetocaloric material (MCM) undergoing an AMR cycle, which consists of subsequent magnetization and demagnetization. Combined with a reciprocating flow of a heat transfer fluid through the matrix, excess heat generated due to the magnetocaloric effect during magnetization can be rejected to a hot reservoir and a cooling load can be accepted from a cold reservoir when the matrix is demagnetized. Reciprocating devices in which a permanent magnet system and a regenerator are moved linearly with respect to each other for magnetization/demagnetization support operating conditions with a high controllability and are well suited for testing different MCMs and regenerator geometries, see e.g. Tusek et al. (2013a) and Tusek et al. (2014). Reciprocating prototypes in which a regenerator and a superconducting magnetic field source are used have been presented by various groups, see e.g. Zimm et al. (1998) and Rowe and Barclay (2002). However, from a commercial standpoint, superconducting magnets are not (yet) viable due to price and energy consumption. Furthermore, the reciprocating motion is difficult to handle at higher operating frequencies. For these reasons the development has been towards permanent magnet designs and rotary systems. With such a system, a temperature span of 33 K has been demonstrated Arnold et al. (2014) with a machine using two regenerators, each with their own rotating magnet system. A promising concept for commercial machines is that of rotary multi-bed regenerators, where a compartmentalized regenerator is rotated relative to a magnet system. Multi-bed rotary regenerator devices have the advantageous possibility of utilizing the magnet for magnetizing MCM at all times, which is crucial due to the fact that the magnet is the single most expensive component in a magnetic refrigerator Bjørk et al. (2010). Recently developed prototypes of this kind are using stationary regenerators and rotating magnet arrangements as this supports an energy efficient operation, see e.g. Aprea et al. (2014), Aprea et al. (2016) and Lozano (2015). Such devices are indeed demonstrating improved efficiencies, see e.g. Jacobs et al. (2014) and Eriksen et al. (2015c). However, the somewhat limited temperature span, compared to vapor compression, may suggest introducing the technology through special niche applications and concepts, see e.g. Aprea et al. (2015).

One of the main technical challenges with rotary multi-bed devices is the flow handling system that continuously distributes the pump-driven heat transfer fluid between the AMR and the hot and cold heat exchangers, while
enabling a reciprocating flow in the regenerator beds timed with the varying magnetic field.

In a recently constructed prototype at the Technical University of Denmark Eriksen et al. (2015c), the flow is controlled by a system of poppet valves on the hot side of the AMR in combination with check valves on the cold side, see Eriksen et al. (2015a). Experiments with this prototype have revealed strong impacts on the AMR performance by small adjustments of the flow resistances of the individual regenerator flow channels, thereby affecting the balancing of the reciprocating flow in the regenerator beds Eriksen et al. (2015b). This issue is inherent to multi-bed rotary regenerator devices since some variations in the resistance of the flow paths through the beds will to some extend be present due to manufacturing tolerances.

In some cases, imposing a controlled unbalance in an AMR may be utilized in special configurations to improve the system performance Zimm and Russek (2009). However, the unbalances in rotary multi-bed devices arising from the varying bed flow resistances are not controlled and their impact is not well understood.

In this paper we present a theoretical study based on numerical AMR modeling of the effects on system performance of unbalanced flow due to varying flow resistances exemplified with experiments on the actual prototype.

5.2 Experimental

The rotary multi-bed AMR prototype considered in this paper and its heat transfer fluid flow system are illustrated in Figure 5.1. The device consists of a cylindrical, compartmentalized regenerator with eleven beds, containing packed spheres of Gd and Gd$_1-x$Y$_x$, $x \in \{0.025, 0.05, 0.10\}$ alloys as MCM. Regenerator bed dimensions with layers, particle sizes and Curie temperatures can be seen in Figure 5.2 and Table 5.1

<table>
<thead>
<tr>
<th>Layer</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x$</td>
<td>0</td>
<td>0.025</td>
<td>0.05</td>
<td>0.10</td>
</tr>
<tr>
<td>$T_C$[K]</td>
<td>291</td>
<td>287</td>
<td>283</td>
<td>272</td>
</tr>
<tr>
<td>$d_p$[µm]</td>
<td>500-600</td>
<td>300-500</td>
<td>300-500</td>
<td>300-500</td>
</tr>
</tbody>
</table>

Table 5.1: Regenerator layers numbered according to Figure 5.2 with yttrium fractions, $x$, Curie temperatures, $T_C$, and particle diameters, $d_p$, of the MCM spheres.

A Halbach-like magnet arrangement is rotated around the regenerator, which is situated on the outside of an iron core, thereby sweeping high and low field regions over the beds. Further details are given in Eriksen et al. (2015c). The poppet valves situated over the regenerator are connected to
the beds with bent plastic hoses and each of them are connected in series with dedicated ball valves before going to the hot side manifolds. The beds are hydraulically connected in parallel with three to four beds being open to flow in each direction at all times. This means that each bed interacts with three other beds from a fluid flow standpoint in both flow directions. Since the device is a one-off construction, plastic hoses with slightly different lengths and bends are used as connectors. The check valves ensuring a uni-directional flow through the cold heat exchanger are installed in a compact
component containing the cold inlet and outlet manifolds. Due to man-
ufacturing tolerances, small variations in the flow resistances through the
different beds, valve arrangements and connectors in both directions will be
present. It is, however, possible to compensate for this via the adjustment
valves.

During an experiment illustrating this, the prototype was operating with

all adjustment valves fully open. Then, the one in the cold-to-hot direction
of bed number four was incrementally closed, until the temperature span,
\( \Delta T \), was maximized. With this setting, an experiment resulting in the tem-
perature span shown in Figure 5.3 was carried out. After reaching \( \Delta T = 19.8 \) K, the valve adjustment was reset, resulting in a rather dramatic
decrease in the span down to \( \Delta T = 16.4 \) K. We expect this effect to be due
to unbalanced flow in the regenerator beds, resulting from the variations in
the flow resistances described above.

\section*{5.3 Modeling AMRs with flow unbalance}

To investigate the effect on AMR performance of flow unbalance, a system
comprising only two coupled regenerator beds, such as the one presented by
Tura and Rowe (2011), is first considered; see Figure 5.4.

In such a system, two regenerator beds are magnetized and demagnetized
in opposite phases by dedicated variable field sources. A heat transfer fluid
is being displaced, such that the demagnetized bed experiences a hot-to-cold
blow while the magnetized bed is subjected to a cold-to-hot blow, see figure
5.4. A system using a displacer cannot experience flow unbalance due to the flow system design. However, adding additional regenerator beds is difficult. If the beds are instead connected to a pump that operates continuously the flow in each bed in each direction is not necessarily equal. However, the displacement may be unbalanced, in which case $\Delta x_1 \neq \Delta x_2$ in Figure 5.4. The flow-wise series coupling of the two beds implies due to continuity, that the volume of the hot-to-cold blow in bed one, $V_{HC_1}$, equals that of the cold-to-hot blow in bed two, $V_{CH_2}$. Likewise, the volume of the hot-to-cold blow through bed two, $V_{HC_2}$ equals that of the cold-to-hot blow of bed one, $V_{CH_1}$. The average absolute displacement during one AMR cycle in the beds is then

$$V_{avg} = \frac{V_{HC_1} + V_{CH_1}}{2} = \frac{V_{HC_2} + V_{CH_2}}{2}.$$ \hspace{1cm} (5.1)

From this we define the unbalance, $u$, in each of the two beds as the volume displacement in the hot-to-cold direction minus that in the cold-to-hot direction, relative to their average:

$$u_1 = \frac{V_{HC_1} - V_{CH_1}}{V_{avg}},$$ \hspace{1cm} (5.2)

$$u_2 = \frac{V_{HC_2} - V_{CH_2}}{V_{avg}} = -u_1.$$ \hspace{1cm} (5.3)

The influence of this unbalance parameter on the AMR performance has been analyzed by the use of a 1D numerical AMR model, based on Engelbrecht (2008).
In this model a single regenerator in which a flow unbalance may be present is considered. The model solves energy balance equations for the heat transfer fluid and the solid porous MCM matrix respectively. The governing equation, considering volume specific energies in an infinitesimal control volume of length $dx$ and cross sectional area, $A_c$, for the fluid, may be formulated as:

$$\frac{\dot{m}_c}{A_c} \frac{\partial T_f}{\partial x} + h_a s (T_f - T_s) + \rho_f c_f \frac{\partial T_f}{\partial t} - k_{\text{disp}} \frac{\partial^2 T_f}{\partial x^2} = \frac{\dot{m}}{\rho_f A_c} \frac{\partial p}{\partial x}. \quad (5.4)$$

The terms represent, from left to right, enthalpy change of the fluid, convective heat transfer from fluid to solid, energy storage, axial dispersion and viscous dissipation. $\dot{m}_c$ is the fluid mass flow rate (positive in the hot to cold direction), $c_f$ is the mass specific heat capacity of the fluid, $T_f$ and $T_s$ are the temperatures of the fluid and solid respectively, $h$ is the convective heat transfer coefficient from fluid to solid, $a_s$ is the volume specific surface area of the MCM, $\rho_f$ is the fluid density, $\epsilon$ is the porosity, $t$ is time, $k_{\text{disp}}$ is the thermal conductivity of the fluid due to axial dispersion and $p$ is the pressure.

Likewise, the governing equation for the solid can be formulated as:

$$h_a s (T_f - T_s) + k_{\text{eff}} \frac{\partial^2 T_s}{\partial x^2} = (1 - \epsilon) \rho_s c_B s \frac{\partial T_s}{\partial t} + (1 - \epsilon) \rho_s c_B s \frac{\partial T_s}{\partial t}, \quad (5.5)$$

where the terms, from left to right, represent convective heat transfer from fluid to solid, axial conduction, transfer of magnetic work and energy storage. $k_{\text{eff}}$ is the thermal conductivity of the combined fluid/solid matrix, $B$ is the internal magnetic field, $\rho_s$ is the density of the MCM and $c_B s$ is the mass specific heat capacity of the MCM for fixed magnetic field.

The mass flow rate and fluid temperature entering the matrix are given as boundary conditions as functions of space, $x$, and time, $t$:

$$\dot{m}(t) \geq 0 \Rightarrow T_f(x = 0, t) = T_H$$
$$\dot{m}(t) \leq 0 \Rightarrow T_f(x = L, t) = T_C, \quad (5.6)$$

where $T_H$ and $T_C$ are the temperatures of the hot and cold reservoirs respectively and $L$ is the regenerator length. A linear temperature ramp between $T_H$ and $T_C$ is used as initial condition. The regenerator modeled in this case corresponds to one bed of the rotary prototype described above. Here, the temperature span between the hot and the cold reservoirs is set to $\Delta T = 20K$, with $T_H = 292K$. The AMR operational frequency is $f = 1Hz$, and the average mass flow rate is $\dot{m} = 1.67 \cdot 10^{-2} \text{kg s}^{-1}$. The heat transfer fluid is assumed as water mixed with 10% ethylene glycol. As properties of the MCM, data from the Weiss mean field model for Gd are used, see Bjørk (2010) , with Curie temperatures shifted to match the Gd$\text{I}_{-x}Y_x$ alloys. Data sets from this model are often used in modeling AMRs as this
supports comparison between different models, see e.g. Zheng et al. (2007), Petersen et al. (2008) and Nielsen et al. (2009).

The resulting cooling power, $\dot{Q}_C$, and heat rejection, $\dot{Q}_H$, in a regenerator bed as functions of the unbalance are plotted in Figure 5.5. When reviewing Figure 5.5, it is important to remember that the system is being supported by outside fluid and energy inputs to support the single regenerator operating with unbalanced flow. Systems with a negative unbalance value require a supply of fluid at $T_C$ provided from an outside source while systems with a positive unbalance require additional fluid flow at $T_H$ but at the same time produce an excess of fluid at $T_C$. The assistance of an outside energy system allows regenerators with a negative unbalance to operate at efficiencies that appear higher than the Carnot limit.

A realistic modeling case is to couple a regenerator with a negative unbalance with one that has a positive unbalance. This gives the overall system an equal fluid flow in both directions and provides realistic predictions for the entire system. For the coupled system, the unbalances of the two regenerators are always of equal magnitudes and opposite signs. That is, in order to obtain the total value of $\dot{Q}_C$ (or $\dot{Q}_H$) for a given absolute value of the unbalance of the system, the values for a single bed corresponding to the positive and negative values respectively of that unbalance are added together. The results are shown in Figure 5.6.

From this it is clear that the cooling power is reduced by almost 30% if the system is operated at an unbalance of only 5% in the modeled case.
5.3. MODELING AMRS WITH FLOW UNBALANCE

The COP, defined as

\[
COP = \frac{\dot{Q}_C}{\dot{W}} = \frac{\dot{Q}_C}{\dot{Q}_H - \dot{Q}_C},
\]

(5.7)

where \(\dot{W}\) is the total work input to the system, as a function of the unbalance is plotted in figure 5.7. In this case, an unbalance of 5% reduces the COP with almost 50%.

5.3.1 Effects on temperature profile of flow unbalance

To illustrate the effects of flow unbalance on the AMR cycle, the temperature profiles in the MCM of three example operating conditions with different unbalances are plotted in Figure 5.8. The wavy nature of the solid temperature profile is caused by concentrations of high specific heat and entropy change associated with the layered bed. In the figure, the profile is plotted at the middle of the hot-to-cold flow period as a representative snapshot of the AMR operation. Here, the modeled flow profile, i.e. flow rate vs. time, is trapezoidal, corresponding to the flow model used in section 5.4 for the rotary multi-bed case. Full videos that show fluid and MCM temperatures throughout a single cycle operating at cyclical steady state are available as supporting material.

Figure 5.8 shows that flow unbalance drastically affects the temperature profile in the regenerator. When the flow is unbalanced in the positive direction, the temperature of the regenerator increases throughout the interior.
of the regenerator and its ability to accept a cooling load is greatly reduced, mostly due to the fact that the fluid exiting the cold end is above the cold reservoir temperature for a portion of the cycle. On the other hand, the regenerator with a negative unbalance is cooler in the interior and is much more capable of accepting a cooling load. The reason the performance is so influenced by flow unbalance is that the enhanced cooling of the negative unbalance beds is only possible when the excess fluid at $T_C$ is provided by
the regenerator bed with a positive unbalance. In the modeled case, the regenerators are all working between the same hot and cold reservoirs, i.e. the fluid inlet temperature is $T_H$ at the hot side and $T_C$ at the cold side in all cases. What is varying as a function of flow unbalance is the fluid exit temperatures - and this is what causes the different values of $\dot{Q}_C$ and $\dot{Q}_H$. This is also evident from Figure 5.9, where the temperature of the fluid exiting the cold end of the regenerator is plotted for the entire hot to cold blow period. In this case it can be seen, that the exit temperature actually gets above the cold reservoir temperature of 272 K at the end of the blow period in all cases. However, the average between the exiting temperatures of the positive and the negative unbalanced beds is higher than that of the balanced bed. Therefore, two coupled beds perform better with no unbalance.
5.4 Rotary multi-bed flow model

In order to analyse the unbalance effects on the performance of a multi-bed AMR machine that may arise from variations in the flow resistances of the different beds/flow paths that are operated in varying combinations, a simple flow model has been set up.

![Schematic diagram representing the flow circuit of a multi-bed AMR machine with n beds corresponding to the rotary prototype illustrated in Figure 5.1. The resistances, $R_{ij}$ refer to bed $i$ including corresponding valves and fittings in the direction $j$, with $j = 1$ denoting the hot-to-cold and $j = 2$ the cold-to-hot direction.](image)

As indicated in Figure 5.10, the flow system of a multi-bed prototype is treated as a circuit in which the flow channels from hot to cold (index $j = 1$) and from cold to hot (index $j = 2$) each have different resistances, $R_{ij}$. In the case of the prototype illustrated in Figure 5.1, $R_{11}$ is the sum of the resistances of an adjustment valve, a poppet valve, regenerator bed 1 in the hot to cold direction, a check valve and the connecting hoses and fittings. The resistances, $R_{ij}$ are modeled as Ohmic. They are functions of the rotational angle, $\phi$, of the magnet/cam rings, that open/close the poppet valves linearly. That is, the rotational angle normalized with the angle of a full AMR cycle, $\phi^*/\phi_{AMR}$, can be divided into parts taken up by flow pauses, $\phi^*_p$, ramps, $\phi^*_r$ and full flow, $\phi^*_f$, that all together add up to one for a complete cycle. This implies that if bed one were tested separately in each direction, corresponding to $R_{ij} = 0, i \neq 1$ in Figure 5.10, the resistance would decrease/increase hyperbolically between infinity and finite values, $R_{1j,open}$, while the flow would ramp up/down linearly, as indicated in Figure 5.11. The total resistances in the two directions are then:
Figure 5.11: Resistance and flow rate as functions of the normalized rotational angle in bed one during an AMR cycle with all other resistors set to zero. The resistances are normalized with the values corresponding to fully open poppet valves, i.e. $R_{ij}^* (\phi) = R_{ij} (\phi) / R_{ij, \text{open}}$. The flow rates are normalized with the average non-zero flow rates in both directions in the bed, i.e. $\dot{V}_{ij}^* (\phi) = \dot{V}_{ij} (\phi) / \dot{V}_{j, \text{avg}}$.

$$R_{j, \text{tot}} (\phi) = \left( \sum_{i=1}^{n} \frac{1}{R_{ij} (\phi)} \right)^{-1}. \quad (5.8)$$

A constant pressure above the atmosphere, $P_1$, is set after the pump, and the circuit is “grounded” (open to the ambient pressure) before the pump, i.e. the total pressure drop is $\Delta P_{\text{tot}} = P_1$, giving a total volumetric flow rate, $\dot{V}_{\text{tot}}$ of

$$\dot{V}_{\text{tot}} (\phi) = \frac{P_1}{R_{1, \text{tot}} (\phi) + R_{2, \text{tot}} (\phi)}. \quad (5.9)$$

From this, the intermediate pressure, $P_2$, can be expressed as

$$P_2 = R_{2, \text{tot}} (\phi) \dot{V}_{\text{tot}} (\phi). \quad (5.10)$$

With $\Delta P_1 = P_1 - P_2$ and $\Delta P_2 = P_2 - P_3$, the flow rates in each bed as functions of $\phi$ are then

$$\dot{V}_{ij} (\phi) = \frac{\Delta P_j}{R_{ij} (\phi)}. \quad (5.11)$$

To illustrate the flow profiles in an actual multi-bed machine, measured resistances in the prototype shown in Figure 5.1 are used, see Table 5.2.
The resistances in a bed in one direction, say hot to cold, is measured by only having this particular bed open to flow in that direction. By means of pressure gauges, the pressure drop over the bed is measured and the corresponding flow rate is measured by an in-line flow meter. The signals are then used simultaneously in a LabView program to calculate the resistance. Due to the accuracies of the gauges, the uncertainty on the resistance measurements is 2%. Although this is quite significant compared to the variations between the beds, these values are used as examples in the present analysis. From these resistances, the corresponding flow rates are calculated using Eqs. 5.8–5.11, with the inlet pressure set to $P_1 = 2$ bar, and the corresponding unbalances are given in Table 5.2. As an example, the resulting flow profiles are plotted for bed number four in Figure 5.12 (solid lines), with the values normalized as in Figure 5.11. In this case, an unbalance of $u = -7.54\%$ is present. In the same figure, the profiles are plotted in the case where the average resistance in the two directions are used respectively, thereby eliminating the unbalance.

### Table 5.2: Measured flow resistances, $R_{ij}$, in bar min L$^{-1}$ and resulting calculated flow unbalances, $u_i$, in percent.

<table>
<thead>
<tr>
<th>$i$</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_{i1}$</td>
<td>0.84</td>
<td>0.83</td>
<td>0.87</td>
<td>0.92</td>
<td>0.89</td>
<td>0.90</td>
<td>0.92</td>
<td>0.83</td>
<td>0.85</td>
<td>0.79</td>
<td>0.88</td>
</tr>
<tr>
<td>$R_{i2}$</td>
<td>1.06</td>
<td>1.02</td>
<td>1.14</td>
<td>1.06</td>
<td>1.00</td>
<td>1.12</td>
<td>1.02</td>
<td>1.13</td>
<td>1.12</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>$u_i$</td>
<td>3.1</td>
<td>-0.2</td>
<td>5.4</td>
<td>-7.6</td>
<td>-7.7</td>
<td>3.0</td>
<td>-7.3</td>
<td>-0.8</td>
<td>6.5</td>
<td>12.5</td>
<td>-6.8</td>
</tr>
</tbody>
</table>

Figure 5.12: Calculated flow profiles with and without unbalance of bed number four in the eleven bed prototype.
5.5. **EFFECT ON MULTI-BED AMR PERFORMANCE**

The actual cam profile has steep ramps (\( \phi^* = 0.05 \)) and a fully open fraction of \( \phi^*_f = 0.5 \). It is seen that during the fully open periods, the flow profile has a jagged shape. This is explained by the number of beds open to flow in the two directions, which varies with the rotation angle. All beds have the same number of spikes and depressions at the same rotational angles relative to the individual beds. In general, the number and magnitudes of spikes and depressions for rotary multi-bed machines are functions of the total number of beds, number of cams per revolution and the cam profiles, which should be taken into consideration when designing such systems. Modeled flow profile spikes and depressions are in good agreement with experimental results, see Eriksen et al. (2015b).

It should be noted that the flow rate as function of \( \phi \) is evaluated at steady flow conditions. During AMR operation the spikes and depressions may be dampened by the means of a pressure vessel. Due to asymmetries in the design, it is clear from Table 5.2 that the resistance is higher in the cold-to-hot than in the hot-to-cold direction, with average resistances of \( R_{1,\text{avg}} = 0.86 \) bar min \( L^{-1} \) and \( R_{2,\text{avg}} = 1.06 \) bar min \( L^{-1} \), respectively. The standard deviations in flow resistances in the two directions are: \( \sigma_{R,1} = 4.0\% \) and \( \sigma_{R,2} = 5.4\% \), respectively. This offset between resistances in the two directions does not in itself give rise to unbalanced flow. Furthermore, if there is a variation in resistance between the beds, but it is independent of the flow direction, i.e. \( R_{i1} = R_{i2} \), no flow unbalances are present, although this leads to a variation in flow rates between beds. Only in the case where the varying resistances become direction-dependent, the flow in the beds becomes unbalanced.

### 5.5 Effect on multi-bed AMR performance

The 1D AMR model described in Section 5.3 has been used with flow profiles calculated by the flow model described in Section 5.4 to investigate the impact of varying flow resistances on multi-bed AMRs. We have modeled the rotary prototype with these flow profiles with and without the spikes and depressions while keeping the average flow constant. We found no significant impact of the “jaggedness” in the present case.

The impact of direction dependent flow resistances has been investigated by modeling a number of cases of AMR machines containing eleven beds, similar to the described rotary prototype. This was done as described in the following.

A data set containing cooling power and heat rejection for a bed was calculated with the AMR model with a frequency of \( f = 1 \) Hz. Here, the flow rate was varied from \( 0.9 \) L min\(^{-1}\) to \( 1.1 \) L min\(^{-1}\) in steps of \( 0.01 \) L min\(^{-1}\) and the unbalance was varied from -20\% to 20\% in steps of 1\%. The
results are shown in Figures 5.13 and 5.14.

Figure 5.13: Modeled single bed cooling power of the rotary prototype as a function of unbalance fraction and flow rate.

Figure 5.14: Modeled single bed heat rejection as a function of unbalance fraction and flow rate.

Next, resistance combinations corresponding to a number of hypothetical eleven-bed machines have been generated with random normally distributed flow channel resistances with mean values of $R_{i1,\text{avg}} = R_{i2,\text{avg}} = 1$ bar min
L⁻¹ and standard deviations, σₐ, varied from 0% to 5% in steps of 1%, to stay consistent with the orders of magnitudes observed in the experiment. The inlet pressure was set to P₁ = 2 bar. The flow rates and unbalances corresponding to this were calculated with the flow model in all cases. For each of these hypothetical machines, the flow rates and unbalances of all beds correspond to cooling powers and heating powers which are found by interpolation in the modeled AMR dataset shown in Fig. 5.13 and 5.14.

![Machine cooling power, 11 beds](image)

Figure 5.15: Distributions of cooling powers from 1000 imaginary prototypes with eleven beds plotted at different standard deviations in the flow resistances

The result of this modeling exercise is a resulting cooling power and COP for 1000 randomly generated devices for each standard deviation of flow resistance. The resulting cooling powers and COPs are shown as box and whiskers plots in Figures 5.15 and 5.16. In these plots, the 25th and 75th percentiles are indicated by the lower and upper box edges respectively, while the median is indicated by a central mark and necking of the boxes. The whiskers extend to the most extreme values not considered outliers, as these are plotted individually. The plot indicates that a device with even relatively small variation in the flow resistance in the regenerator beds can exhibit a large range of possible device performance. It also indicates that some combinations fortuitously perform better than others. It is clear from the figures that the performance is likely to decrease significantly as the standard deviation of the channel flow resistances increases, causing unbalanced flow in the beds. In this case, the median of the cooling powers decreased by more than 30% at a standard deviation of 5% of the resistances compared to a perfect distribution of flow resistances. The me-
Figure 5.16: Distributions of COPs from 1000 imaginary prototypes with eleven beds plotted at different standard deviations in the flow resistances.

median COP dropped by almost 50%. These trends are in agreement with the case of the two-bed device model presented in Section 5.4.

For the actual prototype, the flow resistances were all set to 0.92 bar min L$^{-1}$ in the hot-to-cold direction and 1.14 bar min L$^{-1}$ in the cold-to-hot direction by the adjustment valves, to eliminate the flow unbalances within the measurement uncertainty of 2%. This resulted in a temperature span of 27.3 K for an experiment similar to the one shown in Figure 5.3, where the span was 16.4 K in the unbalanced case. This supports the prediction of strong impact on AMR performance of flow balancing. In the AMR experiments, which are without an applied cooling load, the effects are only evident via the generated no-load temperature spans. With the current setup it is not feasible to do a direct comparison between the modeled effects of resistance variations on cooling powers and COPs with those predicted by the calculations. This would require a setup with a very fine control over the resistances of the beds, as even small variations may result in large impacts as the present analysis suggests.

### 5.6 Conclusion

It is shown by a theoretical study based on 1D AMR modeling that unbalanced heat transfer fluid flow in the regenerators can drastically deteriorate the performance of AMR machines. This is shown to be the case for a modeled device with two coupled regenerators and a fluid displacer impos-
ing an unbalance. For rotary multi-bed devices, it is shown through flow modeling, that variations in the flow resistances between the flow channels transporting flow in and out of the regenerator beds give rise to unbalanced flow in the beds. By modeling a large number of cases with varying standard deviations on the resistances of the flow channels, corresponding to an actual eleven-bed prototype, it is concluded that the performance in terms of cooling power and COP is likely to deteriorate as much for the multi-bed machine with a standard variation in resistances of 5% as it did in the modeled case of the two-bed device with an imposed unbalance of 5%. The results are supported by observed negative impacts on the temperature span by flow resistance variations in experiments with the rotary prototype.

5.7 Addendum

A more recent experiment has resulted in a no-load temperature span of $\Delta T = 29.2$ K. This is, at the time of writing, the highest temperature span achieved with the prototype. It was obtained at a fluid flow rate of $\dot{V} = 3.4$ L min$^{-1}$ and a frequency of $f_{\text{AMR}} = 1.4$ Hz.
Chapter 6

Exploring the efficiency potential for an active magnetic regenerator

Abstract
A novel rotary state of the art active magnetic regenerator (AMR) refrigeration prototype was used in an experimental investigation with special focus on efficiency. Based on an applied cooling load, measured shaft power and pumping power applied to the AMR, a maximum second-law efficiency of 18% was obtained at a cooling load of 81.5 W, resulting in a temperature span of 15.5 K and a COP of 3.6. A loss analysis is given, based on measured pumping power and shaft power together with theoretically estimated regenerator pressure drop. It is shown that, especially for the pressure drop, significant improvements can be made to the machine. However, a large part of the losses may be attributed to regenerator irreversibilities. Considering these unchanged, an estimated upper limit to the 2nd-law efficiency of 30% is given by eliminating parasitic losses and replacing the packed spheres with a theoretical parallel plate regenerator. Furthermore, significant potential efficiency improvements through optimized regenerator geometries are estimated and discussed.

The present chapter is, apart from the addendum, accepted for publication in: Eriksen, D., Engelbrecht, K., Bahl, C. R. H., Bjørk, R. (2016). Exploring the efficiency potential for an active magnetic regenerating refrigerator. *Science and Technology for the Built Environment.* (Eriksen et al. (2016b))
6.1 Introduction

Magnetic refrigeration is a promising emerging alternative to conventional vapor compression refrigeration. It is based on magnetization and subsequent demagnetization of a magnetocaloric material (MCM), which thereby heats up and cools down. By means of a heat transfer fluid, excess heat may be rejected to the surroundings and a cooling load accepted from a cold reservoir in what is known as the active magnetic regenerator (AMR) cycle (Kitanovski et al. (2015)). As the refrigerant is a solid state material and the heat transfer fluid may be a water based liquid, hazardous and environmentally harmful gases are avoided. The absence of a compressor opens the possibility for a more silent operation. However, the main argument in favor of magnetic refrigeration is the potential for high efficiency. Already in 1998, COPs above 6 were demonstrated (Zimm et al. (1998)). This was, however, obtained with a superconducting magnet producing a magnetic field of 5 T and the power consumption of this was not included in the COP, which was furthermore adjusted to ignore seal friction. Since then, an increasing number of AMR prototypes based on permanent magnets have been reported by various groups (Bjørk et al. (2010), Kitanovski et al. (2015)). Besides this, numerical AMR modelling has indicated the possibility of obtaining competitive COPs (Engelbrecht et al. (2006), Engelbrecht (2008)). However, technological and scientific challenges, spanning from development and basic understanding of the MCMs themselves to actual machine design issues, have made the road towards the ultimate goal of a magnetic refrigerator with a competitive COP long and bumpy - and we are not quite there yet. This is reflected in the fact that only a few of the groups presenting prototypes have even reported COPs - the focus has in general been more on obtaining relevant temperature spans and cooling powers. A group at the Tokyo Institute of Technology and Chubu Electric Power Co. presented a rotary AMR device in 2005 (Okamura et al. (2005)) and further development by this group has resulted in an prototype with a COP of 2.5 at a temperature span of 5 K (Okamura and Hirano (2013)). Astronautics Corporation of America has recently presented results from a prototype operating at a COP above 2 at a temperature span of 10 K (Jacobs et al. (2014)). Pushing the development towards more efficient devices makes it crucial to obtain knowledge about issues limiting the performance to reveal opportunities of improved future designs. At the Technical University of Denmark, a thorough loss analysis of a rotary prototype (Lozano et al. (2013)) has revealed a number of significant design issues reducing COP, the most important being friction in the flow system and heat leaks. This knowledge has been used in the design of a recently published novel prototype (Eriksen et al. (2015c)) in which the flow is controlled by poppet valves giving greatly reduced friction. Furthermore, special care has been taken to minimize heat leaks by keeping components compact and well
insulated on the cold side and by including an insulating air gap between regenerator and magnet (Eriksen et al. (2015c)). In the present paper, the most recent results obtained with this device are presented. Furthermore, the work input to the AMR is analyzed in order to map out the different losses, both relating to the pumping power and drive power. Based on this, possibilities for future improvements are discussed.

6.2 Experimental setup

The presented study was carried out on a recently constructed AMR prototype at the Technical University of Denmark (Eriksen et al. (2015c)). The setup is illustrated in Figure 6.1. A cylindrical regenerator is divided into eleven beds filled with a total of 1.7 kg of closely packed spheres of Gd and Gd$_{(1-x)}$Y$_x$ alloys with an average diameter of 460 µm. The beds are subsequently magnetized and demagnetized by a rotating Halbach-like magnet arrangement which is supported on the outside by three HEPCO bearings. On top of the rotating magnet, two cam rings actuate poppet valves controlling the flow in each direction in the beds, timed with the varying magnetic
CHAPTER 6. EXPLORING THE EFFICIENCY POTENTIAL...

field. The rotating part is driven by a motor with a gear box. The rotational speed of the shaft is measured by an optical encoder. On the drive shaft, between gear box and driving gear, a torque transducer is installed. On the cold side of the regenerator, a compact manifold comprising a system of check valves ensures a unidirectional flow through an insulated electrical heater, in which a controlled heat load can be applied. The temperature on the hot side is controlled by a temperature controlled bath (not shown) connected to the hot heat exchanger. The flow is driven by a gear pump. For further details regarding the machine design, see Eriksen et al. (2015c). Temperatures are measured on both the hot and cold side of the AMR by thermocouples, pressure is measured by pressure gauges, and the fluid flow rate is measured by an in-line flow meter as indicated in Figure 6.1.

6.3 Analysis

In the present investigation the AMR performance is characterized by considering the shaft work and pumping power as inputs to the system consisting of magnet, regenerator and flow control components, indicated as "AMR machine" in Figure 6.1. The pumping power, $W_{\text{pump}}$, is evaluated as the total pressure drop over the AMR machine times the volumetric flow rate, whereas the shaft power, $W_{\text{shaft}}$, is calculated as

$$W_{\text{shaft}} = 2\pi \cdot \tau \cdot f_{\text{rot}},$$

where $\tau = \text{torque}$ and $f_{\text{rot}} = \text{rotational frequency}$. When a certain cooling load, $Q_C$, is applied via the electric heater during an AMR experiment, the $COP$ at steady state may be evaluated as

$$COP = \frac{Q_C}{W_{\text{pump}} + W_{\text{shaft}}}$$

(6.2)

In the present analysis we consider the temperature span, $\Delta T$, to be the temperature difference between the time averaged temperatures exiting the regenerator at the hot end ($T_H$) and cold end ($T_C$). However, to calculate a 2nd-law efficiency, we will use the temperatures entering the hot side ($T_{H,\text{in}}$) and cold side ($T_{C,\text{in}}$) of the AMR machine as this would resemble the reservoir temperatures of a corresponding Carnot machine:

$$\eta_{2\text{nd,AMR}} = \frac{COP}{COP_{\text{Carnot}}} = \frac{T_{H,\text{in}} - T_{C,\text{in}}}{T_{C,\text{in}}} \cdot COP.$$  

(6.3)

6.4 Experimental AMR results

A series of AMR experiments were carried out with varying hot side temperatures, but with fixed AMR frequency, $f_{\text{AMR}}$, of 1 Hz, a fluid flow rate
of $\dot{V} = 2.5$ L/min and an applied cooling load of $\dot{Q}_C = 81.5$ W. The ambient temperature was 293 K ± 1 K during the tests. In each experiment, experimental values were recorded once steady state was reached. This was obtained once the average temperatures in and out of the AMR machine were unchanged for several minutes.

![Figure 6.2](image)

The resulting temperature spans and COPs are shown in Figure 6.2 (a). A maximum temperature span of $\Delta T = 16.7$ K was achieved at a hot side temperature of 19.5°C. By keeping the hot side temperature fixed, a second series of experiments was conducted by varying the AMR frequency and keeping everything else unchanged. The results are shown in Figure 6.2 (b). For these experiments, a highest second law efficiency of $\eta_{2nd,AMR} = 18\%$ was achieved at $f_{AMR} = 0.61$ Hz. In this case, the temperature span was 15.5 K and the COP was 3.6. The total pumping power was $W_{pump} = 8.9$ W and the shaft power was $W_{shaft} = 14$ W. An overview of the experimental uncertainties can be found in Table 6.1.

<table>
<thead>
<tr>
<th>Measurement uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
</tr>
<tr>
<td>--------------</td>
</tr>
<tr>
<td>0.1 K</td>
</tr>
</tbody>
</table>

6.5 Loss analysis - what is limiting the efficiency?

The experiment with $\eta_{2nd,AMR} = 18\%$ at $f_{AMR} = 0.61$ Hz and $\dot{V} = 2.5$ L/min described above will in this section be used as a starting point for a loss analysis to identify the reasons why the actual work is more than
five times that of a corresponding Carnot cooling machine, although the magnetocaloric effect is reversible in second order MCMs, like the ones used in the present regenerator.

6.5.1 Mechanical losses

In order to evaluate losses increasing the necessary mechanical work which is input as shaft power to the AMR machine, experiments were conducted in which the machine was operated without pumping (no AMR cycles) at different rotational speeds. For practical reasons, the regenerator was installed during all of these experiments, which were conducted at room temperature. Previous experiments have shown a very minor influence of the presence of the regenerator on the shaft power under such conditions, and this is neglected here. The resulting measured shaft powers are therefore considered equal to what is dissipated during AMR operation due to drive gear losses and friction in bearings and poppet valve system. These experiments were then repeated without the poppet valves installed.

Figure 6.3: Power dissipation based on measured torque at different frequencies, with and without the poppet valves installed.

The resulting shaft powers with and without the valves installed are plotted as functions of corresponding AMR frequencies and fitted with power laws as phenomenological models, see Figure 6.3. The difference between the value of the two functions at a given frequency then corresponds to the power dissipation due to valve friction. In the case of $f_{AMR} = 0.61$, the dissipation due to bearings and gear corresponds to 1.6 W, while the valve
friction accounts for 1.1 W.

### 6.5.2 Pumping losses

The total measured pressure drop over the AMR machine includes that of the regenerator itself as well as those caused by the flow system components. The flow system components include the poppet valves, needle valves used to normalize flow resistance, the cold manifold, check valves, fittings and connecting tubing. For the packed sphere bed, a theoretical estimate of the pressure drop can be given (Ergun and Orning (1949)):

\[
\Delta P = \left( \frac{18\pi^2(1 - \varepsilon)}{d_{sp}} + \frac{1.8v_s\rho_f}{\mu} \right) \frac{v_s^2\rho_f}{\varepsilon^3Re_{sp}}L
\]

where 
- \( \varepsilon \) = porosity
- \( \mu \) = fluid viscosity
- \( v_s \) = superficial velocity
- \( d_{sp} \) = sphere diameter
- \( \rho_f \) = fluid density
- \( L \) = length of bed

Here, the superficial velocity is defined as

\[
v_s = \frac{\dot{m}}{\rho_f A_c}
\]

where
- \( \dot{m} \) = mass flow rate
- \( A_c \) = cross sectional area of the bed

and the Reynolds number is defined as

\[
Re_{sp} = \frac{d_{sp}v_s\rho_f}{(1 - \varepsilon)\mu_f}.
\]

For the viscosity and density of the fluid, values corresponding to the average regenerator temperature are used. The porosity is set to \( \varepsilon = 0.36 \), corresponding to closely packed spheres. For the sphere diameter, the mean value of \( d_{sp} = 460 \mu m \) is used. Apart from minor spikes in the flow profiles (Eriksen et al. (2015b)) which are neglected here, three beds are open to flow in both directions in the AMR machine at all times. For the considered AMR experiment, one bed carries one third of \( \dot{V} = 2.5 \text{ L/min} \) which corresponds to a superficial velocity of \( 3.87 \times 10^{-2} \text{ m/s} \). From Equation 6.4 this implies an estimated pressure drop of \( \Delta P = 0.49 \text{ bar} \) through the beds in one direction, corresponding to \( \Delta P_{\text{AMR}} = 0.98 \text{ bar} \) over the entire AMR machine (hot to cold and cold to hot). This estimated pressure drop caused by the packed sphere beds yields an estimated pumping power of \( W_{\text{pump,reg}} = 4.1 \text{ W} \). Hence, the pumping power due to external components accounts for an estimated \( W_{\text{pump,ext.}} = 4.8 \text{ W} \).
6.5.3 Distribution of input power to the AMR machine

Table 6.2: Consumption of power input to the AMR machine, W

<table>
<thead>
<tr>
<th></th>
<th>Carnot work</th>
<th>Bearings and gear</th>
<th>Poppet valve friction</th>
<th>Pumping, regenerator beds</th>
<th>Regenerator losses</th>
<th>Motor power</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4.0</td>
<td>1.6</td>
<td>1.1</td>
<td>4.1</td>
<td>4.8</td>
<td>7.2</td>
</tr>
</tbody>
</table>

The results and estimates given in the above section are summarized in Table 6.2 and the relative distribution is shown in Figure 6.4.

Figure 6.4: Relative distribution of power input to the AMR machine corresponding to experiment with a second law efficiency of 18%.

The 7.2 W, corresponding to 31% of the input power, which are not accounted for above, are considered regenerator losses. These will inevitably arise from heat leaks, flow bypass or flow channeling, and entropy generation due to axial conduction, dispersion and heat transfer between fluid and solid in the regenerator.

6.6 Discussion

Mapping out the losses as in the case study presented above is crucial to get an insight into which improvements might be done in future designs to maximize efficiency. But furthermore, it may be used to estimate an upper limit for the efficiency that might be reached based on actual experiments rather than (just) modeling.
6.6. DISCUSSION

6.6.1 Machine design

In the present case, the external HEPCO bearings support the magnet via a steel ring. The dynamical contact between bearings and ring implies not only rolling but also sliding. The associated dissipation due to friction might be greatly reduced by choosing a different bearing design using low friction centralized roller bearings. Mechanical losses from the gear drive and poppet valve friction may also be reduced by design engineering efforts. As for the pumping power, a very significant part is consumed by external components. As can be seen from Figure 6.1, each bed is connected to different valves with systems of bent hoses. The cold manifold is very compact to minimize heat leaks as mentioned, but it also introduces a significant flow resistance. The fluid flows through two check valves, each with a crack pressure, and associated pressure loss, of 0.07 bar. Again, much of these losses may be avoided by relatively simple design improvements.

6.6.2 Regenerator design

When it comes to the work required to drive the fluid through the regenerator beds themselves, this might be greatly reduced by going from the packed spheres currently used to parallel plates (Nielsen et al. (2012)). Detailed analysis comparing beds of different geometries, including spheres and parallel plates, can be found in (Trevizoli (2015)). To give an estimate of the potential reduction of the pressure drop in the present case, a parallel plate regenerator with the same heat transfer effectiveness and amount of magnetocaloric material as the current packed sphere regenerator could be considered. This is done in the Appendix. In the present case, this results in a regenerator with a plate thickness of 159 $\mu$m and a plate spacing of 89.5 $\mu$m. Realizing such a geometry with the relevant MCMs may pose a greater challenge, as even small variations in the plate spacing will greatly reduce the regenerator performance (Nielsen et al. 2012). However, such a regenerator would reduce the pumping power associated with the porous beds from 4.09 W to 2.24 W as estimated in the Appendix. To give an estimated upper limit for the achievable efficiency based on the present case, an imaginary corresponding AMR machine could be considered, in which the mechanical losses are eliminated as well as the pressure drop in the external components. Furthermore, the corresponding plate regenerator with perfect stacking and lower pressure drop is considered. For simplicity, the regenerator losses are considered unchanged. This situation would correspond to a second law efficiency of 30%. However, the assumption of unchanged regenerator losses can indeed be questioned. The regenerator losses are not straightforward to quantify and require evaluation through numerical AMR modeling. In literature, detailed treatments of the entropy generation in regenerators are presented along with suggested minimization
methods (Trevizoli and Barbosa Jr. (2015), Li et al. (2008)). In the real system with non-zero pressure drop in the external components, another complication arises from the fact that the regenerator losses are somewhat interlinked with the pumping losses, as heat dissipated in the fluid takes part in the regeneration process. An ideal version of the current experiment where parasitic losses are disregarded (but regenerator pumping work and regenerator losses are not) would correspond to a second-law efficiency of 26%. Recently, a study of regenerator geometries based on numerical AMR simulations has been carried out (Lei et al. (2016)). The analysis was based on conditions quite similar to those of the experiment considered here and the results may be used to roughly estimate potential improvements of the current experimental efficiency, disregarding parasitic losses due to pressure drop over external components and mechanical losses. Firstly, the current regenerator shape is a compromise between many design choices including availability of MCM and magnet geometry. Therefore, the combination of hydraulic diameter, \( d_{h,sp} = 173 \, \mu m \) (see Appendix), and aspect ratio, \( R_a = L/\sqrt{A_c} = 4.7 \), is not optimal. If the sphere diameter is fixed, reducing the aspect ratio to 3.4 and increasing the frequency to 1.9 Hz would increase the COP by approximately 27% according to the results of the numerical study, due to both reduced pumping power and regenerator losses. Secondly, if smaller spheres could be used, corresponding to a sphere diameter of 200 \( \mu m \), an increase in COP of approximately 69% could be achieved. This would require a rather "short and fat" regenerator with an aspect ratio of 0.95 operating at a frequency of 2.3 Hz. Finally a regenerator geometry different from the packed sphere bed may be considered. If parallel plates are disregarded due to difficulties of manufacturing, a micro-channel matrix might be an option. Based on the results from the considered numerical study, this could increase the COP by approximately 120%.

### 6.6.3 Potential improvements - summary

<table>
<thead>
<tr>
<th>No.</th>
<th>Situation</th>
<th>( \eta_{2nd,AMR}, % )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Current experimental value</td>
<td>18</td>
</tr>
<tr>
<td>2</td>
<td>As No. 1 without mechanical losses and pumping power due to external components</td>
<td>26</td>
</tr>
<tr>
<td>3</td>
<td>As No. 2 with reduced pumping power due to corresponding plate regenerator</td>
<td>30</td>
</tr>
<tr>
<td>4</td>
<td>As No. 1 without mechanical losses and pumping power due to external components, regenerator with optimized aspect ratio</td>
<td>33</td>
</tr>
<tr>
<td>5</td>
<td>As No. 4 with optimized sphere diameter</td>
<td>44</td>
</tr>
<tr>
<td>6</td>
<td>As No. 4 with micro channels</td>
<td>57</td>
</tr>
</tbody>
</table>
The experimentally achieved second law efficiency of 18% may be improved in different ways as discussed above. The achievable second-law efficiencies based on the estimated improvements are given in Table 3. Here, the percentwise improvements of the second-law efficiency related to improved regenerator geometry (No. 4-6 in Table 6.3) that may be achieved according to the considered numerical study, are given with the experimental value, disregarding external pumping power and mechanical losses, as a basis. It should be noted that values in Table 3 are idealized because they do not include the efficiency of the motor or pump.

6.7 Conclusion

A series of AMR experiments with a rotary AMR prototype with varying hot side temperature and frequency has resulted in temperature spans of more than 16 K with corresponding COPs above 3, based on shaft power and pumping power as work inputs. The cooling power applied in the experiments was 81.5 W. The experiment with the highest 2nd-law efficiency, which was 18%, was used as a basis for a loss analysis. From this it is concluded that significant losses due to pressure drop in external components and to a lesser extent dissipation of mechanical work exist and may be reduced in future designs. Further reduction of pressure drop by going from the packed sphere regenerator to a corresponding parallel plate regenerator is discussed, but realizing this in practice would be a larger technological challenge. Finally, significant potential efficiency improvements due to alternative regenerator designs are estimated and discussed.

6.8 Appendix: Corresponding plate regenerator

To give a theoretical estimate of the maximum potential reduction in pressure drop over the regenerator beds and hence pumping power, a parallel plate regenerator with the same size and amount of magnetocaloric material as the packed sphere regenerator is considered. To characterize the effectiveness in terms of heat transfer of a regenerator, the number of transfer units, NTU, is commonly used:

\[
NTU = \frac{Nu \cdot k_f \cdot a_s \cdot A_c \cdot L}{\dot{m} \cdot c_f \cdot \dot{d}_h}
\]

where

\(Nu = \) Nusselt number

\(k_f = \) thermal conductivity of the fluid

\(a_s = \) volume specific surface area of solid regenerator matrix

\(A_c = \) cross sectional area of bed

\(L = \) length of beds
\[\dot{m} = \text{mass flow rate}\]
\[c_f = \text{mass specific heat capacity}\]
\[d_h = \text{hydraulic diameter}\]

To give a reasonable comparison, this number will be kept the same for the imaginary parallel plate regenerator beds (NTU\text{pl}) as it is for the current packed sphere beds (NTU\text{sp}). For both the sphere bed and plate bed, the volume specific surface area can be expressed as

\[a_s = \frac{4\varepsilon}{d_h}\]  \hspace{1cm} (6.8)

where
\[\varepsilon = \text{porosity}.\]

For the sphere bed, the hydraulic diameter, \(d_{h,sp}\), can be expressed in terms of the porosity and sphere diameter:

\[d_{h,sp} = \frac{2\varepsilon_{sp}}{3(1 - \varepsilon_{sp})}d_s\]  \hspace{1cm} (6.9)

where
\[d_s = \text{sphere diameter}\]
\[\varepsilon_{sp} = \text{sphere bed porosity}\]

Combining Equations 6.7-6.9 yields, for the sphere beds,

\[NTU_{sp} = Nu_{sp} \cdot k_f \cdot A_c \cdot L \cdot \frac{9 \cdot (1 - \varepsilon_{sp})^2}{\dot{m} \cdot c_f \cdot \varepsilon_{sp} \cdot d_s^2}\]  \hspace{1cm} (6.10)

where
\[Nu_{sp} = \text{Nusselt number for the sphere bed}\]

For the plate bed, combining Equations 6.7 and 6.8 yields

\[NTU_{pl} = Nu_{sp} \cdot k_f \cdot A_c \cdot L \cdot \frac{4 \cdot \varepsilon_{pl}}{\dot{m} \cdot c_f \cdot d_{h,pl}^2}\]  \hspace{1cm} (6.11)

where
\[Nu_{pl} = \text{Nusselt number for the plate bed}\]
\[\varepsilon_{pl} = \text{porosity of the plate bed}\]
\[d_{h,pl} = \text{hydraulic diameter for the plate bed}\]

Besides requiring NTU\text{pl} = NTU\text{sp}, it is assumed that the amount of MCM is the same in the two cases, i.e. \(\varepsilon_{pl} = \varepsilon_{sp} = \varepsilon\). By combining Equations 6.10 and 6.11 with these requirements, the hydraulic diameter of the plate bed becomes

\[d_{h,pl} = \frac{2}{3} \cdot \frac{\varepsilon}{1 - \varepsilon} \cdot \sqrt{\frac{Nu_{pl}}{Nu_{sp}}} \cdot d_s.\]  \hspace{1cm} (6.12)
An estimate of the Nusselt number for the sphere bed can be given (Wakao and Kaguei (1982)) as

\[ \text{Nu}_{sp} = 2 + 1.1 \cdot \text{Pr}^{1/3} \cdot \text{Re}^{0.6} \]  \hspace{1cm} (6.13)

where
- \( \text{Pr} = c_f \cdot \mu_l / k_l \) = Prandtl number
- \( \text{Re} = \dot{m} \cdot d_{h,sp} / (A_c \cdot \mu_l) \) = Reynolds number
- \( \mu_l \) = fluid viscosity.

For the case of the experiment considered in the loss analysis in the present paper, the Nusselt number from Equation 6.132 becomes \( \text{Nu}_{sp} = 7.96 \). For the plate bed, an estimated \( \text{Nu}_{pl} = 7.541 \) is used, ignoring entrance effects (Nikolay and Martin 2002). By using \( \varepsilon = 0.36 \) (packed spheres) and \( d_s = 460 \mu m \) (average sphere diameter), the hydraulic diameter for the corresponding parallel plate bed, from Equation 6.12, becomes \( d_{h,pl} = 179 \mu m \). This corresponds to a plate spacing of

\[ s_{pl} = \frac{1}{2} d_{h,pl} = 89.5 \mu m, \]

and a plate thickness of

\[ t_{pl} = s_{pl} \cdot (1/\varepsilon - 1) = 159 \mu m. \]

For such a parallel plate regenerator, an estimate of the pressure drop may be given (Bejan (1995)) as

\[ \Delta P = 4 \cdot f_F \rho \cdot \frac{(\nu_s/\varepsilon)^2}{2 \cdot d_{h,pl}} \cdot L \]  \hspace{1cm} (6.14)

where
- \( \nu_s \) = superficial velocity, see Equation 6.5
- \( f_F = 24 / \text{Re} \) = friction factor

Here, the Reynolds number is defined as

\[ \text{Re} = \rho l \cdot \nu_s / \varepsilon \cdot d_{h,pl} / \mu_l. \]  \hspace{1cm} (6.15)

Based on this, a total pressure drop over the AMR machine in the case of the parallel plate regenerator and the present experiment becomes \( \Delta P_{AMR} = 0.538 \text{ bar} \), corresponding to a pumping power of \( W_{pump} = 2.24 \text{ W} \).

### 6.9 Addendum

#### 6.9.1 Regenerator entropy generation

Much of the power consumption of the AMR machine is assigned to regenerator losses. One way of addressing these losses is by evaluating entropy
CHAPTER 6. EXPLORING THE EFFICIENCY POTENTIAL...

generation from AMR modeling (Lei et al. (2016), Trevizoli and Barbosa Jr. (2015)). The present analysis is primarily based on Lei (2015) and Li et al. (2008). A 1D regenerator model\(^1\) corresponding to Figure 1.5 with Equations (1.5) and (1.6) was used to model a situation roughly corresponding to the experiment considered above, i.e. with \(f_{\text{AMR}} = 0.61 \text{ Hz}, V = 2.5 \text{ L/min} \), \(T_H = 19.5^\circ \text{C}\) and \(\Delta T = 15.5^\circ \text{C}\). As MCM, material properties, data from the Weiss mean field model for gadolinium were used (Bjørk (2010)), with shifted Curie temperatures to match those of the regenerator layers.

From the equations it is clear that entropy is generated inside the regenerator during different processes:

- Heat transfer between fluid and solid,
- Viscous dissipation due to flow resistance,
- Dispersion
- Axial heat conduction.

From the model, entropy generation rates relating to each of these processes may be calculated using expressions established below, using the terminology from Section 1.3.1.

The volume specific heat transfer between fluid and solid in the matrix can be written as

\[
d\dot{Q}''' = h_a(T_f - T_s), \tag{6.16}
\]

i.e. the corresponding rate of entropy generation is

\[
d\dot{S}_{ht}' = \frac{d\dot{Q}'''}{T_f} - \frac{dT}{T_f} = h_a(T_f - T_s)\left(\frac{1}{T_f} - \frac{1}{T_s}\right) \Leftrightarrow d\dot{S}_{ht}' = h_a(T_f - T_s)^2 \frac{1}{T_f T_s}. \tag{6.17}
\]

By integrating over the regenerator length, multiplying with \(A_C\), integrating over and dividing by the AMR cycle time, \(\tau\), the corresponding total entropy generation rate is obtained as

\[
\dot{S}_{ht} = \frac{1}{\tau} \int_0^\tau \int_0^L h_a A_C \frac{(T_f - T_s)^2}{T_f T_r} dx dt. \tag{6.18}
\]

The entropy generation arising from axial heat conduction through the regenerator is treated by considering an infinitesimal section of length \(dx\) experiencing a temperature difference \(dT\) in the \(x\) direction see Figure 6.5.

\(^1\)The model used in this section, including the entropy generation terms, was implemented by T. Lei, DTU Energy.
The corresponding entropy generation rate is then

\[
d\dot{S}'' = \frac{\dot{Q}''}{T + \frac{dT}{dx} dx} \frac{\dot{Q}''}{T} - \frac{\dot{Q}''}{T(T + \frac{dT}{dx} dx)} (6.19)
\]

\[
= \frac{T\dot{Q}'' - (T + \frac{dT}{dx} dx)\dot{Q}''}{T(T + \frac{dT}{dx} dx)} (6.20)
\]

\[
= \frac{T(-k\frac{dT}{dx}) - (T + \frac{dT}{dx} dx)(-k\frac{dT}{dx})}{T^2 + T\frac{dT}{dx} dx} (6.21)
\]

\[
= \frac{k(\frac{dT}{dx})^2 dx}{T^2 + TdT}. (6.22)
\]

As \(TdT << T^2\), this may to a good approximation be reduced to

\[
d\dot{S}'' = \frac{k(\frac{dT}{dx})^2}{T^2} dx. (6.23)
\]

The rate of entropy generation over the entire regenerator length averaged over an AMR cycle due to axial conduction, \(\dot{S}_{\text{cond}}\), and equivalently dispersion, \(\dot{S}_{\text{disp}}\), may then be expressed as

\[
\dot{S}_{\text{cond}} = \frac{1}{\tau} \int_0^\tau \int_0^L k_{\text{eff}} A_c \frac{1}{T_s} \left(\frac{dT_s}{dx}\right)^2 dx dt (6.24)
\]

and

\[
\dot{S}_{\text{disp}} = \frac{1}{\tau} \int_0^\tau \int_0^L k_{\text{disp}} A_c \frac{1}{T_f} \left(\frac{dT_f}{dx}\right)^2 dx dt (6.25)
\]

Finally, the rate of entropy generation due to the flow resistance in the bed can be written as (Li et al. (2008)):
\[ d\dot{S}_{res} = \frac{-\dot{m}}{\rho_f T_f} dp \Rightarrow \] (6.26)

\[ \frac{d\dot{S}_{res}}{dx} = \frac{-\dot{m}}{\rho_f T_f} \frac{dp}{dx}, \] (6.27)

and the corresponding total entropy generation averaged over a complete AMR cycle may be formulated as

\[ \dot{S}_{resist} = \frac{1}{\tau} \int_0^\tau \int_0^L -\dot{m} \frac{dp}{dx} \frac{1}{\rho_f T_f} dx dt. \] (6.28)

Figure 6.6: Simulated entropy generation rates corresponding to the experiment discussed in the loss analysis of the present chapter.

In the present case, the entropy generation in the regenerator is plotted in Figure 6.6. It is seen to be dominated by the convectional heat transfer between solid and fluid. From this, it could be suggested to go to smaller spheres to improve the heat transfer. This would, however, increase the flow resistance. Another way to go is towards other regenerator geometries as discussed in Section 6.6. Parallel plates seem like a relevant option. In this case, the flow is laminar and unidirectional which may be associated with a lower entropy generation rate compared to the packed sphere bed. However, to obtain a good heat transfer, the plate spacing should be low according to the analysis presented in Section 6.8. For such low channel widths, even small variations in plate spacing will significantly decrease the heat transfer (Nielsen et al. (2012)). Promising experimental results have been reported
for a parallel plate gadolinium regenerator which was manufactured using laser welding technology (Tusek et al. (2013b)). While regenerator geometries such as parallel plates or even micro-channels may not be feasible due to tolerance requirements, packed screen beds might be an interesting alternative for future investigations even though new manufacturing challenges arise (Lei et al. (2016)).
Chapter 7

Outlook - perspectives and applications

In the present chapter, plans and ideas for future work and applications are given with the developed prototype, also known as MAGGIE (Mechanical Alternatives Giving Greatly Improved Efficiency), as a starting point. Furthermore, a developed spin-off valve technology, is briefly described. Finally, a more general assessment is made regarding the presented work and status of the technology.

7.1 Future work on MAGGIE

So far, the experiments with the highest 2nd-law efficiencies (up to 18%) are the ones plotted in Figure 6.2. This was obtained after carefully balancing the flow resistances, as described in Chapter 5. At this point, the machine had been operated for hundreds of hours. At some point, a bad combination of lubrication oil and dust from worn aluminium cam rings have lead to the partial failure of some of the poppet valves. These aluminium rings have now been replaced with nylon rings that do not release this kind of dust, and new poppet valves are being installed. Furthermore, a larger pump for the heat transfer fluid is going to be implemented (so far the utilization has not exceeded 0.4). It is expected that these combined actions will allow for a more comprehensive experimental characterization covering the entire parameter space of varying frequency, flow rate, hot side temperature and applied cooling load.

7.1.1 Flow profile tests

As explained in Section 5.7, different flow profiles may be desired in different situations. One profile could be optimized for high cooling power, another for high COP during low cooling power operation. As described in Eriksen
et al. (2015a) (Appendix A.1), a variable cam profile could be applied. This is illustrated in Figure 7.1, where two cam profiles are illustrated. In between these profiles there can be a continuous transition. With such a cam ring, the operating conditions may be altered during operation to match the requirements at any time. It is desirable to test this concept with MAGGIE. Such a variable cam ring may be manufactured by rapid prototyping. However, an extra mechanism for axial displacement of the ring would have to be applied, which will impose significant practical challenges. Therefore, the short term plan is to test several fixed-profile rings to experimentally investigate the influence of flow profiles on the AMR performance. In the future, it would also be interesting to test a plate regenerator with a low pressure drop, allowing for flow profiles with longer ramp fractions as discussed in Section 3.7.2.

Figure 7.1: Variable cam profile concept. From Eriksen et al. (2015a)

7.2 Modeling

So far, numerical modeling in both 1D and 2D has been used as a tool for design and optimization as well as for investigating various issues emerging during the experimental work with MAGGIE. Although the models capture the trends of the performance as function of operating parameters, no attempts of accurately predicting the experiments one-to-one have been
made, primarily due to the fact that the exact material properties of the Gd$_{1-x}$Y$_x$ alloys are not known. Although the magnetic entropy changes as functions of temperature and applied field were measured, see Section 3.7.3, further characterization of the MCMs is necessary. The described setup for measuring entropy change can also be applied for measuring heat capacity versus temperature and magnetic field (Jeppesen (2008)), automatized with the mechanically variable field source described in Section 3.7.3. Once this is completed, the measured data will have to be prepared for use in the AMR models. The aim of this is to obtain a model with the capability of realistically predicting the performance of MAGGIE for varying operating conditions. To do this, it is expected to include parasitic losses which may be estimated from already obtained experimental data.

7.2.1 Refrigerator with a cold storage

Although promising performance with respect to refrigerator efficiency have been demonstrated with MAGGIE and routes towards further improvements have been discussed, the Achilles heel of magnetic refrigeration is still the cost of permanent magnets. Assuming that the efficiency of magnetic refrigerators in the future may exceed those obtained by conventional technologies, the capital cost per unit is expected to be significantly larger. One way of bringing this down is to look for applications in which it makes sense to operate via a thermal storage. One example of a possible application is a restaurant which is open for a number of hours every night. During this period the required cooling power would be much higher than during the remaining time. Rather than having a large AMR machine capable of providing the high cooling load, it might be beneficial to have a smaller machine running continuously, cooling down a thermal storage medium for the application to draw on during peak load conditions. The feasibility of this concept is confirmed by a recent numerical study by Bjørk et al. (2016). A demonstration setup in which MAGGIE is connected to a small cooling cabinet via a tank of water acting as a storage medium has been made, see Figure 7.2. Via a secondary loop, water from this tank is circulated through a heat exchanger with a fan inside a refrigeration cabinet. When the temperature in the cabinet drops below a certain value set by a thermostat, the pump driving this secondary loop kicks in. This demonstration setup has been used for serving magnetically cooled beverages at various events at DTU Energy. Temperatures measured on the hot and cold sides of MAGGIE together with the cabinet temperature during such an event are plotted in Figure 7.2. The temperature of the cold storage, which was in between the cold “MAGGIE temperature” and the “Cabinet temperature” was not measured in this case. While this setup is merely meant for demonstration, a detailed study of a refined version is planned as future work.
7.3 The Curie valve - a spin-off technology

During the activities within magnetic refrigeration, an alternative idea for utilizing the phase transition between ferromagnet and paramagnet around
7.3. THE CURIE VALVE - A SPIN-OFF TECHNOLOGY

The basic idea is attributed to C. R. H. Bahl, A. Smith and N. Pryds

The Curie temperature came up\(^1\). The idea was to utilize the variations in force between a magnet and a piece of MCM that occur due to this phase transition to actuate a valve as a response to a temperature change. Based on this idea, a novel shunt valve technology was developed by Eriksen et al. (2013), see Appendix B.1. This small detour is included here to serve as an example a spinoff technology popping up unexpectedly from the main research activities.

![Diagram of the developed shunt valve technology](image)

**Figure 7.3**: Principle of the developed shunt valve technology with two different set points. The figure is published in Eriksen et al. (2012).

The basic principle of the shunt valve is illustrated in Figure 7.3. It has one inlet and two outlets. The distance between the MCM block and magnet is a function of the temperature dependent magnetic force and a biasing spring force. This distance determines the distribution of fluid flow between the two outlets. The operating temperature range of the valve may be altered by aligning the magnet with different parts of the internal MCM having different Curie temperatures. Based on this concept, two different prototype valves were built and tested (Eriksen et al. (2012) and Eriksen et al. (2013)). This technology ensures reliable temperature controlled fluid distribution with a fast response time without the need for a power supply. Furthermore, a unique feature is the possibility of mechanically altering the valve set-point from the outside without having a penetration of the valve housing, which might point towards applications involving volatile or toxic

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\(^1\)The basic idea is attributed to C. R. H. Bahl, A. Smith and N. Pryds
Despite these seemingly obvious advantages this technology has, however, not found its niche of application at the time of writing.

7.4 Perspectives of magnetic refrigeration

The development of MAGGIE and subsequent analysis described in the previous chapters has indeed demonstrated a significant improvement of experimentally obtained AMR device efficiency and numerous ways of further improvements have been identified. The rotary multi-bed concept has proven its viability and can be recommended for future applications. However, it is stressed that special care has to be taken to ensure a very accurately balanced flow, in order not to compromise performance.

It should be noted, that the presented prototype is designed with a quite limited amount of resources, and some aspects were prioritized higher than others. For future designs it is suggested to put more emphasis on the choice and characterization of magnetocaloric materials, which will also allow for better numerical models. After the overall conceptual layout of regenerator and magnet system, AMR modeling in combination with magnet optimization modeling should be used in as much detail as possible in the design process (i.e. by inclusion of loss terms as mentioned above). Special attention should be paid to the choice of flow profile, as a bed in a rotary multi-bed device will experience a non-homogeneous magnetization, which should also be included in the modeling as described. Such efforts will undoubtedly enable the design of better machines and hence achievable efficiencies than what has now been demonstrated, which bodes very well for the future of the technology.

However, the capital cost barrier posed by permanent magnets has to be addressed, even when considering the thermal storage concept described above. Currently, the biggest driver of the demand for NdFeB permanent magnets is the wind turbine industry. Based on the increasing demand for rare earth elements, such as neodymium, during the present transition of the energy system towards renewables, there seems to be a bottle-neck issue of opening new mines fast enough to meet the demands(Habib and Wenzel (2014)) and some system of recycling must be applied. It could be considered to utilize the fact that a permanent magnet does not degrade over time. Rather than just considering rare earth element recycling it may be suggested to consider a model in which the end consumer could lease a magnet during the life time of the refrigerator. This magnet could then be recycled as is. Either way, a very thorough life cycle assessment of a magnetic refrigerator is definitely needed.

As a final note, it is considered that the maturity of magnetic refrigeration is now at a stage, where capital from industrial players is needed to go all the way to commercialization.
Chapter 8

Conclusions

Throughout the thesis it is described how the rotary multi-bed AMR refrigeration technology is pushed further towards the objectives of attractive combinations of cooling power, temperature span and COP, as formulated in the working hypothesis. In order to address this combined objective of temperature span and COP, the 2\textsuperscript{nd}-law efficiency has been used. The maximum experimentally demonstrated value obtained with the 2\textsuperscript{nd} generation prototype (MAGGIE) was 18\% in the presented performance analysis. This is, however, not considered the global maximum for MAGGIE as the full parameter space of operating conditions is far from completely mapped out due to practical issues such as valve failures and pump limitations. Due to the latter, the system has not been tested at utilizations above approximately 0.4, which will be necessary in order to explore its potential cooling power. A highest temperature span of 29.2 K has been achieved at the time of writing.

Before it was possible to obtain these experimental results, a number of issues that were found to be crucial to the AMR performance were addressed. Special considerations were given to heat leakage as this was found to greatly limit the performance of the first DTU rotary prototype. Based on simulations and heat transfer calculations, it was chosen to include a 2 mm air gap between regenerator and magnet system components as this was shown to be beneficial from a COP standpoint even though it reduces the maximum cooling power.

Losses related to heat transfer fluid flow handling components were greatly diminished by the means of a novel poppet valve based flow system, actuated by cam rings rotating along with the magnet. The friction arising from this system accounts for only 5\% of the irreversibilities in the above-mentioned performance analysis.

The flow system furthermore enables the possibility of choosing different flow profiles to match the desired performance characteristics. So far, profiles with short ramps have been tested and most experiments were conducted
with relatively short blow periods taking up half of the cycle time, as this was, via numerical simulations, found to enable high cooling powers. Going to longer blow periods is expected to enable higher efficiency.

Through experimental studies considering the flux-conducting iron core it was shown that in the present case it did not pay off to trade iron for insulating non magnetic material. Furthermore, eddy current losses in the core were very minor with the 2 mm plate thickness used, which was confirmed by subsequent numerical simulations.

A main finding that came up during the experiments was the influence of unbalanced heat transfer fluid flow in the regenerator beds. This important effect revealed itself by the fact that very small adjustments of a valve controlling the flow in one direction through a bed had a dramatic impact on the resulting temperature span. Theoretical investigations of the influence of variations in flow resistance between the flow paths in and out of the regenerator beds indicated an average drop in cooling power of 30% and a COP reduction of 50% resulting from flow resistance variations with a standard deviation of only 5% for a typical set of operating conditions.

Although the combined work presented in the thesis has brought the technology far in the direction of higher performance as outlined in the working hypothesis, further challenges remain, the most important being the need for improved regenerator geometries. With the experiments presented in the performance analysis as a starting point, it was discussed how to achieve a 2\textsuperscript{nd}-law efficiency of 30% by replacing the packed sphere regenerator with one of parallel plates. Further improvements, up to an estimated 2\textsuperscript{nd}-law efficiency of 57%, were discussed.

A concept for utilizing the inherent characteristics of AMR refrigerators was demonstrated experimentally, by operating the machine via a thermal storage. This concept enables the use of a smaller, continuously running AMR device for a given cooling power requirement, as the peak loads may be delivered by the storage rather than the AMR machine directly.

As a special application of magnetocaloric material/permanent magnet systems, a spin-off shunt valve technology was developed alongside the magnetic refrigeration activities, enabling the distribution of a fluid flow between two paths according to its temperature. Unique flow controllability eliminating requirements for power supply or penetration of the valve housing was demonstrated with this technology.
Appendix A

Patent application

A.1 An active magnetic regenerator device

-International application published under the patent cooperation treaty (PCT)
Title: AN ACTIVE MAGNETIC REGENERATOR DEVICE

Abstract: A rotating active magnetic regenerator (AMR) device comprising two or more regenerator beds, a magnet arrangement and a valve arrangement. The valve arrangement comprises a plurality of valve elements arranged substantially immovably with respect to the regenerator beds along a rotational direction. A cam surface is arranged substantially immovably with respect to the magnet arrangement along the rotational direction, and comprises a plurality of cam elements arranged to cooperate with the valve elements in order to control opening degrees of the valve elements, in accordance with a relative position of the cam elements and the valve elements. Thereby the opening degree of each valve element is controlled in accordance with a relative angular position of the regenerator beds and the magnet arrangement.

Fig. 2
AN ACTIVE MAGNETIC REGENERATOR DEVICE

FIELD OF THE INVENTION

The present invention relates to an active magnetic regenerator device of the kind comprising two or more regenerator beds and a magnet arrangement arranged at least partly circumferentially with respect to the regenerator beds. In the active magnetic regenerator device of the invention, a flow profile of fluid flowing through the regenerator beds can be controlled to closely follow variations in the magnetic field across the magnet arrangement.

BACKGROUND OF THE INVENTION

Magnetic refrigeration and heating relies on the magnetisation and demagnetisation of magnetocaloric materials and the subsequent removal of the generated heat by a fluid flow. Active magnetic regenerator devices normally comprise a number of regenerator beds, each comprising magnetocaloric material. The regenerator beds are sequentially passed through a magnetic field generated by a magnet arrangement, e.g. comprising one or more permanent magnets. Thereby the magnetocaloric material of the regenerator beds is alternatingly magnetised and demagnetised, and heat is generated. A fluid flow is passed through each regenerator bed in order to remove the generated heat from the regenerator beds. The heat is subsequently removed from the device by means of a heat exchanger, through which the fluid passes.

The performance of such an active magnetic regenerator device is partly determined by the flow rate of the fluid passing through the regenerator beds, and also the exact timing of the fluid flow. Previous attempts to control the flow rate have relied on designing valves, which control the supply of fluid to the regenerator beds.

WO 03/050456 A1 discloses a rotating active magnetic regenerator comprising a valve system which ensures reciprocating fluid flow through the regenerator beds, in synchronization with the rotating movements of the magnet. This is obtained by means of individually rotating valves connected to the mechanism which moves the magnet.

EP 0 187 078 B1 discloses a rotating active magnetic regenerator comprising a valve system which ensures reciprocating fluid flow through the regenerator beds, in synchronization with the rotating movements of the magnet. The valve system comprises discs provided with orifices, the discs being arranged to rotate along with the magnet.
US 8,037,692 B2 discloses a rotating active magnetic regenerator comprising a valve system which ensures reciprocating fluid flow through the regenerator beds. Synchronization between the rotating movements of the magnet and the fluid flow is obtained by carefully controlling one or more solenoid valves.

Common to the prior art documents described above is, that none of the active magnetic regenerator devices described therein provide an easy manner of ensuring that the flow profile matches variations in the magnetic field.

DESCRIPTION OF THE INVENTION

It is an object of embodiments of the invention to provide an active magnetic regenerator which has an improved performance as compared to prior art active magnetic regenerators.

It is a further object of embodiments of the invention to provide an active magnetic regenerator in which the risk of leaking is minimised.

It is an even further object of embodiments of the invention to provide an active magnetic regenerator comprising a valve arrangement, in which valve friction is low.

It is an even further object of embodiments of the invention to provide an active magnetic regenerator in which flow control is fast acting.

The invention provides an active magnetic regenerator device comprising:

- two or more regenerator beds, each regenerator bed comprising magnetocaloric material, a first flow passage allowing fluid to pass through the regenerator bed along a first flow direction, and a second flow passage allowing fluid to pass through the regenerator bed along a second flow direction, the second flow direction being substantially opposite to the first flow direction,

- a magnet arrangement arranged at least partly circumferentially with respect to the regenerator beds, the magnet arrangement comprising at least two sections comprising permanent magnets,

- a first heat exchanger arranged to exchange heat with fluid received from the first flow passages of the regenerator beds, and a second heat exchanger arranged to
exchange heat with fluid received from the second flow passages of the regenerator beds, and

- a valve arrangement arranged to control fluid flow through the regenerator beds,

wherein the regenerator beds and the magnet arrangement are arranged to perform rotational movements relative to each other, and wherein the valve arrangement comprises:

- a plurality of valve elements, each valve element being arranged to control a supply of fluid to at least one regenerator bed, via the first flow passage or the second flow passage, the plurality of valve elements being arranged substantially immovably with respect to the regenerator beds along a direction of relative rotational movement, and

- at least one cam surface arranged substantially immovably with respect to the magnet arrangement along the direction of relative rotational movement, the cam surface(s) comprising a plurality of cam elements arranged along the cam surface, each cam element being adapted to cooperate with a valve element in order to control an opening degree of the valve element, in accordance with a relative position of the cam element and the valve element,

- the valve elements and the cam surface thereby being arranged to perform rotational movements relative to each other, said movements corresponding to the relative movements of the regenerator beds and the magnet arrangement, the opening degree of each valve element thereby being controlled in accordance with a relative angular position of the regenerator beds and the magnet arrangement.

The active magnetic regenerator device of the invention comprises two or more regenerator beds, a magnet arrangement, a first heat exchanger, a second heat exchanger and a valve arrangement arranged to control fluid flow through the regenerator beds.

Each of the regenerator beds comprises magnetocaloric material. The magnetocaloric material is capable of generating heat, thereby providing heating or cooling, when the magnetocaloric material is alternatingly magnetised and demagnetised. The regenerator beds may comprise a single magnetocaloric material. As an alternative, two or more different magnetocaloric materials may be applied.

Each regenerator bed further comprises a first flow passage allowing fluid to pass through the regenerator bed along a first flow direction, and a second flow passage allowing fluid to pass through the regenerator bed along a second flow direction. The first flow direction is
substantially opposite to the second flow direction. Accordingly, fluid is allowed to pass through the regenerator beds, and thereby along the magnetocaloric material. One of the fluid flows is used for removing heat generated by the magnetocaloric material in response to the magnetocaloric material being magnetised, while the other one of the fluid flows is used for supplying heat to the magnetocaloric material when the magnetocaloric material is demagnetised. This may be regarded as removing cooling from the regenerator beds.

The magnet arrangement comprises at least two sections comprising permanent magnets. The permanent magnet sections may be arranged spaced apart, e.g. with sections of another material arranged there between. This will be described further below. As an alternative, the permanent magnet sections may be arranged immediately adjacent to each other. The permanent magnet sections may be designed to provide magnetic fields which vary from one section to the other. For instance the direction of the magnetic field generated by the permanent magnet of one section may differ from the direction of the magnetic field generated by the permanent magnet of a neighbouring section.

The magnet arrangement is arranged at least partly circumferentially with respect to the regenerator beds. This should be interpreted to mean that the regenerator beds and the magnet arrangement are arranged adjacent to each other, in such a manner that either the magnet arrangement, completely or partly, surrounds the regenerator beds, or the regenerator beds, completely or partly, surround the magnet arrangement. Thus, the magnet arrangement may be arranged on the inside and/or on the outside with respect to the regenerator beds. Furthermore, the magnet arrangement may be arranged along the entire circumference (inside or outside) of the regenerator beds, or it may be arranged along only a part of the circumference of the regenerator beds. For instance, the magnet arrangement may have a ‘C’ shape, leaving a part of the circumference of the regenerator beds, which is not ‘covered’ by the magnet arrangement.

Soft magnetic material may be arranged according to the magnet arrangement in such a manner that it acts to ensures magnetic flux closure of the magnetic circuit. Thus, in the case where the magnet arrangement, completely or partly, surrounds the regenerator beds the soft magnetic material may be arranged inside the regenerator beds. Alternatively, in the case where the regenerator beds, completely or partially, surrounds the magnet arrangement the soft magnetic material may be arranged at least partly circumferentially with respect to the regenerator beds.

The first heat exchanger is arranged to exchange heat with fluid received from the first flow passages of the regenerator beds, and the second heat exchanger is arranged to exchange heat with fluid received from the second flow passages of the regenerator beds. As described
above, one of the fluid flows removes heat from the regenerator beds, while the other fluid flow supplies heat to, or removes cooling from, the regenerator beds. Thus, the heat exchanger which is arranged to exchange heat with the fluid flows which remove heat from the regenerator beds, is a heat rejecting heat exchanger, in the sense that it ensures that the heat removed from the regenerator beds is transferred out of the system. Similarly, the heat exchanger which is arranged to exchange heat with the fluid flows which remove cooling from the regenerator beds, is a heat consuming heat exchanger, in the sense that it ensures that the cooling effect removed from the regenerator beds is transferred out of the system.

When the fluid has passed through one of the heat exchangers it may advantageously be led back through the regenerator beds, via the other flow passage. Thus, the fluid flow through the system may be as follows: First flow passage of a regenerator bed; first heat exchanger; second flow passage through a regenerator bed; second heat exchanger; etc.

The valve arrangement is arranged to control the fluid flow through the regenerator beds. This is necessary, because, for a given regenerator bed, whether fluid is allowed to pass through the first or the second flow passage, must be synchronized with the magnetisation and demagnetisation of the magnetocaloric material of the regenerator bed.

The regenerator beds and the magnet arrangement are arranged to perform rotational movements relative to each other. This could be obtained by allowing the regenerator beds to be stationary, while allowing the magnet arrangement to rotate; by allowing the magnet arrangement to be stationary, while allowing the regenerator beds to rotate; or by allowing the magnet arrangement as well as the regenerator beds to rotate, e.g. in opposite directions and/or at different rotational speeds. Accordingly, the active magnetic regenerator device of the invention is of a rotational type.

When the regenerator beds and the magnet arrangement rotate relative to each other, and due to the circumferential relative position of the regenerator beds and the magnet arrangement, each of the regenerator beds will sequentially be arranged in the magnetic field generated by each of the permanent magnets of the permanent magnet sections. This causes the magnetocaloric material of the regenerator beds to be sequentially magnetised and demagnetised.

The valve arrangement comprises a plurality of valve elements and at least one cam surface. Each of the valve elements is arranged to control a supply of fluid to a regenerator bed, via the first flow passage or the second flow passage. Thus, each of the flow passages through the regenerator beds is provided with a valve element which controls the supply of fluid to that passage. Accordingly, for a given regenerator bed and at a given time, the
corresponding valve elements determine whether fluid is passing through the first flow passage or the second flow passage, as well as the flow rate of the fluid passing through the first or second flow passage.

The valve elements are arranged substantially immovably with respect to the regenerator beds along a direction of relative rotational movement. Thus, when the regenerator beds and the magnet arrangement perform relative rotational movements, the valve elements, as a general rule, move along with the regenerator beds or remain stationary along with the regenerator beds, i.e. the relative position of the valve elements and the regenerator beds remains substantially fixed. This is an advantage, because thereby it is not necessary to provide sealing between inlets to the flow passages of the regenerator beds and valve elements which are moving with respect to the inlets. Thereby the risk of leaking from the active magnetic regenerator device is minimised. However, it is not ruled out that it is possible to perform small adjustments of the position of the valve elements relative to the regenerator beds, along the direction of relative rotational movement. This could, e.g., be used for adjusting the timing of the operation of the valves. This could, e.g., be relevant in order to take fluid inertial effects into account during high frequency operation. Furthermore, the valve elements may not be fixed relative to the regenerator beds along an axial direction defined by an axis of rotation of the relative rotational movement.

The cam surface(s) comprise(s) a plurality of cam elements arranged along the cam surface. Each cam element is adapted to cooperate with a valve element in order to control an opening degree of the valve element, in accordance with a relative position of the cam element and the valve element. Thus, the opening degree of each of the valve elements can be adjusted by adjusting the relative position between the cam surface, and thereby each of the cam elements, relative to the valve elements.

The cam surface is arranged substantially immovably with respect to the magnet arrangement along the direction of relative rotational movement. Thus, when the regenerator beds and the magnet arrangement perform relative rotational movements, the cam surface, and thereby the cam elements, as a general rule, move along with the magnet arrangement or remain stationary along with the magnet arrangement, i.e. the relative position of the cam surface and the magnet arrangement remains substantially fixed. However, it is not ruled out that it is possible to perform small adjustments of the position of the cam surface relative to the magnet arrangement, along the direction of relative rotational movement. This could, e.g., be used for adjusting the timing of the operation of the valves. This could, e.g., be relevant in order to take fluid inertial effects into account during high frequency operation. Furthermore, the cam surface may not be fixed relative to the magnet arrangement along an axial direction defined by an axis of rotation of the relative rotational movement.
Accordingly, when the regenerator beds and the magnet arrangement perform rotational movements relative to each other, the valve elements and the cam surface perform corresponding rotational movements relative to each other. Since the opening degree of each of the valve elements is determined by the relative position of the valve element and the corresponding cam element, the opening degree of each of the valve elements is determined by the relative position of the regenerator beds and the magnet arrangement. As a consequence, the opening degree of each of the valve elements, and thereby the flow rate of fluid supplied to the flow passages of the regenerator beds, is automatically synchronized with the relative position of the magnet arrangement and the regenerator beds, and thereby with the sequential magnetisation and demagnetisation of the magnetocaloric material of the regenerator beds. Furthermore, the cam elements can be designed in a manner which closely matches the magnetic field profile provided by the permanent magnets. This improves the performance of the active magnetic regenerator device.

Providing a fluid supply to the flow passages of the regenerator beds by means of interacting valve elements and cam elements is, furthermore, a very simple design, which reduces the risk of faults occurring during operation.

The regenerator beds may be stationary and the magnet arrangement may be adapted to perform rotational movements. According to this embodiment, the regenerator beds and the valve elements do not perform rotational movements. This is an advantage, because thereby the fluid supply to the valve arrangement can also be kept stationary, thereby further reducing the risk of leaks. However, as described above, it could also be envisaged that the regenerator beds could perform rotational movements.

Each cam element may have a shape which reflects variations in a magnetic field generated by a permanent magnet section of the magnet arrangement, the valve elements thereby defining a fluid flow profile which is chosen according to a magnetic field profile of the permanent magnet. According to this embodiment, the cam surfaces are carefully designed to provide a desired fluid flow profile which closely follows the variations in the magnetic field as the regenerator bed passes through the magnetic field generated by a permanent magnet of the magnet arrangement. For instance, the cam element may have a profile which causes the opening degree of the valve element to increase gradually in a manner which follows an increase in the magnetic field as the regenerator bed is moved into alignment with a permanent magnet section.

The cam surface may comprise at least a first region and a second region, and the first region may comprise a plurality of cam elements having a first shape, and the second region may comprise a plurality of cam elements having a second shape, the second shape differing
from the first shape, and the cam surface may be movable between a first position in which the first region is arranged in contact with the valve elements and a second position in which the second region is arranged in contact with the valve elements. According to this embodiment, the cam surface defines at least two different flow profile patterns, one corresponding to the cam elements of the first region and one corresponding to the cam elements of the second region. It is possible to choose between the flow profile patterns, simply by moving the cam surface between the first position and the second position. For instance, one of the flow profile patterns may be designed to provide a “high cooling power” flow mode, while another flow profile pattern may be designed to provide a “high efficiency” flow mode. As an alternative, one of the flow profile patterns may be designed to meet requirements during start-up of the active magnetic regenerator device, while another flow profile pattern may be designed to meet requirements during normal operation of the active magnetic regenerator device.

The cam surface may further comprise an intermediate region arranged between the first region and the second region, said intermediate region defining substantially continuous cam sections interconnecting the cam elements of the first region with cam elements of the second region. According to this embodiment, the active magnetic regenerator device may be operated in two modes, i.e. the mode defined by the cam elements of the first region and the mode defined by the cam elements of the second region. However, due to the intermediate region, the cam surface can be moved between first position and the second position, thereby switching between the two modes, without having to stop operation of the active magnetic regenerator device.

As an alternative, each of the cam elements may have a shape which varies substantially continuously along a substantially axial direction. According to this embodiment, the part of the cam elements, which is arranged in contact with the valve elements, can be continuously changed by moving the cam surface along the substantially axial direction. Since the shape of the cam elements varies continuously, this causes a continuous change in the flow profile of the fluid flow provided by the valve elements. Thus, according to this embodiment, the flow profile can be continuously adjusted to meet the exact requirements under the given circumstances, simply by selecting an appropriate axial position of the cam surface.

The valve elements may be poppet valves. Poppet valves are very suitable for this purpose, because they can easily be controlled by means of a cam surface. As an alternative, other suitable kinds of valves could be used.

The cam surface may form an inner surface or an outer surface of a ring shaped member.

According to this embodiment, the active magnetic regenerator device may advantageously
have a cylindrically symmetrical shape, where the regenerator beds and/or the magnet arrangement is/are arranged to rotate about the axis defined by the cylindrical shape, and with the regenerator beds and the magnet arrangement arranged adjacent to each other along the radial direction defined by the cylindrical shape. In the case that the regenerator beds are arranged closer to the axis than the magnet arrangement, then the cam surface may advantageously form an inner surface of a ring shaped member, since such an inner surface will, in this case, face the regenerator beds, and thereby the valve elements. On the other hand, in the case that the magnet arrangement is arranged closer to the axis than the regenerator beds, then the cam surface may advantageously form an outer surface of a ring shaped member.

The magnet arrangement may further comprise at least two sections of soft magnetic material, arranged between the permanent magnet sections. According to this embodiment, the regenerator beds will alternatingly be moved past a permanent magnet section, thereby magnetising the magnetocaloric material, and a soft magnetic material section, thereby demagnetising the magnetocaloric material. Furthermore, the permanent magnet sections are spaced apart by the sections comprising soft magnetic material.

The regenerator beds may be arranged annularly. The regenerator beds may further be arranged angularly equidistantly.

Similarly, the permanent magnet sections may be arranged annularly. The permanent magnet sections may further be arranged angularly equidistantly.

The active magnetic regenerator device of the invention is equally suitable for being used as a cooling system, in the form of a refrigeration system or the like, or as a heating system, in the form of a heat pump, such as a ground source heat pump.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will now be described in further detail with reference to the accompanying drawings in which

Fig. 1 is an isometric view of an active magnetic regenerator device according to an embodiment of the invention,

Fig. 2 is a cross sectional view of the active magnetic regenerator device of Fig. 1,
Fig. 3 is an isomeric view of a valve arrangement for the active magnetic regenerator device of Figs. 1 and 2,

Fig. 4 is a diagrammatic view of an active magnetic regenerator device according to an embodiment of the invention, illustrating the flow of fluid through the regenerator beds, and

Fig. 5 illustrates a valve arrangement for an active magnetic regenerator device according to an embodiment of the invention.

DETAILED DESCRIPTION OF THE DRAWINGS

Fig. 1 is an isomeric view of an active magnetic regenerator device 1 according to an embodiment of the invention. The active magnetic regenerator device 1 comprises a stationary part comprising a number of regenerator beds (not visible) and a valve arrangement 2. The valve arrangement 2 is fluidly connected to the regenerator beds in such a manner that it supplies fluid to flow passages defined in the regenerator beds. The valve arrangement 2 is, thus, connected to an external flow circuit (not shown) via fluid connections 3.

The valve arrangement 2 further comprises a number of poppet valves 4, each being connected to a flow passage of a regenerator bed. A given poppet valve 4 thereby defines the fluid flow supplied to the corresponding flow passage. This will be described in further detail below.

The active magnetic regenerator device 1 further comprises a rotating part comprising a magnet arrangement (not shown) and a cam ring 5 defining a cam surface formed on an inner surface of the cam ring 5. The magnet arrangement and the cam ring 5 rotate together relatively to the regenerator beds and the valve arrangement 2 during operation of the active magnetic regenerator device 1. The cam surface formed on the cam ring 5 comprises a number of cam elements (not shown). The cam elements are arranged in contact with the poppet valves 4 of the valve arrangement in such a manner that the relative angular position of the cam ring 5 and the poppet valves 4 defines an opening degree of each of the poppet valves 4. Thereby the relative angular position of the cam ring 5, and thereby the magnet arrangement, and the poppet valves 4, and thereby the regenerator beds, determines the opening degree of each of the poppet valves 4.

Fig. 2 is a cross sectional view of the active magnetic regenerator device 1 of Fig. 1. In Fig. 2 the regenerator beds 6 and the magnet arrangement 7 can be seen.
Each of the regenerator beds 6 comprises magnetocaloric material, and a first flow passage and a second flow passage. Furthermore, soft magnetic material 16 is arranged in each regenerator bed 6, in such a manner that the magnetocaloric material is arranged circumferentially with respect to the soft magnetic material in each regenerator bed 6. The soft magnetic material 16 ensures magnetic flux closure of the magnetic circuit comprising the magnets and the soft magnetic material of the magnet arrangement 7 and the magnetocaloric material in the regenerator beds 6.

One of the flow passages extends from the valve arrangement 2 towards an opposite end of the regenerator bed 6. The other of the flow passages extends from the opposite end of the regenerator bed 6 towards the valve arrangement 2.

The magnet arrangement 7 comprises a number of sections comprising permanent magnets, spaced apart by means of sections of soft magnetic material. Thus, when the magnet arrangement 7 is rotated relative to the regenerator beds 6, each regenerator bed 6 is alternatingly arranged adjacent to a permanent magnet section and a soft magnetic material section of the magnet arrangement. Thereby the magnetocaloric material of the regenerator beds 6 is alternatingly magnetised and demagnetised. Furthermore, when some of the regenerator beds 6 are arranged adjacent to a permanent magnet section, other regenerator beds 6 will be arranged adjacent to a soft magnetic material section of the magnet arrangement.

When the magnetocaloric material is magnetised, it generates heat, and when it is demagnetised, it absorbs heat. Thus, when a regenerator bed 6 is arranged adjacent to a permanent magnet section, a fluid flow must be provided through the flow passages of the regenerator bed, which removes the generated heat. Similarly, when the regenerator bed 6 is arranged adjacent to a soft magnetic material section of the magnet arrangement, a fluid flow must be provided through the flow passages of the regenerator bed 6, which supplies heat to, or removes cooling from, the regenerator bed 6. Therefore it is important that the supply of fluid to the first and second flow passages of each regenerator bed 6 is synchronized with the movements of the magnet arrangement 7, relative to the regenerator beds 6, in order to provide an efficient active magnetic regenerator device 1. Accordingly, the heat transfer between the regenerator beds 6 and the fluid is driven by the change in temperature of the magnetocaloric material of the regenerator beds 6, due to the alternating magnetisation and demagnetisation of the magnetocaloric material.

Since the cam ring 5 rotates along with the magnet arrangement 7, the cam elements of the cam surface are moved relative to the poppet valves 4 in the same manner. Furthermore, since the cam elements and the poppet valves 4 cooperate in controlling the opening degrees
of the poppet valves 4, according to their relative position, the opening degree of each poppet valve 4 is automatically controlled in a manner which is synchronized with the magnetisation and demagnetisation of the magnetocaloric material of the regenerator beds 6. Thereby it is also automatically ensured that the fluid flow to the first and second flow passages of each regenerator bed 6 is synchronized with the magnetisation and demagnetisation of the magnetocaloric material.

Furthermore, since the poppet valves 4 are stationary, i.e. they are immovable with respect to the regenerator beds 6, and thereby with respect to flow passages of the regenerator beds 6, the valves do not comprise parts which are moving relative to each other, and which therefore need sealing there between. This minimises the risk of leaking.

Fig. 3 is an isometric view of a valve arrangement 2 for the active magnetic regenerator device 1 of Figs. 1 and 2. A cam ring 5 can be arranged adjacent to the valve arrangement 2. The cam ring 5 may be divided into two parts, one arranged in abutment with a first subset of the poppet valves 4a, and the other arranged in abutment with another subset of the poppet valves 4b. The first subset of poppet valves 4a control fluid supply to the first flow passages of the regenerator beds, and the second subset of poppet valves 4b control fluid supply to the second flow passages of the regenerator beds.

It is easy to imagine how cam surfaces formed on the inner side of the cam ring can be arranged in abutment with the poppet valves 4, in such a manner that cam elements formed on the cam surfaces determine how much the poppet valves 4 are depressed, and thereby the opening degrees of the poppet valves 4.

Fig. 4 is a diagrammatic view of an active magnetic regenerator device 1 according to an embodiment of the invention, illustrating the flow of fluid through the regenerator beds 6. Four regenerator beds 6 are shown. A pump 8 is arranged to drive the fluid flow through the active magnetic regenerator device 1. The active magnetic regenerator device 1 illustrated in Fig. 4 could, e.g., be the active magnetic regenerator device 1 shown in Figs. 1 and 2.

Each of the regenerator beds 6 comprises a first flow passage 9 and a second flow passage 10. Fluid flowing through the first flow passages 9 is arranged to remove heat generated by the magnetocaloric material of the regenerator beds 6, and fluid flowing through the second flow passages 10 is arranged to supply heat to, i.e. to remove cooling from, the magnetocaloric material of the regenerator beds 6. The fluid flow through the active magnetic regenerator device 1 is as follows.
When a given regenerator bed 6 is moved into the magnetic field generated by one of the permanent magnets of the magnet arrangement, the cam surface simultaneously rotates to operate a poppet valve 4 which supplies fluid to the first flow passage 9 of that regenerator bed 6. As the regenerator bed 6 is moved into the magnetic field, the poppet valve 4 is opened in a manner which corresponds to the magnetic field experienced by the regenerator bed 6.

The fluid is then allowed to pass through the regenerator bed 4, via the first flow passage 9. Thereby heat generated by the magnetocaloric material, which has been magnetised by the magnetic field, is removed by the fluid, i.e. the fluid is heated. When the fluid leaves the first flow passage 9, it is passed, via the pump 8, through a first heat exchanger 11, where the fluid is cooled, thereby removing the heat from the system.

Simultaneously, some of the other regenerator beds 6 are arranged adjacent to a section of the magnet arrangement which comprises soft magnetic material, and the magnetocaloric material of these regenerator beds 6 is therefore demagnetised. The cam surface is in a position which opens the poppet valves 4 which supply fluid to the second flow passages 10 of these regenerator beds 6.

Therefore, the fluid leaving the first heat exchanger 11 is allowed to pass through these regenerator beds 6, via the second flow passages 10. Thereby heat is supplied to, or cooling is removed from, the magnetocaloric material of these regenerator beds 6. Finally, the fluid is passed through a second heat exchanger 12, before it is once again passed through first flow passages 9 of some of the regenerator beds 6. In the second heat exchanger 12 the fluid is heated. Check valves 15 are provided opposite to the poppet valves 4 in order to ensure unidirectional flow in the second heat exchanger 12.

The fluid flow through a given regenerator bed 6 is alternated between being passed through the first flow passage 9 and the second flow passage 10, in synchronization with the regenerator bed 6 being arranged adjacent to a permanent magnet section or adjacent to a section of the magnet arrangement comprising soft magnetic material, due to the cooperation between the poppet valves 4 and the cam elements of the cam surface, as described above.

Fig. 5 illustrates a valve arrangement 2 and a cam surface 13 for an active magnetic regenerator device according to an embodiment of the invention. A cam ring 5 is arranged adjacent to the valve arrangement 2. The inner surface of the cam ring 5 forms a cam surface 13 which is arranged in abutment with the poppet valves 4 of the valve arrangement 2.
The cam surface 13 defines two cam profiles 13a and 13b. A first cam profile 13a is shown as a solid line, and a second cam profile 13b is shown as a dashed line. The cam ring 5 can be moved along an axial direction in order to position the first cam profile 13a or the second cam profile 13b in abutment with the poppet valves 4. In Fig. 5 the first cam profile 13a is arranged in abutment with the poppet valves 4.

It is clear from Fig. 5 that the relative position of the cam ring 5 and a given poppet valve 4 determines the position of a follower 14 of the poppet valve 4, and thereby determines the opening degree of the poppet valve 4.
CLAIMS

1. An active magnetic regenerator device comprising:

   - two or more regenerator beds, each regenerator bed comprising magnetocaloric material, a first flow passage allowing fluid to pass through the regenerator bed along a first flow direction, and a second flow passage allowing fluid to pass through the regenerator bed along a second flow direction, the second flow direction being substantially opposite to the first flow direction,

   - a magnet arrangement arranged at least partly circumferentially with respect to the regenerator beds, the magnet arrangement comprising at least two sections comprising permanent magnets,

   - a first heat exchanger arranged to exchange heat with fluid received from the first flow passages of the regenerator beds, and a second heat exchanger arranged to exchange heat with fluid received from the second flow passages of the regenerator beds, and

   - a valve arrangement arranged to control fluid flow through the regenerator beds,

wherein the regenerator beds and the magnet arrangement are arranged to perform rotational movements relative to each other, and wherein the valve arrangement comprises:

   - a plurality of valve elements, each valve element being arranged to control a supply of fluid to at least one regenerator bed, via the first flow passage or the second flow passage, the plurality of valve elements being arranged substantially immovably with respect to the regenerator beds along a direction of relative rotational movement, and

   - at least one cam surface arranged substantially immovably with respect to the magnet arrangement along the direction of relative rotational movement, the cam surface(s) comprising a plurality of cam elements arranged along the cam surface, each cam element being adapted to cooperate with a valve element in order to control an opening degree of the valve element, in accordance with a relative position of the cam element and the valve element,

   - the valve elements and the cam surface thereby being arranged to perform rotational movements relative to each other, said movements corresponding to the relative
movements of the regenerator beds and the magnet arrangement, the opening
degree of each valve element thereby being controlled in accordance with a relative
angular position of the regenerator beds and the magnet arrangement.

2. An active magnetic regenerator device according to claim 1, wherein the regenerator beds
are stationary and the magnet arrangement is adapted to perform rotational movements.

3. An active magnetic regenerator device according to claim 1 or 2, wherein each cam
element has a shape which reflects variations in a magnetic field generated by a permanent
magnet section of the magnet arrangement, the valve elements thereby defining a fluid flow
profile which is chosen according to a magnetic field profile of the permanent magnet.

4. An active magnetic regenerator device according to any of the preceding claims, wherein
the cam surface comprises at least a first region and a second region, and wherein the first
region comprises a plurality of cam elements having a first shape, and the second region
comprises a plurality of cam elements having a second shape, the second shape differing
from the first shape, and wherein the cam surface is movable between a first position in
which the first region is arranged in contact with the valve elements and a second position in
which the second region is arranged in contact with the valve elements.

5. An active magnetic regenerator device according to claim 4, wherein the cam surface
further comprises an intermediate region arranged between the first region and the second
region, said intermediate region defining substantially continuous cam sections
interconnecting the cam elements of the first region with cam elements of the second region.

6. An active magnetic regenerator device according to any of claims 1-3, wherein each of the
cam elements has a shape which varies substantially continuously along a substantially axial
direction.

7. An active magnetic regenerator device according to any of the preceding claims, wherein
the valve elements are poppet valves.

8. An active magnetic regenerator device according to any of the preceding claims, wherein
the cam surface forms an inner surface or an outer surface of a ring shaped member.

9. An active magnetic regenerator device according to any of the preceding claims, wherein
the magnet arrangement further comprises at least two sections of soft magnetic material,
arranged between the permanent magnet sections.
10. An active magnetic regenerator device according to any of the preceding claims, wherein the regenerator beds are arranged annularly.

11. An active magnetic regenerator device according to claim 10, wherein the regenerator beds are arranged angularly equidistantly.

12. An active magnetic regenerator device according to any of the preceding claims, wherein the permanent magnet sections are arranged annularly.

13. An active magnetic regenerator device according to claim 12, wherein the permanent magnet sections are arranged angularly equidistantly.
### INTERNATIONAL SEARCH REPORT

**International application No**

PCT/EP2015/052294

#### A. CLASSIFICATION OF SUBJECT MATTER

**INV. F25B21/00**

**ADD.**

According to International Patent Classification (IPC) or to both national classification and IPC

#### B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F25B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

#### Electronic database consulted during the international search (name of database and, where practicable, search terms used)

EPO-Internal, WPI Data

#### C. DOCUMENTS CONSIDERED TO BE RELEVANT

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Further documents are listed in the continuation of Box C. See patent family annex.

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Date of the actual completion of the international search

27 April 2015

Date of mailing of the international search report

12/05/2015

Name and mailing address of the ISA/

European Patent Office, P.B. 5818 Patentlaan 2 NL-2280 HV Rijswijk
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Appendix B

Curie valve

B.1 Utilizing materials with controllable Curie temperatures for magnetic actuation purposes

- Paper published in *IEEE Transactions on Magnetics*
Utilizing Materials With Controllable Curie Temperatures for Magnetic Actuation Purposes

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The magnetic force between a permanent magnet and different blocks of ferromagnetic materials was measured and calculated as a function of distance and temperature in the vicinity of the Curie temperature of the materials. The calculations were carried out using a 3-D finite-element model of the system. On the basis of forces predicted by the model a number of equilibrium points were calculated for a system where the magnetic force on a ferromagnetic block of material is balanced by a linear spring force. It is shown how these calculation procedures can be used as a tool for designing autonomous temperature dependent and temperature adjustable actuation systems. A shunt valve utilizing such a system was designed, built and tested.

Index Terms— Magnetic forces, magnetic materials, permanent magnet (PM) machines.

I. INTRODUCTION

MagNetic materials with tuneable Curie temperatures are subjects to intensive research in the area of magnetic refrigeration where they are used for active magnetic regenerators [1] Examples of such materials that are well known from previous work within the area of magnetic refrigeration are the ceramic La$_{1-x}$(Ca, Sr)$_x$MnO$_3$ [2], [3] and the intermetallic compound La$_3$(Fe, Co, Si)$_13$ [4]. The previous research has mainly been concerned with the magnetocaloric effect near the Curie temperature of such materials. Basic ideas about utilizing the change in magnetization that occurs when a magnetized ferromagnetic (FM) material is heated above its Curie temperature and/or cooled down again for mechanical actuation purposes have appeared a few times in patents [5]–[7]. With the materials mentioned above as a starting point, this paper presents a detailed investigation into the temperature dependent forces that arise when a FM block is placed at a variable distance from a permanent magnet in the vicinity of its Curie temperature. This investigation both considers FM blocks with a single $T_C$ and graded blocks that contain regions of different $T_C$’s that are altered by tuning. Furthermore, we demonstrate the implementation in a magnet/spring actuation system and finally in an actual shunt valve prototype. The hereby demonstrated novel valve technology opens the possibility of designing valves that are temperature adjustable from the outside without penetrating the wall between the flowing medium and the surroundings while at the same time being autonomously actuated without power supply or sensor input of any kind. This combination of properties is not achievable with any conventional valve technology.

II. MAGNETIC FORCES

The magnetic force between a permanent magnet and a block of ferromagnetic material is given by

$$F = \int_V \nabla (m(r) \cdot B(r)) \, dV$$

where $m$ is the magnetization of the block and $B$ is the flux density produced by the permanent magnet, $r$ is a position vector and $V$ is the volume of the block. This is complicated by the fact, that $m = m(B)$. As all the FM blocks used have negligible anisotropy it is assumed that $m \parallel B$, hence $m = m(B)$. A high magnetic flux density leads to a high magnetization. This means together with (1) that in order to obtain high magnetic forces one needs a high flux density but with a steep gradient.

For calculation of the actual forces on the FM blocks a three dimensional model was implemented in the commercially available finite element multiphysics program Comsol Multiphysics [8]. The model is fed by data describing the magnetization of the material as a function of applied field and temperature. These data were obtained using a LakeShore 7407 vibrating sample magnetometer (VSM). The applied field, $H$, was measured in the range $\mu_0 H = 0$–1.5 T in a range of temperatures around the Curie point of each material. The modeled system consists of a cylindrical permanent magnet which is magnetized along its length axis with a remanence of $B_r = 1.3$ T. A FM block is placed at a distance from the end of the magnet. A table describing $B$ as a function of $H$ from the VSM data is in Comsol assigned to the domain representing the FM block. The two domains are modeled in a volume with a relative permeability of $\mu_r = 1$ corresponding to air. The resulting field is evaluated by Comsol and finally the force on the FM block is calculated using the Maxwell stress tensor formulation [9]. A Matlab program was made to repeat the calculations for different values of temperature and displacement of the FM block away from the magnet. To validate the model a series of force measurements were carried out by pulling an actual FM block away from a magnet using a spring dynamometer reading the force necessary to overcome the magnetic force. The spacing was controlled and varied using a number of plastic sheets between magnet and FM block and the temperature was measured directly at the FM block using a thermocouple. An example of modeled and measured forces versus displacement is shown in Fig. 1 where it can be seen that the modeled forces based on the magnetization data are in fair agreement with the directly measured forces.
Fig. 1. Temperature dependent magnetic forces versus displacement. The initial gap between magnet and FM block is 3 mm. Each of the blue curves represent the simulated force versus displacement for a certain temperature ranging from 255 K (top curve) to 330 K (bottom curve). The crosses represent experimentally measured values at $T = 255 \, \text{K}$ and the green curve represents an interpolation in the calculated data corresponding to this temperature. Mean deviation of the measured force values from the calculated is $\pm 10\%$. $T_r$ is 14 °C.

Fig. 2. Spring/magnet system. Equilibrium points at a fixed temperature. In the illustration the FM block is in a stable equilibrium point (B) where the magnetic force is balanced by a linear spring force from a compression spring. The other equilibrium point (A) is unstable.

III. SPRING/MAGNET SYSTEM

The temperature and distance dependent magnetic forces on a FM block in conjunction with a restoring force from a linear spring have been investigated using the calculated magnetic force data described above. In such a system a number of equilibrium points may exist in which the spring force balances the magnetic force on the FM block.

The magnetic force versus distance based on the measured magnetization data has for each temperature been approximated with a quadratic polynomial. When a force from a linear compression spring is biasing the attracting magnetic force at a given temperature, a maximum of two equilibrium points can exist. This is illustrated in Fig. 2. It can be seen that between the two equilibrium points the repulsive force from the spring is dominating whereas the attractive magnetic force is dominating elsewhere. As a consequence only one of the equilibrium points is stable.

Fig. 3. Displacement due to temperature change. The FM block is initially in a stable equilibrium point (A). If the block is fixed while the temperature is changed to $T > T_r$ and released it will move to the new stable equilibrium point $B_2$. Alternatively if the temperature is decreased to $T < T_r$ it will move to $B_1$.

When the temperature is changed the FM block will move to the stable equilibrium point corresponding to the new temperature. This is illustrated in Fig. 3 where it is shown what will happen if the FM block is fixed at a given stable equilibrium point, while the temperature is either increased or lowered, and then released. In reality this displacement will happen continuously as the temperature is changed.

IV. GRADED MATERIAL

To investigate the possibility of using graded FM blocks where sections with different FM blocks are placed directly adjacent to each other, a simple experiment has been conducted and compared with predictions made from finite element modeling as previously described. Three FM blocks measuring $15 \times 15 \times 20 \, \text{mm}^3$ are composed of different compounds of the material $\text{La}_x(\text{Fe}, \text{Co}, \text{Si})_{13}$ corresponding to the ones used in the prototype described later. These have been placed next to each other in a nylon holder which is also guiding a cylindrical magnet with a diameter of 20 mm and a length of 40 mm, see Fig. 4. The magnet and FM blocks are spaced by a 3 mm wall in the holder. The FM blocks are positioned in different positions along the x-axis as shown in the figure. For each position the force necessary to pull the magnet away from the FM blocks was measured using a spring dynamometer. The results from these tests are shown in the graph in Fig. 4 together with the model predictions. As it can be seen, at high forces there is some discrepancy between experimental results and model predictions. However, there is a clear trend indicating an almost linear regime when the magnet is not close to the outer edge of the high $T_{C_{1}}$ region. This result indicates that it will be possible to design actuation systems with a continuous and linear regulation of the switching point temperature by using graded FM blocks.

V. VALVE ACTUATION

A spring/magnet system as described above makes it possible to design novel valve systems that are actuated according
Fig. 4. Graded FM block consisting of three sections with \( T_C \)’s of 10 °C, 20 °C, and 30 °C, respectively. At this initial position, the edge of the \( T_C = 30 \) °C block is aligned with center axis of the magnet. The actual temperature in this case was \( T = 22 \) °C. (a) Results. (b) Experimental setup.

to temperature changes in either the flowing medium or the surroundings. By using graded FM blocks containing regions with different Curie temperatures it even becomes possible to adjust the working temperature by moving a magnet assembly and graded FM block relative to each other without penetrating the wall separating the flowing medium from the surrounding environment. Many different types of valves could be made using this principle. A particularly interesting possibility is shunt valve applications, i.e., valves that are distributing an incoming flow between multiple outlets dependent on the fluid temperature. This gives the advantage of always having a flow through the valve which implies good thermal contact between the FM block and the flowing medium.

VI. SHUNT VALVE PROTOTYPE

A shunt valve prototype has been designed and built, see Fig. 5. The valve has been designed for conditions relevant for domestic water applications or central heating. The flowing medium is water between 0 °C and 80 °C and the flow rates are between 0 and 250 L/h. For simplicity and easy comparison with model prediction the valve has been designed with three distinct FM blocks not directly adjacent to one another. In this way only the magnetic force on one FM block at a time has to be considered. The three FM blocks are different compositions of \( \text{La(Fe,Co)}_{13} \) made by Vacuumschmelze [10] and \( \text{La}_0.67\text{Ca}_{0.26}\text{Sr}_{0.07}\text{Mn}_{1.06}\text{O}_3 \) made by EMPA [11] has been tested in the valve as this material also has good magnetization properties and a tuneable Curie temperature. This material is noncorrosive in water as opposed to the intermetallic \( \text{La(Fe,Co, Si)}_{13} \) materials that had to be coated by a thin layer of polymer. A 20 × 20 × 40 mm\(^3\) permanent NdFeB magnet is used and the spring forces are supplied by two leaf springs. This gives a shunt valve with three different temperature switching set points corresponding to three positions of the external magnet.

VII. PROTOTYPE TESTS

A special test rig was built in which the shunt valve has been tested under conditions similar to what can be expected in a conventional central heating system in a regular house. Water was sent through the inlet at a chosen flow rate and temperature and distributed by the integrated magnetic actuation system between the two outlets after which the resulting flow rates were measured. A series of experiments have been conducted to characterize the valve under steady state and steady flow conditions.
Fig. 6 shows the results of such a test. Each of the measured data points represents a flow rate through an inlet/outlet after the temperature has been set and the system has reached thermal equilibrium. In this case the valve has initially been below the switching point. The temperature has then been increased and a new equilibrium point reached. This has been repeated a number of times until the temperature was well above the switching point. Then the temperature was incrementally lowered until the valve was below the switching point again. It can be seen from the figure, that there is a fair agreement between the measurements and model predictions. The actual switching point where the two outlet flow rates are equal as well as the shape of the curves are very well captured. However, there is a minor systematical discrepancy above the switching point due to an internal leak at the hot valve outlet. Besides this the experimental plot shows a small hysteresis when it goes from cold to hot and back again. This lag is considered to be due to internal static friction forces that have to be overcome when the magnet/spring force balance changes.

VIII. CONCLUSION

Temperature dependent attraction forces between a permanent magnet and materials with different Curie temperatures have been investigated. By finite-element modeling supported by experiments it has been shown how these forces depend on distance and temperature in the vicinity of the Curie points of the materials. A system where the magnetic attraction force is biased by a force from a linear compression spring has been analyzed. It has been shown how both stable and unstable equilibria exist and how the stable equilibrium is moving with temperature. It was discussed how such a system could be utilized for valve actuation. An actual shunt valve prototype distributing an incoming flow between two outlets dependent on the temperature of the flowing medium has been developed and tested. The results generally demonstrate a good agreement between predictions based on the numerical model and test results. This provides a basis for the developed model to be used as a design tool for future valve applications based on this principle. The developed valve prototype has three different switching points which can be chosen by moving the external magnet relative to the internal material. Finally, the possible use of graded blocks of materials where regions with different Curie temperatures are directly adjacent to one another has been discussed. A preliminary experiment together with results from a finite element model indicate that this will make an almost linear regulation of the switching point of a valve utilizing such a system possible.

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