A two-dimensional investigation about magnetocaloric regenerator design: parallel plates or packed bed?

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Abstract. Magnetic Refrigeration (MR) is a novel refrigeration technique based on eco-friendly solid materials as refrigerants, whom react to the application of magnetic fields, with warming and cooling by magnetocaloric effect. The thermodynamical cycle which best suits the magnetic refrigeration is Active Magnetic Regenerator cycle (AMR). Regenerator is the core of a magnetic refrigerator, since that it plays a dual-role: it operates both as refrigerant and regenerator in an AMR cycle. An AMR cycle consists of two adiabatic stages and two isofield stage. In this paper an investigation is conducted about the magnetocaloric refrigerator design through two-dimensional multiphysics numerical models of two different magnetocaloric regenerators: (1) a packed bed and (2) a parallel plates magnetic regenerators made of gadolinium, operating at room temperature under a 1.5T magnetic field induction. Both models employ water as secondary fluid. The tests were performed with variable fluid flow rate at fixed AMR cycle frequency. The results obtained are presented in terms of temperature span, cooling power, coefficient of performance and mechanical power of the circulation pump, and they indicate under which operating conditions packed bed configuration is to be preferred to parallel plates and vice versa.

1. Introduction

In the last fifteen years the scientific community has oriented more and more interest toward non-vapor compression refrigeration, in order to develop new refrigeration techniques based on solid-state materials [1]. Magnetic refrigeration [2] is the one that most has undergone progressive developments, since it has demonstrated to be an innovative, promising and ecologic technology, based on solid state magnetic materials [3] exhibiting MagnetoCaloric Effect (MCE). Compared to conventional vapor compression systems [4], magnetic refrigeration can be an environment-friendly and efficient technology [5]. Magnetic refrigerant are solid materials, with essentially zero vapor pressure, and therefore they are ecological with no direct Ozone Depletion Potential (ODP) and zero direct Global Warming Potential (GWP) [6].

Specifically MR founds its operation MCE, a physical phenomenon observed in every material with magnetic properties and it consists in an increment of temperature in the magnetic material as a consequence of magnetization under adiabatic conditions. Furthermore, magnetic refrigeration exhibits high potential energetic efficiencies (30-60% of Carnot COP). On the other side MR presents at least two relevant disadvantages: the high costs associated both with the magnetic field generation...
and magnetic materials [9]. The Active Magnetic Regenerator (AMR) is the core of a magnetic refrigerator system. It is a special kind of thermal regenerator made of magnetic material which works both as a refrigerating and as a heat regenerating medium. The performance of an AMR system strongly depends on the magnetocaloric effect of the magnetic material which the regenerator is made of, on the geometry of the regenerator and on the secondary fluid.

In this paper an investigation is conducted about the magnetocaloric refrigerator design through two-dimensional multiphysics numerical models of two different AMR regenerators. We compare the results of two numerical two-dimensional (2D) multiphysics models [10]: (1) a packed bed and (2) a parallel plates magnetic regenerators made of gadolinium, operating at room temperature under a 1.5T magnetic field induction. Both models employ water as secondary fluid. The tests were performed with variable fluid flow rate, at fixed AMR cycle frequency. The results obtained are presented in terms of temperature span, pressure losses, cooling power, coefficient of performance and they indicate under which operating conditions packed bed configuration is to be preferred to parallel plates and vice versa.

2. Magnetocaloric effect and the AMR cycle
The magnetocaloric effect is a physical phenomenon detected for the first time in iron, by Warburg in 1881[11] as temperature change in the material by varying the intensity of the magnetic field applied, under adiabatic conditions. The entropy of a magnetic solid at constant pressure \( S(T,H) \) is a function of both the magnetic field intensity \( H \) and the absolute temperature \( T \) and it consists of three contributions: magnetic \( (S_M) \), lattice \( (S_{lat}) \) and electronic \( (S_{el}) \) ones:

\[
S = S_M(T,H) + S_{lat}(T) + S_{el}(T)
\]  
(1)

In equation (1), the magnetic contribution is function of both \( H \) and \( T \), whereas the electronic and lattice contributions only depend on \( T \). Consequently, only the magnetic entropy \( S_M \) can be controlled by varying the magnetic-field strength. When the magnetic field is applied to a magnetic material, the magnetic dipoles become oriented according to the direction of the field. If this is done isothermally, it carries to a decrement of the material’s magnetic entropy by the isothermal entropy decreasing \( (\Delta S_M)_T \). The entropy change can be evaluated as:

\[
(\Delta S_M)_T = \int_{H_i}^{H_f} \left( \frac{\partial S_M}{\partial H} \right)_T dH
\]  
(2)

On the other side, if the magnetization is done adiabatically, the total sample entropy remains constant and the decrease in magnetic entropy is countered by an increase in the lattice and electron entropy. This causes a heating of the material and it takes an increasing of its temperature given by the adiabatic temperature change, \( \Delta T_{ad} \). Dually, an adiabatic demagnetization involves the increasing of the material's magnetic entropy, causing a decreasing in lattice vibrations and by that a temperature decrease. The expression for \( \Delta T_{ad} \) of a magnetic material can be evaluated as:

\[
(\Delta T_{ad})_S = -\int_{H_i}^{H_f} \frac{T}{C_H} \left( \frac{\partial M}{\partial T} \right)_H dH
\]  
(3)

where the specific heat can be defined as:

\[
C_H = T \left( \frac{\partial S}{\partial T} \right)_H
\]  
(4)

At the Curie temperature a magnetic material exhibits its maximum MCE. There are two types of magnetic phase changes that may occur at the Curie point: First Order Magnetic Transition (FOMT) and Second Order Magnetic Transition (SOMT). A FOMT material shows a discontinuity in the first
derivative of the Gibbs free energy, whereas in a SOMT material the first derivative is a discontinuous function and discontinuity takes place in the second derivative. Therefore, in a FOMT material the magnetization, i.e. the first derivative of the free energy with respect to the applied magnetic field strength, is discontinuous. In a SOMT material the magnetic susceptibility, i.e. the second derivative of the free energy with the field strength, changes discontinuously. Most of the magnetic materials order with a SOMT from a paramagnet to a ferromagnet, ferrimagnet or antiferromagnet [12]. The benchmark material of magnetic refrigeration is considered to be gadolinium (Gd), a rare earth which exhibits a SOMT at $T_{\text{Curie}} = 294$ K and it belongs to lanthanide group of elements.

The referring thermodynamical cycle of magnetic refrigeration is the reverse Brayton cycle [13] and, as a matter of fact, in 1982 Barclay introduced an innovative way to follow it up by the introduction of the Active Magnetic Regenerative refrigeration cycle (AMR) [14]. The AMR couples into a single concept what had been before two separate processes: instead of using a different material as a regenerator to recuperate heat from the magnetic material, the AMR concept made use of the refrigerant magnetic material itself. In essence, a temperature gradient is established throughout the AMR and a fluid is used to transfer heat from the cold to the hot end. The working principle of an AMR is presented in Figure 1 where: the dashed line represents the initial temperature profile of the bed in each process, the solid line depicts the final temperature profile of the process.

Assuming that the bed is in steady state condition with the hot heat exchanger at $T_H$ whereas the cold heat exchanger is at $T_C$, four processes constitute the AMR cycle:

(a) adiabatic magnetization: each particle of the magnetic material which constitute the regenerator, warms up by increasing progressively the intensity of the magnetic field applied, under adiabatic conditions;

(b) isofield cooling: with the magnetic field present at maximum value, the fluid blows from the cold to the hot end of the regenerator. Therefore the regenerator cools down transferring heat to the fluid, whom is expelled in the hot heat exchanger;

(c) adiabatic demagnetization: by decreasing adiabatically the intensity of the magnetic field applied from a maximum until a minimum value, the regenerator cools anymore and a decrement of temperature, equal to $\Delta T_{\text{ad}}(T)$ is registered;

(d) isofield heating: while magnetic field has kept to a minimum value, the fluid flows from the hot to the cold end; as a result the fluid, hotter, cools itself by crossing the regenerator and reaching a temperature lower than $T_C$. At this stage the secondary fluid absorbs heat from the cold heat exchanger at $T_C$, producing a cooling load.
3. Regenerator geometries and model description
The performances of a magnetic refrigerator AMR based, are mostly influenced by:

- the MCE properties of the magnetic material employed;
- the regenerator geometry and related operating conditions;
- the irreversible heat losses due to non-idealities of the refrigerators.

The criteria to identify the best magnetic material for magnetic refrigeration are reported by Aprea et al. [9,15] and by Franco et al. [16]. The benchmark material is still considered to be gadolinium due to its excellent magnetocaloric properties.

The irreversible heat losses [17] of AMR regenerators have also great influence on the performances of the AMR cycle. Main irreversible heat losses are:

- heat transfer between regenerator and heat transfer fluid;
- pressure drop by flow resistance;
- thermal conduction along the magnetic material
- heat leakage;
- losses due to magnetic hysteresis and eddy currents;
- viscous dissipation in the secondary fluid;
- “dead volume” loss which is due to the heat transfer fluid trapped in the connecting lines between the bed and the heat exchangers.

In the cooling system, the regenerator geometry assumes paramount importance [18-20] for gaining efficient heat transferring between magnetic material and fluid flow, thus minimizing the major loss in the cooling system. Two different regenerators are both widely used: a parallel plate regenerator, which yields lower pressure drops; a packed bed regenerator which can produce a larger temperature span, due to the larger extension of the heat transfer surface.

This paper compares the results of two numerical two-dimensional (2D) multiphysics models: (a) a packed bed and (b) a parallel plates magnetic regenerators made of gadolinium, operating at room temperature under 1.5 T as magnetic field induction. Both models employ water as secondary fluid.

![Figure 2. The packed bed AMR regenerator geometry: a 20x45mm wrapper contains 3600 spheres.](image)

The regenerator have a rectangular shape (20x45mm) which acts as a wrapper. The area of the packed bed regenerator is filled with a regular matrix of 3600 circles that constitute the magnetocaloric material, i.e. gadolinium; every circle has a diameter of 0.45mm and the amount of the area occupied by all of the circles is about 60% of the total rectangular area. A group of channels is
formed by stacking particles in the regenerator area: the fluid flows through these interstitial channels. In Figure 2 is shown a scheme representing the AMR packed bed regenerator considered.

The area of the parallel plates AMR regenerator contains 27 parallel plates made of gadolinium: every plate has a thickness of 0.5mm. The amount of the area occupied by all of the plates is 60% of the total rectangular area. A group of channels is formed among the plates in the regenerator area, where the secondary fluid flows. Figure 3 reports the parallel plates geometry of the regenerator.

![Figure 3](image)

**Figure 3.** The parallel plates AMR regenerator geometry: a 20x45mm wrapper contains 27 plates and 26 channels.

Both the models take into account both the fluid flow and heat transfer between the solid and the fluid; the fluid flow in the positive x direction during the isofield cooling process and in the opposite direction during isofield heating. The two models are supported by the same mathematical formulation which describes the AMR cycle, including several distinct groups of equations according to the different processes of the AMR cycle that the regenerator experiences.

The equations that rule the regenerative fluid flow processes, in both directions, are: the Navier-Stokes for the heat transfer fluid and the energy equations for both the fluid and the solid particles. The fluid is assumed incompressible.

The equations during the fluid flow phases are:

\[
\begin{align*}
\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} &= -\frac{1}{\rho_f} \frac{\partial p}{\partial x} + v \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \\
\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} &= -\frac{1}{\rho_f} \frac{\partial p}{\partial y} + v \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \\
\frac{\partial \tau_f}{\partial t} + u \frac{\partial \tau_f}{\partial x} + v \frac{\partial \tau_f}{\partial y} &= \frac{k_f}{\rho_f C_{fp}} \left( \frac{\partial^2 \tau_f}{\partial x^2} + \frac{\partial^2 \tau_f}{\partial y^2} \right)
\end{align*}
\]

(5)

The equations that model magnetization and demagnetization processes are:

\[
\begin{align*}
\rho_f C_{fp} \frac{\partial \tau_f}{\partial t} &= k_f \left( \frac{\partial^2 \tau_f}{\partial x^2} + \frac{\partial^2 \tau_f}{\partial y^2} \right) \\
\rho_s C_H \frac{\partial \tau_s}{\partial t} &= k_s \left( \frac{\partial^2 \tau_s}{\partial x^2} + \frac{\partial^2 \tau_s}{\partial y^2} \right) + Q
\end{align*}
\]

(6)
The model takes into account magnetocaloric effect which elevates or reduces the temperature of the solid by the variation of the external magnetic field applied to the regenerator. Hence the MCE temperature variation $\Delta T_{ad}$ is converted \[21\] into a heat source $Q$:

\[
Q = Q(H, T_s) = \frac{\partial_s C_{V}(H, T_s) \Delta T_{ad}(H, T_s)}{\Delta t}
\]

which has the dimensions of a power density and included in the solid energy equation, only for magnetization and demagnetization phases. The term $Q$ is positive during magnetization, negative during demagnetization. $\Delta t$ is the period of the magnetization/demagnetization process. The coupled equations that govern the AMR cycle, imposed on this model, are solved using Finite Element Method.

The AMR cycle is modeled as four sequential steps. The same time step $\Delta t$ has been chosen for the resolution during all the four periods of the cycle. The cycle is repeated several times with constant operating frequency until the regenerator reaches steady state operation.

The model has been validated with a Rotary Permanent Magnet Magnetic Regenerator (RPMMR) \[10,21,22\], developed at the Refrigeration Lab in the University of Salerno in collaboration with the University of Naples Federico II by numerical simulation and experimental tests operating in the same conditions.

4. Numerical simulations and results

Several AMR cycles have been simulated on both the models in order to compare their performances by varying fluid flow rate at fixed AMR frequency. All the results have been obtained working with 1.25 Hz as AMR frequency, in the 288-298 K temperature range and investigating the regenerators behaviors while the secondary fluid velocity changes in 0.0595-0.25 m/s range. The applied magnetic field has a variable intensity from 0 to 1.5 T. The secondary fluid employed is pure water, whereas the magnetocaloric material involved is gadolinium. Cold and hot heat exchangers temperature ($T_C$=288 K; $T_H$=298 K) have been chosen to characterize the performance sensitivity of the regenerator in a range which embraces the gadolinium Curie temperature, i.e. 294 K.

The results obtained are presented in terms of temperature span, cooling power, coefficient of performance (COP) and pump power, in order to quantify the pressure losses influence on COPs, for both the regenerator geometries. As a matter of fact the results provide an indication about the operating conditions under which packed bed configuration has to be preferred to parallel plates and vice versa.

![Figure 4. $\Delta T_{span}$ evaluated for both the regenerator configuration for variable fluid flow velocity and therefore fluid flow rate.](image)

In Figure 4 it has shown the $\Delta T_{span}$ detected for both the regenerator geometries under each fluid flow velocity investigated. Such parameter has been obtained by evaluating the difference between $T_H$
and the cold side temperature of the secondary fluid averaged in the last process of AMR cycle (fluid flow from hot to cold side of the regenerator), as shown in the following equation:

\[ \Delta T_{\text{span}} = T_H - \int_{t_M + t_{CF} + t_D}^{t_M + t_{CF} + t_{HF}} T_f(y, t) \, dt \]  

(8)

Figure 4 reveals that for both the model, the \( \Delta T_{\text{span}} \) decreases with the fluid velocity. Indeed, corresponding to a small fluid mass flow rate, the fluid can be regenerated to reach lower cold side temperature. The maximum \( \Delta T_{\text{span}} \) is reached in correspondence of a fluid velocity of about 0.1 m/s. Furthermore one can observe that although, for low speed, packed bed configuration lets to \( \Delta T_{\text{span}} \) greater than parallel plates ones, for high values of fluid velocity parallel plates geometry ensure larger temperature spans. The latter regenerator configuration seems to exhibit a less sensitive behavior with respect to velocity than the former, in terms of \( \Delta T_{\text{span}} \).

Figure 5 reports the refrigeration power as a function of fluid velocity, obtained for packed bed and parallel plates AMR regenerator. The refrigeration power has been calculated according to the following equation:

\[ Q_{\text{ref}} = \frac{1}{\tau} \int_{t_M + t_{CF} + t_D}^{t_M + t_{CF} + t_{HF}} \dot{m}_f C_p(T_c - T_f(y, t)) \, dt \]  

(9)

Figure 5 clearly shows that the cooling power increases with the fluid flow rate and then decreases reaching a maximum. Indeed, corresponding to a small fluid mass, although the fluid can be regenerated to reach lower exit temperature, a small amount of transfer fluid can produce only a little refrigerating capacity. On the other hand, excessive quantities of transfer fluid perturb the temperature profile of the AMR, decreasing the temperature difference between the fluid and the bed. The bed quickly becomes overwhelmed by the fluid flow and the efficiency of the heat transfer decreases quickly causing a loss in cooling power. From figure 5 one can observe that the points where the geometries confer the maximum refrigeration power differs from each other. As a matter of fact, if the secondary fluid has moved into the regenerator with a velocity of 0.15 m/s, packed bed and parallel plates configurations provide quite the same \( Q'_{\text{ref}} \).

![Figure 5](image)

**Figure 5.** The refrigeration power evaluated for both the regenerator configuration for variable fluid flow velocity and therefore fluid flow rate.

Furthermore, with lower flow speeds, parallel plates shows a drastic reduction of its cooling capacity, higher than packed bed; whereas for values greater than 0.15 m/s, where a slump in packed
bed’s performances happens, parallel plates responds in a very satisfactory manner, providing refrigerant powers 3 times higher than the former geometry.

To estimate the performance of the two models, the Coefficient of Performance has been introduced as follows:

\[
COP = \frac{\dot{Q}_{ref}}{\dot{Q}_{H} - \dot{Q}_{ref} + \dot{W}_p}
\]  

(10)

whereas \(\dot{Q}_H\) is the power related to the heat supplied in the environment which has been evaluated according to:

\[
\dot{Q}_H = \frac{1}{\tau} \int_{t_M}^{t_C} \dot{m}_f C_f (T_f (L, y, t) - T_H) \, dt
\]  

(11)

\(\dot{W}_p\) is the mechanical power associated to the circulation pump:

\[
\dot{W}_p = \frac{\dot{m} (\Delta p_{CF} + \Delta p_{HF})}{\eta_{CF} \rho_f} (t_{CF} + t_{HF})
\]  

(12)

Figure 6. COP estimated for both the geometries for different flow speed.

In Figure 6 the COP evaluated for both the geometries by varying fluid flow rate, are reported. Figure 6 clearly shows that COPs increase until reaching the maximum whom, in both the geometries, assumes the same value (about 3), but the velocity where it is achieved are different. COPs show the same trends of refrigeration power: packed bed is better with low speed, parallel plates are to be preferred for high ones. A packed-bed regenerator is characterized by the best heat transfer surface, with a more marked pressure drop. A flat plates regenerator is characterized by a lower heat transfer surface and lower pressure drops. Therefore another parameter is the responsible of such COP fall in packed bed configuration, for high velocity: pressure losses and, therefore, the mechanical power of the circulation pump.

Figure 7, which exhibits the work of the pump employed to allow the circulation of secondary fluid, reveals a significant reality: despite in the parallel plates configuration the pressure losses are negligible and not influenced by a change of velocity, a packed bed regenerator is strongly affected by pressure losses that seriously limit its operation at high flow speeds.
5. Conclusions

In the present paper, two numerical two-dimensional (2D) multiphysics models have been investigated. A packed bed and a parallel plates geometries of magnetic regenerators have been compared. They are either made of gadolinium, working in room temperature range, under 1.5 T as magnetic field induction. A practical model for predicting the performance and efficiency of an AMR refrigerator system for different magnetocaloric materials has been introduced. The results obtained are presented in terms of temperature span, cooling power, coefficient of performance and mechanical power of the circulation pump, when the fluid velocity is varying.

After an accurate analysis of all the above mentioned parameters, a general trend, which describes the performance of the two geometries under a general framework, has been carried out. We can assert that for low values of fluid velocity, packed bed configuration has to be preferred since it offers \( \Delta T_{\text{span}} \), refrigeration power and COP higher than parallel plates one and therefore a more satisfactory behaviour. For high flow speed the situation is completely upside down: parallel plates geometry provides good results whereas packed bed suffers a significant slump in its performance, caused also by an huge increment of pressure losses whom limits its use.

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