

Review on the developments of active magnetic regenerator refrigerators – evaluated by performance

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Abstract

Magnetic/magnetocaloric refrigeration is an energy-efficient and environmentally safer cooling technology with the potential to be an alternative to conventional vapor compression systems in the future. Magnetocaloric effect (MCE) is a measure of relative temperature rise/drop of certain ferromagnetic materials upon the application/removal of a magnetic field. The technology uses MCE of some materials such as Gd to produce temperature difference/span relative to the ambient via a four-stage regenerative cycle known as active magnetic regenerative (AMR) cycle. Research in this area has been thriving especially during the last two decades focussing on different aspects of technology such as materials, magnetic field sources, and system design. On the system design, studies investigating the effect of different magnetic, thermal-hydraulic, and geometric parameters on the performance have been found in the literature. The present work offers a chronological review and comparison of recent advances in AMR refrigerators. Findings and results reported in the literature are compared in terms of magnetocaloric materials, geometric parameters (such as regenerator geometry); operating parameters e.g. cycle frequency, utilization, heat transfer fluid, heat rejection temperature, and cooling load, etc. Besides, performance indicators such as no-load temperature span, cooling capacity, and/or system coefficient of performance have been considered. Parametric sensitivity and performance trends have been identified and discussed. Major barriers to achieving system peak performance and hence the marketability of the technology are also highlighted.

Highlights

- Magnetic field and magnetocaloric material are crucial parameters determining overall performance.
- Regenerator geometry and heat transfer fluid strongly influence the regenerator temperature gradient.
- Utilization and cycle frequency require optimization to meet a given cooling load requirement.

Keywords:

- Review magnetic refrigeration; active magnetic regenerator; magnetocaloric effects; performance analysis.

Nomenclature

Latin

A	area [m^2]
a_p	heat transfer area per unit vol. [1/m]
c_P	specific heat capacity [J/kg.K]
d	diameter [m]
d_h	hydraulic diameter [m]
f	frequency [Hz]
f_F	Fanning friction factor
$\mu_0 H$	magnetic field intensity [T (Tesla)]
j	Colburn modulus
L	length [m]
m	mass [kg]
$\dot{m}$	mass flow rate [kg/s]
M	magnetization
Nu	Nusselt number
Pr	Prandtl number
p	pressure [Pa]
Q	heat transfer rate [W]
Re	Reynolds number
St	Stanton number
T	temperature [K]
u	flow velocity [m/s]
V	volume [m^3]

Greek

φ	utilization
δ	spacing [m]
ρ	density [kg/m^3]
τ	period [s]
ε	porosity
Δ	change/difference/span

Subscript

ad	adiabatic
C	cold end
H	hot end
r	regenerator
f	fluid, flow

Abbreviations/Acronyms

AMR	active magnetic regenerator
COP	coefficient of performance
MCE	magnetocaloric effect
MCM	magnetocaloric material

1. Introduction

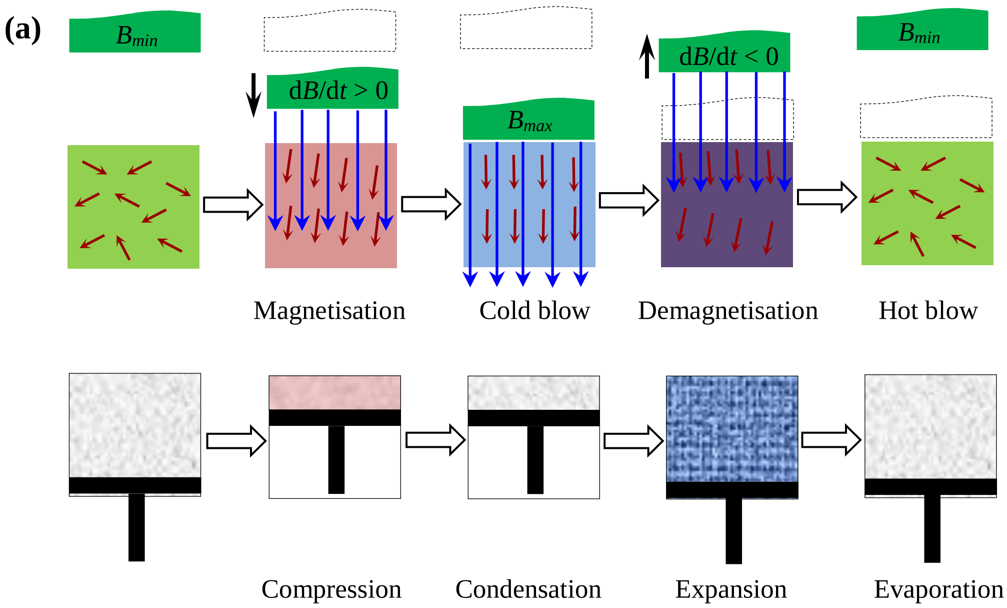
The world is becoming increasingly conscious of the existing ways of energy conversion and its utilization. There has been a strong global drive toward energy conservation and energy-efficient technologies, in addition to prioritizing energy harnessing through renewable and sustainable techniques and resources. Many aspects of modern society are unimaginable without cooling and thermal management technologies. Refrigeration, for example, finds its applications ranging from the healthcare and food industry to the energy sector. International Institute of Refrigeration (IIR) in its 29th informatory note reported that around 17% of the worldwide electricity was being consumed by ~3 billion refrigeration units. Refrigeration systems are primarily based on the vapor-compression cycle (VCC) which is a highly energy-intensive process. The relative higher energy input to run the compressor in VCC hinders the wide-scale affordability and accessibility of the technology, especially in the developing world facing a severe energy crisis. Also, continuous research to optimize vapor-compression based systems since the last several decades have brought the energy efficiency of the system to almost saturation. Moreover, although significant emphasis and legal framework, e.g. Montreal Protocol, have been put in place to regulate and allow refrigerants with minimum global warming and ozone depletion potentials, the inherently hazardous nature of the conventional refrigerants remains a source of concern. There has always been a quest among engineering researchers to discover/invent alternative ways to produce cooling/heating relative to space. Vapor absorption and thermo-electric cooling do offer the alternate but with a relatively lower coefficient of performance.

In recent years, caloric cooling technologies have gained popularity and are undergoing rigorous research, exploration, and development phases. Few worth mentioning caloric technologies are (a) magnetocaloric: employs the change in temperature (known as the magnetocaloric effect, MCE, discovered by Warburg in 1881) caused by a change in the applied magnetic field; (b) barocaloric/mechanocaloric: employs change in temperature due to change in applied pressure/stress and (c) electrocaloric: employs the change in temperature caused by a change in the applied electric field. Magnetocaloric technology has been able to capture the attention of a relatively wider research community when compared with other caloric technologies. It employs iso-magnetic processes that can have minimum irreversibilities for ferromagnetic materials, enabling the system to achieve higher efficiency i.e 30% to 60% of the Carnot cycle (Yu et al. [1], Zimm et al. [2]). Some of the key features are (a) water (with anti-freeze) as the only heat transfer fluid, the complete absence of conventional refrigerants and thus no environmental concerns; (b) pump is used to circulate the fluid in the system and no use of compressor means higher energy efficiency. Also, the associated CO_2 emissions are significantly for lower pumping power requirements of the system; (c) relatively lower number of moving parts making the system more compact and quieter. The application of a regenerative cycle to achieve a useful large temperature span and hence higher cooling capacity while using relatively lower magnetic field change (permanent magnet-based field sources) was first proposed by Brown in 1976 (Brown [3]). Steyert introduced the concept of Active Magnetic Regenerator (AMR) refrigeration in 1978 followed by the development of AMR refrigerator by Barclay and Steyert in the 1980s (Pecharsky and Gschneidner [4]).

In the last two decades, researchers in the fields of heat transfer, system design, materials, and numerical modeling have reported their findings on magnetocaloric cooling. Several review articles have been published describing different aspects of progress in magnetic refrigeration. Yu et al. [1] reviewed the development of the magnetic materials, magnetic refrigeration cycles, magnetic field sources, and regenerator characteristics for application in magnetic refrigeration. The review also included the description of room-temperature magnetic refrigeration prototypes of Brown (1976), Steyart (1978), Kirol (1987), and Zimm (1997). Brück et al. [5] presented a review of magnetocaloric materials (MCMs) specifically first-order magnetic transition materials and compared their MCE with that of Gd considering their industrial applications. Gschneidner et al. [6] published an article reviewing the research in magnetocaloric materials and compared the magneto-thermal properties of different families of MCMs such as the lanthanide laves phases, Gd-Si-Ge, Mn-As-Sb, Mn-Fe-P-As, La-Fe-Si, their hydrides, and manganites. They also discussed the feasibility of these materials as refrigerants for magnetic refrigeration. Engelbrecht et al. [7] presented a review of the research trends in the system design and MCMs. Design aspects that could influence the performance of the refrigeration system were also discussed. Brück et al. [8] presented a review of the works carried out by different research groups regarding the magnetocaloric properties of Mn-based inter-metallic compounds. They concluded that the most promising inter-metallic compounds are of Mn-Fe-P-As but the presence of As limits their application in magnetic refrigeration. Gschneidner and Pecharsky [9] reviewed the progress of near room temperature magnetic refrigeration. They have discussed the prospects of magnetic refrigeration research, application, and commercialization. Yu et al. [10] presented a comprehensive review of the forty-one prototypes of magnetic refrigerators and heat pumps which were built till 2010. Starting from the first device built by Brown in 1976 till 2008, twenty-five prototypes were reviewed. The review covered the description and operation of each prototype in detail. Similarly, Bjork et al. [11] compiled various designs of magnetic field sources used in magnetic refrigerators till then and compared their performance by considering design and operation parameters such as the generated magnetic flux density, volume of the high flux density region, volume of the magnet used and magnetic field operation time as a fraction of the cycle time. They introduced a parameter Λ_{cool} to quantify the performance of the magnetic field source and to select the optimal magnet design. Engelbrecht et al. [12] presented a review of the numerical models which have been developed by 2011. The models were categorized based on the dimensions (1D, 2D, and 3D) and the major modeled terms e.g. inclusion of MCE, flow and magnetic field profiles, temperature and magnetic field dependence on the properties of fluids and solid materials, thermal conductivity, flow-channeling effect in particle-beds, parasitic thermal losses, magnetic hysteresis, and demagnetizing fields. Although the aforementioned review articles encapsulated investigations of magnetocaloric materials, experimental prototypes, and numerical modeling, however a detailed review based on the evaluation of the performance parameters has not been effectively addressed. The present work, therefore, provides a review of the relative sensitivity of various design and operating parameters. An attempt is made to develop a summary of the temporal developments in AMR based cooling devices. Investigations on the effect of regenerator geometric configurations and operating conditions (e.g. frequency, utilization, and hot end temperature) on the AMR performance carried out by different research groups have been compared and evaluated.

2. Active magnetic regenerative cycle

The operating cycles can be categorized based upon the conditions under which magnetization and demagnetization of the material are brought about. In the magnetic Ericsson cycle, magnetization and demagnetization are carried out under isothermal conditions while in the Brayton cycle magnetization and demagnetization are carried out adiabatically (Cross et al. [13]). The majority of the magnetic refrigerator prototypes operate on the Brayton cycle (e.g. Tagliafico et al. [14], Trevizoli et al. [15], and Kawanami et al. [16]). Figure 1(a) shows the four stages of the AMR cycle and their corresponding processes in the VCC. Figure 1(b) presents a graphical view of the AMR operation and Figure 1(c) shows a T-S representation of the AMR cycle. The four stages of the cycle are: (a) adiabatic magnetization: under adiabatic conditions, the regenerator is exposed to the magnetic field and experience a temperature rise; (b) iso-magnetic heat transfer (also referred to as *cold blow*): under constant magnetic field, heat transfer fluid flows of from heat transfer fluid from the cold side heat exchanger to hot side; (c) adiabatic demagnetization: the regenerator is moved of the magnetic field resulting in temperature drop and (d) iso-magnetic heat transfer (also referred to as *hot blow*): under a constant magnetic field, heat transfer fluid flows in reverse direction i.e. from hot side to cold side. These four stages are repeated and the performance is measured when system reaches a steady-state.



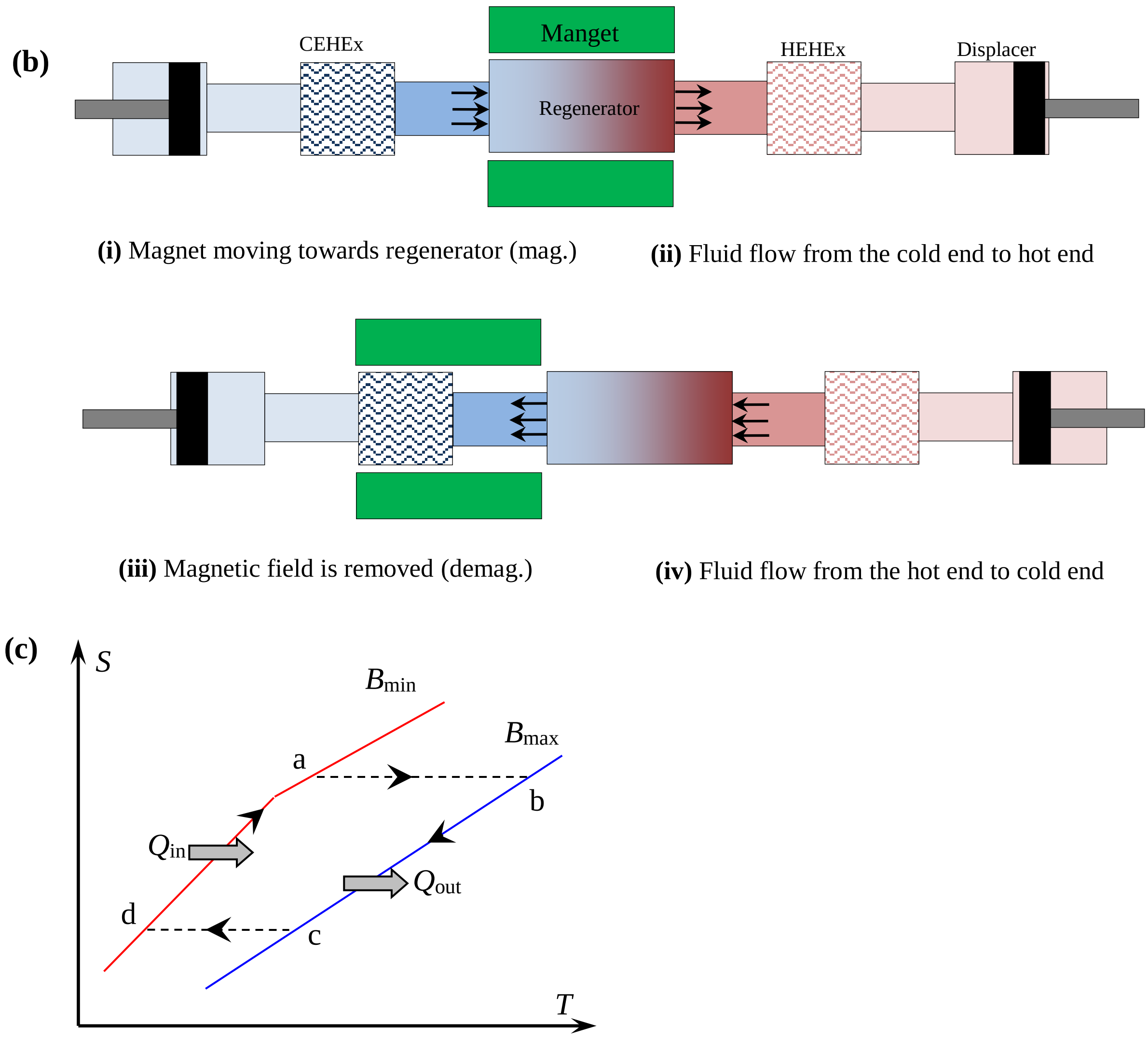


Figure 1 (a) Stage-wise comparison of AMR with VC cycle; (b) alternative schematic of four stages of the AMR operation; (c) *T*-*S* diagram representation for the AMR cycle

Based on the movement of the magnetocaloric material into the magnetic field, two kinds of configurations are possible: (a) Reciprocating: the regenerator moves in and out of the magnetic field in a reciprocating manner. Reciprocating motion can be provided either by moving the magnetic material or by moving the magnetic field source. Several reciprocating experimental systems have been developed (e.g. Clot et al. [17], Bahl et al. [18], Trevizoli et al. [15]). (b) Rotary: regenerator rotates about a fixed axis and moves in and out of the field in a circular path. The rotary arrangement can be very compact and can operate at relatively higher frequencies as compared to the reciprocating systems. Tura and Rowe [19], Zimm et al. [20], and Okamura et al. [21] have built rotary systems. A typical magnetic refrigerator prototype has the following main components: (1) regenerator: made of magnetocaloric material to work as a solid refrigerant; (2) heat transfer fluid: transports heat to and from the regenerator; (3) Magnetic field source: provides the magnetic field for MCE to take place; (4) hot end heat exchanger; maintains the hot-end temperature to ambient conditions, (5) cold end heat exchanger; provides a cooling load for the refrigeration system, and (6) hydraulic system: pumps heat transfer fluid in the system.

Like any other refrigeration system, temperature span, cooling capacity, and coefficient of performance are the major AMR performance indicators. The regenerative process in the AMR builds up a temperature gradient between its hot and cold ends, and in the absence of any cooling load, the steady-state temperature difference across two ends of the regenerator known as *the no-load temperature span* indicates AMR performance. The enthalpy flux at the cold end (usually a function of the mass flow rate of heat transfer fluid and MCE of the regenerator material) determines the cooling capacity of the system (Tura and Rowe [19]). Cooling capacity is, thus, the rate of heat absorbed by the heat transfer fluid from the cold end heat exchanger. After determining the amount of heat transfer taking place at the cold and hot ends, the COP of the system can be determined as the ratio of cooling capacity to work input (Aprea and Maiorino [22]).

3. AMR performance parameters

The AMR performance is influenced by magnetic, thermal-hydraulic, and operating parameters (Hall et al. [23] and Tura and Rowe [19]). The significance of these parameters is discussed, and the results of different research groups have been compared to examine the effect of each parameter on the AMR refrigeration performance. A chronological review of the geometric parameters and operating conditions along with the performance of different prototypes built so far is presented in Table 1.

Table 1 Chronological summary and review of the geometric. operating, and performance parameters of the AMR systems as reported in the literature
(this table is attached at the end of this manuscript)

3.1 Magnetic parameters

Magnetism and magnetocaloric properties are of key importance in determining the performance of a magnetic refrigerator. The common MCMs are Gadolinium and its alloys. Various aspects of magnetic parameters are given below:

3.1.1 Magnetocaloric material

The type of magnetocaloric materials used is important to the AMR performance. Pecharsky and Gschneidner [24] and Liu et al. [25] described the criteria to select a magnetic material for its application in magnetic refrigeration: Curie temperature is a key property of MCM for its application in magnetic refrigeration. Past Curie temperature, the magnetic domain structure of ferromagnetic material changes, and it becomes paramagnetic. In the vicinity of Curie temperature, the material exhibits the maximum values of the magnetization gradient $\partial M/\partial T$. Magnetic entropy ΔS_M and adiabatic temperature ΔT_{ad} are proportional to this gradient. A magnetic material with Curie temperature around the room temperature is ideal for near room temperature application e.g. Gadolinium Gd has its Curie temperature of 293.6 K which makes it a perfect candidate for magnetic refrigeration in the room temperature range. The magnetic material should exhibit a maximum change in its magnetic entropy and adiabatic temperature. The magnetic refrigeration cycle is based on a reversible cycle (Ericson, Brayton, etc.), the ferromagnetic material being used for the regenerator should exhibit minimum thermal and magnetic hystereses. Since the magnetic material acting as a solid refrigerant comes in direct contact with the heat transfer fluid (e.g. water, water-based mixtures, helium gas, etc.), it is essential that magnetic material does not corrode/react during the operation to ensure the maximum possible heat transfer with the fluid as the development of any chemical layer on the regenerator surface will affect the rate of heat transfer. The availability of magnetic material and the cost involved during its preparation are important factors while selecting the magnetic material. The magnetization factor is sensitive to the geometry or shape of the magnetic material and can result in lower magnetic field intensity inside the material (Dupuis et al. [26]).

Gadolinium is a benchmark working material for magnetic refrigeration and is most commonly employed in AMR prototypes built to date (Yu et al. [10]). As reported in the literature, the temperature span and cooling capacity of a single-material regenerator are significantly lower and the multi-material or layered-bed based regenerators have been reported to yield improved performance. MCMs with different Curie temperatures are layered along the flow direction. MCM with the lowest Curie point is laid near the cold end while MCM with the highest Curie point is laid near the hot end so that each material experiences maximum possible MCE. Arnold et al. [27] used a layered bed of Gd and Gd-Er and found that the average temperature of each material layer must lie near its Curie point for maximum possible temperature spans. FOMT materials undergo a giant magnetocaloric effect (Gschiener and Pecharsky [28]) and their use in the magnetic refrigeration experimental research has not become very common so far, probably due to large magnetic hysteresis of these materials. Engelbrecht et al. [29] conducted an experimental study to compare the performance of a two-material ($La(Fe,Co,Si)_{13}$ compounds) bed with single material (Gd) bed. They concluded that the two-material bed gives better performance only if the material transition temperature is between 286 K and 289 K, otherwise Gd bed gives maximum temperature span. Rowe and Tura [30], Green et al. [31], Gao et al. [32], Zimm et al. [33] and Okamura et al. [34] also employed Gd and its alloys to examine the performance of magnetic refrigeration systems.

3.1.2 Magnetic field intensity

The magnetocaloric effect is directly proportional to field intensity. A relatively larger change in the applied magnetic field would cause a bigger temperature change in the regenerator material leading to a greater temperature span across the hot and cold ends of the regenerator (Spichkin Y.I. [35]).

Magnetocaloric material experiences maximum MCE at its Curie temperature and is a function of field intensity as given by Pecharsky and Gschneidner [24].

$$\Delta T_{\mathrm{ad}} = 3.675\left(\mu_0 H\right)^{0.7} \tag{1}$$

where ΔT_{ad} is the adiabatic temperature change of the regenerator material and $\mu_0 H$ is the magnetic field intensity. A stronger magnetic field will produce larger MCE and hence a larger temperature span across the two ends of the regenerator. A permanent magnet assembly is an efficient, compact in size, and economical way of producing magnetic fields therefore it has commonly been used by research groups (Bjork et al. [11]).

3.2 Thermal-hydraulic parameters

An ideal AMR geometry should possess the following attributes: Uniform magnetization and demagnetization of the material, optimum heat transfer with the fluid to ensure maximum temperature gradient, the minimum fluid pressure drop as the fluid passes through the AMR, and minimum axial/longitudinal conduction. Passive regenerators of different geometric configuration can be used to test the heat transfer and fluid flow performance. Usually, passive regenerators are tested using a single-blow experimental technique. Mullisen and Loehrke [36], Heggs and Burns [37], Sarlah [38], Engelbrecht [39] and Tura A. [40] have conducted experiments on passive regenerators.

3.2.1 Heat transfer coefficient

The heat transfer coefficient depends upon the geometry of the material, thermo-physical properties of the fluid and solid, and the utilization (ratio of the thermal mass of fluid to the thermal mass of solid). Taking into account the effect of geometry on the heat transfer coefficient, packed particle bed gives the maximum contact area, which means a higher heat transfer coefficient but with a large pressure drop as compared to other geometries (Kays and London [41]). A trade-off has to be made while selecting the geometry for MCM. Colburn modulus *j* is a dimensionless parameter used to evaluate the heat transfer performance of a given geometry (Shah and Sekulic [42]). Colburn modulus is defined as:

$$j = \frac{Nu}{RePr^{1/3}} = StPr^{2/3} \tag{2}$$

where *St* is the Stanton number, *Re* is the Reynolds number, *Nu* is the Nusselt number and *Pr* is the Prandtl number.

3.2.2 Pressure drop

Pressure drop directly influences the magnetic refrigerator performance as large pressure drop would result in higher pumping work. Friction factor f is a dimensionless parameter used for comparison of different flow conditions (flow surface, mass flow rate, *Re*). As discussed by Shah and Sekulic [42];

$$f_{\mathrm{F}} = \Delta p \frac{d_h}{2\rho u^2 L_{\mathrm{r}}} \tag{3}$$

where f_{F} is Fanning friction factor, Δp is the pressure drop, d_{h} is the hydraulic diameter ($d_{\mathrm{h}} = 4V_{\mathrm{f}}/A_{\mathrm{ht}} = 4\varepsilon/a_{\mathrm{p}}$), ρ and u are respectively the density and velocity of the fluid, L_{r} is the length of the regenerator, ε is the void volume and a_{p} is heat transfer area per unit volume.

Heat transfer to flow friction ratio for different geometries and flow passages have been compared by Kays and London [41]. They described the heat transfer and pressure drop considering different flow situations in detail and found that parallel plate geometry gives the optimum performance. In a magnetic refrigeration system, the regenerator is exposed to a cyclic flow of hot and cold fluids. An experimental study using a single-blow method was conducted by Sarlah [38] to investigate the heat transfer and flow characteristics and results showed that parallel plate geometry gives the best performance out of the six tested regenerator-geometries.

3.2.3 Longitudinal/axial thermal conduction

Longitudinal/axial thermal conduction is the heat transfer along the direction of flow within the regenerator geometry. Regenerator geometries continuously connected in the flow direction allow higher longitudinal conduction and heat gets dispersed within the regenerator body; for instance, parallel plate geometry has higher longitudinal conduction as compared to the particle bed where the axial conduction is less due to discontinuous particles geometry. Minimum longitudinal conduction is required within the regenerator to have a high convective heat transfer with the heat transfer fluid. A system operating at relatively low cycle frequency and mass flow rate would increase the longitudinal heat conduction (Nielsen et al. [43]).

3.2.4 Regenerator porosity

Regenerator porosity is a very important geometric factor and has a major influence on the AMR performance. In principle, the regenerator in a magnetic refrigerator should have minimum porosity which means a maximum mass of the magnetic material resulting in larger MCE. Porosity ε is given as:

$$\varepsilon = 1 - \frac{V_{\mathrm{reg}}}{V} \tag{4}$$

The temperature span achieved in a regenerator is inversely proportional to its porosity. Porosity should have an optimum value to achieve larger temperature spans.

3.3 Operating parameters

The effects of cycle frequency, utilization (non-dimensional mass flow rate), heat rejection temperature, and cooling-load are discussed in this section.

3.3.1 Cycle frequency

The number of operating cycles completed in one second is the cycle frequency. The total time taken for the four stages of the cyclic repetition of the pre-defined stages (as explained under Section 2) can be written as:

$$\tau = \tau_1 + \tau_2 + \tau_3 + \tau_4 \tag{5}$$

where τ_1 and τ_3 are the periods of magnetization and demagnetization while τ_2 and τ_4 are the periods of cold and hot blows, respectively. These values are generally taken as $\tau_1 = \tau_3$ and $\tau_2 = \tau_4$. The increasing frequency will increase the number of MCEs undergone by MCM per unit time and hence would increase the system cooling power (Zhang et al. [44]). Equation (11) shows the relation of frequency and the specific exergetic cooling power of the system (Rowe [45]).

$$\mu_{\mathrm{ex,sp.}} = f\left(\frac{Ex_Q}{B_0 V_{MCM}}\right) \tag{6}$$

where f is the frequency of operating thermodynamic cycle. Experimental results of Russek et al. [46], Richard et al. [47], Yao et al. [48], Huang et al. [49], Lozano et al. [50], and Velazquez et al. [51] have been presented in Figure 2 to overview the range and the influence of cycle frequency on AMR temperature span. Although both Richard et al. [47] and Yao et al. [48] have determined the variation of temperature span for different gas pressures of helium being used as heat transfer fluid, Yao et al. [48] have employed comparatively higher gas pressures leading to a relatively sharper rise in temperature span. Velazquez et al. [51] has found a very slight variation in temperature span for utilizations of 0.19 and 0.29 using an aqueous solution of ethylene glycol as heat transfer fluid. The detailed geometric and operating parameters adopted by Velazquez et al. [51] are presented in Figure 2. Huang et al. [49] presented a sharp increase in temperature span for a small range of cycle frequency (between approx. 0.05 and 0.2). Lozano et al. [50] compared the experimental and numerical results using gadolinium as MCM and an aqueous solution of EG as heat transfer fluid. They revealed that the temperature span begins to decrease slowly after a cycle frequency of 2.0 Hz. Although, the performance is found to be proportional to the cycle frequency for all the discussed results, very high frequencies are not possible because of the mechanical limitations of fluid displacer or pump and movement of the magnetic field source. A maximum frequency of 4.68 Hz has been obtained by Russek et al. [46]. Another drawback of high cycle frequency is the high dead volume of heat transfer fluid (Kitanovski and Egolf [52]).

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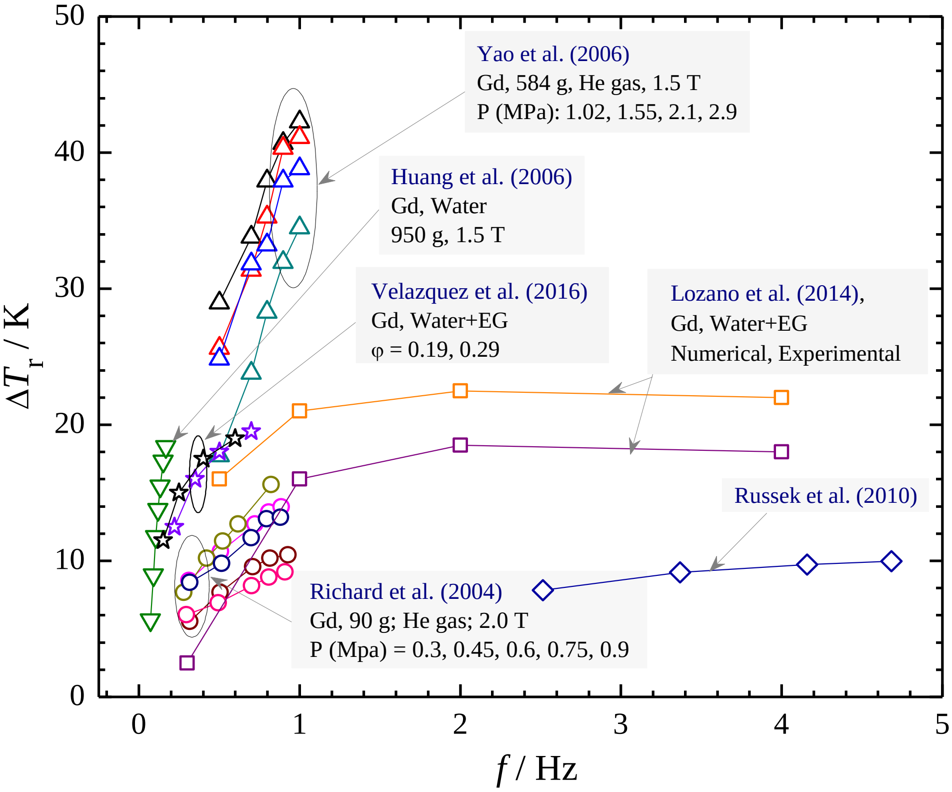

Figure 2 Temperature span vs cycle frequency

3.3.2 Utilization

Utilization is the key parameter in the design of AMR-setup. For a given mass of magnetocaloric material, the fluid mass flow rate in the system can be determined by selecting an appropriate range of utilization factor (usually from 0.01 to 1.5). Utilization is defined as the ratio of the thermal mass of the heat transfer fluid to the thermal mass of MCM.

$$\varphi = \frac{m_{\mathrm{f}}\, c_{\mathrm{p,f}}\, \tau_{\mathrm{flow}}}{m_{\mathrm{MCM}}\, c_{\mathrm{P,MCM}}} \tag{7}$$

where $c_{\mathrm{p,\,MCM}}$ is the heat capacity of regenerator material and is considered at $B = 0$, $T = T_{\mathrm{C}}$, and τ_{f} is the time period for hot or cold flow. Utilization can be increased by increasing the fluid mass flow rate in the system. Results of Richard et al. [47], Tagliafico [53], Engelbrecht [39], Trevizoli et al. [15], Aprea et al. [54], Lozano et al. [55], Velazquez et al. [51] and Ezan et al. [56] are plotted in Figure 3 to examine the variation of no-load temperature span with respect to utilization. In the operating range of utilization of Richard et al. [47] given in Figure 3, which is much below 0.1, the temperature span sharply increases with the utilization. Both, Tagliafico [53] and Ezan et al. [56] observed the variation of no-load temperature span with utilization for different cycle frequencies but the operating range of cycle frequencies and the obtained no-load temperature spans presented by Tagliafico [53] are relatively much higher. Engelbrecht [39] performed an experimental investigation to determine the variation of no-load temperature spans with utilization using a parallel plate regenerator with different spacing between them. It can, further, be observed from the figure below that Aprea et al. [54] and Velazquez et al. [51] presented no-load temperature spans for different heat rejection temperatures and mass flow rates, respectively. The overall trend for most of the results is that temperature span is maximum when utilization ranges from ≈0.05 to ≈0.2 and decreases with further increase in utilization. Although, increasing utilization gives better heat transfer but simultaneously it is associated with high pressure drop which increases the required pumping power (Richard et al. [47], Tura and Rowe [57]).

For gaseous heat transfer fluids, utilization can be varied by changing the mass flow rate or pressure of the gas. Changing operating pressure changes the density of the gas and hence the thermal mass (Tura et al. [58]). The performance of AMR using gaseous fluids increases as the charging pressure is increased as shown in Figure 2 above. For an AMR system under given geometric and operating conditions, there exists an optimum utilization which gives the optimum performance contrary to the efficiency of the passive regenerators which is inversely related to the utilization (Richard et al. [47]).

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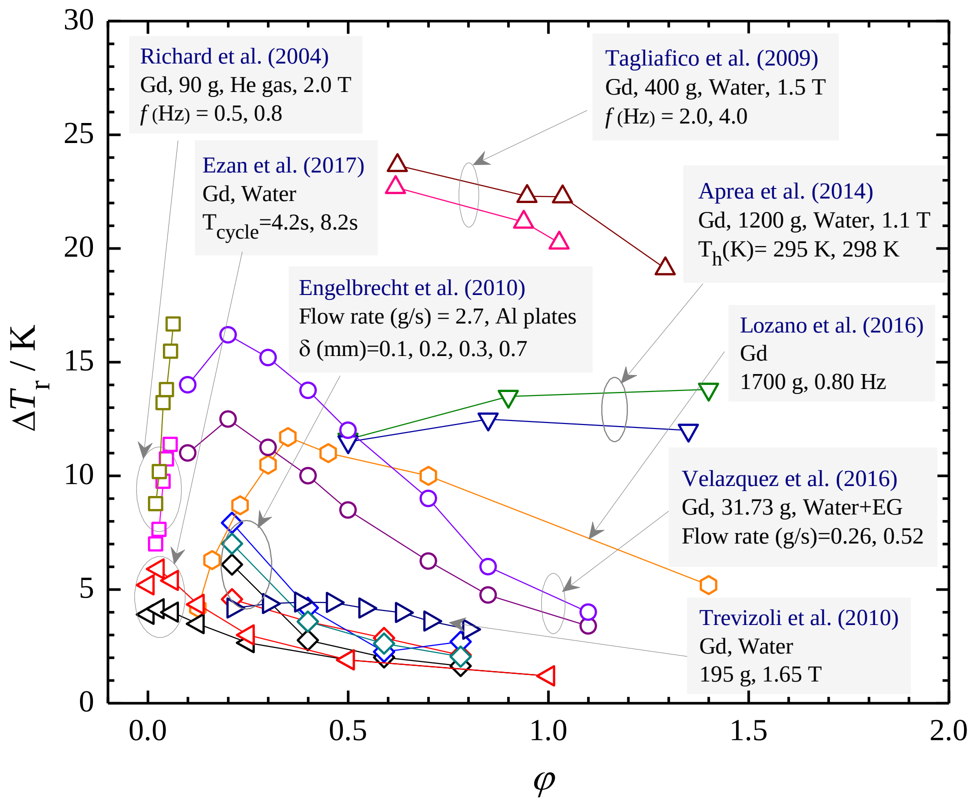

Figure 3 Temperature span vs utilization

3.3.3 Heat transfer fluid

A high heat transfer rate is essential in the AMR devices and heat transfer fluid to be used should possess high thermal conductivity k to maximize volumetric heat transfer, high heat capacity c to minimize flow rate, and low viscosity μ to minimize pressure drop and frictional losses (Zimm et al. [59]). Hence kc/μ value determines the suitability of certain heat transfer fluid for AMR application. Mostly water or water-based heat transfer fluids have been used while gaseous fluids (Helium, Air, and Nitrogen) have also been used by Tura et al. [60] and Zhang et al. [44]. Zhang et al. [44] conducted experiments with helium and nitrogen as heat transfer fluids and concluded that helium gives better performance than nitrogen. Bahl et al. [18] employed four different types of heat transfer fluids in the regenerator; water-ethanol, ethylene glycol, propylene glycol, and olive oil. Water-based heat transfer fluids gave the best performance. Kitanovski and Egolf [52] discussed the desired fluid and flow characteristics for magnetic refrigeration applications. Along with the above-mentioned desired properties of heat transfer fluid; laminar flow in the regenerator with minimum entrance and exit losses is also required to minimize the overall regenerator pressure losses.

3.3.4 Heat rejection temperature

Heat rejection temperature significantly affects the cooling performance. Tura et al. [58] found that the sensitivity of temperature span to hot end temperature increases as the cooling load increases. Experimental results of Rowe and Tura [30], Aprea et al. [54], Lozano et al. [50], Trevizoli et al. [61], Tura et al. [60], and Tagliafico [53] are compared in Figure 4 to examine the effect of heat rejection temperature on temperature span. Rowe and Tura [30] and Aprea et al. [54], both have studied the variation of temperature span with heat rejection temperature for cycle frequencies of 0.65 Hz and 1.0 Hz but the temperature span obtained by Rowe and Tura [30] is significantly higher which is mainly due to application of gadolinium alloys and increased strength of the magnetic field. Some other factors leading to a higher temperature span developed by Rowe and Tura [30] are presented in Table 1. For gaseous fluid, the temperature span strongly depends upon the gas pressure in case of gaseous heat transfer fluids, the same has also been presented by Tura et al. [60] for gadolinium alloys as MCM at magnetic field intensity of 2.0 T. Although, we can observe nearly the same parabolic trend of temperature span against heat rejection temperature for majority of results, for Trevizoli et al. [61] it is found to increase continuously in the entire range of rejection temperature. The detailed geometric and operating parameters adopted by Trevizoli et al. [61] and the performance achieved are also reflected in Table 1. It can, further, be noted that there lies an optimum heat rejection temperature for given magnetic and geometric conditions of AMR system. Optimum heat rejection temperature is seen to lie in the range of 295 – 300 K so the temperature span is found to decrease before and after this temperature range.

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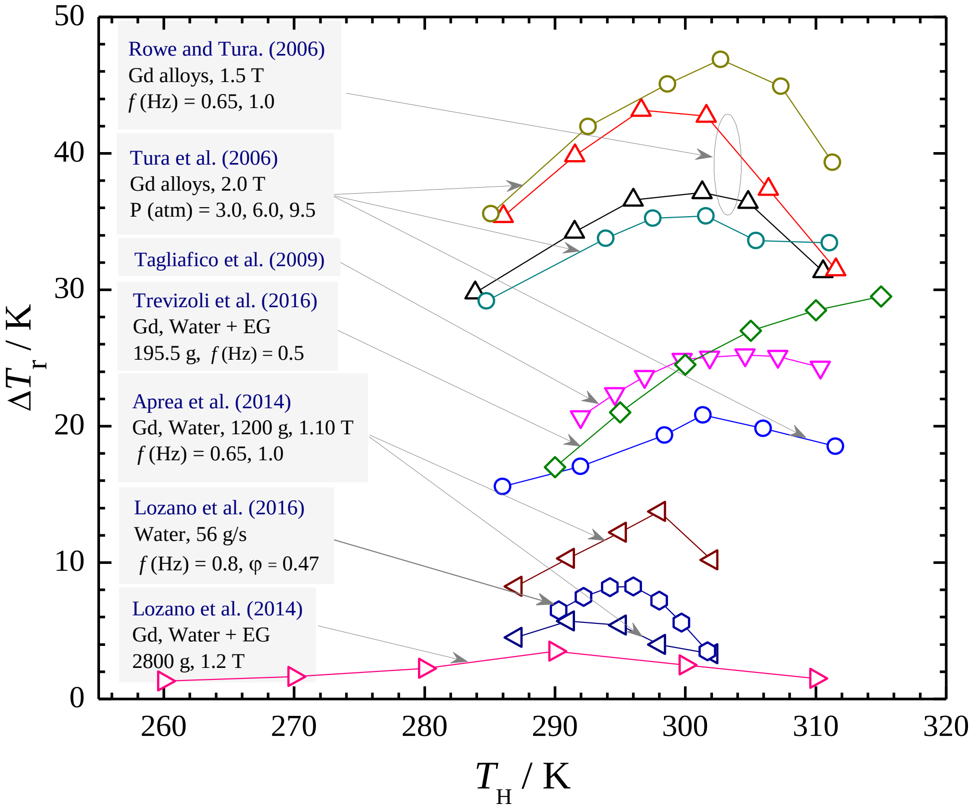

Figure 4 Temperature span vs hot end temperature

3.3.5 Cooling load

Cooling load, being the amount of heat absorbed at the cold heat exchanger, is the most important parameter for performance quantification of the system. The variation of temperature span with the cooling load for some of the research studies is compared and shown in Figure 5. For instance, Zimm et al. [33], Arnold et al. [27], Bahl et al. [62], Eriksen et al. [63], Saito et al. [64], Trevizoli et al. [61], You et al. [65], Lozano et al. [55] and Aprea et al. [66] have reported the variation of cooling load on the temperature span for their respective experimental setups. For Arnold et al. [27] and Trevizoli et al. [61] a sharp decline can be observed in temperature span in the range of cooling load from 0-30 W for different magnetic field strengths. Gd and its alloys were employed by Zimm et al. [33] and Saito et al. [64] to examine the variation in temperature span with the cooling load. Bahl et al. [62], Eriksen et al. [63] and Aprea et al. [66] have determined and presented the performance over a relatively larger range of cooling loads. The research results depict a gradual reduction (as expected) in temperature span when the cooling load increases, however, the slope of decline differs from one dataset to another. A lower negative slope is desired to ensure noticeable cooling against larger temperature span. Negligible temperature span is produced at very high cooling loads as presented by Bahl et al. [62] and Aprea et al. [66].

Double column figure

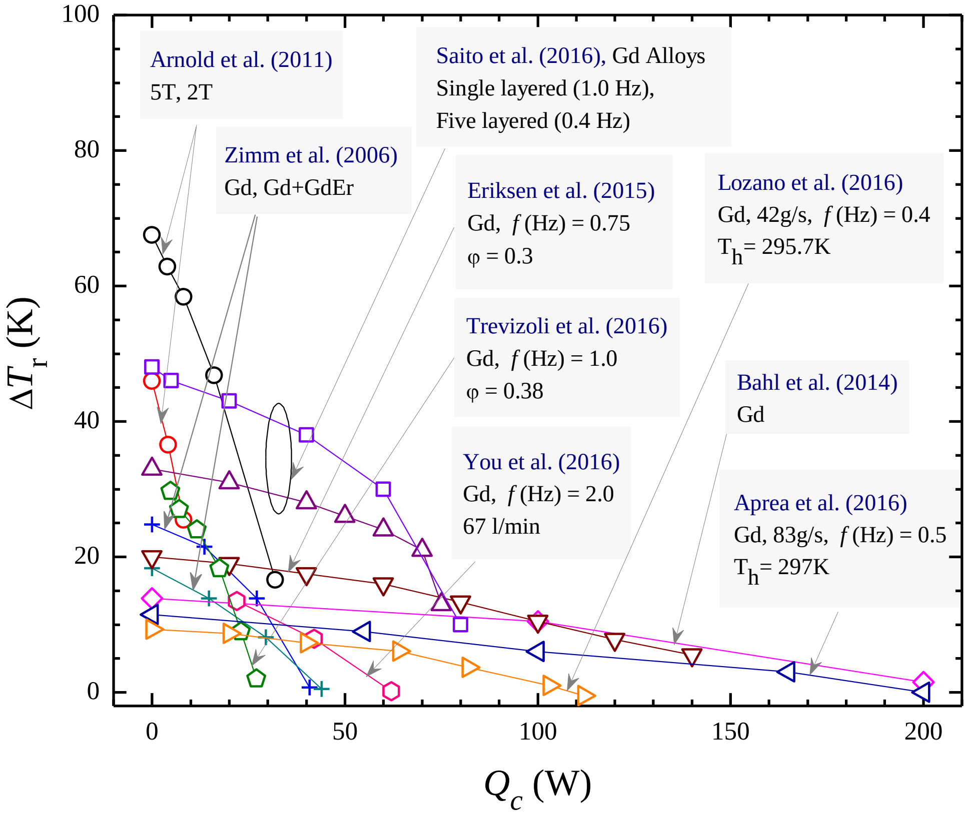

Figure 5 Variation of temperature difference of two reservoirs with the cooling load

3.3.6 Regenerator sequence and phase angle

In reciprocating AMR system with two regenerator beds (e.g. Tura et al. [60], Zhang et al. [44]), mass flow rate and magnetic field profiles are synchronized to maintain the stage sequence of the operating thermodynamic cycle. During the fluid flow through the beds, the magnetic field source is stationary and vice versa. The phase angle between the two profiles may influence system performance. Zhang et al. [44] investigated the effect of the phase angle. For multi regenerator device, the effect of flow and magnetization/demagnetization sequence, as well as phase angle, is relatively under-reported in the literature.

4. Challenges and opportunities

A comparison of the few prominent prototypes is presented in Table 2. The system performance is compared in terms of cooling capacity and temperature span. The geometric and operating parameters such as geometry, material, size, and no. of regenerators, cycle frequency, and magnetic field change are also listed. Superconducting magnets, at present, can provide a noteworthy temperature span (Teyber et al. [85]) of 100 K. However, the use of a superconducting magnet has its challenges when considering the overall size, system performance, and market competitiveness of the refrigeration unit, which might hamper the ability of AMR refrigerators to replace the conventional vapor-compression systems. The comparison below reveals that although the relative system performance has improved over time, it is still not ready to compete against vapor-compression. The technology is in dire need of major research breakthroughs (a) in the materials domain to obtain high MCE materials; (b) in magnetics to achieve higher rates of magnetic field change and (c) in system design to increase cycle frequency.

Table 2 A comparative summary of few AMR prototypes

Research group	**Geometry**	**MCM**	V_{reg} **cm**^{3}	n_{reg}	***f*** **Hz**	ΔB **T**	$Q_{c,max}$ **W**	ΔT_{max} **K**
Astronautics Laboratory, USA [33, 67, 68]	Spheres, Spheres, Plates	Gd, (Gd, GdEr), Gd	600, 33, 242	2, 6, 12	0.167, 4	5, 1.5, 1.5	(100, 600), 15, (27, 0)	(38,0), 14, (14, 25)
Chubu Electric, Toshiba, Japan [69]	Spheres	Gd	484	2	0.167	4, 2	100, 40	26, 24
Institute of Tech., Chubu, Japan [70]	Spheres	GdDy	844	4	0.39, 0.42	1.1	540	0.2
Nanjing University, China [71]	Spheres	Gd, GdSiGe, GdDy	200	2	0.25	1.4	0, 0, (0, 40)	23, 10, (25, 5)
Riso Lab, Denmark [18, 50, 62, 72-74]	Spheres	Gd	23	24	2.25	1.24	200	18.9
POLO Research Lab, Brazil [15]	Plates	Gd	34	1	0.143	1.65	3.9	4.45
University of Victoria, Canada [30, 75-78]	Powder/ Spheres	Gd, GdTb, GdEr, Gd, Gd	74, 74, 74, 49, 25	2	1, 1, 1, 0.8, 0.6	2	0, 0, 0, 0, 7	50, 50, 50, 15.5, 14
University of Ljubljana, Slovenia [79-81]	Plates, Cylinders, Powder, Spheres	LaFeCoSi, Gd	32	1	0.15 - 0.45	1.15	0	23
University of Salerno, Italy [54]	Particles	Gd	31.5	8	0.36- 1.79	1.25	0	13.5
G2E Lab, Grenoble, France [82]	Plates	Gd $PrSrMnO_3$ LaFeCoSi	14, 21, 24	1	0.1 - 1.43	0.8	0	11.5, 10.5, 5
University of Tokyo, Japan [83]	Particles	Gd	-	1	0.25	1.07	0	6.5
Wroclaw Uni. of Technology, Poland [84]	Particles	Gd	-	1	0.025	1.0	0	1.6
Teyber et al. [85]	Spheres	Gd	-	2	0.25	3.3	-	100

5. Concluding remarks

Magnetic refrigeration is a ‘greener’ cooling technology since it has zero ozone-depletion potential due to the use of magnetic material as a solid refrigerant. Magnetic refrigeration has high energy efficiency as the compression and expansion stages of the VCC are replaced with magnetization and demagnetization stages. An electromagnet or a permanent magnet provides the required magnetic field and thus there is no need for a compressor with moving parts having large vibrations, noise, and high rotational speeds. The performance of the AMR refrigerator is sensitive to various geometric and operating parameters. The sensitivity factor varies majorly depending on system configuration (utilization and cycle frequency), applied magnetic field change, and regenerator geometry. The influence of key parameters has been evaluated based on the findings reported in the literature and the following concluding remarks are being made:

- It has been found that the AMR performance is proportional to the cycle frequency, but very high cycle frequencies cannot be employed owing to the limitations regarding movements of the displacer or pump. A maximum cycle frequency of 4.68 Hz has been used by Russek et al. [46].
- The performance is very sensitive to utilization maximum temperature span has been obtained in a range of utilization from ~0.05 to ~0.2 by the majority of the research groups.
- The type of heat transfer fluid significantly affects the AMR performance and water-based heat transfer fluids have been found to yield optimal performance.
- A trend closer to parabolic in nature was found to exist between heat rejection temperature and temperature span, implying that there is an optimum value of heat rejection temperature which gives the maximum temperature span at given geometric and operating conditions. A reduction in temperature span as cooling load increases with nearly zero temperature span at very high cooling loads has been reported by several research studies.
- The peak heat rejection temperature was seen to lie in-between 25 – 30 ^{o}C for the studied prototypes. This appears to restricts the applicability of the technology especially in hotter climates where the ambient temperature is ~40^{o}C for a significant fraction of the year.
- The absence of an energy-demanding compression stage makes AMR refrigerator highly energy-efficient, however, the research focus is required to improve the cooling performance of the system to broaden the commercial horizons of this technology.
- For its marketability, the technology requires major research breakthroughs in the field of system design, higher magnetic fields, and advanced magnetocaloric materials. Multi-material regenerator, high strength magnetic fields, and higher cycle frequency are the potential promising features which may contribute to achieving the higher performance of the system.
- The literature review reveals a strong dependence on rare earth metals in the regenerator (e.g. Gd) as well as in the magnetic field source (e.g. NdFeB). This compromises/undermines the sustainability of the technology. The development of sustainable materials is thus a major challenge to unleash higher system performance as well as for a lower initial cost of the system.

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Table 1 Chronological summary and review of the geometric. operating, and performance parameters of the AMR systems as reported in the literature

References	Geometric parameters						Operating parameters					Performance		
	Regenerator geometry	V_{reg} (mm^3)	MCM	m_s (g)	d_p (μm)	n_{beds}	ΔB (T)	HTF	$\dot{m}$ (g/s)	φ	f (Hz)	P (kPa)	Q_C (W)	COP
(1976) ↓														
Brown [3]	Plates (1mm thick)	-	Gd	-	-	-	7.0	Water + Ethanol	-	-	-	-	-	-
(1984) ↓														
Kirol and Mills [86]	Foils 0.076 mm δ=0.127	-	Gd	-	-	-	0.9	Water	-	-	0.5	-	-	-
(1990) ↓														
Green et al. [31]	Ribbon	-	Gd-Tb	-	-	-	7.0	N_2 gas	-	-	0.02	-	-	-
(1998) ↓														
Zimm et al. [67]	Particles	-	Gd	1500	130 -150	-	5.0	Water	41-100	-	0.167	-	500	6
(2000) ↓														
Bohigas et al. [87]	Ribbon	wheel d=75, 110 w=8	Gd	-	-	-	0.95	Water	-	-	0.8	-	-	-
(2002) ↓														
Blumenfeld et al. [88]	Particles	d = 25 L = 160	Gd	-	-	-	1.7	Water	4	-	0.033	-	3	-
Hirano et al. [69]	Particles	d = 25, 80	Gd	2200	300	2	4.0	Water	83	-	0.0167	-	100	
Rowe et al. [89]			Gd				2.0	Helium						
(2003) ↓														
Clot et al. [17]	Plates 1mm thick δ=0.15	-	Gd	223	-	-	0.8	Water	0.5 – 4.8	-	-	-	8	2.2
(2004) ↓														
Richard et al. [47]	Flakes ε=0.57	d = 25	Gd, Gd-Tb	90, 80	-	2	2.0	He gas	-	0.01-0.07	0.2 – 1	-	-	-
(2005) ↓														
Lu et al. [71]	Particles	-	Gd	-	200	-	1.4	Water	-	-	-	-	-	-
Shir et al. [90]	Particles	-	Gd	-	200		1.4	Helium	-	-	-	-	-	-
(2006) ↓														
HUANG et al. [49]	Particles	-	Gd, $LaFe_{13-x-y}Co_xSi_y$	950	500 – 2000	2	1.5	Water (alkaline)	-	-	0.178	200	20	-
Kawanami et al. [91]	Chips	4×8×60	Gd	1.5	-	10	0.9	Air	0.017 – 0.02	-	-	-	-	-
Okamura et al. [21]	Particles ε=0.37	-	4 Gd-based alloys	-	-	-	0.77	Water	33 – 66	-	0.6	-	60	-

References	Geometric parameters						Operating parameters					Performance		
	Regenerator geometry	V_{reg} (mm^3)	MCM	m_s (g)	d_p (μm)	n_{beds}	ΔB (T)	HTF	$\dot{m}$ (g/s)	φ	f (Hz)	P (kPa)	Q_C (W)	COP
Rowe and Tura [30]	Particles ε=0.55	-	Gd and its alloys	270	550	-	1.5, 2	Helium	-	-	0.65 – 1.0	300, 600, 950	-	-
Yao et al. [48]	Particles	-	Gd	584	-	-	1.5	Helium	-	-	-	-	-	-
Zimm et al. [33]	Particles	-	Gd, $Gd_{0.94}Er_{0.06}$	-	425 – 355	-	1.5	-	-	-	0.5 – 4	-	-	-
(2007) ↓														
Buchelnikov et al. [92]	-	-	Gd, Mn-alloy	-	-	-	1.0	Water	-	-	10	-	40	-
Chen et al. [93]	Particles	-	Gd	1000	500	-	1.5	Water	-	-	0.7	-	40	-
Muller et al. [94]	Particles	-	Gd	-	-	-	1.3	Water	-	-	1	-	-	-
Petersen et al. [95]	Plates δpl = 1mm	-	Gd	-	-	-	1.2	-	-	-	0.56 – 0.83	-	-	-
Tagliafico et al. [14]	Particles	-	Gd	1167	-	2	1.5	Helium	-	-	0.4 – 1.0	500 – 2900	51	-
Zimm et al. [2]	Plates	-	Gd	-	-	-	1.5	Water	-	-	4	-	-	-
(2008) ↓														
Bahl et al. [18]	Plates 0.9×25×40 mm^3 δ_{ch} = 0.8 mm	-	Gd	92	-	-	1.4	WE,EG, PG,OO	-	-	0.56 – 0.83	-	-	-
Nakamura et al. [96]	Particles ε=0.374	-	Gd	33.4	300	2	2.0	Water, Air	-	-	0.2	-	-	-
(2009) ↓														
Bour et al. [97]	Sheets 0.63×12×100 mm^3 δ_{ch} =0.1-0.2 mm	-	Gd	250	-	-	1.1	Zitrec S	-	-	0.05 – 0.3	-	-	-
Coelho et al. [98]	Pins 1×1×60	Pvc pipe d = 250	Gd	960	-	-	2.3	Ethyl alcohol	-	-	0.4 – 0.5	-	-	-
Dupuis et al. [26]	Sheets 1mm thick	L = 50 d= 30	Gd	-	-	-	0.8	-	≤ 23	-	1	-	-	-
Hirano et al. [99]	-	-	La alloy	-	-	-	2.0	Air	-	-	-	-	-	-
Pryds et al. [100]	Sheets 0.3mm thick δ = 0.2	-	La-Mn LCMS	-	-	-	2.0	Water+ ethanol	-	-	-	-	-	-
Tura and Rowe [57]	Particles ε = 0.36	L = 110 d_i = 16	Gd	110	300	-	1.5	Water + glycol	-	0.62 – 1.28	0 – 5	300	-	1.6
(2010) ↓														
Okamura et al. [34]	Particles ε = 0.36	h =10,16 L =200,320 w = 8	Gd $Gd_{0.89}Dy_{0.11}$ $Gd_{0.92}Y_{0.08}$	4000	600	-	1.1	Water	50 – 100	-	-	-	540	2.5
Tagliafico et al. [101]	Particles ε = 0.46	-	Gd	400	300	2	1.5	Water	20	-	≤ 2	-	-	-
Tušek et al. [102]	Plates	h = 10	Gd	600	-	34	0.98	Water	-	-	0.25 –	-	-	-

References	Geometric parameters						Operating parameters					Performance		
	Regenerator geometry	V_{reg} (mm^3)	MCM	m_s (g)	d_p (µm)	n_{beds}	ΔB (T)	HTF	$\dot{m}$ (g/s)	φ	f (Hz)	P (kPa)	Q_C (W)	COP
	0.3 mm $\varepsilon = 0.59$	$L = 50$ $w = 10$									4.0			
Zhang et al. [44]	Particles $\varepsilon = 0.64$	-	Gd	320	800–1400	2	1.5	He, N_2	-	-	0.45, 0.6	-	20.5, 11	-
(2011) ↓														
Arnold et al. [27]	Particles	$L = 25.4$ $d_i = 25.4$ $d_o = 28.6$	Gd, Gd-Er	33, 35	550	2	2 and 5	Helium	-	0.16	1	850 – 900	-	0.7, 2.4
Engelbrecht et al. [29]	Plates 1mm thick $\delta = 0.5$	$h = 17$ $L = 40$ $w = 23$	$La(FeCoSi)_{13}$ (LaCaSr)MnO Gd	71.3 34.1 78.2	-	1	1.4	Water + EG	-	0.55	0.125	-	-	-
Kawanami et al. [16]	Particles	$L = 60$ $d_i = 12$	Gd	-	-	-	2.0	Water, air	-	-	-	-	-	-
Kim and Jeong [103]	Particles $\varepsilon = 0.5$	$L = 47$ $d = 12.7$	Gd	21	325–500	1	1.57	Helium	-	-	1	1700	-	-
Trevizoli et al. [15]	Plates 0.85x126	$L = 160$ $h = 7$ $w = 30.3$	Gd	195	-	-	1.65	Water	-	-	-	-	3.9	-
Tura and Rowe [19]	particles	$L = 47$ $d = 12.7$	Gd	55	300	1	1.4	Water+ glycol	-	-	-	-	-	10
(2012) ↓														
Bahl et al. [18]	Plates $n_{pl} = 28$ 0.3×25×40 mm^3	20	LaCaSrMn-1 LaCaSrMn-2	51.1	-	1	1.1	Water-glycol	-	0.3-0.7	0.11	-	-	-
Engelbrecht et al. [53]	Particles	5.7×10^5 (0.57 dm^3)	Gd	2800	250–800	24	1.24	Water-glycol	55–190	-	≤ 8	-	100 (21 K)	1.8
Tura et al. [77]	Plates 0.1×15×80 mm^3	-	Gd	60	-	1	1.47	Water	-	0.3–0.8	0.5–2	-	-	-
(2013) ↓														
Park et al. [104]	Spheres	-	Gd	186	300–700	2	1.4	Water	5–15	-	0.5	-	33	-
Tagliafico et al. [105]	Parallel-plate 0.8×8×100 mm^3	50×8×100	Gd	388	-	2	1.55	Water-ethanol	5–20	-	≤ 0.25	-	-	-
Tušek et al. [80]	Plates & spheres	-	Gd	93 –176	-	-	1.15	Water-glycol	0.8–40	-	≤ 0.33	-	-	-
(2014) ↓														
Arnold et al. [78]	Spheres	22 cm^3, 57cm^3	Gd	650	-	2	1.47 – 1.54	Water-glycol	-	-	≤ 4	-	-	-
Legait et al. [82]	Single & multi MCM plates 0.5×10×80 mm^3	-	Gd $Pr_{0.65}Sr_{0.35}MnO_3$ $La(FeCo)_{13-x}Si_x$	190, 150, 91	-	1	0.8	Water	1 – 5	< 1	0.18 – 1.4	-	-	-
Tušek et al. [81]	Plates 0.5×10×80 mm^3 0.25×10×80 mm^3	40×10×80	Gd, $La(FeCo)_{y-x}Si_x$	176, 144	-	1	1.2	Water-antifreeze	0.6 – 20	-	0.15 – 0.45	-	-	-
Aprea et al. [54]	Spheres	-	Gd	1200	400 -	8	1.10-1.25	Distilled	117	0.50	0.36 -	170	-	-

References	Geometric parameters						Operating parameters					Performance		
	Regenerator geometry	**V_{reg} (mm^3)**	**MCM**	**m_s (g)**	**d_p (μm)**	**n_{beds}**	**ΔB (T)**	**HTF**	**$\dot{m}$ (g/s)**	**φ**	**f (Hz)**	**P (kPa)**	**Q_C (W)**	**COP**
					500			water		-2.72	1.79			
Bahl et al. [62]	Spheres	L = 50 L = 100	Gd	1400 2800	350 - 850 250 - 800	24	1.0	Water + EG	56 – 168	0.20 -0.55	≤ 2.25	-	1010	-
Lozano et al. [50]	Spheres	L = 12.5 w = 18.6	Gd	2800	250 - 800	24	0.3-1.2	Water + EG	≤170	-	≤ 10.0	-	640	-
Jacobs et al. [68]	Spherical Particles	L = 38 $30cm^3$	$LaFe(Si)_{13}H$	1520	177 - 246	12	1.44	Water	208 – 353	-	4.0	124	3042	3.35
(2015) ↓														
Lei et al. [106]	Packed Spheres	L = 100	La(Fe,Mn,Si)13Hy	2010	300	20	1.2	-	-	-	2.0	-	-	6.70
Eriksen et al. [63]	Spheres	L = 90.2 w = 32.7 h = 11	Gd	1700	500 - 600	11	0-1.2	Water + EG	50	< 1	0-4	-	102.8	3.1
Lee [107]	Spheres	L = 100 d_o =15.8 d_i =14.2	Gd	-	250	2	0.8	Water	10	-	0.25	-	5.0	-
Lei et al. [108]	(i) Packed bed (ii) Parallel-plate (iii) Microchannel (iv) Packed screen	2.24×10^4	Gd	-	140 - 570	20	1.2	Water + EG	-	-	0.3 - 10	-	100 W/kg	(i) 7.6 (ii) 11.2 (iii) 10.1 iv) 9.6
Lionte et al. [109]	Parallel Plates δ = 0.17 ε = 0.215	L = 100 w = 20 h = 23.7	Gd	-	-	-	0-1.1	Water	-	-	0.3 -2.0	0-6.0	18.5	5.2
Aprea et al. [110]	Packed bed	L = 45 h = 20	(i) Gd (ii) $Pr_{0.45}Sr_{0.35}MnO_3$ (iii) $Gd_5Si_2Ge_2$ (iv) $LaFe_{11.3}Mn_{0.3}Si_{1.2}H_{1.5}$ (v) $MnFeP_{0.45}As0_{.55}$		450		0-1.5	Water	83.35	-	1.25	-	(i) 265 (ii) 45 (iii) 465 (iv) 360 (v) 50	(i) 3.0 (ii) 0.5 (iii) 5.0 (iv) 4.0 (v) 0.6
Vuarnoz et al. [111]	Parallel plate 1.0mm thick	30×8×50	Gd	0.057	-	20	-	Water	-	1.5	0.25	-	-	-
(2016) ↓														
You et al. [65]	Spheres ε = 0.362	L = 51.44 w = 15.9 h = 6.6	Gd and $Gd_{0.73}Tb_{0.27}$	-	250	3	0-1.8	Water + EG	1.67 – 23.3	-	2.0	-	874.7 kW/m^3	9.7
Trevizoli et al. [61]	Spheres ε = 0.36	L = 100 d_i =14.2	Gd	195.5	500 - 600	-	0.04-1.69	Water + EG	1.3-10.7 2.6-21.4 5.2-28.6	0.14-1.15	0.25 0.50 1.0	80	53.7	4.65
Aprea et al. [66]	Packed bed spheres 45×35×20 mm^3	31.5 cm^3	Gd	1200	400 - 500	8	1.25	Water	83	-	0.1- 1.0	-	50-163	0.6-1.8
Lozano et al. [55]	Spheres ε = 0.403	L = 80 w = 29.7 h = 10	Gd	1700	425 - 600	16	0-1	Water – antifreeze	28-56	0.12-1.4	0.8 – 1.4	-	150	0.9

References	Geometric parameters						Operating parameters					Performance		
	Regenerator geometry	V_{reg} (mm^3)	MCM	m_s (g)	d_p (μm)	n_{beds}	ΔB (T)	HTF	$\dot{m}$ (g/s)	φ	f (Hz)	P (kPa)	Q_C (W)	COP
Velazquez et al. [51]	Spheres $\varepsilon = 0.39$ & 0.34	60×10×5.4 3.19 cm^3	Gd	31.73	200 - 400	2	0-1.4	Water+ EG	0.26-2.61	0.1-1.1	0.04 - 0.82	-	6.0	0.7
Saito et al. [64]	Particles	-	(i) Gd (ii) $Gd_{0.985}Y_{0.015}$ (iii) $La(Fe_{0.86}Si_{0.14})_{13}H_{1.2}$ (iv) $La(Fe_{0.85}Co_{0.07}Si_{0.08})_{13}$	1000	500 - 850	-	0.8	Water, Water+EG	-	-	0.4-1.0	-	0-3.6	-
(2017) ↓														
Ezan et al. [56]	Parallel plates 1.0mm thick	$L = 200$ $w = 100$ 0.00025 m^3	Gd	-	-	-	0.27-0.98	Water, EG, Ethanol-Water	-	0.0032 – 1.0129	0.24, 0.16, 0.12	-	60, 40, 30 W/m	0.7
You et al. [112]	Parallel plates 0.9mm thick	$L = 200$ (i) w = 0.2 (ii) w = 0.4 (iii) w = 0.6 (iv) w = 0.8	Gd	-	-	-	0.16-1.0	Water	-	-	1.0	(i) 1.1 (ii) 0.6 (iii) 0.3 (iv) 0.085	-	-
Trevizoli et al. [113]	(i) Parallel plates (ii) Pin Array (iii) Packed spheres $\varepsilon = 0.36$-0.37	$L = 100$ $w = 13.12$ $h = 13$	Gd	-	(i) 750 (ii) 795 (iii) 780	-	0.04-1.69	Water+EG	-	0.26-2.7	0.25, 0.50, 1.0	-	(i) 19.0 (ii) 30.5 (iii) 33.5	(i) 1.4 (ii) 1.4 (iii) 1.2
(2018) ↓														
Monfared [114]	Packed beds 15 cm length	*31.2±0.1 cm3*	La(Fe,Mn,Si)13Hz particles.	-			1	Water+EG (15%)			0.5, 2.2 ,2.8		4.1	
Navickaite [115]	Multi-layered regenerators	*D=30* *D=34* *H=40* *(Dxh)* *30X40* *34X40*	La(Fe,Si,Mn)13Hy	-	250-500	2,5,9 layered	1.1	water		0.42-0.45 0.3-0.9	0.15		3.5 W	
Fortkamp et al. [116]	particles	*L= 90.2* *W= 32.7* *H= 11*	Gd-Y	-		11	1.34				1		50w,80w	1.6,6.1
Tian Lei [117]	packed particle beds.		La(Fe,Mn,Si)13Hy	-				Water+EG (20%)			0.5-1.25		5.7	
Taskaev et al. [118]	Stacked plates, stacked wires, packed bed structures		Gd-Y, Gd-In and Gd-Zr	-			2-3				1-10			

Highlights

- Magnetic field and magnetocaloric material are crucial parameters determining overall performance.
- Regenerator geometry and heat transfer fluid strongly influence regenerator temperature gradient.
- Utilization and cycle frequency require optimization to meet a given cooling load requirement.