The energy performances of a rotary permanent magnet magnetic refrigerator

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Abstract

The interest of the scientific community regarding magnetic refrigeration at room temperature is constantly growing. In recent years, there has been an increase in the experimental work relating to both new prototypes and new magnetocaloric materials. The magnetic refrigerators built to date still have some limitations that make them uncompetitive when compared with conventional vapour compression systems. However, among the different configurations realized, one can recognize that rotary devices, having rotating magnets and static regenerators, are of particular interest because of their good energy performances. In this paper, we report an experimental investigation on the identification of the energy performances of a Rotary Permanent Magnet Magnetic Refrigerator. Employing 1.20 kg of gadolinium and operating at a temperature of heat rejection equal to 296 K, the system was subjected to different operating conditions obtained by varying the thermal load, volumetric flow rate of the regenerating fluid and cycle frequency.

Keywords: magnetic refrigeration; regenerator; experimental; coefficient of performance; prototype

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Nomenclature

Acronyms

AMR	active magnetic regenerator
COP	coefficient of performance
MCM	magnetocaloric material

Symbols

ΔT_{span}	temperature span at ends of the device	[K]
f_{AMR}	cycle frequency	[Hz]
P_{in}	pressure at the inlet of the pump	[Pa]
P_{out}	pressure at the outlet of the pump	[Pa]
$\dot{Q}_{ref}$	cooling capacity	[W]
T_H	temperature of heat rejection	[K]
T_{HFi}	temperature of the water outgoing from the geothermal probe	[K]
T_{HFo}	temperature of the regenerating fluid outgoing from the hot side of the refrigerator	[K]
T_{LFo}	temperature of the regenerating fluid outgoing from the cold side of the refrigerator	[K]
φ	utilization factor	[-]
$\dot{W}_{el,h}$	electrical power absorbed by the electric heater	[W]
$\dot{W}_{el,p}$	electrical power absorbed by the pump	[W]
$\dot{W}_{el,m}$	electrical power absorbed by the electric motor	[W]

1. Introduction

Approximately 15% of the worldwide energy consumption involves the use of refrigeration. Modern refrigeration is actually based on vapour compression plants. Most of the commonly used refrigerants are ozone-depleting substances that can also be powerful greenhouse gases.

Magnetic refrigeration (MR) is an emerging cooling technology based on the magnetocaloric effect (MCE) in solid-state materials. Compared with conventional vapour compression systems, magnetic refrigeration can be an environmentally friendly and efficient technology. The magnetic refrigerant is a solid and has essentially zero vapour pressure; therefore, it is ecologically sound, with no Ozone Depletion Potential (ODP) and zero direct Global Warming Potential (GWP).

The MCE originates from the interaction of the magnetic field with the molecular magnetic moments of a substance. MCE is associated with warming as the magnetic moments of the atoms align with the application of a magnetic field and, correspondingly, with cooling upon removal of the magnetic field. MCE can be defined as an adiabatic temperature change due to magnetization/demagnetization of a magnetic material or, alternatively, as an isothermal magnetic entropy change. MCE is strongly dependent on magnetic field intensity and temperature and is maximized at the magnetic material ordering temperature, i.e., the Curie temperature.

In this paper, attention is paid to the near room temperature MCE. Its applications at near room temperature have been the object of several studies over the past thirty years. The study of the magnetic refrigeration started with the discovery of the magnetocaloric effect attributed to Weiss and Piccard (Smith, 2013). Because of the small temperature increment carry out of magnetization, the employment of a regenerative thermodynamic cycle is necessary to produce a useful temperature span. In 1982, Barclay brought in the use of reciprocating thermal regenerators coupled with a magnetocaloric cycle, well known as Active Magnetic Regenerators (AMR) (Barclay, 1982).

In an AMR (Active Magnetic Regenerative refrigeration) cycle, instead of using a separate material as a regenerator to recuperate the heat from the magnetic material, the magnetic material matrix works both as a refrigerating medium and as a heat regenerating medium, while the fluid flowing in the porous matrix works as a heat transfer medium. Regeneration can be accomplished by blowing the heat transfer fluid in a reciprocating fashion through the regenerator, which is made of magnetocaloric material and is alternately magnetized and demagnetized. Each particle of the magnetic material in the packed bed undergoes a unique magnetic Brayton cycle, and the whole bed undergoes a cascade Brayton cycle.

The magnetic field of the magnetic refrigeration cycle can be supplied by an electromagnet, superconductor or permanent magnet. Practical application in the room temperature range uses permanent magnets to produce the magnetic field. Variations in the magnetic field can be made by physically moving either the magnetic regenerator or the magnet relative to one another, either linearly in a reciprocating device or rotationally in a rotary device.

The secondary fluid exchanges thermal energy with the environment thanks to two heat exchangers.

An AMR cycle (Bingfeng et al., 2006; Kitanovski et al. 2006; Sari et al., 2014; He at al., 2008) consists of the four following processes: (1) bed magnetization, (2) iso-field cooling, (3) bed demagnetization and (4) iso-field heating.

In the magnetization process, the magnetic field in the bed is increased with no fluid flow, which causes the temperature of the material to increase due to the magnetocaloric effect. The temperature of the magnetic material at the hot end of the bed rises above the hot heat exchange temperature. In the iso-field cooling process, with the high magnetic field, the fluid is blown from the cold end to the hot end of the bed. The magnetic material temperature decreases because the fluid absorbs heat from the bed and afterwards expels heat in the hot heat exchanger. In the bed demagnetization, the magnetic material temperature decreases with no fluid flow. Finally, in the last step, with a zero field, the fluid is blown from the hot end to the cold end of the bed. The magnetic material temperature increases because the fluid expels heat to the bed and afterwards absorbs heat in the cold heat exchanger, producing the cooling capacity.

In recent years other thermodynamic cycles have been proposed. Plaznik et al. (2013) have introduced the Hybrid Brayton-Ericsson cycle, Aprea et al. (2011 (a)) have proposed a cascade AMR cycle, and Kitanovski et al. (2014)have presented guidilnes for future work on new thermodynamic cycles.

In this paper, attention is paid to AMR thermodynamic cycle near room temperature MCE. A reference magnetocaloric material for refrigeration at room temperature is gadolinium, which is a member of the lanthanide group of elements. At the Curie temperature T_C of 294 K, Gd undergoes a second-order paramagnetic–ferromagnetic phase transition. Althoug, different researchers are focusing on new materials and new compounds, with the aim to carry out materials having greater adiabatic temperature change and cheaper than the gadolinium (Pecharsky and Gschneidner, 2006; Gschneidner and Pecharsky, 2008). As noticed in Aprea et al. (2015 (b)), the First-order magnetic materials present a Giant Magnetocaloric effect; nevertheless, at this moment they seem impractical for commercial use because of the highest costs (in particular for the rare earth compunds) and because of the hysteresis in the phase transformation, the narrow MCE curve in the Curie temperature vicinity and the volume variation of the materials. Anyway, as the literature reflects, there is a great interest on the part of the scientific community, which bodes well for the future.

After 1976, researchers all over the world started working on their own magnetic refrigerator prototypes (Brown, 1976). A number of devices based on the AMR cycle have been developed since, each with unique design approaches. Additionally, while most devices have been tested using gadolinium as the refrigerant, some explored the implementation of layered regenerator beds with gadolinium-based alloys and first-order transition refrigerants. In many experimental devices, the magnetic field is generated by an electromagnet or a superconducting electromagnet. Therefore, these configurations are not of practical interest for medium- to small-scale refrigeration for near room temperature applications.

A review of magnetic refrigerator and heat pump prototypes is reported in Yu et al. (2010) and in Kitanovski et al. (2015). The existing prototypes are divided into reciprocating and rotating design. Most of the reciprocating prototypes use a superconducting magnet and typically operate at low frequency, whereas the rotating prototypes are characterized by a higher frequency of operation and permanent magnets (Bjørk et al., 2010).

A summary of the most significant prototype systems developed is reported in Table 1. Research group, type of relative motion between magnet and regenerator, maximum magnetic field, magnetic material mass, maximum process frequency, magnetic material, cooling power, maximum temperature-span, regenerator geometry and maximum COP are all specified. With some exceptions (University of Victoria BC, Astronautics Corporation and RisØ Laboratory Denmark), most of the experimental devices developed have low cooling power, low energy performances and low operating frequencies. In any case, the devices built until now are not suitable for practical applications because of the high capital costs and the high ratio between dimensions and cooling power.

To the best of our knowledge, a magnetocaloric refrigerator or heat pump for commercial use has not yet been built, even though some interesting ideas exist. Therefore, attention should be drawn to the development of new experimental prototypes with a cooling power suitable for commercial applications and with energetic performances higher than those of traditional vapour compression plants.

Our research group has worked on magnetic refrigeration since 2010 (Aprea and Maiorino, 2010; Aprea et. al. 2011(a), 2011 (b), 2012, 2013, Aprea et. al. 2013, Aprea et. al. 2014). In this paper, experimental tests have been conducted on a Rotary Permanent Magnet Magnetic Refrigerator (RPMMR) named 8Mag that has been described in previous works (Aprea et al. 2014; Aprea et. al. 2015 (a)). The prototype presents a rotating magnet with a stationary magnetocaloric material (MCM). The name comes from both the octagonal shape of the magnetic system and the total number of regenerators. The temperature span, COP and refrigerating power have been measured under a wide range of operating conditions.

Table 1. Summary of the most significant prototype systems developed; S, superconducting magnet; E, electromagnet; P, permanent; RE, Reciprocating; RO, Rotary; (PRE), predicted values by means of simulations.

TABLE 1 HERE

2. The experimental apparatus

The device developed at the Refrigeration Lab (LTF) of the University of Salerno is a rotary permanent magnet magnetic refrigerator (RPMMR) with a stationary magnetocaloric material (MCM) and a rotating magnet. The prototype, named 8Mag, was extensively discussed in a previous paper (Aprea et al. 2014). It is a magnetic refrigerator that is characterized by a rotating group of permanent magnets realized through a Halbach array configuration that was modified and that is able to guarantee a maximum magnetic field of 1.25 T in two areas with a high magnetic field and a magnetic field of 0.01 T in two areas with a low magnetic field when the free air gap (magnetization area) is 43 mm. Gadolinium is selected as the magnetic refrigerant and demineralized water is employed as the regenerating fluid. To prevent the corrosion of gadolinium and as suggested by Forchelet et al. (2014), we have adopted a corrosion inhibitor. The magnetocaloric material is housed in eight regenerators, each of them having a height of 20 mm, a length of 45 mm, and a width of 35 mm; as a result, the aspect ratio is 1.5 and the avalaible volume is 31.5 cm^3. with an available volume of 31.5 cm^3. The total mass of the gadolinium is 1.20 kg, shaped as packed bed spheres (of 400-500 microns). The regenerators are alternatively magnetized and demagnetized with the rotation of the magnets and are supported by an aluminium structure with a 45 degree spacing. This assembly is constrained to the stator of the rotary valve and to the frame of the device. A rotary valve mechanically coupled with the magnetic field generator imparts the direction of heat transfer fluid through the regenerators. The cycle frequency (f_{AMR}) is determined by rotating the magnets; specifically, for each rotation of the magnets, each regenerator experiences two AMR cycles. A hydraulic system obtained by the combination of a rotary valve and a vane pump ensures the proper distribution of the regenerating fluid in each component of the apparatus in accordance with the AMR cycle phases. The total fluid flow rate entering in the rotary valve is partitioned into two equal shares and then is transported to the regenerators. At each instant, there are four regenerators hydraulically connected to each other that are subject to the fluid flow: a couple is magnetized and another one is demagnetized. At the same time, the remaining four regenerators are disconnected from the hydraulic circuit to experiment on their adiabatic temperature change. Based on the design specifications, the ratio between the magnetization period and the fluid flowing period is 1:1 and their duration is imposed by the rotational speed of the magnets.

The drive system consists of a brushless DC motor rotating the magnets at variable speed between 0.1-1 Hz. A digital encoder and a programmable speed controller complete the drive system. This results in an available maximum continuous torque of 70 Nm at 54 rpm. A photo of 8Mag is shown in Figure 1. Compared with the configuration shown in Aprea et al. (2014), 8Mag has undergone a slight modification to the piping system. To reduce the overall pressure losses, we adopted pipes with larger diameters. Specifically, we replaced the pipes that had internal diameters of 4 mm with pipes that had an inner diameter of 6 mm, with the exception of the connection sections of the valve-regenerator.

FIGURE 1 HERE

Fig. 1 Picture of 8Mag

To measure the refrigeration duty provided by 8Mag, a cold heat exchanger has been realized by the combination of electric resistance with a thermally insulated pressure vessel. A variable voltage supply feeds the electrical resistance to provide a thermal load that is variable from 0 to 500 W. Water, the temperature of which can be adjusted by an electrical heater managed by a PID controller, is used as a secondary fluid in the hot heat exchanger. A scheme of the prototype core principle of operation is shown in Figure 2.

FIGURE 2 HERE

Figure 2. Prototype core details. Longitudinal (A-A) and axial section (B-B): 1) permanent magnet assembly, 2) mounting support, 3) shaft-rotary valve combination, 4) regenerators.

3. Experimental procedure and data analysis

The experimental investigation involved the measurement of: temperature span, pressure drop, AMR frequency, volumetric flow rate of the regenerating fluid, thermal load, electrical power and COP.

Using PT100 thermo-resistances placed between the input and output of each component, the temperature of the regenerating fluid is measured. In particular, referring to Figure 3:

- the temperature span (ΔT_{span}) is the average time between the temperature of the regenerating fluid exiting the hot side (T_{HFo}) and the temperature of the regenerating fluid exiting the cold side (T_{LFo});
- the temperature of heat rejection (T_H) is the average time between the temperature of the water exiting the hot side of the magnetic refrigerator (T_{HFo}) and the water exiting the geothermal probe (T_{HFi}), both measured under steady state condition.

By using a magnetic flow meter placed between the pump and the hot heat exchanger, the volumetric flow rate of the regenerating fluid is measured. To estimate the pressure drop of the entire device, two piezoelectric pressure sensors placed between the input and the output of the pump are employed. With reference to Fig. 3, the total pressure loss is evaluated as the average time over the difference between the pressure at the outlet (P_{out}) and the pressure at the inlet of the pump (P_{in}). The pressure measurements are also performed at the ends of the regenerators, with the objective of measuring the blow duration and the pressure drop through the regenerators.

FIGURE 3HERE

Figure 3. Elementary scheme of 8Mag.

Using the digital encoder on the motor, the AMR cycle frequency is also measured.

The test apparatus is equipped with 32 bit A/D converter acquisition cards with a sampling rate of up to 10 kHz.

To measure the cooling capacity developed by the device, the outer surface of the cold heat exchanger is thermally isolated. In this way, it is possible to consider the following equation to be valid:

$$\dot{Q}_{ref} = \dot{W}_{el,h} \tag{1}$$

where $\dot{W}_{el,h}$ is the electrical power that is absorbed by the electric heater inserted into the exchanger and measured using a watt transducer. Using a PT100, temperature measurements on the outer surface of the insulation coat of the exchanger are performed to verify the quality of the insulation level. Moreover, using a two-watt transducer, the electrical power absorbed by the pump ($\dot{W}_{el,p}$) and the electrical power absorbed by the electric motor used for magnet rotation ($\dot{W}_{el,m}$) are evaluated. For the evaluation of the COP, the following equation is used:

$$COP = \frac{\dot{Q}_{ref}}{\dot{W}_{el,m}+\dot{W}_{el,p}} = \frac{\dot{W}_{el,h}}{\dot{W}_{el,m}+\dot{W}_{el,p}} \tag{2}$$

In Tab. 2, the characteristics of the sensors used and the accuracy of the measurements performed are reported.

Table 2 Characteristics of the sensors used and the accuracy of the measurements performed.

TABLE 2 HERE

4. Experimental results

A number of tests were conducted for the characterization of the device performances. All of them were performed using gadolinium spheres as the refrigerant and demineralized water as an auxiliary regenerating fluid. To reduce the influence of the temperature of the air surrounding the device, the connecting pipes between the heat exchangers and the magnetic refrigerator were insulated. Moreover, as suggested by Engelbrecht et al. (2012), during all of the tests, the magnetic refrigerator was introduced into a climatic room, where the air temperature was maintained in a small range between 293 and 298 K and the relative humidity was equal to 50% ± 5%.

As noticed by Tušek et al. (2013), the fluid flow rate affects the performance of a magnetic refrigerator; consequently, the 8Mag performances have been carried out at three different fluid flow rates: 5.0, 6.0 and 7.0 l min^{-1}. In accordance with the literature (Lozano et al., 2014, Eriksen et al., 2015), all the tests were performed while keeping the hot side temperature at approximately the Curie temperature. In particular, T_H was fixed at 296 K. Consequently, for each fluid flow rate, a series of tests were conducted while varying the AMR cycle frequency (f_{AMR}) and the thermal load.

A large number of parameters affect the behaviour of an AMR cycle, one of which is the utilization factor, defined as:

$$\varphi = \frac{M_S C_S}{M_f C_f} \tag{3}$$

where M_f is the fluid mass blown over one cycle through a single regenerator, C_f is the fluid specific heat, M_s is the mass of a single regenerator and C_s is the refrigerant specific heat in the demagnetized state at the Curie temperature. The φ values in our experimental tests were evaluated according to the procedure identified in a former study (Aprea et al. 2014).

Starting from a condition of no load, for each AMR cycle frequency, the thermal load was increased step by step. For each test, the following expected steady-state conditions were achieved: the temperature span change was less than the measurement accuracy for a time longer than 300 s.

The experimental pressure losses through the overall device were almost constant for each fluid flow rate value, as reported in Tab. 3.

Table 3 Pressure drops at different fluid flow rates.

TABLE 3 HERE

Fig. 4 shows ΔT_{span} as a function of the thermal load for different frequencies f_{AMR} at fluid flow rates of 5.0, 6.0 and 7.0 l min^{-1} at T_H equal to 296 K. The figure clearly shows that at a fixed frequency, ΔT_{span} decreases as the cooling capacity increases. This behaviour, which is in accordance with other works (Lozano et al., 2014; Tura and Rowe, 2011; Tušek et al., 2013), is due to the thermal load supplied at the cold side of the regenerator. The maximum temperature span corresponds to the zero load tests and is 10.9 °C (at f_{AMR} = 0.77 Hz and $\dot{V} =$ 5.0 l min^{-1}), 11.9 °C (at f_{AMR} = 0.93 Hz and $\dot{V} = 6.0$ l min^{-1}) and 10.5 °C (at f_{AMR} = 1.08 Hz and $\dot{V} = 7.0$ l min^{-1}). The maximum refrigeration power reached by 8Mag corresponds to a ΔT_{span} close to 0 K and is approximately 200 W.

FIGURE 4 HERE

Figure 4 (a,b,c) ΔT_{span} as a function of the thermal load for different frequencies and different fluid flow rates.

Furthermore, in Fig. 4, we can note that the iso-frequency lines cross over each other at a certain point. This behaviour is better shown in Fig. 5, where we have reported the temperature span for different thermal load values as a function of cycle frequency. The top axis of the graphs gives the corresponding φ. We can easily note that for each thermal load value, there is an optimal cycle frequency for which ΔT_{span} is maximized.

FIGURE 5 HERE

Figure 5 (a,b,c) ΔT_{span} as a function of f_{AMR} for different thermal loads and different fluid flow rates.

As noticed by Lozano et al. (2014), this phenomenon happens because at low frequencies, there is a large influence of the longitudinal thermal conduction and the regenerator utilization factor becomes too high. However, it can be seen that at lower frequencies, the temperature span increases with increasing frequency until a certain optimum frequency, at which the irreversible losses (friction inner to the rotary valve, bad fluid distribution inside the regenerators) and the dead volumes become significant and the heat transfer is affected; thus, the regenerator is not capable of maintaining a high temperature span. At a higher cycle frequency, the fluid is not capable of utilizing all of the energy available from the magnetocaloric effect, thus lowering the maximum temperature span. Meanwhile, at an excessive cycle frequency, the fluid is not able to utilize all of the energy available from the magnetocaloric effect, thus lowering ΔT_{span}.

In addition, during our experimentation we have observed that for each fluid flow rate the optimal cycle frequency decreases as the thermal load increases. We can explain this phenomenon considering that keeping constant the fluid flow rate a decrease of cycle frequency leads to an increase of the total mass flowing through the regenerators. This involves an increase of the utilization and then the device can absorb a greater thermal load. Consequently, to maximize the temperature span, similarly to other devices (Bahl et al., 2014; Russek et al., 2010; Tura and Rowe, 2011) 8Mag has to work at the optimal cycle frequency. For all of the experimental tests, the optimal frequency falls in the range between 0.4 and 0.8 Hz, corresponding to an optimal utilization factor between 2.36 and 1.18.

Fig. 4 and Fig. 5 also show that at fixed thermal load and frequency, the best ΔT_{span} is achieved for a fluid flow rate of 6.0 l min^{-1}. Therefore, there is an optimum fluid mass flow rate that leads to the largest temperature span at each operating frequency. If the regenerating fluid flow rate is small, only part of the energy generated by the magnetocaloric effect can be utilized. With a greater value of fluid flow rate, the fluid can utilize the entire energy available. Further fluid flow perturbs 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.

Fig. 6 reports the cooling capacity as a function of temperature span at a fixed T_H (296 K) and φ (1.76) for the three different fluid flow rates and corresponding cycle frequencies (5.0 l min^{-1} at 0.38 Hz, 6.0 l min^{-1} at 0.46 Hz and 7.0 l min^{-1} at 0.54 Hz). The figure clearly shows that the cooling capacity decreases when the temperature span increases. In addition, we can note that for a fixed ΔT_{span} an increase of the water mass flow rate improves the cooling capacity due to the increase of the heat transfer coefficient. The refrigeration power ranges between 200 W (at an almost zero ΔT_{span}) and 50 W (at a ΔT_{span} between 6 and 6.7 K).

FIGURE 6 HERE

Figure 6 $\dot{Q}_{ref}$ as a function of ΔT_{span} for three different couples of fluid flow rate and f_{AMR} corresponding to the same utilization factor.

Fig. 7 presents the torque as a function of the temperature span at a fixed T_H (296 K) and φ (1.76) for the three different fluid flow rates and corresponding cycle frequencies (5.0 l min^{-1} at 0.38 Hz, 6.0 l min^{-1} at 0.46 Hz and 7.0 l min^{-1} at 0.54 Hz). The mechanical torque due to the rotation of the magnetic system is measured directly at the shaft of the drive system and takes into account the mechanical power required to rotate the magnets and the magnetic work. The figure clearly shows that the torque increases with the temperature span together with the magnetic work at a fixed frequency and fluid flow rate. In particular, a maximum value of 29 Nm was measured at the maximum temperature span of 6.7 K. We can attribute the increase of the mechanical torque with the temperature span to the second order transition phase of the gadolinium. A high termperature span causes a greater mass of magnetocaloric material has a ferromagnetic behaviour; consequently, the magnetic work and then the mechanical torque increase. The torque, at a fixed temperature span, increases as the fluid flow rate and frequency increase. Indeed, increasing the fluid flow rate also increases the cycle frequency to make a comparison at fixed φ. The rotating magnetic field induces eddy currents in the regenerator that generate forces that oppose the changing magnetic field, thus increasing the magnetic work. The strength of this term increases with frequency. Furthermore, mechanical frictions also increase with the frequency. Moreover, Aprea et al. (2014) also noticed that a significant dead volume between the regenerators and the valve ports affects the performance of 8Mag. The negative effect of dead volume increases with the frequency.

FIGURE 7 HERE

Figure 7 The mechanical torque as a function of ΔT_{span} for three different couples of fluid flow rate and f_{AMR} corresponding to the same utilization factor.

Fig. 8 presents COP as a function of ΔT_{span} at a fixed T_H (296 K) and φ (1.76). At a fixed frequency and fluid flow rate, COP decreases with the temperature span. Indeed, as ΔT_{span} increases, the cooling power decreases and the torque increases. At a fixed temperature span, the COP increases as the fluid flow rate and cycle frequency decrease, reaching the maximum value of 2.5 at 0 ΔT_{span}. The reason for this increase is related to both the decrease of the fluid flow and the concurrent decrease of the cycle frequency. The first effect leads to a decrease of the pressure drops and therefore of the work of the pump. The second effect leads to a decrease of both mechanical and eddy current losses.

FIGURE 8 HERE

Figure 8 COP as a function of ΔT_{span} for three different couples of fluid flow rate and f_{AMR} corresponding to the same utilization factor.

Fig. 9 presents COP as a function of the cooling power at a fixed T_H (296 K) and φ (1.76). Here, it is clearly shown that COP increases with the cooling capacity. Indeed, as shown in Eq. 2, the COP is directly proportional to the cooling capacity and inversely proportional to the electrical power absorbed by the pump and by the electrical motor. Indeed, increasing the thermal load at a fixed fluid flow rate does not change $\dot{W}_{el,p}$. Figure 7 shows that the torque decreases with the temperature span. Therefore, because increasing the thermal load decreases the temperature span, $\dot{W}_{el,m}$ decreases. The best value of the COP is 2.5, corresponding to a thermal load of 200 W.

FIGURE 9 HERE

Figure 9 COP as a function of $\dot{Q}_{ref}$ for three different couples of fluid flow rate and f_{AMR} corresponding to the same utilization factor.

5. Conclusions

In this paper, we have reported the results of new experimental tests conducted on a Rotary Permanent Magnet Magnetic Refrigerator (RPMMR) named 8Mag. Its Halbach array configuration produces a peak flux density of 1.25 T and an average flux density of 1.10 T. It uses eight radially placed regenerators, which are currently filled with gadolinium spheres with a total refrigerant mass of 1.20 kg. Demineralized water is employed as the regenerating fluid. The fluid distribution is managed by a rotary valve positioned inside the magnetic system, while the fluid circulation is obtained with a commercial rotary vane pump. To characterize the energy performances of 8Mag, we have performed various tests under different operating conditions. In particular, we have tested the behaviour of 8Mag by changing both the cycle frequency and the fluid flow rate while we have fixed the hot side temperature at 296 K. For each test, we have measured ΔT_{span}, the cooling capacity, the mechanical torque and the COP. Regarding ΔT_{span}, it is worth noting that we have reported a pessimistic measurement. Indeed, we have measured the actual ΔT_{span} available at the ends of the device, including the thermal losses typical of 8Mag. The tests have shown, at 296 K, a maximum temperature span at the zero load tests of 11.9 °C (at f_{AMR} = 0.93 Hz and $\dot{V} = 6.0$ l min^{-1}) and a maximum COP of 2.5 corresponding to a thermal load of 200 W (at f_{AMR} = 0.38 Hz and $\dot{V} = 5.0$ l min^{-1}). Moreover, for each condition, 8Mag exhibits an optimal f_{AMR}, for which it is possible to maximize ΔT_{span}.

As a result of the experimental analysis, we have found that the behaviour of 8Mag is in accordance with those of other devices built around the world. Furthermore, considering that 8Mag offers a reduction of thermal losses, we can conclude that the tests presented here show promising performances. Although we have to note that these results can be easily overcome by vapour compression systems, the device met the objectives of 1) continuous cooling, 2) simple operation mode, 3) compact design of the machine, 4) wide range of operating conditions, 5) high operation frequencies and 6) intense magnetic flux. To improve our prototype, we have scheduled new experimental tests focused on the identification of the worst causes of energy losses.

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