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DESIGNING AN ACTIVE MAGNETIC REGENERATOR FOR A FIXED MAGNETIC CIRCUIT

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Abstract. *Magnetic refrigeration presents itself as one of the most promising emerging technologies for application in heat pumps and cooling systems. The Active Magnetic Regenerator (AMR) has been studied by numerous works and the importance of this component to the performance of magnetic refrigerator devices is well established. The aim of the present work is to numerically evaluate the effect of the regenerator geometry and system operating conditions on the performance of a magnetic refrigeration device. The applied magnetic field waveform is taken as an input to the model, and a practical outcome of this study will be the design of a more efficient active magnetic regenerator for an existing prototype developed at Polo/UFSC. Improvements in terms of the power consumption of a novel electrovalve manifold and an upgraded drive system were also considered. The analysis resulted in a regenerator composed of 11 beds, capable of providing 76 W of cooling capacity for a system temperature span of 15 K and a COP of 0.8. Improvements on the power consumption of the electrovalves and the drive system could lead to a significant enhancement in the COP, particularly for low utilization factors.*

Keywords: *magnetic refrigeration, active magnetic regenerator*

1. INTRODUCTION

Magnetic refrigeration stands as one of the most promising non-vapor compression technologies for cooling applications at room temperature. The potential for reducing the power consumption of its components combined with the theoretical large efficiency of magnetic refrigerators, makes this technology very attractive (Kitanovski *et al.*, 2015). The use of a solid refrigerant also presents benefits, since there is no risk of gas leakage, which often represents an environmental hazard. However, this technology presents several challenges until it turns commercially viable, highlighting among them: providing a cooling capacity while reaching a considerable temperature span between the hot and cold environments; supplying such cooling capacity as efficiently as possible in order to reduce power consumption and producing both items listed before with an small amount of magnetocaloric material (Trevizoli *et al.*, 2016a).

One of the fundamental phenomena applied in magnetic refrigerators is the magnetocaloric effect, which is defined as the thermal response of a magnetic material when subjected to a variation in the applied magnetic field. Magnetic refrigerators can work according to different thermodynamic cycles, such as Carnot, Brayton or Eriksson, although to overcome the small temperature change caused by the magnetocaloric effect, a thermodynamic cycle based on thermal regeneration is usually employed. In the active magnetic regenerator (AMR) (Barclay and Steyert, 1982) the magnetocaloric material is usually applied as the solid matrix in a porous media, working as both regenerative matrix and refrigerant. A thermal fluid is then used to exchange heat with the solid matrix, as well as with the hot and cold temperature reservoirs. The use of regenerative cycles enables the achievement of increased temperature spans between reservoirs, while keeping considerable cooling capacity.

Several prototypes have been developed in the literature Engelbrecht *et al.* (2012); Eriksen *et al.* (2016), including the rotary device designed and built at Polo/UFSC (Lozano *et al.*, 2016). A thermodynamic evaluation of this device showed that the hydraulic subsystem and the AMRs are the largest contributors to the performance losses of the system (Capovilla *et al.*, 2016). In order to improve that, Cardoso *et al.* (2016) proposed improvements in the hydraulic system through the application of electrovalves, rather than the previously applied rotary valves, which can enhance the performance of the system by consuming less power and providing better control over the fluid flow profile. However, there is still room for

improvement on the Polo/UFSC prototype through the optimization of the AMRs.

This work evaluates some of the factors involved in the performance of AMRs, focusing on the design of a new regenerator geometry for the Polo/UFSC prototype. The analysis targeted at the geometric parameters of the regenerator, as well as on the operating conditions of the system, while keeping the magnetic circuit, hydraulic profile, porous media geometry and magnetocaloric material fixed. The dependent variables are the cooling capacity (\dot{Q}_C) and the coefficient of performance (COP) of the system, while regenerator losses due to demagnetization factor, viscous dissipation, axial heat conduction and regenerator ineffectiveness are evaluated parametrically. The aimed operating point is a cooling capacity of at least 75 W while reaching a system temperature span of 15 K. A system loss analysis is carried out to evaluate the influence of the electrovalves and the transmission system power consumption in the COP of the system.

2. METHOD

As mentioned before, when it comes to thermodynamic cycles, there are several options for application in magnetic refrigeration devices. However, the Brayton cycle is the most common thermo-magnetic regenerative cooling cycle (Trevizoli, 2015). The idealized Brayton cycle is composed of four steps:

1. **Adiabatic Magnetization:** By increasing adiabatically the modulus of the applied magnetic field over the regenerator, there is an increase in the temperature of the magnetocaloric material, which composes the solid matrix, due to the magnetocaloric effect.
2. **Constant Magnetic Field Cold Blow:** After the adiabatic magnetization, which has increased the temperature of the magnetocaloric material fluid coming from the cold reservoir, flows through the regenerator at a constant magnetic field, absorbing heat from the solid matrix, which reduces its temperature and increases the temperature of the fluid. After it crosses all the solid matrix, the fluid has its temperature risen above the temperature of the hot reservoir and rejects heat on this heat sink.
3. **Adiabatic Demagnetization:** Analogously to the adiabatic magnetization, the adiabatic demagnetization consists in reducing the modulus of the applied magnetic field over the AMR, which reduces the temperature of the magnetocaloric material through the magnetocaloric effect.
4. **Constant Magnetic Field Hot Blow:** Similarly to the cold blow, fluid coming from the hot reservoir, at a temperature relative to this heat source, returns to the regenerator and crosses the solid matrix at a constant magnetic field. The solid matrix removes heat from the fluid phase, which cools down to a temperature below that of the cold reservoir. As it flows through the regenerator, the fluid absorbs heats from the cold reservoir, which represents the cooling capacity of the system.

Due to limitations in the magnetic circuit design and the flow distributing system, magnetic refrigeration devices usually do not operate in an ideal Brayton cycle. The interaction between the fluid flow and magnetic profiles also have an effect over the performance of the system. The superposition of the fluid flow profile and magnetic profile for the Polo/UFSC prototype is show in Fig. 1 (Lozano *et al.*, 2016). It can be noticed that the cold and hot blow processes do not occur at constant magnetic field, and the magnetization and demagnetization steps are performed while there is fluid flow in the regenerator, thus, differing the operation of this device from an ideal Brayton cycle.

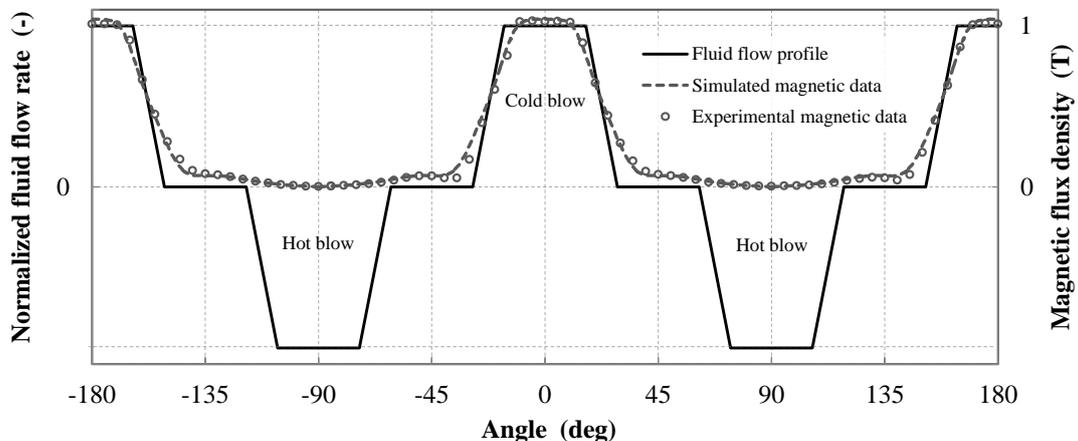


Figure 1: Magnetic and fluid flow profile coupling of the Polo/UFSC prototype (Lozano *et al.*, 2016).

In order to evaluate the effect of AMR dimensions and system operating parameters on the performance of a magnetic system, numerical simulations were carried out using the one-dimensional mathematical model developed at Polo/UFSC

by Trevizoli *et al.* (2016b). The system was simulated as an array of multiple prismatic regenerators, defined by three characteristic dimensions: length, height and width. Each regenerator allowed for a longitudinal oscillatory flow of a mixture of water (80%) and ethylene-glycol (20%) through a solid porous matrix composed of gadolinium (Gd) spheres.

The magnetic circuit considered was already developed for the Polo/UFSC prototype (Lozano, 2015), which is able to generate two regions of high magnetic field of up to roughly 1 T and two regions of low magnetic field of almost 0 T, and that has a profile given by Fig. 1. This constrains the internal radius of the outer magnet and the external radius of the inner magnet, which limits the magnetic gap to place the regenerator beds, fixing their height, as well as the magnetic profile. The effect of the fluid profile was also not of interest to the current study and was kept fixed, while the fluid flow distribution system is assumed to be done by a set of electrovalves as those studied and already implemented in an actual AMR apparatus by Cardoso *et al.* (2017); Hoffmann *et al.* (2017); Dutra *et al.* (2017). Losses due to casing heat transfer, heat exchangers and dead volume were not taken into account at this stage of the design process. A summary of the fixed parameters of the analysis is presented in Table 1.

Table 1: Fixed parameters of the analysis.

Magnetocaloric Material	Gadolinium
Porous Media Geometry	0.5 mm diameter sphere
Regenerator Cross-Section Geometry	Rectangular
Flow Profile	25% per blow
Hot Heat Source Temperature	300 K
Temperature Span	15 K
External radius of inner magnet	90 mm
Internal radius of outer magnet	110 mm
Length of the magnetic circuit	150 mm
Regenerator Height	10 mm

Given the restrictions caused by the fixed magnetic circuit and the fluid flow profile, the analysis of the geometric parameters of the AMR and the operating system conditions could be defined. The evaluated parameters were the length (L) and width (W) of the regenerator beds, the number of regenerator beds (N), the operating frequency of the system (f), and the utilization factor (ϕ).

The utilization factor is defined as the thermal mass of the fluid in each blow over the thermal mass of the solid, and can be calculated by Eq. (1), where c_f is the specific heat of the fluid phase, c_s is the specific heat of the solid phase, \dot{m}_f , is the mass flow rate of the fluid and m_s is the mass of magnetocaloric material in the AMR. Once the regenerator dimensions and the operating frequency of the system are fixed, the utilization factor is changed by a change in the mass flow rate of the fluid.

$$\phi = \frac{\dot{m}_f c_f}{2f m_s c_s} \quad (1)$$

The constraints applied previously are known to effect the variables in the analysis. The maximum length of the regenerator cannot exceed the length of the magnetic circuit, and it is actually desired for the regenerator to be shorter than the magnetic circuit, as the field intensity is reduced near its edges. Besides that, a sharp reduction of the length of the regenerator would mean a reduction on the mass of magnetocaloric material, heat transfer area and an increase in the axial conduction loss, which are known to deteriorate the performance of the AMR. Therefore, the maximum regenerator length considered in the analysis was 125 mm and the minimum of 50 mm.

Furthermore, the magnetic field profile has a high field period that corresponds to 45° , which when converted for an arc corresponding to the circle of radius 100 mm (magnetic gap perimeter) gives a dimension of 78 mm. It was established that the distance, in that same radius arc, between two consecutive regenerators inside the magnetic gap would be the same as the regenerator width. Hence, the maximum width of the regenerators was defined as 35 mm. The major interest on the analysis was to evaluate the effect of the demagnetization factor on the cooling capacity of the regenerators, as well as the effect of the number of valves on the COP. Thus, the mass of magnetocaloric material was kept fix by changing the number of regenerators according to the change in their width. These two variables are correlated by Eq.(2), where R represents the mean radius of the magnetic gap, which is equal to 100 mm.

$$NW = \pi R \quad (2)$$

Each regenerator requires two electrovalves for its operation, which not only increases the power consumption, but also raises the cost of the device, complexity of the fluid distribution and control system, maintenance, weight and noise. Therefore, it is desired that the number of regenerators should be as low as possible to reduce the number of electrovalves,

even though it may somewhat decrease the performance of the system. This leads to the establishment of a maximum number of regenerators of 15 beds.

Without considering the losses in the AMR, one would expect as a result, the cooling capacity to increase with the mass flow rate (utilization factor) and operating frequency. That, however, does not occur, because viscous dissipation and axial conduction, reduce the effectiveness of the regenerator and increase the power consumption associated with fluid pumping and transmission system. As a result, the cooling capacity and the COP have an optima with respect to the frequency and the mass flow rate (utilization factor). For engineering purposes, it is desired for the system to operate at low frequencies and mass flow rates, which would mean smaller pumps and electric motors, thus, less noise. All those reasons caused the utilization factor and the operating frequency to be evaluated under the intervals ranging from 0.25 to 1 and 0.5 Hz to 4 Hz, respectively.

Given the restrictions for the aforementioned variables, to evaluate the effect of each one of them over the performance of a magnetic refrigeration device, simulations were carried out according to the values presented in Table 2.

Table 2: Varied parameters of the analysis.

Regenerator Length [mm]	50, 75, 100, 125
Regenerator Width [mm]	35, 29, 25, 21
Number of Regenerators [-]	9, 11, 13, 15
Operating Frequency [Hz]	0.5, 1, 2, 4
Utilization Factor [-]	0.25, 0.5, 0.75, 1

The numerical model evaluates the performance of the AMR by solving one dimensional energy and momentum equations on the fluid phase and one dimensional energy equation in the solid phase. The equations are modelled through finite volume method approach. Further information about the mathematical model and numerical solution can be found at Trevisoli *et al.* (2016b).

An important parameter in the characterization of regenerators is the Number of Transfer Units (NTU), which can be calculated through Eq. (3), where h is the interstitial heat transfer coefficient and A_{HT} is the heat transfer area.

$$NTU = \frac{hA_{HT}}{\dot{m}_f c_f} \quad (3)$$

The interstitial heat transfer coefficient is given by the correlation for porous media composed of spheres proposed by Pallares and Grau (2010). The heat transfer area can be calculated by the correlation given by Eq. (4), where V_s is the volume of the solid phase of the regenerator and d_p is the particle diameter.

$$A_{HT} = \frac{6V_s}{d_p} \quad (4)$$

To evaluate the system power consumption (\dot{W}), four different contributions were considered: the fluid pumping power (\dot{W}_{pump}), the power consumption by a manifold of electrovalves (\dot{W}_{valv}), the power required to drive the mechanical transmission system (\dot{W}_{driv}) and the magnetic power required to magnetize and demagnetize the solid (\dot{W}_{mag}), as given by Eq. (5).

$$\dot{W} = \dot{W}_{valv} + \dot{W}_{pump} + \dot{W}_{driv} + \dot{W}_{mag} \quad (5)$$

The pumping power is obtained by Eq. (6), where Δp is the pressure drop over the regenerator bed, and ρ_f is the density of the fluid mixture and t^* is the flow profile fraction, kept at 25% in this study. The friction factors was calculated by the correlation for porous media proposed by Ergun (1952).

$$\dot{W}_{pump} = N \frac{\dot{m}_f \Delta p}{\rho_f} t^* \quad (6)$$

The power consumption by the electrovalves is given by Eq. (7), where \dot{W}_{sv} is the nominal power consumption by a single electrovalve. The nominal power consumption of the electrovalves currently available at Polo/UFSC is approximately 4 W (Cardoso *et al.*, 2017).

$$\dot{W}_{valv} = 2N \dot{W}_{sv} t^* \quad (7)$$

The power consumption of the mechanical transmission drive system is given by Eq. (8). The values of the constants a and b where calculated through a fitting in experimental data of power consumption versus operating frequency for the

prototype Polo/UFSC obtained by Capovilla *et al.* (2016). For frequencies ranging from 0.25 Hz to 4 Hz, the values of a and b are 12.411 and -1.04, respectively.

$$\dot{W}_{driv} = af + b \quad (8)$$

The magnetic power is obtained through an energy balance over the AMR. The COP can be calculated by the ratio between the cooling capacity and the total power consumption by the system, according to Eq. (9).

$$COP = \frac{\dot{Q}_C}{\dot{W}} = \frac{\dot{Q}_C}{\dot{W}_{valv} + \dot{W}_{pump} + \dot{W}_{driv} + \dot{W}_{mag}} \quad (9)$$

The contributions regarding pumping and magnetic work are related to the AMR operation itself, meaning that changes in them would result in direct changes on the AMR operation and performance. On the other hand, the nominal power consumption of the electrovalves and the drive system are components of the total power consumption external to the AMR, which would affect only the COP. It was evaluated that the mechanical transmission system designed for the prototype is oversized, meaning that the power consumption by this system can be reduced. Besides that, it is also known that the nominal power consumption of the electrovalves can be considerably reduced from the actual 4 W.

Therefore it is interesting to evaluate the effect of improvements on the power consumption of both of these fractions in the COP of the system. To do so, three different nominal power consumption of each electrovalves were employed (0, 1 and 4 W), as well as three different power consumption of the driving system (0, 50% and 100% of the actual power consumption of the Polo/UFSC prototype) were considered, and their effect over the COP was evaluated.

3. RESULTS AND DISCUSSION

Following the definition of the parameters that were kept constant and the ones that would be evaluated, numerical simulations were carried out. Several scenarios were simulated and some of the results are presented in this section. The results are displayed in plots of the cooling capacity and coefficient of performance versus the utilization factor for different values of the parameter, while the others are kept fixed. Firstly the power consumption of the electrovalves and the drive system were fixed, and the operating frequency, regenerator length and width and the number of regenerator beds were evaluated. Later, the effect of the reduction of the power consumption of the electrovalves and the drive system on the coefficient of performance is evaluated for the selected regenerator.

3.1 Regenerator Performance Analysis

For the regenerator performance analysis, the power consumption of the electrovalves was considered as 4 W, while the drive system power consumption was considered as the results given by Eq. (8). The first parameter to be evaluated was the operating frequency. For this analysis, the length and the width of the regenerator beds were defined to be 100 mm and 29 mm, respectively, and the system was considered to be composed of 11 regenerator beds. The \dot{Q}_C and COP as a function of utilization factor for different operating frequencies are shown in the Figs. 2(a) and (b), respectively.

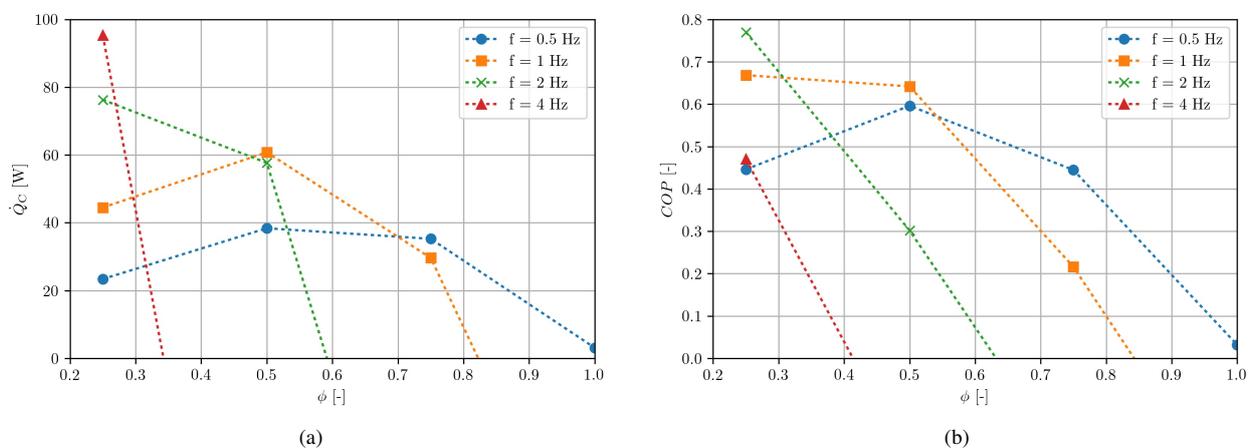


Figure 2: Effect of operating frequency on a magnetic refrigeration system composed by 11 regenerator beds with 29 mm of width and 100 mm of length each in terms of (a) cooling capacity and (b) Coefficient of Performance.

Through Fig. 2(a) it can be noticed that are optimum utilization factors for 0.5 and 1 Hz operating frequency that maximizes the cooling capacity. A trend given by the figure is that the higher the frequency, the lower the optimum utilization factor. This is caused by the higher mass flow rates associated with higher frequencies for the same utilization.

An increase in the mass flow rate generates a decrease in the NTU , an increase in the viscous dissipation, but also an increase in the mass of fluid in contact with the heat reservoirs, which explains the existence of a condition that maximizes the cooling capacity.

After the optimum point is attained, a fast drop in the cooling capacity occurs with an increase in the utilization factor. This trend will happen for all the following analysis, and it is caused by the decrease in the AMR NTU and effectiveness and the increase in the viscous losses that follow the rise in the mass flow rate. For the frequency interval analyzed, the higher operating frequency corresponded to higher cooling capacities.

Regarding the coefficient of performance, an increase in the frequency will lead to an increase in the pumping power, as well as, an increase in the drive system power, and not necessarily an increase in the cooling capacity, which explains the lack of a trend with respect to the effect of the operating frequency in the COP. Usually there is a reduction in the COP with the rise of the utilization factor except for a regenerator operating at 0.5 Hz, where there is an increase in the cooling capacity that surpasses the increase in pumping power. The fall that happens after the optimum utilization point in the cooling capacity is reflected into the COP, which is even more accentuated because of the rise in the pumping power.

The following analysis was performed in terms of the width of the regenerators and the number of regenerator beds. The operating frequency was kept fixed at 2 Hz and the length of the regenerator at 100 m. Since the mass of magnetocaloric material was maintained constant through Eq. (2), changes in the number and width of regenerator beds are expected to have effect in the demagnetization factor, pumping power, viscous losses and axial heat conduction in the regenerator and in the electrovalves power consumption. As can be observed in Fig. 3(a), the highest cooling capacities were achieved by the narrowest AMRs for all the utilization factors considered in this analysis. This is caused because there is no change in the mass of magnetocaloric material, while there is a reduction in the demagnetization factor for AMRs with smaller widths and changes in other losses were not as considerable as the one caused by the demagnetization factor.

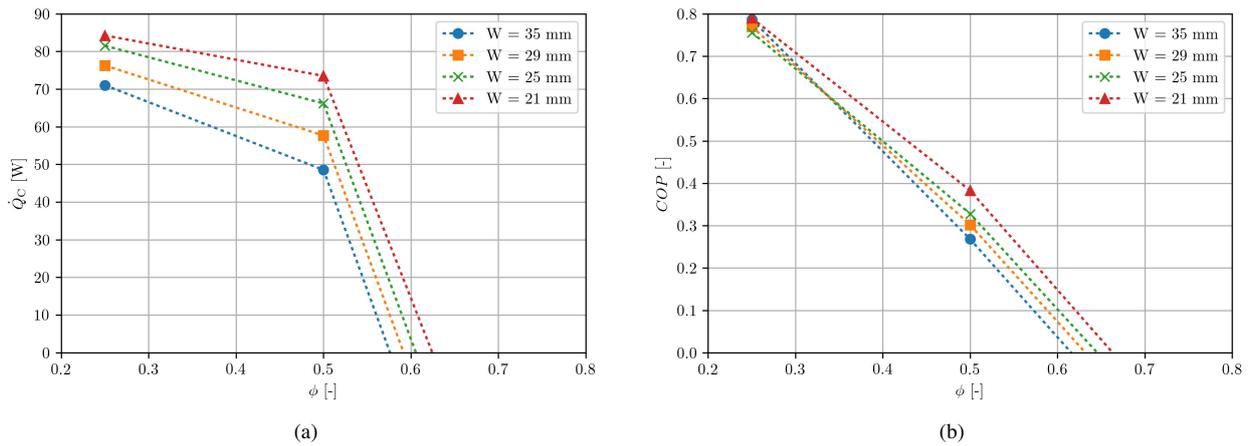


Figure 3: Number of regenerator beds and regenerator width effect on a magnetic refrigeration system composed by AMR beds of 100 mm each operating at 2 Hz in terms of (a) cooling capacity and (b) Coefficient of Performance.

For the COP, Fig. 4(b) shows that the increase in cooling capacity surpasses the increase in the electrovalves power consumption, leading to an increase in the COP, for an utilization of 0.5. This is due to the dominance of the pumping power over the electrovalves power consumption at this operating point. For smaller utilization factors it is expected that the system operate with less electrovalves to reach higher COPs, due to the reduction in the pumping power, and higher influence of the electrovalves power consumption over the COP.

The last analysis carried out was regarding the length of the regenerator. This parameter has a complex influence over the regenerator once it will have effect on the total mass of magnetocaloric material, heat transfer area, pressure drop and defines the regenerator effectiveness, viscous dissipation, axial conduction and pumping power. Such complex interaction between all those factors corroborates for the results exposed in Fig. 4(a). An increase in the length of the regenerator will increase the mass of magnetocaloric material, heat transfer area of the regenerator and decrease the axial conduction in the solid, which all lead to an increase in the cooling capacity. However, it also enhances the viscous dissipation on the AMR, which can be highly harmful for the regenerator cooling capacity, specially at higher utilization factors. The simultaneous action of these aspects listed previously corroborates for the existence of optimum points in terms of each length, and avoid establishing trends with the length variation as could be done in the previous analyses.

When it comes to the COP, Fig. 4(b) shows a marked effect of the increase in the utilization factor in the COP, always decreasing its value, which is more notable for AMR with higher lengths. This is caused by the direct effect of the length over the pressure drop and fluid pumping power, causing it to generate rapid decreases in the COP as the utilization factor increases. Also, there is always a decrease in the cooling capacity for utilization factors higher than 0.5, which combined with the increase in the pumping power, generate the decrease in the COP.

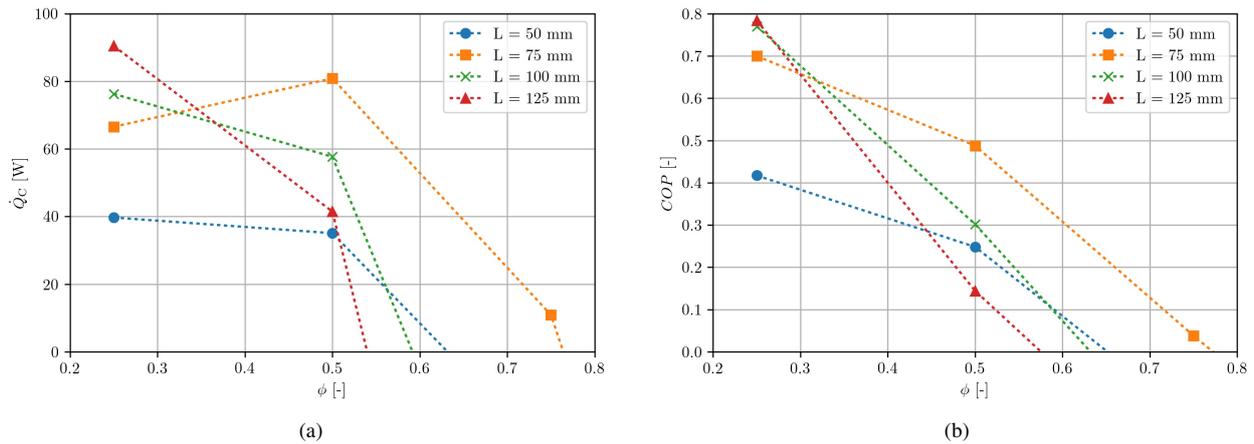


Figure 4: Regenerator length effect on a magnetic refrigeration system composed by 11 AMR beds with 29 mm of width each operating at 2 Hz in terms of (a) cooling capacity and (b) Coefficient of Performance.

An important trend regarding the latter analyses is the lack of regenerators that achieved positive cooling capacities at higher utilization factors. This is caused as a result of the loss in effectiveness, due to a decrease in the NTU , and increase in viscous dissipation over the AMR by the increase in the mass flow rate over the AMR. The values of this parameter are considered still high for a domestic magnetic refrigerator (above 100 kg/h), specially due to the high frequencies considered in the analysis.

3.2 AMR selection and external losses analysis

Given the results obtained in the previous section, an AMR geometry and operation condition can be selected in order to fulfill the requirements settled previously for the magnetic refrigerator to operate at a cooling capacity of 75 W in a system temperature span of 15 K and with a COP as high as possible. The selected AMR has a length of 100 mm, a width of 29 mm, composed by 11 beds, and operating at a frequency of 2 Hz and a utilization factor of 0.25 (which corresponds to a mass flow rate of 47.1 kg/h). Each one of the regenerator beds is composed of 146 g of Gd. Such configuration would attain a cooling capacity of approximately 76 W and a COP of 0.8. A further analysis was developed by decreasing the utilization factor for values below 0.25. This led to a decrease in the cooling capacity, caused by a less amount of fluid in contact with the heat reservoirs over time. Operating in utilization factors between 0.25 and 0.5 can increase the cooling power, however this operation decreased the COP, therefore, it was selected an utilization factor of 0.25.

As mentioned before, the power consumption of electrovalves and driving system can be reduced. The effect on the performance of the selected AMR due to the reduction in the power consumption of the electrovalves and the driving system is plotted as a function of the utilization factor in Fig. 5(a) and (b), respectively. For both cases, the power

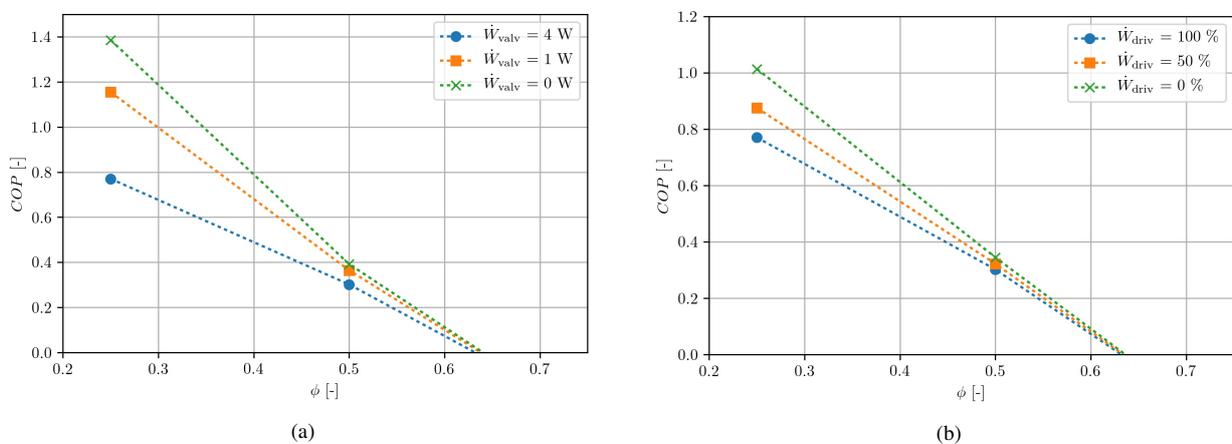


Figure 5: External power consumption effect on the Coefficient of Performance for a magnetic refrigerator composed by 11 AMR beds with 29 mm of width and 100 mm of length operating at 2 Hz. Results for a reduction in the power consumption of (a) valves and (b) driving system.

consumption decrease can cause a considerable improvement in the COP at an utilization factor of 0.25. This is due to the fact that at low utilization factors, the pumping power fraction over the total power consumption is smaller, making

improvements on the other parts of the overall power consumption to have higher effect on the COP. For higher utilization factors, the effects of the reduction on those power consumption is the same, while the pumping power increases, meaning that those improvements have less influence over the total power consumption of the system.

When combined improvements in both, electrovalves and driving systems power consumption, the resulting COP is plotted in Fig. 6. Upgrades in electrovalves and driving system performance can lead to significant increase in the COP, increasing from 0.8 to approximately 2.5 in the ideal case. The gains proposed promote an increase in the second law efficiency from roughly 4.2% to 13.2%.

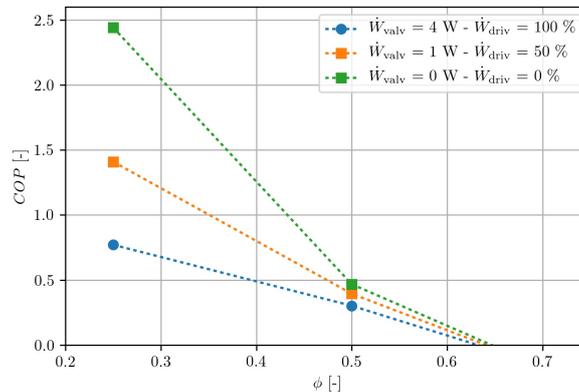


Figure 6: External power consumption effect on the Coefficient of Performance for a magnetic system composed by 11 AMR beds with 29 mm of width and 100 mm of length operating at 2 Hz.

4. CONCLUSIONS

The effect of the geometric and operating conditions on the overall performance of an AMR was evaluated in the present study. The design of an active magnetic regenerator for applications in magnetic refrigeration depends on a complex interaction between the regenerator geometry variables and the system operating parameters – the mass of magnetocaloric material, temperature profile in the solid and fluid, displaced mass of fluid per blow, duration of each blow and porous media characteristic dimension are a few of the parameters that will have influence over the performance. Those parameters will define a set of thermal, magnetic and mechanical losses associated with the operation of the device that will play an important role over the achieved efficiency.

A novel regenerator geometry and operating condition were selected for the fixed magnetic circuit of the magnetic refrigerator developed at Polo/UFSC. From the results obtained in this study through numerical simulations, it was selected a regenerator composed by 11 beds, each with a width of 29 mm, a length of 100 mm and a height of 10 mm. Assuming that the AMR is operating at 2 Hz and with a mass flow rate of 47.1 kg/h, the configuration would attain a cooling capacity of approximately 76 W over a system temperature span of 15 K and an overall COP of 0.8. Improvements on the power consumption of the electrovalves and the driving system can lead to COP values up to 2.5 and a second law efficiency of 13.2%. Further studies on AMR insulation, porous media particle diameter and multilayered AMRs can lead to larger improvements on the performance of the system.

5. ACKNOWLEDGMENTS

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6. REFERENCES

- Barclay, J.A. and Steyert, W.A., 1982. "Active magnetic regenerator".
- Capovilla, M.S., Lozano, J.A., Trevizoli, P.V. and Barbosa Jr., J.R., 2016. "Performance evaluation of a magnetic refrigeration system". *Science and Technology for the Built Environment*, pp. 534–543.
- Cardoso, P.O., Destro, M.C., Alvares, M.G., Hoffmann, G., Lozano, J.A.C. and Barbosa Jr., J.R., 2017. "Estimating energy consumption and experimental analysis of electro valves applied in magnetic refrigerators". In *Proceedings of the 24th ABCM International Congress of Mechanical Engineering - COBEM 2017*. Curitiba - Brazil.
- Cardoso, P.O., Destro, M.C., Alves, M.G., Lozano, J.A., DeNegri, V.J. and Barbosa Jr., J.R., 2016. "Modeling of a novel digital hydraulics system for magnetic refrigerators". In *7th International Conference on Magnetic Refrigeration at Room Temperature*.
- Dutra, S.L., Nakashima, A.T.D., Hoffmann, G., Lozano, J.A.C. and Barbosa Jr., J.R., 2017. "Using electrovalves as a flow

- distribution system for an active magnetic regenerator”. In *Proceedings of the 24th ABCM International Congress of Mechanical Engineering - COBEM 2017*. Curitiba - Brazil.
- Engelbrecht, K., Eriksen, D., Bahl, C.R.H., Bjørk, R., Geyti, J., Lozano, J.A., Nielsen, K.K., Saxild, F., Smith, A. and Pryds, N., 2012. “Experimental results for a novel rotary active magnetic regenerator”. *International Journal of Refrigeration*, Vol. 35, pp. 1498–1505.
- Ergun, S., 1952. “Fluid flow through a packed column.” *Chemical Engineering Progress*.
- Eriksen, D., Engelbrecht, K., Bahl, C.R.H., Bjørk, R. and Nielsen, K.K., 2016. “Effects of flow balancing on active magnetic regenerator performance”. *Applied Thermal Engineering*, Vol. 103, pp. 1–8.
- Hoffmann, G., Cardoso, P.O., Nakashima, A.T.D., Dutra, S.L., Lozano, J.A. and Barbosa Jr., J.R., 2017. “Actuation and control of electric valves for a magnetic refrigerator”. In *Proceedings of the 24th ABCM International Congress of Mechanical Engineering - COBEM 2017*. Curitiba - Brazil.
- Kitanovski, A., Tušek, J., Tomc, U., Plaznik, U., Ožbolt, M. and Poredoš, A., 2015. *Magnetocaloric Energy Conversion: From theory to applications*. Springer.
- Lozano, J.A., Capovilla, M.S., Trevizoli, P.V., Engelbrecht, K., Bahl, C. and Barbosa Jr., J.R., 2016. “Development of a novel rotary magnetic refrigerator”. *International Journal of refrigeration*, Vol. 68, pp. 187–197.
- Lozano, J.A., 2015. *Designing a rotary magnetic refrigerator*. Ph.D. thesis, Universidade Federal de Santa Catarina.
- Pallares, J. and Grau, F.X., 2010. “A modification of a nusselt number correlation for forced convection in porous media.” *International Communications in Heat and Mass Transfer*.
- Trevizoli, P.V., Christiaanse, T., Govindappa, P., Niknia, I., Teyber, R., Barbosa Jr., J.R. and Rowe, A., 2016a. “Magnetic heat pumps: An overview of design principles and challenges”. *Science and Technology for the Built Environment*.
- Trevizoli, P.V., Nakashima, A.T. and Barbosa Jr., J.R., 2016b. “Performance evaluation of an active magnetic regenerator for cooling applications - part ii: mathematical modeling and thermal losses”. *International Journal of refrigeration*, Vol. 72, pp. 192–205.
- Trevizoli, P.V., 2015. *Development of Thermal Regenerators for Magnetic Cooling Applications*. Ph.D. thesis, Universidade Federal de Santa Catarina.

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