PERFORMANCE OF A MAGNETIC REFRIGERATOR OPERATING WITH FINITE THERMAL CONDUCTANCE HEAT EXCHANGERS

Ricardo S. Calomeno, ricardo@polo.ufsc.br
Jaime A. Lozano, jaime@polo.ufsc.br
Jader R. Barbosa Jr., jrb@polo.ufsc.br
POLO - Research Laboratories for Emerging Technologies in Cooling Thermophysics, Department of Mechanical Engineering,
Federal University of Santa Catarina, Florianópolis, SC, 88040-900, Brazil

Paulo V. Trevizoli, paulot@uvic.ca
IESVic - Institute for Integrated Energy Systems, Department of Mechanical Engineering, University of Victoria, 3800 Finnerty Rd,
Victoria, B. C. V8W 396, Canada.

Abstract. *In magnetic refrigeration systems using an active magnetic regenerator (AMR) the thermal performance may have a significant variation according to the heat exchangers employed in the system. The finite overall thermal conductance (UA) of both heat exchangers can limit the cooling capacity obtained by the AMR for a certain temperature span. A mathematical model based in the effectiveness and the number of transfer units method was developed and implemented in a former AMR numerical model. The obtained results have indicated that the performance of the magnetic refrigerator decreases when the overall thermal conductance of the heat exchangers decreases. Another numerical simulations were performed to evaluate the influence of the operating parameters in the thermal performance of the AMR with real heat exchangers. Also, the influence of the heat loss through the AMR casing was analyzed. The results revealed a significant deterioration of the system performance as the thermal conductances are decreased. The system performance is more sensitive to the overall thermal conductance of the hot-side heat exchanger. This should be considered in the design of actual AMR operating with real heat exchangers.*

Keywords*: Heat exchanger, thermal conductance, thermodynamic performance, magnetic refrigerator.*

1. INTRODUCTION

Most of the magnetic cooling prototypes developed so far employ an electric heater to emulate the thermal load in direct contact with the working fluid at the cold heat exchanger (Kitanovski et. al, 2015). However, the thermal interaction between the cooling system and the refrigerated environment is not well represented by this method. In a real magnetic refrigeration system, the heat exchangers should have a finite overall thermal conductance (UA) which will affect the operating conditions of the active magnetic regenerator (AMR) and the thermodynamic performance of the device (Trevizoli et. al, 2016).

In this paper, the influence of the finite heat exchanger thermal conductances on the thermodynamic performance of a magnetic refrigeration system are analyzed. A mathematical model based on the method of effectiveness (ε) and the number of transfer units (NTU) was developed and implemented in a 1-D AMR numerical model developed by Trevizoli (2015). Numerical analyses of the AMR thermal performance were carried out considering a gadolinium (Gd) packed-sphere regenerator at different operating conditions. Tests evaluating the impact of the overall thermal conductance of the cold and the hot heat exchanger in the thermodynamic performance of the system were carried out. Also, the impact of the heat losses through the AMR casing in the performance of the system were analyzed.

2. METHODOLOGY

The magnetocaloric effect (MCE) is the thermal response of a magnetocaloric material when subjected to a changing in the magnetic field. An AMR is composed by a magnetocaloric porous matrix and it makes use of the magnetic work to transfer heat from a low-temperature environment to a high-temperature one (Barbosa Jr. et al, 2014).

In an AMR cycle, firstly, the magnetocaloric material is magnetized adiabatically and its temperature rises due to the MCE. While the magnetic field is maintain applied, a cold-to-hot blow of the heat transfer fluid is performed by passing the fluid through the porous matrix from the cold heat exchanger (CHEx) at temperature T_{CHEx} to the hot heat exchanger, cooling the AMR solid matrix, and exiting it with a temperature T_{HE}. This temperature is higher than the hot heat exchanger (HHEx) temperature, thus, it rejects heat to the hot environment and exits the HHEx at T_{HHEx}. Then, the AMR is demagnetized adiabatically and the solid temperature decreases, and a hot-to-cold blow is carried out by passing the heat transfer fluid from the HHEx to the CHEx and exiting the regenerator with a temperature of T_{CE}. An schematic diagram of the AMR cycle and its related variables can be observed in Fig. 1.

A system temperature span (ΔT_{sys}) is defined as the difference between the hot and cold environment temperatures, respectively, T_H and T_C. And a regenerator temperature span (ΔT_{reg}) is defined as the difference between the temperatures of the fluid exiting the hot and the cold ends of the regenerator, T_{HE} and T_{CE}, respectively. Due to the finite overall thermal conductance of the hot and the cold heat exchangers, UA_{HHEx} and UA_{CHEx}, respectively. The

temperature difference created at each heat exchanger to the environment temperature is respectively ΔT_{CHEx} and ΔT_{HHEx}.
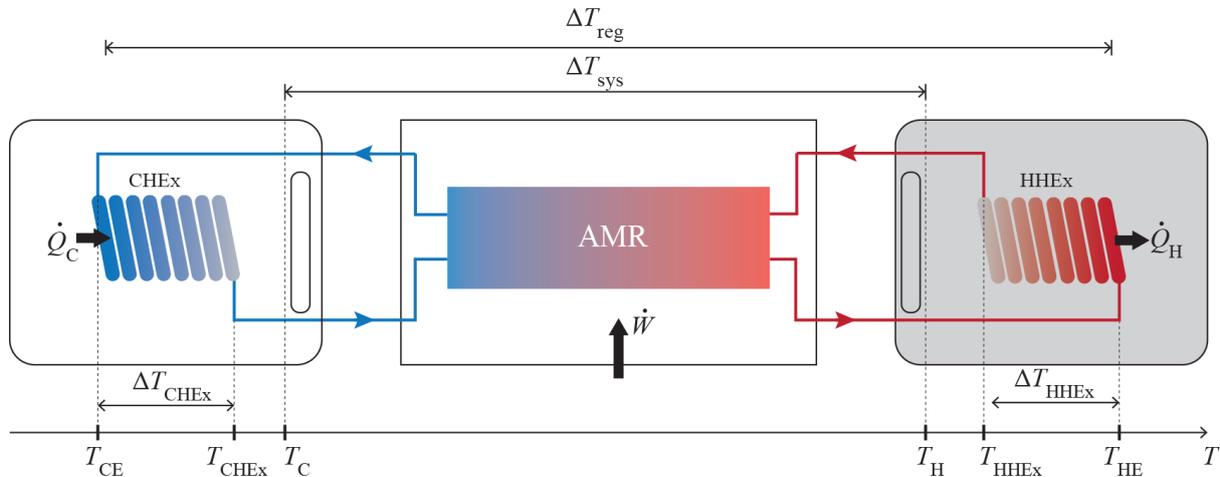

Figure 1. Schematic diagram of a refrigeraton system operated by an AMR and its related variables.

In this work, the heat exchangers are considered to have unidirectional flow of the heat transfer fluid. From the AMR numerical model, the temperatures of the fluid exiting the regenerator are known, thus, the ε-NTU method (Kays & London, 1984; Incropera & DeWitt, 2006) can be applied to predict the temperatures of the fluid exiting the heat exchangers.

The effectiveness (ε) of a heat exchanger can be defined as the ratio between the actual heat transfer rate and the maximum possible heat transfer rate. The effectiveness of the cold and hot heat exchangers are, respectively:

$$\varepsilon_{CHEx} = \frac{T_{CE} - T_{CHEx}}{T_{CE} - T_C} \tag{1}$$

$$\varepsilon_{HHEx} = \frac{T_{HE} - T_{HHEx}}{T_{HE} - T_H} \tag{2}$$

The number of transfer units (NTU) is defined as the ratio of the heat exchanger overall thermal conductance (UA) and the lowest thermal capacity between the two streams. In this work, it was assumed that the air thermal capacity is higher than the thermal capacity of the heat transfer fluid. The NTU of the cold and hot heat exchangers are, respectively:

$$NTU_{CHEx} = \frac{UA_{CHEx}}{\dot{m}.c_p} \tag{3}$$

$$NTU_{HHEx} = \frac{UA_{HHEx}}{\dot{m}.c_p} \tag{4}$$

where $\dot{m}$ is the mass flow rate and c_p the specific heat of the heat transfer fluid.

Due to the higher values of the air thermal capacity rate, the air temperature variations tend to be very small. In those situations, the effectiveness can be written in terms of the NTU for the cold and hot heat exchangers by the following expressions, respectively:

$$\varepsilon_{CHEx} = 1 - e^{NTU_{CHEx}} \tag{5}$$

$$\varepsilon_{HHEx} = 1 - e^{NTU_{HHEx}} \tag{6}$$

The aforementioned ε-NTU method equations were implemented in the AMR numerical model developed by Trevizoli (2015). After the cold and the hot blow, the temperature of the heat transfer fluid is obtained. The overall thermal conductance of each heat exchanger is treated as an input data. Therefore, the NTU and the effectiveness of each heat exchanger are known parameters and the temperature of the heat transfer fluid exiting the heat exchangers can be calculated. The temperature of the fluid entering the regenerator is updated until the convergence is reached.

At the present work, some numerical simulations were carried out to evaluate the thermal performance of an AMR connected with heat exchangers with finite overall thermal conductance. A gadolinium packed-sphere bed regenerator was employed, its geometry was based in Trevizoli (2015) and it is detailed in Table 1. The heat transfer fluid is a mixture of water and ehylene-glycol (20% vv.). The fluid is pumped in a square-wave (instantaneous) profile and the cold and hot blow are assumed to have the same period. Different values for the operation frequency, sphere diameters and overall thermal conductance of the heat exchangers were simulated. The simulation parameters and the regenerator geometry are presented in Table 1.

Table 1. Simulation parameters and regenerator geometry.

Parameter	Value
Magnetic field profile	Cosine wave form
Magnetic field (Min./Max.)	0 T /1.5 T
Hot Reservoir Temperature	300 K
Mass flow rate	100 kg/h
Regenerator length	100 mm
Regenerator diameter	22.22 mm
AMR matrix porosity	0.36
Sphere diameter	0.55 mm
Operation frequency	1 Hz

3. RESULTS

Initially, the effect of the finite overall thermal conductance of the heat exchangers in the thermodynamic performance of the AMR was evaluated. Here, the cold heat exchanger was assumed to have different overall thermal conductances (from 5 to 20 W/K) while the thermal conductance of the hot heat exchanger was held constant. In Figs. 2(a) and 2(b) are shown the performance curves for the AMR operating with the conditions summarized in Table 1 and the hot heat exchangers with overall thermal conductances of 10 and 15 W/K, respectively. These values for UA_{HHEx} represent commonly found thermal conductances in real domestic refrigerators.
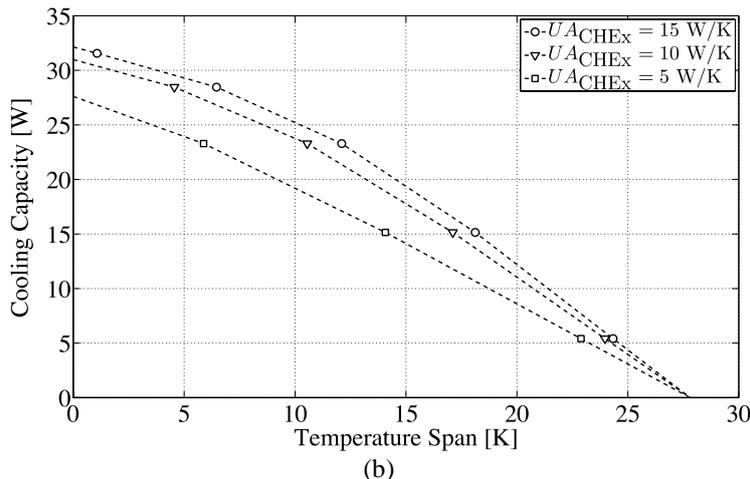

(a)

(b)
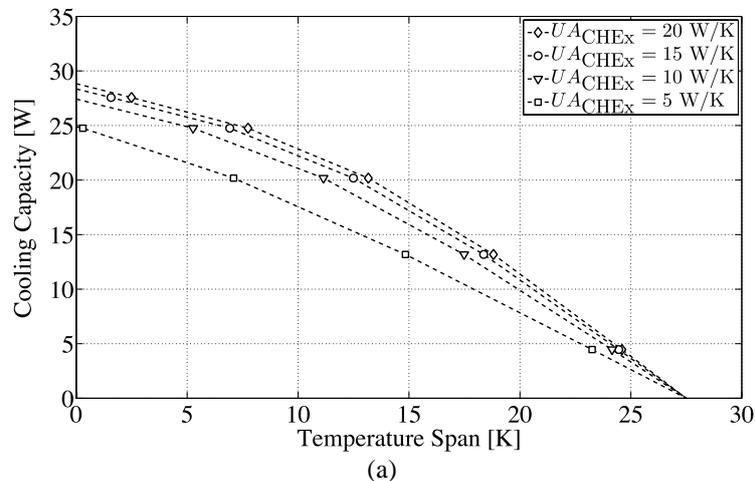
Figure 2. Performance curves for an AMR with real cold heat exchanger with different thermal conductances (from 5 W/K to 20 W/K) and real hot heat exchanger with overall thermal conductance of (a) 10 W/K and (b) 15 W/K.

The results show that the obtained cooling capacity increases as the overall thermal conductance of the cold heat exchanger increases. As UA_{CHEx} is varied, the AMR can absorb the same cooling capacity but at different system temperature spans. The difference between curves at the same cooling capacity increases as the temperature span becomes smaller. The increase of UA_{CHEx} results in an improvement of the AMR performance, but this is very small at the largest values of temperature span where the desired operating points of the refrigerator are located. A similar behavior is found when UA_{HHEx} is increased from 10 to 15 W/K. Nevertheless, having a large UA_{CHEx} does not compensate for a small UA_{HHEx}, as a higher heat rate is transferred to the hot reservoir. Therefore, a balance between the design of the heat exchangers and the performance of the AMR must be found so that a competitive (low-priced) refrigerator can be developed.

In a second analysis, the influence of the AMR frequency in the performance of an AMR operating with heat exchangers with overall thermal conductance of $UA_{CHEx} = 10$ W/K and $UA_{HHEx} = 15$ W/K was evaluated. The operating AMR frequency was varied from 0.5 Hz to 2 Hz and the results are shown in Fig. 3. For the mass flow rate of 100 kg/h used in those simulations there thermal performance of the AMR was decreased as the operating frequency was decreased. For higher values of frequency, such as 1.5 and 2 Hz, the temperature span is increased and the cold heat exchanger can achieve higher values of the cooling capacity in a reasonable ranges of the temperature span.
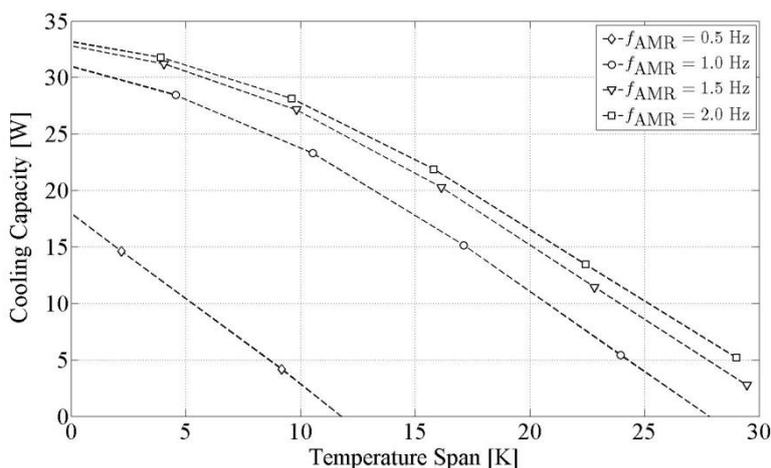

Figure 3. Influence of the AMR frequency in the performance of an AMR operating with heat exchangers with overall thermal conductance of $UA_{CHEx} = 10$ W/K and $UA_{HHEx} = 15$ W/K.

The influence of the Gd spheres diameter (d_{sp}) in the thermal performance of the AMR was verified. In this analysis, three scenarios were carried out, each with a different d_{sp} of 0.35, 0.45 and 0.55 mm. For these simulations the operating AMR frequency was held at 1.0 Hz and the overall thermal conductance of the cold and hot heat exchangers were, respectively, 10 and 15 W/K, and the obtained results are shown in Fig. 4. For such spheres diameters, the performance is slightly decreased as the diameter is increased. Inversely, the pressure drop is much more increased as the sphere diameters are decreased. Smaller spheres have a higher surface area which will improve the heat transfer

between the porous matrix and the heat transfer fluid. However, the spheres diameter affects its manufacturing cost and the pressure drop that should be compensated by the hydraulic system (pumping power) of the magnetic refrigerator.
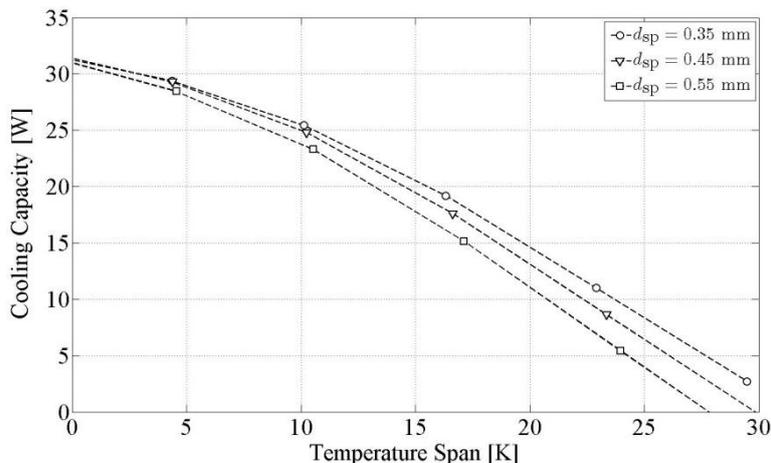

Figure 4. Influence of the Gd spheres diameter in the performance of an AMR operating with heat exchangers with overall thermal conductance of $UA_{CHEx} = 10$ W/K and $UA_{HHEx} = 15$ W/K.

Finally, the impact of the heat losses through the AMR casing in the performance of the AMR was evaluated. In this case, three scenarios were carried out, analogous to those presented in Fig. 2(b) which don't have heat losses, the operating frequency was considered of 1.0 Hz, spheres diameter of 0.550 mm, and the overall thermal conductance of the hot heat exchanger was 15 W/K, while the overall thermal conductance of the cold heat exchanger was varied from 5 to 15W/K, and the obtained results are shown on Fig. 5. Comparing Figs. 2(b) and 5, it can be seen that, due to the heat losses, the cooling capacity for each temperature span and thermal conductance is decreased substantially, and this effect is larger as the temperature span is higher.
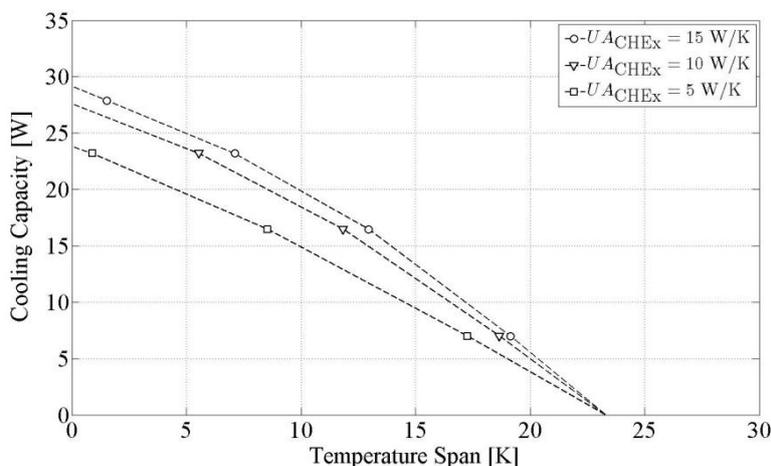

Figure 5. Influence of the cassing heat losses in the performance of the AMR operating with heat exchangers with overall thermal conductance of $UA_{HHEx} = 15$ W/K and UA_{CHEx} varied from 5 to 15 W/K.

4. CONCLUSIONS

In this work, the influence of the overall thermal conductances of the cold and the hot heat exchanger were evaluated in terms of the AMR thermodynamic performance. The effect of some operating parameters, such the AMR frequency and the spheres diameter, were simulated. Also, an analysis was carried out to evaluate the cooling capacity in the situation with heat losses through the AMR casing. To evaluate the performance of an AMR with heat exchanger with a finite overall thermal conductance for the cold and hot heat exchanger a mathematical model based on the ε-NTU method was developed and implemented in a former developed AMR numerical model. The obtained results indicates the AMR thermal response for different conditions according to the chosen values for the operation frequency, spheres diameter and the overall thermal conductance of the cold and hot heat exchanger. The design of the heat exchangers and the operating conditions to improve the AMR thermal performance is subject of further studies.

5. ACKNOWLEDGEMENTS

6. REFERENCES

Barbosa Jr., J. R., Lozano, J. A., Trevizoli, P. V., 2014. Magnetocaloric Refrigeration Research at the INCT Cooling and Thermophysics. 15th Brazilian Congress of Thermal Science and Engineering, Belém, PA, Brazil.

Incropera, F. P. and DeWitt, D. P., 2006. Fundamentals of Heat and Mass Transfer, John Wiley & Sons, New York.

Kays, W. M. and London, A. L., 1984. Compact Heat Exchangers, Krieger Publishing Co.

Kitanovski A., Tusek, J., Tomc, U., 2015. Magnetocaloric Energy Conversion: From theory to applications, Springer.

Trevizoli, P.V., 2015. Development of Thermal Regenerators for Magnetic Cooling Aplications. Ph.D. thesis, Federal University of Santa Catarina. Avaible in: http://tede.ufsc.br/teses/PEMC1634-T.pdf

Trevizoli, P.V., Christiannse, T. V., Govindappa, P., Niknia, I., Teyber R., Barbosa Jr., J. R., Rowe, A. 2016. Magnetic heat pumps: An overview of design principles and challenges. Science and Technology for the Built Environment. DOI: 10.1080/23744731.2016.1171632

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.