INFLUENCE OF THE HEAT EXCHANGERS THERMAL CONDUCTANCE ON THE THERMODYNAMIC PERFORMANCE OF A MAGNETIC REFRIGERATOR

R.S. Calomeno\textsuperscript{(a)}, J.A. Lozano\textsuperscript{(a),*}, P.V. Trevizoli\textsuperscript{(b)}, J.R. Barbosa Jr.\textsuperscript{(a)}

\textsuperscript{(a)} POLO — Research Laboratories for Emerging Technologies in Cooling and Thermophysics, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, SC, 88040-900, Brazil

\textsuperscript{(b)} IESVic — Institute for Integrated Energy Systems, Department of Mechanical Engineering, University of Victoria, 3800 Finnerty Rd, Victoria, B.C. V8W 3P6, Canada

*Corresponding author. E-mail: jaime@polo.ufsc.br

ABSTRACT

In the majority of the magnetic refrigeration devices developed so far, the thermal load is provided by an electrical heater in direct contact with the working fluid in the cold heat exchanger. This is acceptable in laboratory devices, but does not represent the real thermal interactions taking place in the heat exchanger in contact with the refrigerated environment (e.g., the air inside the cabinet of the refrigerator). In this work, the influence of the thermal conductances of the heat exchangers on the thermodynamic performance of a magnetic refrigerator has been analyzed. A mathematical model based on the $\varepsilon$-NTU method was developed and implemented in an AMR numerical model. The results reveal 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.

DOI: 10.18462/iir.thermag.2016.0129

1. INTRODUCTION

Most magnetic cooling prototypes developed to this date emulate the refrigerator thermal load via electrical heaters in direct contact with the working fluid in the cold heat exchanger [1]. However, this does not represent the actual thermal interaction between the cooling system and the refrigerated environment in real cold-side heat exchangers. In a real magnetic refrigerator, for instance, the cold and hot heat exchangers (CHEX and HHEX, respectively) have finite overall thermal conductances ($UA$) that affect the thermodynamic performance of the AMR refrigeration system. The thermal design of the cabinet itself and of the heat exchangers is restricted by space (volume) and cost limitations.

In this paper, we analyze the influence of the cabinet and heat exchanger thermal conductances on the thermodynamic performance of a magnetic refrigerator. A mathematical model based on the effectiveness ($\varepsilon$) and number of transfer units (NTU) method was developed and implemented in an active magnetic regenerator (AMR) numerical model developed in [2]. Numerical analyses of the AMR thermal performance were carried out considering a gadolinium (Gd) packed-sphere regenerator at specified operating conditions. Different values of the overall thermal conductance were assigned to the cold and hot heat exchangers to quantify their impact on the system thermodynamic performance.

2. METHODOLOGY

The various temperatures and other relevant variables of a magnetic refrigeration system are schematically represented in Fig. 1. The system temperature span ($\Delta T_{sys}$) is defined as the difference between the hot and cold environment temperatures, $T_H$ and $T_C$, respectively. The regenerator temperature span ($\Delta T_{reg}$) is defined as the difference between the temperature 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 conductances of the hot and cold heat exchangers ($UA_{HHEX}$ and $UA_{CHEX}$, respectively) there are temperature differences associated with each heat exchanger.
(\(\Delta T_{\text{HHEX}}\) and \(\Delta T_{\text{CHEX}}\), respectively). Thus, the fluid temperatures entering the regenerator are, respectively, \(T_{\text{HHEX}}\) and \(T_{\text{CHEX}}\).

\[ \Delta T_{\text{reg}} \]
\[ \Delta T_{\text{sys}} \]

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

\(T_{\text{HE}}\) and \(T_{\text{CE}}\) are usually output parameters in AMR models, and correspond to the temperatures entering the heat exchangers. In this work, unidirectional flow of the heat transfer fluid is assumed in the heat exchangers, so the \(\varepsilon\)-NTU method [3,4] is employed to predict the outlet fluid temperature in each heat exchanger for different values of overall thermal conductance. The NTU is the ratio of the heat exchanger overall thermal conductance and the lowest thermal capacity rate between the two streams. Assuming that the air in hot and cold environments has a much higher thermal capacity, the NTUs of the hot and cold heat exchangers can be defined as:

\[
NTU_{\text{HHEX}} = \frac{UA_{\text{HHEX}}}{\dot{m}_f c_f}
\]

\[ (1) \]

\[
NTU_{\text{CHEX}} = \frac{UA_{\text{CHEX}}}{\dot{m}_f c_f}
\]

\[ (2) \]

where \(\dot{m}_f\) is the mass flow rate and \(c_f\) is the specific heat capacity of the fluid.

The effectiveness of a heat exchanger is defined as the ratio of the actual heat transfer rate and the maximum possible heat transfer rate. In terms of temperatures, the effectiveness of the hot and cold heat exchangers are given, respectively, by:

\[
\varepsilon_{\text{HHEX}} = \frac{T_{\text{HE}} - T_{\text{HHEX}}}{T_{\text{HE}} - T_{\text{H}}}
\]

\[ (3) \]

\[
\varepsilon_{\text{CHEX}} = \frac{T_{\text{CE}} - T_{\text{CHEX}}}{T_{\text{CE}} - T_{\text{C}}}
\]

\[ (4) \]

For very large thermal capacity rates of the air in the hot and cold environments, their temperature changes along the heat exchangers are expected to be small. In this situation, the effectivenesses of the hot and cold heat exchangers are related to the numbers of transfer units by the following expressions:

\[
\varepsilon_{\text{HHEX}} = 1 - e^{-NTU_{\text{HHEX}}}
\]

\[ (5) \]

\[
\varepsilon_{\text{CHEX}} = 1 - e^{-NTU_{\text{CHEX}}}
\]

\[ (6) \]

The above \(\varepsilon\)-NTU relationships were implemented coupled with the one-dimensional AMR numerical model developed in [2]. After each blow, the temperatures of the fluid exiting the regenerator are updated and
become the inlet temperatures of the heat exchangers. Since $U_{A_{HHEX}}$ and $U_{A_{CHEX}}$ are treated as input data, the NTU for each heat exchanger are also known so the temperatures of the fluid entering the regenerator are updated until the convergence is obtained.

In this work, numerical simulations were carried out to evaluate the thermal performance of an AMR connected to heat exchangers with different thermal conductances and subjected to different operating conditions. The simulations assumed a Gd packed-sphere bed regenerator with a geometry based on those analyzed in [2]. The heat transfer fluid was a water/ethylene-glycol mixture (20% vv.). A square-wave (instantaneous) fluid pumping profile with equal blow periods was assumed. A summary of the regenerator geometry and the simulation conditions are presented in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Regenerator length</td>
<td>100 mm</td>
<td>Porosity of AMR matrix</td>
<td>0.36</td>
</tr>
<tr>
<td>Regenerator diameter</td>
<td>22.22 mm</td>
<td>Mass flow rate</td>
<td>100 kg/h</td>
</tr>
<tr>
<td>Magnetic field Profile</td>
<td>Cosine wave form</td>
<td>Operating frequency</td>
<td>1 Hz</td>
</tr>
<tr>
<td>Magnetic field (Min./Max.)</td>
<td>0 T / 1.5 T</td>
<td>Hot reservoir temperature</td>
<td>300 K</td>
</tr>
<tr>
<td>Sphere diameter</td>
<td>550 μm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 1. Regenerator geometry and simulation parameters.

3. RESULTS

The first analysis consisted of verifying the performance of the AMR when the overall thermal conductance of both heat exchangers is varied. The performance curves (cooling capacity as a function of the system temperature span) for the AMR with ideal heat exchangers (infinite $UA$) and for real heat exchangers with $U_{A_{HHEX}} = U_{A_{CHEX}} = 25$, 15 and 10 W/K are shown in Fig. 1. In most AMRs developed so far, the heat exchangers have been considered ideal. In small capacity refrigeration systems (e.g., a domestic refrigerator), overall thermal conductances of 25, 15 and 10 W/K correspond to values that are typically twice those obtained with static (i.e., natural convection) condensers and evaporators [5,6]. The results have demonstrated that the performance of the regenerator deteriorates significantly as the thermal conductance of the heat exchangers is decreased; that is, for the same temperature span the cooling capacity is lower as the $UA$ values are smaller. Therefore, it is very important to find optimal designs (in terms of cost and volume) of the heat exchangers for an actual cabinet. Generally, the heat exchanger cost is directly proportional to the $UA$, as this increases with the amount of solid material (i.e., metal) in the heat exchanger.

![Figure 2](image)

Figure 2. Performance curves for an AMR with ideal and real heat exchangers with different thermal conductances.

Usually, in an actual cabinet, the cold heat exchanger has less space available in comparison to the hot heat exchanger (which is placed outside the cabinet). The thermal performance of an AMR operating with real heat exchangers was evaluated through numerical simulations assuming fixed thermal conductances of the hot heat exchanger and different thermal conductances of the cold heat exchanger. Two scenarios were simulated, $U_{A_{HHEX}}$ was maintained at 10 W/K and at 15 W/K while varying $U_{A_{CHEX}}$ from 5 to 20 W/K. The resulting performance curves for an AMR operating with these combinations are shown in Figs. 3(a) and 3(b), respectively. Here, $U_{A_{HHEX}}$ represents two values of thermal conductance commonly used in real
domestic refrigerators. As can be seen in Figs. 3(a) and 3(b), the performance of the AMR is reduced as $U_{A_{\text{CHEX}}}$ is decreased. As $U_{A_{\text{CHEX}}}$ 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 $U_{A_{\text{CHEX}}}$ 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 $U_{A_{\text{HHEX}}}$ is increased from 10 to 15 W/K. Nevertheless, having a large $U_{A_{\text{CHEX}}}$ does not compensate for a small $U_{A_{\text{HHEX}}}$, 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.

4. CONCLUSIONS

The influence of the heat exchangers thermal conductance on the thermodynamic performance of a magnetic refrigerator has been analyzed in this work. A mathematical model based on the $\varepsilon$-NTU method was developed and implemented in an AMR numerical model. Numerical simulations assuming an AMR with hot and cold heat exchangers with different overall thermal conductances were carried out to evaluate the influence on the thermodynamic performance. It has been demonstrated that special care has to be taken when designing an AMR that will operate an actual cabinet with real heat exchangers. The design of heat exchangers and the thermodynamic performance evaluation, in terms of the coefficient of performance and the second law efficiency, for an actual AMR will be subjects of further studies.

ACKNOWLEDGMENTS

Financial support from CNPq, Embraco and the EMBRAPII Unit Polo/UFSC is duly acknowledged.

REFERENCES