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DESIGN AND EXPERIMENTAL VALIDATION OF A FLUID FLOW MANAGEMENT SYSTEM BASED ON SOLENOID VALVES FOR A LARGE MAGNETIC REFRIGERATOR

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Abstract.*Magnetic refrigeration stands out as one of the most promising emerging cooling technologies. It is based on the magnetocaloric effect (MCE) which is the thermal response of a magnetic material to a change in magnetic field. A magnetocaloric device is composed mainly of a magnetic circuit, a active magnetic regenerator (AMR) and a hydraulic system. The latter is based mainly on a fluid flow management system (FFMS) that allows alternating fluid flows through the AMR and it has an important role in the overall performance of the magnetic refrigerator as it is generally one of the major energy-consuming components of the device. A novel approach to achieve a more efficient FFMS for a large magnetic refrigerator (LMR) is proposed by using electronically-controlled (solenoid) valves. In the development of a LMR, several operating requirements arise that the FFMS has to overcome. To emulate the operating conditions of a LMR, an experimental apparatus has been developed and experimental analyses of a selected set of commercial solenoid valves were carried out. Their performance has been evaluated in terms of mass flow rates and operating frequencies analogous to those require by a LMR. The experimental results show that solenoid valves could be successfully applied to a LMR as the opening and closing time responses obtained by the electronic valves and the resulting mass fluid flow profile do fulfill the requirements needed to operate a LMR.*

Keywords: *Magnetocaloric refrigeration, large magnetic refrigerator, fluid flow management system, solenoid valve*

1. INTRODUCTION

Magnetic refrigeration is an emerging cooling technology that is arising as an alternative to mechanical vapor compression (Dutra *et al.*, 2017). This technology is based on the magnetocaloric effect (MCE), which is the thermal response presented by some materials to a variation of the applied magnetic field. The magnetocaloric materials (MCMs) are applied as solid refrigerants in an active magnetic regenerator (AMR) as a porous medium. In most of the prototypes the AMR operation is based on the idealized thermo-magnetic Brayton cycle (Barclay and Steyert Jr, 1982), which can be divided in four main steps: (i) *adiabatic magnetization*: the magnetic field applied to the AMR is adiabatically changed and the MCM temperature increases as a consequence of the MCE; (ii) *cold blow*: fluid from the cold source flows across the AMR, removing heat and rejecting it at the hot source, so the MCM temperature decreases; (iii) *adiabatic demagnetization*: the magnetic field applied to the AMR is adiabatically removed and the porous medium temperature decreases (reversible MCE characteristic); and (iv) *hot blow*: fluid from the hot source flows through the AMR and is cooled down by the porous medium, so it can absorb heat from the cold source; the MCM temperature increases.

A magnetic refrigerator apparatus is composed, basically, by: a magnetic circuit, heat exchangers, active magnetic regenerators and a fluid flow management system (FFMS). Flow management is a critical issue in AMR systems, as unidirectional flow through the heat exchangers (thermal reservoirs) is required, despite the oscillatory nature of the blow steps in the regenerator (Dutra *et al.*, 2017). The complexity of the FFMS increases with the number of AMRs used in the system. As the cooling capacity scales with the mass of MCM in the system, the FFMS directs affects the total power consumption and, consequently, the coefficient of performance (COP) of the system. As a result, design guidelines are needed for the fluid flow management system that take into account the overall efficiency and power consumption. Besides that, a crucial aspect is the significant accuracy level required in time response of the FFMS regarding the volume of fluid displaced, especially at high operation frequencies. The FFMS must guarantee the synchrony between the magnetic field applied to the AMR by the magnetic circuit and the adequate fluid flow direction through the AMR during the cycle. So, it is crucial to enable a correct operation of a large magnetic refrigerator (LMR). Therefore, the main goal of this paper is to experimentally evaluate and validate a possible more efficient fluid flow management for a large magnetic refrigerator system using electronically-controlled (solenoid) valves.

Literature Review

A proposal of an ideal synchronization between the magnetic and fluid flow profiles in an AMR, according to Teyber *et al.* (2017), Fortkamp *et al.* (2018) and Nakashima *et al.* (2018b), is presented in Fig. 1. The dotted curve (trapezoid) shows the magnetic field profile, while the hydraulic fluid flow is presented in the dashed curve (rectangular) where there are instantaneous opening and closing steps and a stratified fluid flow plateau. A positive fluid flow (from cold side to hot side) can be noted when there is a magnetic field applied to the AMR and negative fluid flow (from hot side to cold side) when there is no magnetic field applied ($H = 0$). However, it is not easy to achieve a hydraulic profile with these pointed characteristics, hence it is necessary to look for alternative possibilities, which would reproduce as close as possible the ideal fluid flow profile during operation.

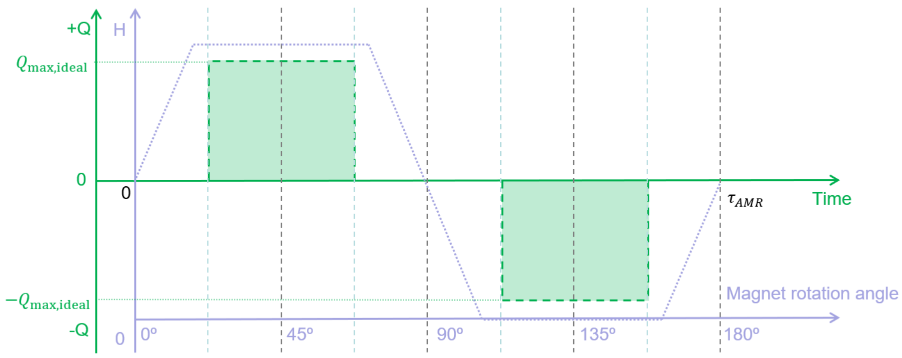


Figure 1: Ideal synchronization between the magnetic and the fluid flow profiles.

Over the last years, several solutions to perform the fluid flow management of magnetic refrigerators have been evaluated. Among them, different solutions to perform hydraulic management, have been presented the literature: (i) *double effect pumps and check valves*, which according to Trevizoli and Barbosa (2015) are indicated for systems with a small number of regenerators due to require a piston for each pair of AMRs. In addition, the amount of fluid displaced per cycle is directly dependent of the piston chamber volume, therefore fixed once the piston is selected. Then, to modify the amount of displaced fluid volume it is necessary to replace the component; (ii) *Rotary valves (face-to-face sealing)* (Lozano *et al.* (2016) and Nakashima *et al.* (2017)), this solution presents problems with recirculation and leakages, in addition to high energy consumption, so these facts are against the objectives of the project. However, the rotary valves bring a greater versatility in terms of fluid flow, *i.e.* displaced fluid flow volume, once the flow period can be controlled by the size of the seal oblong; (iii) *Mechanical valves driven by cams* (Eriksen *et al.*, 2016), this solution commonly demands a complete development in terms of design, not only concerning the drive mechanism but also for the hydraulic sealing components. It is also necessary to develop a system to activate them through an external element, usually a rotary one; and (iv) *solenoid valves* (Ebel *et al.* (2016), Cardoso *et al.* (2016), Hoffmann *et al.* (2017), Dutra *et al.* (2017), Teyber *et al.* (2017) and Nakashima *et al.* (2018a)), which have showed to be the most promising solution in terms of practicality and cost of design. This solution greatly makes the FFMS practical, once it allows the adjust of the controlled variables in real time by changing the operating parameters, which can be easily modified by an interface. Therefore, a software development is usually necessary to apply this solution, in terms of a prototype it is interesting in characterization tests phases. In addition, this solution requires a system to ensure its synchrony with the magnetic profile, when in Magnetic Refrigeration applications, hall sensor or encoders can be applied.

As previously exposed, solenoid valves have become the most promising solution to be used as the main component of the FFMS in refrigeration systems applying caloric technologies, with emphasis on magnetic refrigeration. Solenoid Valves are commercially available devices, therefore easily interchangeable, providing a great capacity for adjustments when linked, mainly, to an electronic control interface. There is a wide range of solenoid valves available on the market for applications in fluid flow control devices, however it is necessary that the commercial valve properly meets the specific design requirements, essentially, in terms of response time, fluid flow profile, frequency and pressure operation. In addition, not only the requirements regarding fluid flow and pressure drop when in application are important, but also other requirements, such as low noise and affordable cost should also be considered during the design evaluation.

The decision to apply solenoid valves as a flow control device in the FFMS was justified by dos Santos (2020). The selection was made by a comparison between different solutions: several solenoid valve models and mechanical valves operated by cam. The selected valve was the Asco model SC8210-112/220V-NC, which showed the best balance of desirable results between the candidates evaluated. The hydraulic systems applied in most studies on caloric cooling technologies are quite similar, so they can share the same principles, which emphasizes the importance of this work. So, the results obtained here can be extended, in order to be an alternative solution not only for magnetic refrigeration devices but also for other new caloric technologies that require the application of a FFMS. The main requirements for the valve operation are shown in Tab. 1.

Table 1: Requirements for the solenoid valves operation.

Number of regenerators	16 units
Operation frequency	3 Hz
Mass flow rate per AMR	750 kg/h

The parameters presented in Table 1 were provided by the other components of the project of which this research is included, so they are input parameters for the selection of the valve to be applied in a MR prototype. It is important to keep in mind that given the oscillatory flow and arrangement in the magnetic circuit, that are two options to perform the flow management of the MR: (i) each pair of valves operates only one regenerator or (ii) each pair of valves operates two regenerators. There is an oscillatory behavior concerning inlet and outlet blows in the regenerators, to manage them it is possible to arrange the AMRs and use a pair of valves per regenerator or group them in couples and control it, two valves for each couple of regenerators. So, there are two main possibilities to apply a set of solenoid valves in the final prototype and the number of valves can increase the complexity and the flexibility of the system.

2. EXPERIMENTAL APPARATUS

As a sequence of the selection and comparison carried out by dos Santos (2020) a validation of the selected commercial solenoid valve was required to evaluate the possibility of the application of this solution as the FFMS. Thus, an experimental apparatus was developed in this work to provide the relevant information about the candidate's operation, allowing the characterization and subsequent standardized validation during the tests. Figures 2 and 3 show, respectively, a schematic diagram and a photograph of the experimental apparatus developed. When in operation, the fluid flow leaves the tank (T1), crosses through the control and hydraulic power unit and the fluid flow left the filter (F1) and, if it does not return to the reservoir by any of the lines, the by-pass (G1 and A1) or the pressure control valve (G2 and E1), it goes through an air purge valve and reaches a fork. Then, the fluid flow can cross through one of the two test lines placed in parallel, depending on the set of valves that are activated (V1 and V2 or V3 and V4). In each line, the fluid flows through a solenoid valve under test, a gate valve (R1 or R2), crosses another valve under test and arrives at a union of the two previously bifurcated lines. One of the lines has pressure transducers (from PT02 to PT05), placed before and after each valve under test. The needle valve is used to impose a pressure drop in order to emulate the components of the final MR system, which will be positioned between the solenoid valves. After the union, there is a Coriolis flow transducer (FM01) located in the return line to the reservoir.

The system operation tests were aimed to emulate the hydraulic circuit designed for a MR and the experiments have been performed in similar conditions as those expected in the final application. The hydraulic performance was evaluated in order to understand the candidate's behavior during each test period, where the periods were divided in 3 steps: closed valve, opened valve and transition period (between opening and closing periods). Evaluating each instrumented valve individually, using valve V1 as an example, the procedure performed to check out their behavior was stipulated as follows: if the instantaneous pressure downstream of the valve, in this case the equivalent pressure signal of the pressure transducer PT03, was greater than 90% of the defined operating pressure (adjusted with V1 and V2 opened, in steady state conditions), the valve was considered opened. However, if the equivalent pressure signal read by PT03 was lower than 110% of atmospheric pressure (1 bar), the valve was considered closed, and if the pressure value was between the two limits shown, the valve was considered in transition period.

Then, the main goal was to verify the ability of the selected solenoid valve to operate with a satisfactory balance between closed and opened conditions to be able to compose the FFMS in a MR device. Also, it is interesting to select a valve which has the shortest time in transition period, aiming to obtain the experimental profile which better approaches the ideal profile as shown in Fig.1. Once a MR is required to work at multiple operation points, with several combined conditions in terms of operating frequency, pressure drop and mass flow rate, tests were carried out covering a wide range of these mentioned parameters. Table 2 shows the spectrum of them, as a total of 120 tests, which were obtained through all combinations of mass flow rate, ΔPs and operation frequency, in order to achieve a better characterization of the candidate's performance.

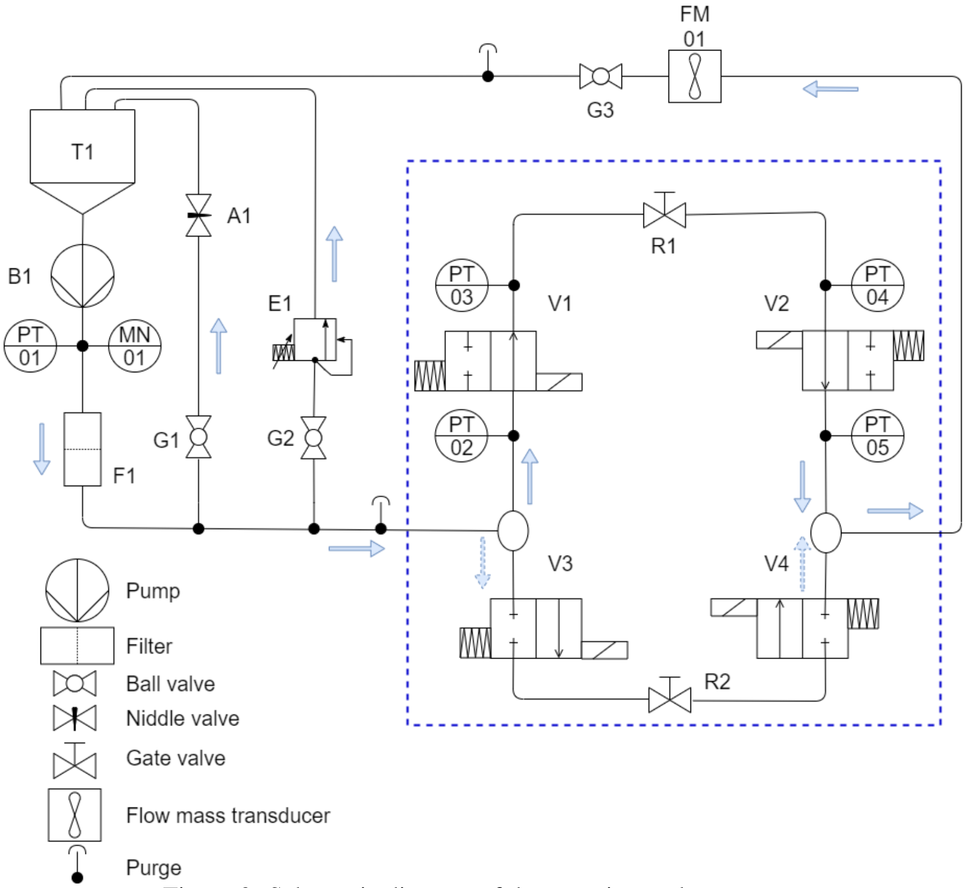


Figure 2: Schematic diagram of the experimental apparatus.

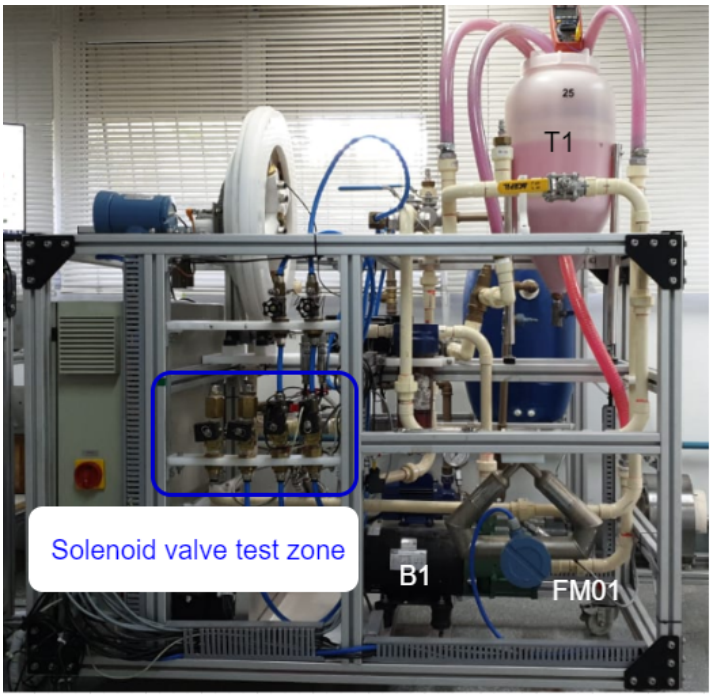


Figure 3: Photograph of the experimental apparatus.

3. RESULTS

An example of the analysis of the valve steps characterization is presented in Figure 4, where is possible to see the Opened, Closed and Transition stages for an experimental test. This shows that when in Open mode the valve tends to

Table 2: Test parameters to characterize the selected valve.

Mass flow rate [kg/h]	ΔP [bar]	Operation Frequency [Hz]
500	2	1
750	3	2
1000	4	3
1250	5	4
1500		5
1750		

behave above a Plateau previously adjusted; when in Closed mode the valve does not experience pressure variations; in Transition mode to opening is just a quick jump for the plateau but in Transition for closing the valve spends more time, this happens because of the type of valve, Normal Closed with Piloting, who uses the solenoid for opening and piloting for pressure equalization in shutdown.

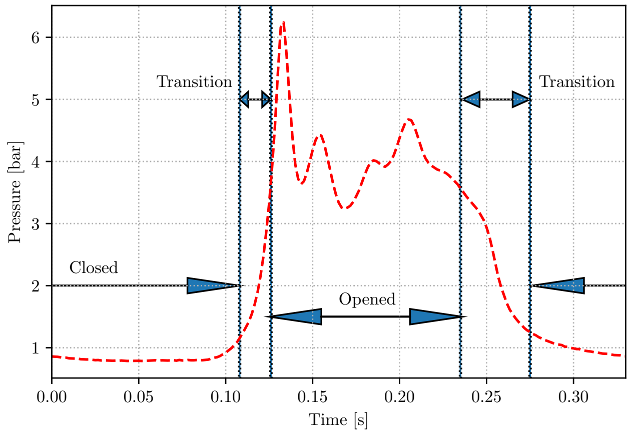
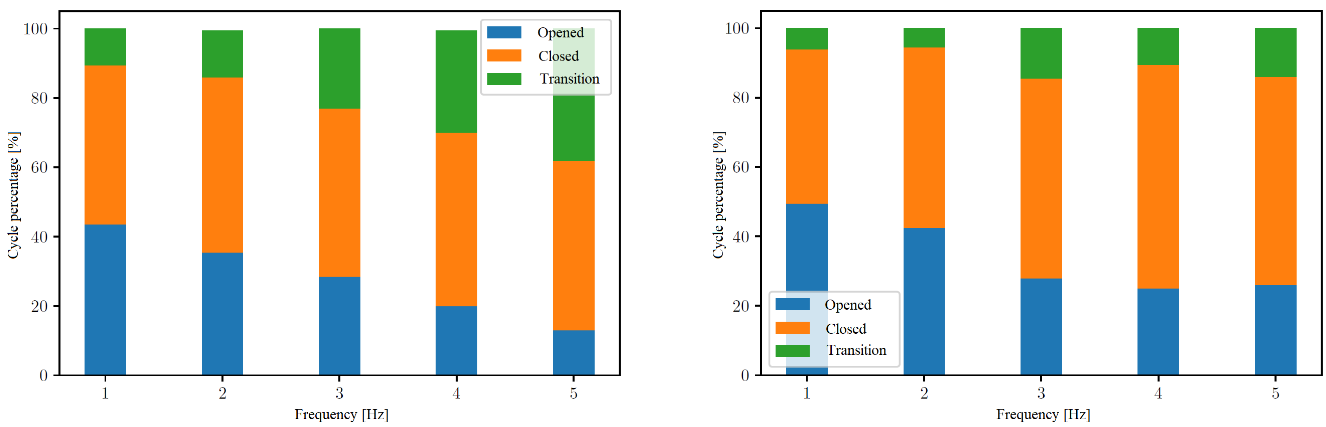


Figure 4: Example of valve steps characterization.

As a result of the analysis of the valves characterization, Figure 5 shows the distribution of the solenoid valve positions in a bar plot for mass flow rates of 750 and 1500 kg/h, (a) and (b) respectively, to different operation frequencies, the pressure drop between the evaluated valves was kept around 4 bar in these tests. Comparing Figs. 5a and 5b, it can be seen that, for a higher mass flow rate, the valve tends to spend less percentage of the cycle in transition phase and, consequently, spends more time in opening and closing positions, so it indicates a promising behavior when 16 valve units are used, once a couple of valves manage two AMRs and the mass flow rate should be twice comparing with the other possibility.



(a) Operating at 750 kg/h. (b) Operating at 1500 kg/h.

Figure 5: Cumulative bar graph showing the percentage distribution of the valve position in a valve cycle.

A disadvantage of this choice would be the fact that the head loss is less expressive, or it reduces when the mass flow rate decreases. In this case comparing the 750 kg/h to the 1500 kg/h solution, as can be seen in Fig. 6, the difference between them would be less than 0.15 bar, which is negligible when compared to the other components of the system, such as the heat exchange or the AMR, being the pressure drop more than 1 bar for these component.

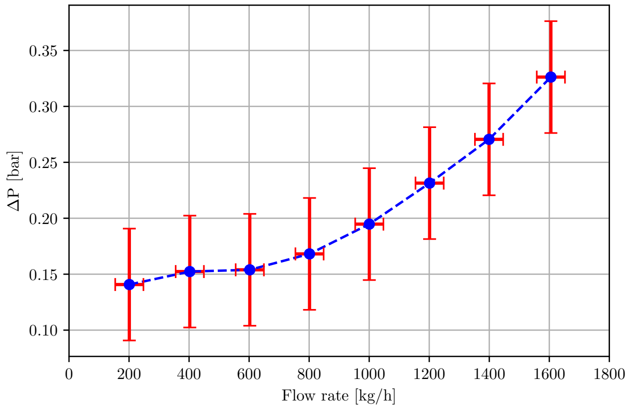


Figure 6: Experimental pressure drop on the solenoid valve for different mass flow rates.

Figure 7 shows the results for the fluid flow profile as a function of the opening time of the valve mass flow rates in discussion. It can be noted that they display very similar results, being the temporal response behavior when the valve is opened at 1500 kg/h closer of the rectangular (much like a trapezoid) geometry (ideal) than that one shown at 750 kg/h. It is also notable that for a greater mass flow rate there is less pressure fluctuation immediately after the valve opening, which is advantageous to avoid recirculation and pressure peaks in the hydraulic system.

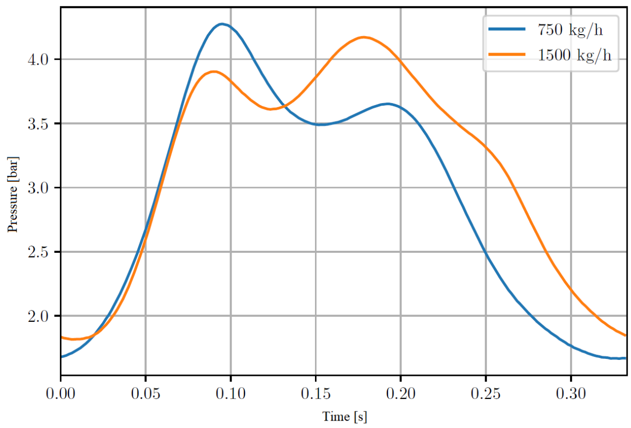


Figure 7: Flow profiles of the chosen candidate operating at 3 Hz.

4. CONCLUSION

Through the measurements of the solenoid valve operating periods, it was noted that, for higher mass flow rate values, the valve has had lower time in transition phase. At 1500 kg/h the results shown lower transition times when compared with the operation at 750 kg/h. So, as close as the experimental mass flow rate profile is to the ideal profile more interesting it is for a LMR operation. Thus, given the results obtained in this work, it was possible to evaluate and validate the behavior of the selected solenoid valves and to propose the application in a set of 16 valves to compose the Hydraulic Management System of the Magnetic Refrigeration Unit. So, the AMRs should be arranged in pairs as described previously and a pair of valves will be responsible to manage a pair of AMRs.

5. ACKNOWLEDGEMENTS

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