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EXPERIMENTAL EVALUATION OF THE FLUID FLOW MANAGEMENT SYSTEM OF A MAGNETIC AIR CONDITIONER

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Abstract. *Magnetic refrigeration stands out as one of the most promising emerging cooling technologies. It relies on the magnetocaloric effect - the thermal response some magnetic materials present upon a variation in the magnetic field applied over them. A magnetocaloric device is composed mainly of a magnetic circuit, an active magnetic regenerator, heat exchangers and a hydraulic system. A fundamental feature of such systems is the precision of the hydraulic system, especially regarding the fluid flow management system, given its high influence on the cooling capacity and performance. Additionally, this system has a large role in the overall performance of magnetic refrigerators since it represents one of the largest power consumption shares of the device. An approach previously defined to achieve a more efficient fluid flow management system for the magnetic refrigerators by using electronically controlled valves is applied. An experimental apparatus was applied to provide quantitative data on the operation of the valves, allowing a standardized validation to be made. The performance has been evaluated in terms of mass flow rates, operating frequencies and blow fractions. The experimental results show that solenoid valves were successfully applied as the fluid flow management system of a magnetic refrigeration system. Its performance is highly dependent on the operating conditions of the system, presenting better results as the mass flow rate and blow fraction rises, and as the frequency drops.*

Keywords: *magnetocaloric refrigeration, fluid flow management system, solenoid valve, magnetocaloric air conditioner*

1. INTRODUCTION

The utilization of technologies apart from conventional mechanical vapor compression in refrigeration system has been a challenge for both industry and academia for decades. Caloric technologies arise as one of the most promising alternatives for that, even though several obstacles still prevent the commercialization of systems operating through such principles. Among several alternatives, Magnetic Refrigeration (MR) stands out as the most promising one. This technology relies on the Magnetocaloric Effect (MCE), the thermal response presented by some materials when exposed to a variation of the magnetic field applied over them. Materials that present such property are called Magnetocaloric Materials (MCM).

Given the low magnitude of the MCE (below 5 K), MCM are usually applied in MR as both solid refrigerants and regenerative matrices, characterizing an operation of the so called Active Magnetic Regenerator (AMR). The idealized thermodynamic cycle of the AMR is based on the idealized thermo-magnetic Brayton-cycle (Barclay and Steyert Jr, 1982), which has four main stages: (i) *adiabatic magnetization*: when the magnetic flux density applied on the AMR is adiabatically increased and the MCM temperature rises as a consequence of the MCE; (ii) *cold blow*: when the fluid coming from the cold heat exchanger flows across the AMR, absorbing heat from the solid phase, decreasing the temperature of the MCM, and later on rejecting it at the hot source via the hot heat exchanger; (iii) *adiabatic demagnetization*: when the magnetic flux density applied on the AMR is adiabatically reduced and the porous medium temperature decreases; and (iv) *hot blow*: when the fluid from coming from the hot heat exchanger flows through the AMR, rejects heat in the MCM, and later on absorbs heat from the cold source via the cold heat exchanger. Hence, MR are usually composed of four main subsystems: the AMR, where the MCM are housed and the MCE occurs; the Magnetic Circuit, responsible to provide the magnetic field for the AMR; the Cold Hot Heat Exchangers, responsible to enable the heat exchange between the refrigeration system and the heat reservoirs; and the Fluid Flow Management System (FFMS), responsible to assure the proper management and distributions of the fluid flow through the AMR.

Flow management is a critical and complex issue in MR, given that the AMR requires an oscillatory flow coupled to the magnetization and demagnetization processes, while the remaining hydraulic components, especially the heat exchangers, demand a unidirectional flow. Moreover, a common strategy to increase the cooling capacity of MR is to increase the number of AMRs employed in the system, which further increases the complexity of the flow management task and of the FFMS. Lastly, the maximum frequency achieved by MR is usually limited by the FFMS. As the system operates faster the requirements of valve actuation accuracy, displaced volume of fluid and synchronization with the magnetic profile rises, which are all dictated by the FFMS. For that, the FFMS must guarantee the synchronization between the magnetic and hydraulic profiles according to the demands of the AMR and MR, while consuming the least amount of power.

Several works have been developed regarding the synchronization between the hydraulic and the magnetic profiles for AMR application over the last years (Teyber *et al.*, 2017; Fortkamp *et al.*, 2018; Nakashima *et al.*, 2018b). An illustration of the coupling between the magnetic and hydraulic profiles is displayed in Fig. 1, where the grey dotted curve (trapezoid) shows the magnetic field profile, while the hydraulic fluid flow is presented in the green dashed curve (rectangular). Once these are idealized profiles, the fluid flow profile has instantaneous opening and closing steps and keeps a stratified plateau after opened. Positive values in the fluid flow axis represent the cold blow, while negative ones represent the hot blow.

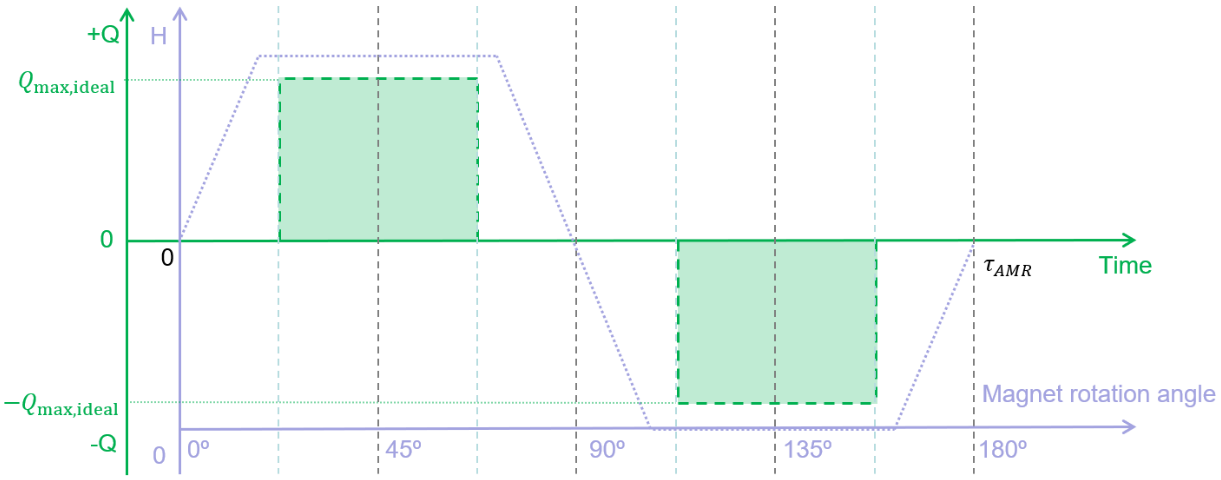


Figure 1: Ideal synchronization between the magnetic and the fluid flow profiles. The grey dotted curve displays the magnetic profile, while the green dashed curve displays the hydraulic profile.

Several strategies and solutions for FFMS for MR applications have been developed over the last years, having each one of them advantages and disadvantages. In general, the solutions are based on:

- *Double effect pumps and check valves* (Trevizoli and Barbosa, 2015): this strategy is indicated for systems with a small number of regenerators, due to the requirement of a piston to operate as a pump for each pair of AMRs. In addition, the fluid displacement is dependent on the piston chamber volume, therefore fixed once the piston is selected. Changes on the fluid displacement rate, profile or blow fraction all demand a replacement of the whole subsystem;
- *Rotary valves with face-to-face sealing* (Lozano *et al.*, 2016; Nakashima *et al.*, 2017): this solution brings a bigger versatility in terms of fluid flow control and displaced fluid flow volume, once the flow period can be controlled by the size of the seal oblong. Nevertheless, it presents issues associated with fluid recirculation and leakages and high energy consumption, which are all contrary to the objectives of the subsystem;
- *Mechanical valves driven by cams* (Eriksen *et al.*, 2016): this configuration presents higher versatility and lower power consumption than the previous ones, but commonly demands a complete development in terms of design, not only concerning the drive mechanism but also for the hydraulic sealing components;
- *Solenoid valves* (Cardoso *et al.*, 2016; Hoffmann *et al.*, 2017; Dutra *et al.*, 2017; Teyber *et al.*, 2017; Nakashima *et al.*, 2018a; Dos Santos *et al.*, 2020): this solution has been demonstrated to be the most promising one for

application in MR (Ebel *et al.*, 2016). It provides the FFMS great flexibility since it allows the adjustment of the fluid flow profile and blow fraction easily and even during the operation of the system. The coupling of the hydraulic and magnetic profiles could be performed via the mechanical coupling of the FFMS and AMR or magnetic circuit, as done in the previous FFMS alternatives. Instead, the profiles coupling is performed by hall sensors or encoders to measure the magnetic flux density or position of the AMR or magnetic circuit, ensuring the synchronization between the magnetic and hydraulic profiles.

Given the available alternatives, solenoid valves present the most favorable features to be employed as the FFMS not only for MR, but also for other caloric technologies. Their commercial availability enables an easy replacement and a large range of validated operating conditions. Hence, the selected option must meet the design requirements of the AMR and MR, in terms of response time, fluid flow profile, and frequency and pressure operation ranges. In addition, not only the requirements regarding fluid flow and pressure drop must be considered, but also requirements such as low noise and affordable cost should be considered during the design evaluation.

In a previous work developed by dos Santos (2020), a FFMS for MR applications was designed. The design and selection was made by a comparison between two different solutions: a set of commercially available solenoid valves and a custom mechanically actuated valves operated by cams, designed and manufactured for the specific application. The results, which are detailed in dos Santos (2020), demonstrated by the individual evaluation of the alternatives, that the solenoid valves presented better results in comparison with the mechanically actuated ones, as previously demonstrated in the literature (Ebel *et al.*, 2016; Dutra *et al.*, 2017). The selected solenoid valve was the Asco model SC8210-112/220V-NC, which presented the best results regarding the desirable characteristics among the evaluated candidates.

The results for the operation of a single Asco SC8210-112/220V-NC solenoid valve have been characterized by Dos Santos *et al.* (2020). The authors evaluated the performance of an arrange of solenoid valves regarding the hydraulic profile, pressure drop, time response and opening, closing and transition periods through the assessment of an individual valve. The results demonstrated that the solenoid valves were able to operate according to the specified conditions, but further evaluations were required to demonstrate the capabilities of the arrange for MR operations. The main issues which required further evaluation concern the simultaneous assessment of a pair of solenoid valves with a coupled operation, according to the FFMS proposed by dos Santos (2020), and the comparison between the hydraulic profiles in the oscillatory and unidirectional components of the system.

Therefore, the main goal of this paper is to experimentally evaluate and validate the design proposed by Dos Santos *et al.* (2020) for a efficient fluid flow management for a large magnetic refrigerator system using electronically-controlled solenoid valves. 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. Hence, the results obtained here can be extended, 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.

2. EXPERIMENTAL APPARATUS

The FFMS designed by dos Santos (2020) was designed to be assembled in Magnetic Air-Conditioner apparatus being developed in Polo - UFSC. The schematic diagram of the system is presented in Fig. 2, where the main components are the heat exchangers, hydraulic power unit, FFMS, AMRs, Magnetic Circuit and measuring instruments.

The path of the fluid through the apparatus goes as following: the fluid stored in the tank (bottom left side of the diagram) is displaced by the pump through the whole system. Given the head difference generated by the pump, the fluid flows from the tank through the pump and the mass flow meter, where its mass flow rate is measured. After that, it goes through the hot heat exchanger and rejects heat in the hot reservoir. Following, the fluid reaches the high pressure valves manifold, where the flow is divided through the regenerator beds (1A and 1B, or 2A and 2B,... until 8A and 8B) by the simultaneous actuation of the high and low pressure solenoid valves pair according to the requirements of the hot blow of the AMR cycle. After crossing the regenerators, the fluid flow from each one of the AMR beds is integrated and flows to the cold heat exchanger, where it absorbs heat from the cold heat reservoir. After that, the fluid flow goes back to the cold side distributor, where it is distributed through the regenerator beds (1A and 1B, or 2A and 2B,... until 8A and 8B) according to the requirements of the cold blow of the AMR cycle. After that, the fluid goes through the low pressure manifold and solenoid valves and, lastly, back to the tank. Besides the fluid flow line, the diagram shows the magnet circuit, responsible for providing the magnet field over the regenerators, and the cold and hot reservoirs, where the heat exchange should occur. It is worth mentioning that the arrange presented in Fig. 2 displays a configuration where each pair of valves controls the hot or cold blow of a pair of regenerator beds (1A and 1B, or 2A and 2B,... until 8A and 8B). The system can also operate in a configurations where each pair of valves controls only one regenerator bed, which entails in two times the number of valves as displayed in Fig. 2.

Given restrictions in the availability of materials and to focus on the experimental evaluation and the understanding of the FFMS, a simplified hydraulic version of the magnetic air-conditioning apparatus was built. This configuration employs only two regenerator beds, operating contrariwise regarding the AMR cycle, and four solenoid valves, two in the

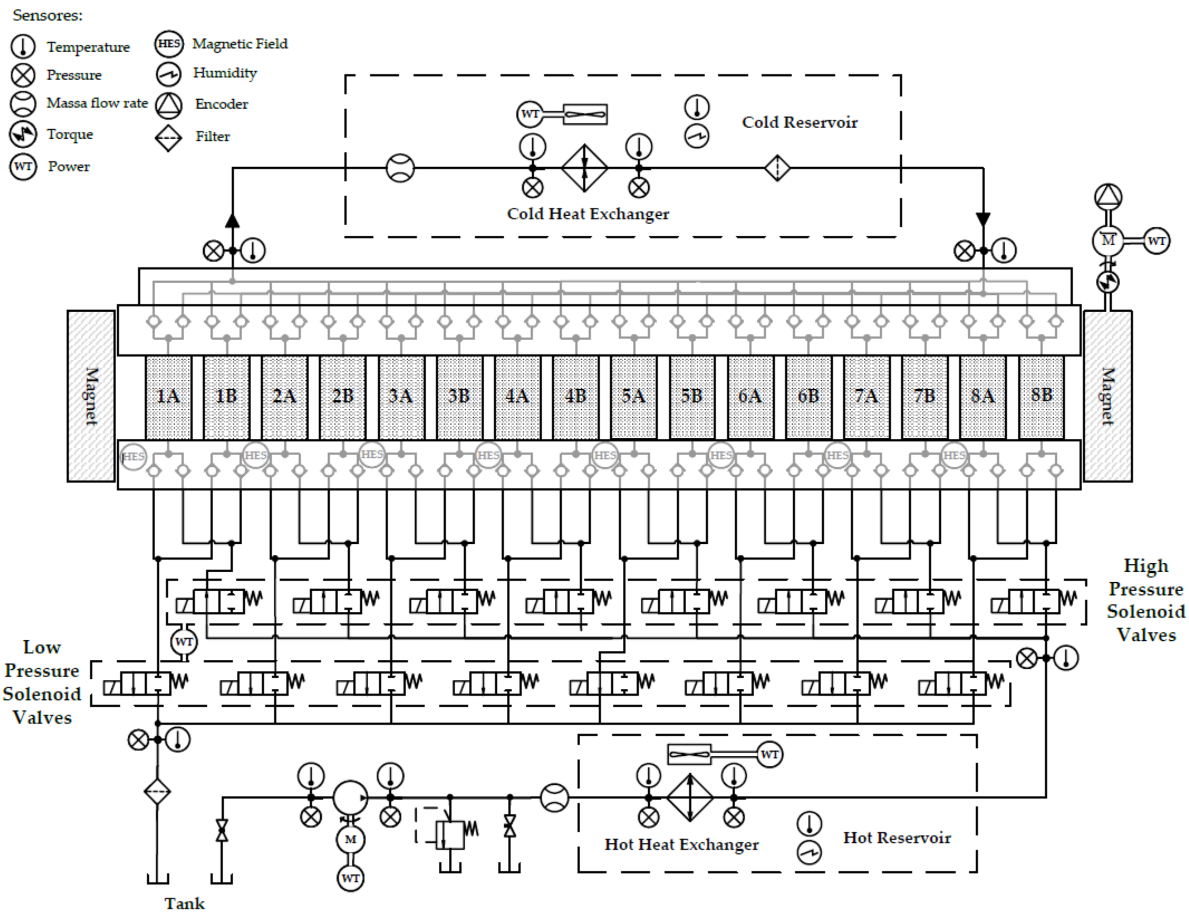


Figure 2: Schematic diagram of the complete FFMS.

high pressure manifold and two in the low pressure one. For this configuration, one valve in the high pressure manifold and one valve in the low pressure manifold performs the hydraulic management of one regenerator bed. The pressure measurements are taken at 8 spots along the fluid line: the inlet and outlet of the pump, the inlet of the high solenoid valve manifold, the outlet of each high pressure solenoid valve, the outlet of the cold side high-pressure manifold, the inlet of the cold side low pressure manifold and the outlet of the hot side low pressure manifold. The schematic diagram of the simplified FFMS is depicted in Fig.3.

The hydraulic performance was evaluated to claim the candidate's behavior during each test condition, focusing on the maintenance of the synchronization between the blows according to the AMR cycle requirements. For that, it was utilized pressure measurements at the outlet of each high pressure solenoid valve to characterize the transient response of the fluid flow and the ones at the inlet of the high solenoid valve manifold, the outlet of the cold side high-pressure manifold, the inlet of the cold side low pressure manifold and the outlet of the hot side low pressure manifold to characterize the continuous response of the system.

Using this metrics to evaluate the performance, the tests were defined to cover a zone around the working point previously determined for the operation of the magnetic refrigeration system. Tab. 1 presents a set of points for AMR Frequency, Fluid flow rate and blow fraction (which is defined as the ration between the period that the valve is actuated and the total cycle period) combined represent the 18 tests. The process to set theses conditions was done directly on the software interface of the MRU, to achieve a better validation of the candidate's performance in a real condition.

Table 1: Parameters for the solenoid valves operation's test.

AMR Frequency [Hz]	Fluid flow rate [L/h]	Blow fraction [%]
0.75	350	25
1	525	40
1.5	700	

The main goal was to verify the ability of the selected solenoid valve to operate with whole apparatus and conditions

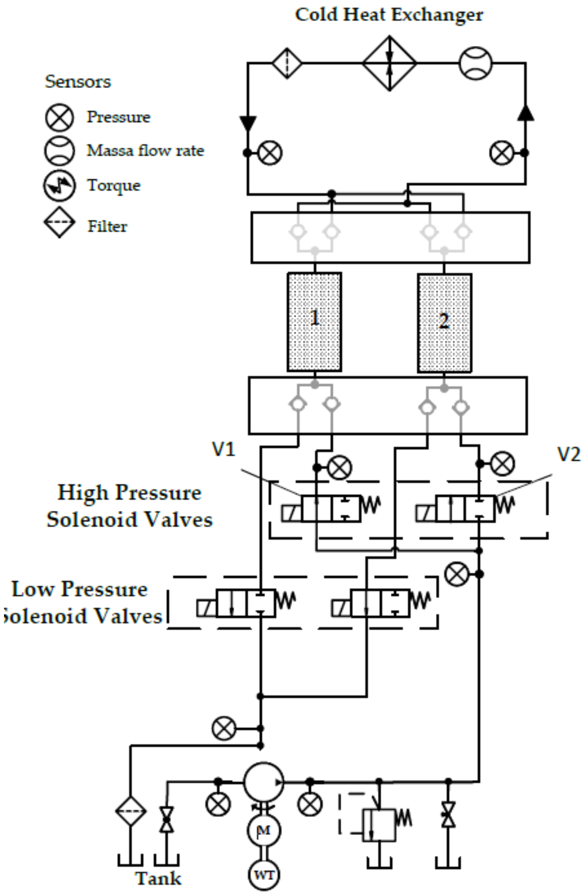


Figure 3: Schematic diagram of the simplified FFMS.

needed to compose the FFMS in the Magnetic air-conditioner. Also, it is interesting to confirm if the selected valve has the shortest time in transition period, aiming to obtain the experimental profile which better approaches the ideal profile as shown in Fig. 1. In addiction to the validation of the performance, the objective to find the best point of synchronization between the AMR cycle and the FFMS, aiming a higher COP for the refrigeration apparatus.

3. RESULTS

The performance of the FFMS is evaluated based on the pressure measurements taken at the inlet of the high solenoid valve manifold, the outlet of each high pressure solenoid valve, the outlet of the cold side high-pressure manifold, the inlet of the cold side low pressure manifold and the outlet of the hot side low pressure manifold. These pressure points are referred on the Figures depicted in this section as $HB\ in$, $V1\ out$, $V2\ out$, $HB\ out$, $CB\ in$ and $CB\ out$, respectively.

The results for 10 cycles are recorded with an acquisition frequency that provides 500 measurement points per cycle. The full data set is divided into 5 subsets, and an average value of the 5 sets of 2 consecutive periods is performed. After that, a filter is applied to remove the signals with frequencies higher than 60 Hz.

The first result to be displayed is regarding a comparison between the electric actuation signal of the solenoid and the transient fluid flow response, which are displayed in Fig. 4.

Figure 4 displays the electric actuation signal and the corresponding pressure at the outlet ports for valves V1 and V2. According to the schematic diagram presented in Fig. 3 and Fig. 4, when valve V1 is open and valve V2 is closed there is a rise in the pressure at the outlet port of the first one and a drop in the same parameter on the last one. Such conditions result in AMR 1 and 2 undergoing the hot and cold blow, respectively. Conversely, when V1 is closed and V2 is open there is a drop in the pressure of the outlet port of the first valve and a rise in the same parameter of the last valve. This condition results in AMR 1 and 2 undergoing the cold and hot blows, respectively.

It is worth highlight the coupling between the actuation signals of the solenoid valves and the resulting pressure profiles. The time response for opening the valves is faster than the one for its closing. As soon as the actuation signal occurs, there is a response in the pressure at the outlet ports of both valves. After the opening, it can be observed a noisy behavior on the pressure signals, which is characteristic of the operation of solenoid valves. Later, there is a plateau in the pressure signal, where it remains virtually constant, and the fluid flow occurs. At some point, according to the blow

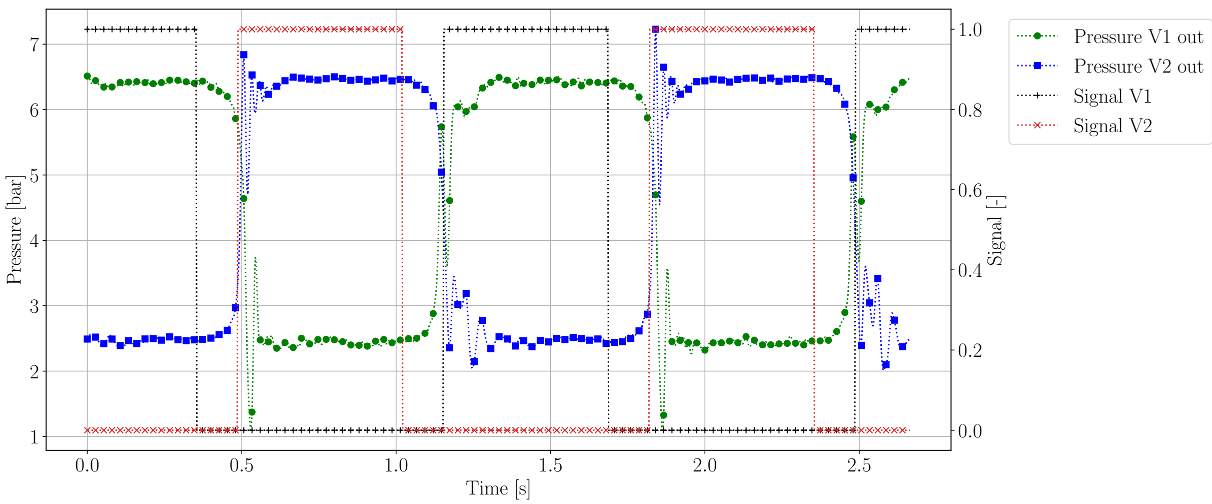


Figure 4: Comparison between the electric actuation signal of the solenoid and the transient fluid flow response for a mass flow rate of 700 kg/h, a frequency of 0.75 and a blow fraction of 40%.

fraction specification, the actuation signal in the solenoid valve is removed, but the flow response for that is rather slow, taking up to 0,2 seconds for the pressure to attain the low plateau values. That slow closing behavior is also characteristic of the operation of solenoid valves, being employed to avoid water hammer effects – which can damage the solenoid valve.

For the operation in caloric systems, the slow closing behavior of the solenoid must be considered, since they result in an increase of the cold and hot blow fractions in comparison with the control blow fraction, *i. e.*, the valve actuation fraction. That deviation can be corrected by evaluating the time response of the valve closing and correcting such period by advancing the removal of the actuation signal in the solenoid valves.

Hence, a preliminary assessment of the pressure signals demonstrates that the proposed configuration of FFMS can perform alternate hot and cold blows in a pair of regenerators operating contrariwise to each other according to the specifications of Fig. 1. For a deeper understanding of the results, evaluation of the hydraulic profiles over several operating conditions were performed.

It is valuable to understand the effect of the mass flow rate, operating frequency and blow fraction on the hydraulic profiles achieved by the proposed FFMS. Hence, each one of these parameters is going to be evaluated separately, among the values presented in Tab. 1. The first parameter to be evaluated is the mass flow rate. Results for a system operating at a frequency of 0.75 Hz and a blow fraction of 40% for 3 distinct mass flow rate values are displayed in Fig. 5.

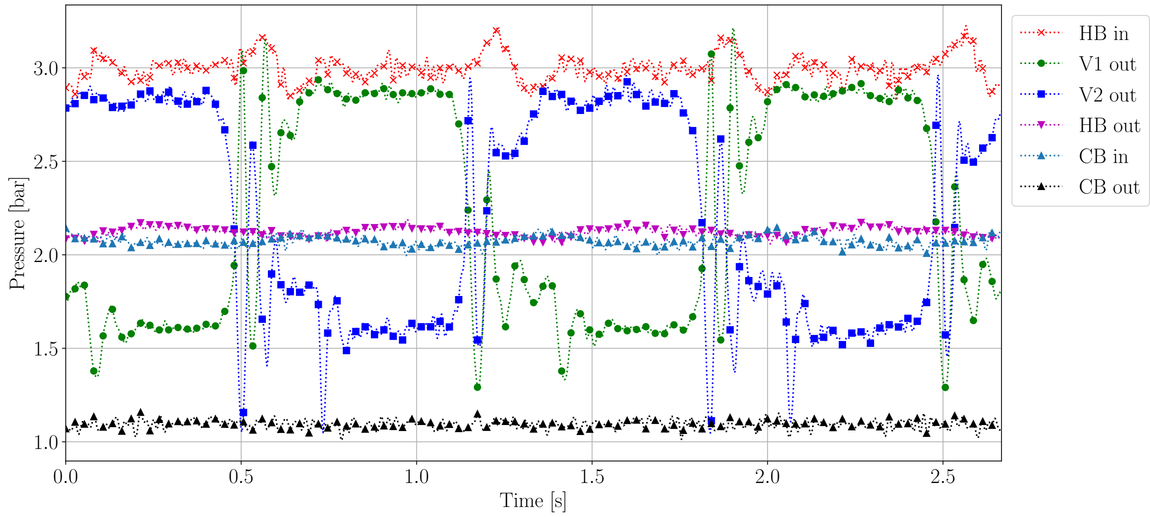
The results displayed in Fig. 5 (a), (b) and (c) demonstrate that the hydraulic profile improves with the rise in the mass flow rate. As one goes from the results to a mass flow rate of 350 kg/h, Fig. 5 (a), to 525 kg/h, Fig. 5 (b), and lastly 700 kg/h, Fig. 5 (c), it is noticeable that most characteristics of the hydraulic profile improves. The noisy behavior for opening and closing the solenoid valves is attenuated in higher mass flow rates. Besides that, for such conditions the attained profiles in the outlet of the solenoid valves during the blow periods are much more stable and restrained within a lower value range, and so is the profile for the inlet of the high pressure hot side manifold, both characteristic of a more stable operation. That can be attributed for the high flow driving force, given by the higher pressure values at the inlet of the hot side high pressure manifold, which overcomes more easily the inertial forces required for the attainment of an oscillatory flow. Also, as the mass flow rate is reduced, the closing period of the valve is reduced, and the noisy behavior of the fluid pressure is enhanced given the decreased period for the fluid flow to decelerate.

Following that, the results for a system operating at a mass flow rate of 525 kg/h and a blow fraction of 25% for 3 distinct frequency values are displayed in Fig. 6.

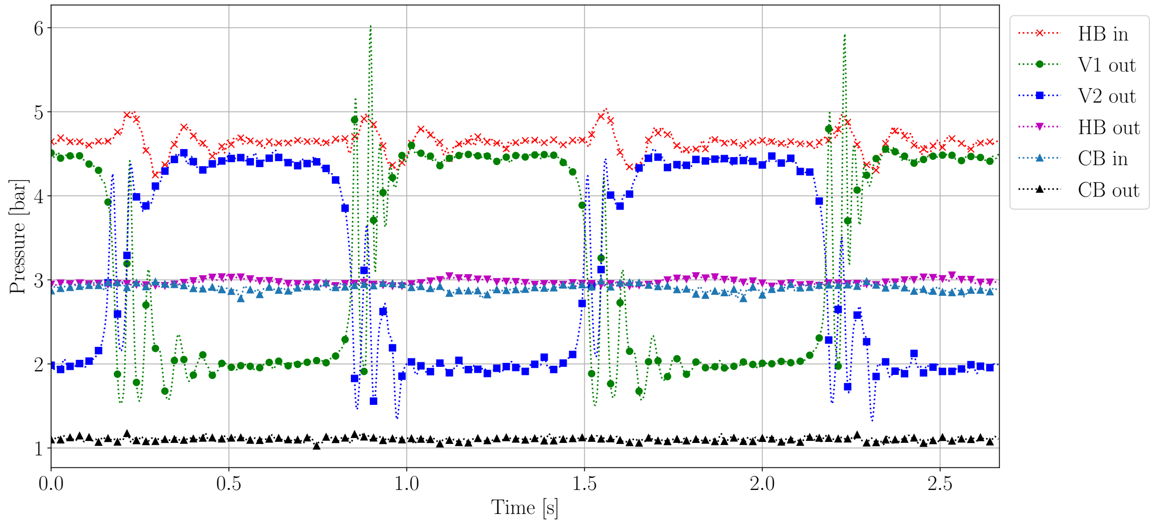
The results displayed in Fig. 6 (a), (b) and (c) demonstrate that the hydraulic profile improves with the drop in the frequency of the oscillatory flow. As one evaluates the rise of frequency following the results of Fig. 6 (a), (b) and (c), it can be noticed that the noisy behavior of the opening and closing periods is enhanced and that the hydraulic profile at the outlet of the solenoid valves and at the inlet of the hot side high pressure manifold is progressively deteriorated. These behaviors are result of the faster requirements for the valve and fluid flow as the frequency increases. Hence, at lower frequencies the valve has more time to open and close and the fluid flow to develop and achieve the plateaus of constant pressure, which generate more stable hydraulic profiles.

Lastly, the results for a system operating at a mass flow rate of 525 kg/h and a frequency of 0.75 Hz for 2 distinct blow fraction values are displayed in Fig. 6.

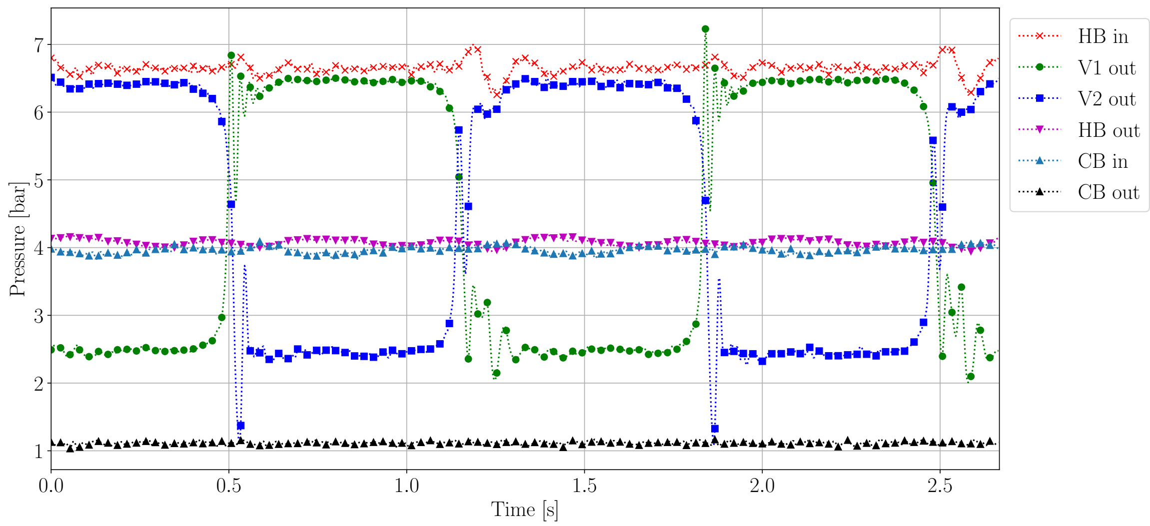
The results displayed in Fig. 7 (a) and (b) demonstrate that the hydraulic profile improves with the rise in the frequency of the oscillatory flow. This result is due to the reduced period where no flow occurs in the regenerator beds associated



(a)



(b)



(c)

Figure 5: Evaluation of the effect of the mass flow rate for a fixed frequency of 0.75 Hz and blow fraction of 40%. The figures depict a mass flow rate of (a) 350 kg/h, (b) 525 kg/h and (c) 700 kg/h.

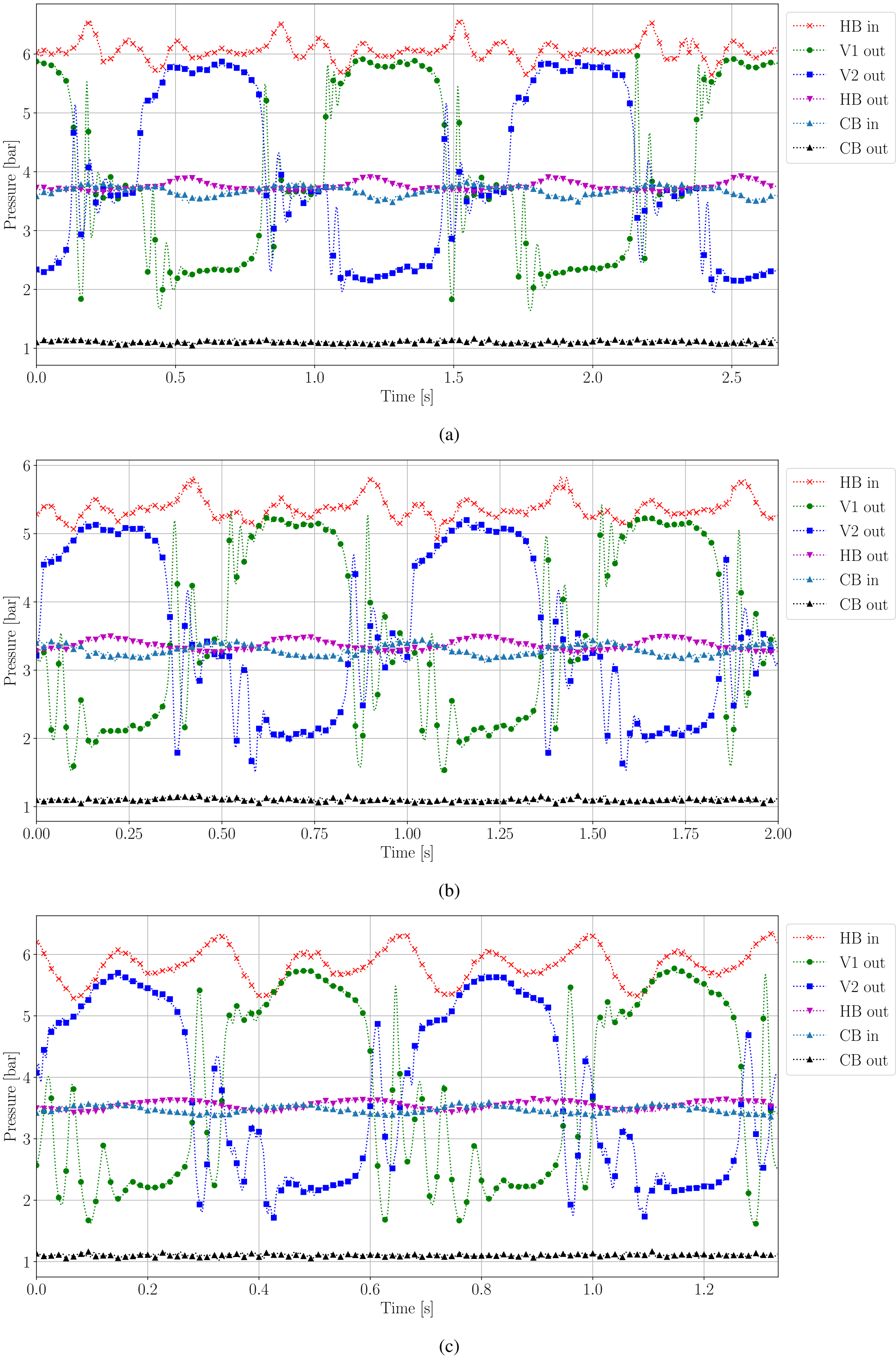
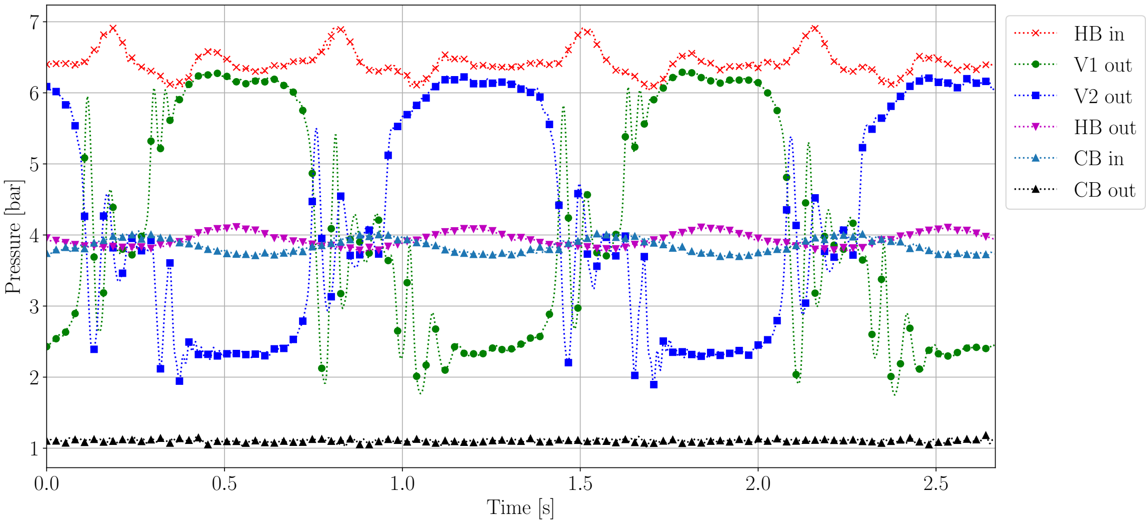
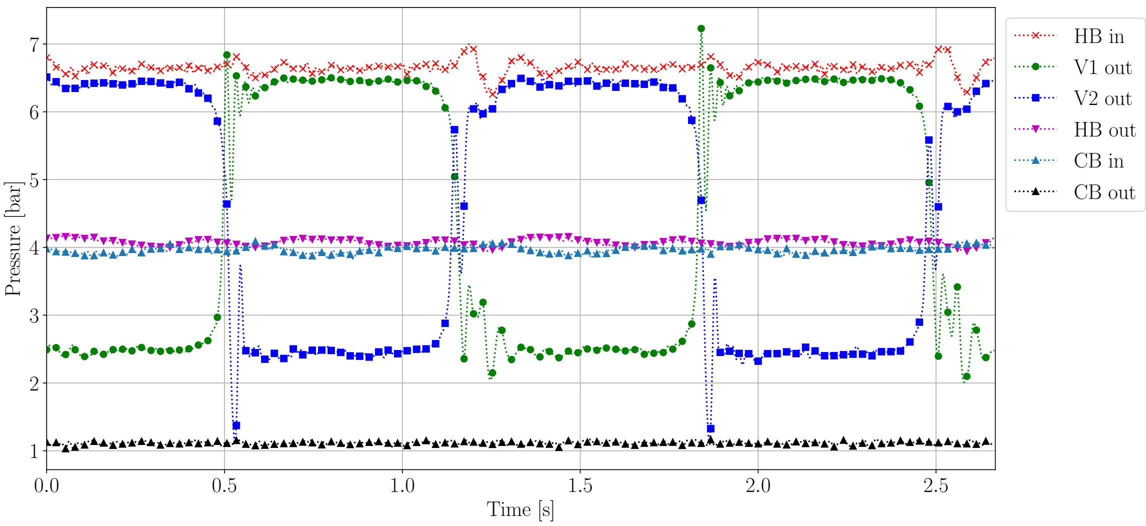


Figure 6: Evaluation of the effect of the frequency for a fixed mass flow rate of 525 kg/h and blow fraction of 25%. The figures depict a frequency of (a) 0.75 Hz, (b) 1 Hz and (c) 1.5 Hz.



(a)



(b)

Figure 7: Evaluation of the effect of the blow fraction for a fixed frequency of 0.75 Hz and mass flow rate of 700 kg/h. The figures depict a mass flow rate of (a) 25% and (b) 40%.

with the higher blow fraction cases once the presence of the fluid flow aids in the attenuation of the pressure variations. This can be especially noticed during the closing periods of the valves.

4. CONCLUSIONS

An experimental apparatus to evaluate the FFMS of a magnetic refrigeration system was design and assembled in this work. Through this apparatus the operation of a FFMS composed of an arrange of solenoid and check valves designed to operate caloric systems was evaluated. The actuation profiles of the solenoid valves were measured by the electric input signals and the hydraulic profiles by measurement of the pressure at determined spots along the liquid line. The experiments demonstrated that the proposed FFMS can perform alternate hot and cold blows in a pair of regenerators operating contrariwise to each other. In general, the performance of the FFMS in terms of the attained hydraulic profile improves as the mass flow rate and blow fraction increases and as the operating frequency decreases.

5. ACKNOWLEDGEMENTS

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