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USING ELECTROVALVES AS A FLOW DISTRIBUTION SYSTEM FOR AN ACTIVE MAGNETIC REGENERATOR

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Abstract. *State-of-the-art magnetic refrigerators for room-temperature applications employ magnetic and hydraulic circuits to perform the thermomagnetic Brayton cycle on an active magnetic regenerator (AMR). The cooling capacity and temperature span developed by AMRs are largely dependent on the hydraulic circuit parameters such as mass flow rate, blow time fraction and magnetic field-fluid flow synchronization. Moreover, the AMR efficiency is also impacted by the pump, motor, valves and control system power consumption. Therefore, the selection of hydraulic components, i.e. pump and valves, plays a vital role in the overall system performance. Most of the AMR devices developed so far employ mechanical concepts in either the pump and the flow distribution system (valve control), which may be a face-to-face sealing (known as rotary valves) or cam-actuated valves. However, recent studies have shown that friction losses, flow imbalance and operational parameters constraints are major issues of such solutions, which require further development. Electronic controlled valves might be a suitable solution for such concerns, as they can greatly increase the flexibility of synchronization parameters and enable comprehensive optimization of flow parameters. In this sense, this paper aims to evaluate an electric solenoid valve design for AMRs applications. A set of four electric valves was installed in an AMR apparatus, and their opening and closing times was synchronized with respect to the magnetic profile via magnetic field measurements of a Hall effect sensor. Thermodynamic performance was assessed in terms of COP for a range of hydraulic circuit parameters and comparison against literature results determined their suitability for the present apparatus. Furthermore, electronic control of blow time fraction enabled real-time correction of blow mass imbalances.*

Keywords: *Magnetic refrigeration, active magnetic regenerator, electronic solenoid valves.*

1. INTRODUCTION

Magnetic refrigerators for room-temperature cooling apply magnetocaloric materials (MCMs) as solid refrigerants in an active magnetic regenerator (AMR), whose operation is based on the ideal thermo-magnetic Brayton cycle (Barclay and Steyert Jr., 1982). The main steps of the cycle are: two adiabatic magnetic field variation processes (magnetization and demagnetization) when the temperature of the MCM changes (rises and decreases, respectively) due to the magnetocaloric effect (MCE), and two iso-field fluid flow steps (cold-to-hot and hot-to-cold blows, respectively) when fluid interacts with the solid refrigerant and then exchanges heat with the thermal reservoirs. By sequentially performing magnetic field variations and fluid flow steps, the AMR is able to transfer a cooling capacity ($\dot{Q}_{\mathrm{C}}$) from the cold reservoir and reject heat ($\dot{Q}_{\mathrm{H}}$) at the hot reservoir over a temperature span (ΔT_{span}), with the expense of mechanical, magnetic, electronic and hydraulic power inputs.

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. In AMR applications where the regenerator is an stationary bed, as in the apparatus of this work, it is necessary to control the flow direction via proper sealing of the inactive fluid lines to ensure that the magnetic-field and fluid blow synchronization is in accordance with the AMR cycle (Nakashima, 2017). Furthermore, in the application of a multiple stationary beds AMR, the hydraulic circuit complexity increases. Several mechanically controlled solutions for blow management have been developed (Lozano, 2015; Eriksen *et al.*, 2015) and their shortcomings usually includes high power consumptions and/or low versatility. In this sense, the present work aims to evaluate the use of an electronic solenoid valve system as a flow distribution control of an AMR test apparatus developed by Trevizoli *et al.* (2016), whose hydraulic circuit at first was composed by piston-pump and check-valves, and then was changed for a gear pump and rotary valves combination (Nakashima *et al.*, 2017). Comparison between present and literature results of cooling capacity and overall COP was performed in order to determine the suitability of the novel hydraulic solution for AMR applications.

2. LITERATURE REVIEW

The hydraulic circuit has an important role in AMR systems, as its performance is greatly affected by flow parameters, which are dependent on the hydraulic components and characteristics. Thus, it is important to describe different hydraulic schemes that have been reported in literature so far, as well as their pros and cons. To this date, four types of flow distribution systems have been proposed: (i) face-to-face sealing (rotary valves) (Engelbrecht *et al.*, 2012; Lozano, 2015; Nakashima *et al.*, 2017), (ii) double effect pump and check valves (Tura and Rowe, 2011; Trevizoli, 2015), (iii) double effect pump and check valves together with directional valves powered by camshaft (Teyber *et al.*, 2017), and (iv) poppet valves driven by cams (Eriksen *et al.*, 2015).

The aforementioned solutions have advantages and disadvantages for applications in AMRs. For example, the double effect pump provides a fixed mass of fluid for each blow and guarantees blow mass balance, however, a robust system is required to withstand the pressure peaks of the sinusoidal fluid flow waveform. On the other hand, poppet valves powered by cams can potentially reduce power consumption, but they are less versatile in terms of flow period adjustment. Face-to-face sealing valves are less restrictive and the resulting trapezoidal fluid flow waveform presents lower pressure peaks, however they require high levels of drive power to rotate. Moreover, such mechanical control of valves are subjected to mechanical torque oscillations generated at the magnetic circuit and transmitted by the single transmission system, that may lead to flow imbalance, which can harm the cooling capacity of the apparatus (Teyber *et al.*, 2017; Nakashima *et al.*, 2017).

The power consumption of the flow distribution system has an important effect in the total apparatus power consumption, mainly when such system is composed by rotary valves. The study of the power losses in magnetic refrigeration systems was previously carried out by Lozano *et al.* (2013) and Capovilla *et al.* (2016). They evaluated the power consumption of each subsystem and concluded that the higher losses are due to the friction losses in the rotary valves. For all operating frequencies, the rotary valve system is responsible for approximately 63% of the power consumed by the electrical motor. Similar results were achieved by Nakashima (2017) in the previous version of the AMR apparatus updated in this work. Moreover for a ΔT_{span} of 10 K and 50% of the blow time fraction the rotary valve consumption was around 4 and 8 W for 0.25 and 0.5 Hz, for all the utilization factors.

Experimental and numerical evaluation of electronic valves for AMRs was proposed by Cardoso *et al.* (2016). This work demonstrated potentials for reduction in the energy consumption and increment in versatility, as well as valve actuation decoupling from the mechanical transmission. Fig. 1 shows a comparison of the measured power consumed by the flow distribution system of the Polo/UFSC prototype developed by Lozano (2015) operating with rotary valves and the expected consumption of a set of electrovalves that could be applied. As can be inferred from Fig. 1, at higher operating frequencies the electronic valves consume less power. Further improvements in valve consumption can lead to efficient operation at lower frequencies. To this date, there is no other AMR apparatus in literature using eletrovalves as a flow distribution system and this work is proposed to be pioneer in doing so.

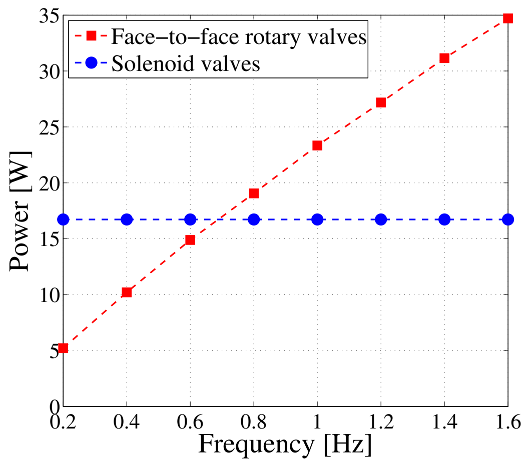


Figure 1. Expected results of the power consumption of solenoid valves in comparison to a hydraulic system operating with face-to-face rotatory valves at different frequencies (Cardoso *et al.*, 2016).

3. EXPERIMENTAL APPARATUS

An schematic diagram of the adapted experimental apparatus originally developed by Trevizoli (2015) with the inclusion of a set of electrovalves is shown in Fig. 2. This device consists of magnetic circuit composed of a Halbach concentric cylinders (HCC), which are driven by a step motor (kalatec Nema 34, model KML-HT34-487). The rotation of the Halbach cylinders promotes variation of the magnetic field intensity, from 0.04 T to 1.69 T, at the inner bore where the AMR is placed. The latter is a porous matrix packed with 195.82 g of commercial grade Gd spheres (0.55 mm in diameter). The regenerator has an internal diameter of 22.2 mm and 100 mm in length. The hydraulic system is composed of a gear pump (Micropump GC-M23) powered by an electric motor (WEG GC M25 PV56), which enables operation at several mass flow rates. Four unidirectional solenoid valves (Burkert 6211) are responsible for delivering the alternative blows to the AMR and returning it to the reservoir. More details regarding the solenoid valves can be found in Cardoso *et al.* (2017). A proportional relief valve (Swagelok SS-RL3M4-F4) diverts the working fluid (water/ethylene-glycol mixture) from the AMR and enables the operation with no-flow periods. Check valves guarantee unidirectional flows in the main and by-pass lines. The operating frequency of the gear pump is kept approximately constant by a frequency inverter (WEG CFW 08), and a needle valve (Swagelok SS-31RS4-G) is employed to control the mass flow rate through the system.

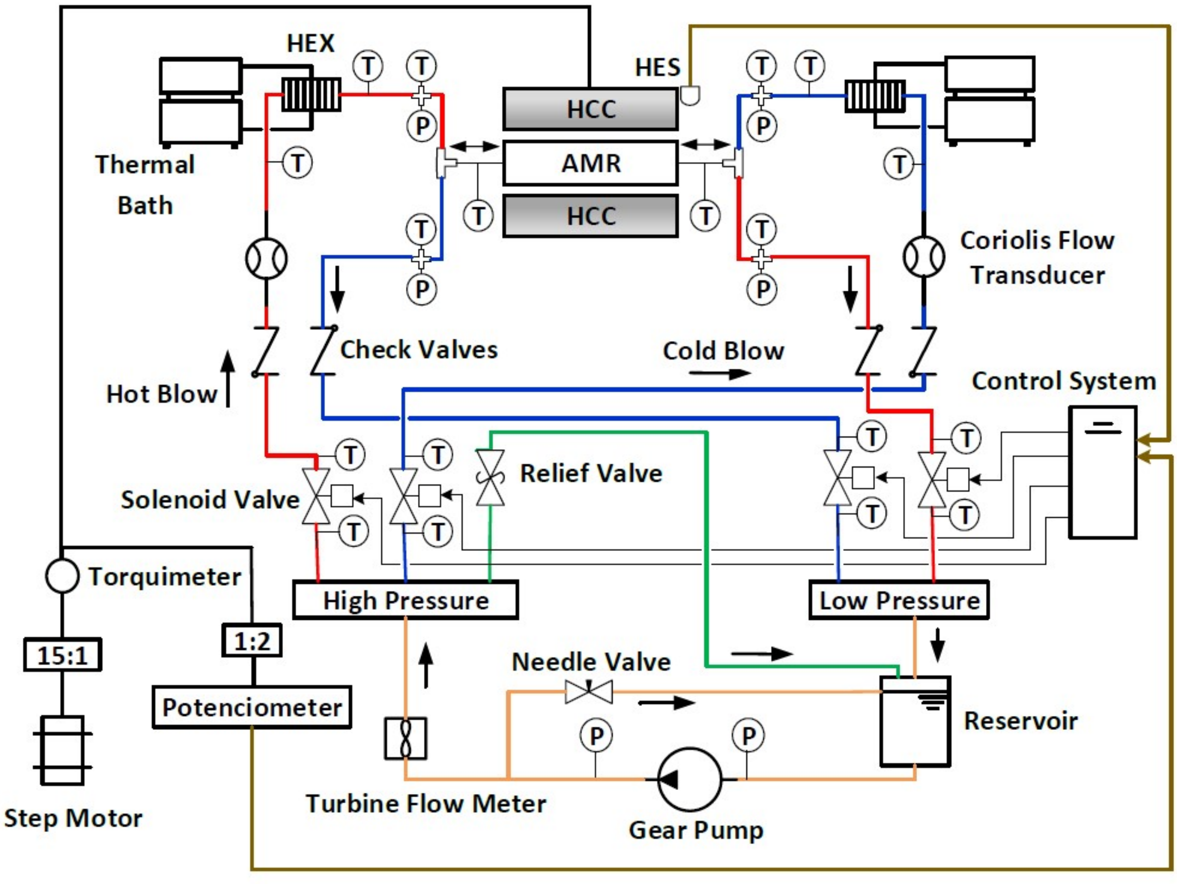


Figure 2. Schematic diagram of the AMR apparatus originally developed by (Trevizoli, 2015) and adapted in this work to operate with a set of four electric valves as the flow distribution system (arrows indicate the flow directions during each blow).

The hydraulic and the magnetic systems are synchronized by a Hall effect sensor (HES) (Alegro A1302) which indicates voltage corresponding to the instantaneous magnetic field waveform applied to the AMR. This data is employed as an input for the control system, which manages the valve opening and closing times and guarantees the desired fluid flow and magnetic field synchronization. More details regarding the control system of the electric valves can be found in Hoffmann *et al.* (2017).

Thermal baths (Thermo Scientific Haake G50 and A25) are used to emulate the cold and hot sources, respectively, and establish the source temperature span ($\Delta T_{\rm span}$). They keep thermal contact with the heat transfer fluid by means of parallel plates heat exchangers (HEx). The instrumentation of the apparatus includes pressure transducers (Omega PX613, uncertainty of 0.5 kPa), which allow pressure drop and pumping work evaluation; a linear potentiometer (Omega LP802-50), which enables cycle frequency calculation; and a torque transducer (HBM T22/50Nm, uncertainty of 0.5%)

for mechanical power assessment. Calibrated thermocouples (Omega TMQSS-020G, uncertainty of 0.15 K) are used to evaluate temperature differences and, together with the flow transducer, the cooling capacity ($\dot{Q}_{\mathrm{C}}$). The mass flow rate is measured by Coriolis flow transducers (Krone Optimass 3300C, uncertainty of 1%) and two turbine flow transducers in parallel (Sponsler MF100 and MF20, uncertainty of 0.5 and 0.1 L/h, respectively) which are selected in accordance with the mass flow rate range. The results obtained in this work were evaluated in terms of the COP and $\dot{Q}_{\mathrm{C}}$ and compared with the results achieved by the previous versions of this apparatus (Trevizoli, 2015; Nakashima, 2017).

4. EXPERIMENTAL METHOD

The experimental tests were carried out in different cycle frequencies (f), utilization factors (ϕ) and blow time fractions (F_{B}). The latter is defined as the ratio of blow and cycle periods, while the utilization factor is defined as the ratio of the thermal capacities of the fluid crossing the regenerator during each specific blow and the thermal capacity of the solid matrix:

$$\phi = \frac{\dot{m}_{\mathrm{B}} c_{\mathrm{p,f}}}{m_{\mathrm{Gd}} c_{\mathrm{Gd}}} \frac{F_{\mathrm{B}}}{f} \tag{1}$$

where the $\dot{m}_{\mathrm{B}}$ and $c_{\mathrm{p,f}}$ are the average mass flow rate in a blow and the fluid specific heat capacity, respectively. Furthermore m_{Gd} is the solid mass and c_{Gd} is the solid specific heat capacity. The specific heat capacity of the solid and fluid phases are evaluated at 295 K, which is the average temperature of the hot and cold sources. The average expanded uncertainties of the frequency, temperature span and utilization factor have been calculated as 0.0002 Hz, 0.3 K and 0.006, respectively (Nakashima *et al.*, 2017).

The temperature span (ΔT_{span}) and the ambient temperature (T_{amb}) were maintained constant during the experimental tests. Table 1 shows the range of parameters previously mentioned. The influence of the ambient temperature in AMR performance was studied by Trevizoli *et al.* (2016) and the more realistic condition is the equivalence of the ambient and hot reservoir temperatures. Here, both were kept at 300 K.

Table 1. Experimental test parameters.

Variable	Value or range	Unit
T_{amb}	300	K
T_{HHEX}	300	K
T_{CHEX}	290	K
f	0.25, 0.5	Hz
ϕ	0.3, 0.5, 0.7, 1	-
F_{B}	50, 35	%

Test routines in the experimental apparatus begin with the set up of the experimental parameters, shown in Table 1. In this work, a single temperature span is evaluated, and this is the first adjusted parameter. Afterwards, the HCC frequency is set by the step motor and the desired mass flow rate is imposed by frequency inverter and needle valve. Later, the electronic control system is activated and the blow time fraction is set. The HES detects the magnetic field position and controls the simultaneous opening of one pair of solenoid valves and closing of the other one, allowing the hot or the cold blow to circulate across the regenerator bed. After the apparatus reaches a periodic steady state condition, the cooling capacity ($\dot{Q}_{\mathrm{C}}$) and the heat rejected ($\dot{Q}_{\mathrm{H}}$) are calculated, respectively, as follows (Trevizoli *et al.*, 2016):

$$\dot{Q}_{\mathrm{C}} = \frac{1}{\tau} \int_{\tau_{\mathrm{HB}}} \dot{m}_{\mathrm{HB}}(t) c_{\mathrm{p,f}} (T_{\mathrm{C}}(t) - T_{\mathrm{CHEX}}) dt \simeq \frac{\tau_{\mathrm{HB}} \overline{\dot{m}}_{\mathrm{HB}} c_{\mathrm{p,f}}}{\tau} \overline{\Delta T}_{\mathrm{C}} \tag{2}$$

$$\dot{Q}_{\mathrm{H}} = \frac{1}{\tau} \int_{\tau_{\mathrm{CB}}} \dot{m}_{\mathrm{CB}}(t) c_{\mathrm{p,f}} (T_{\mathrm{H}}(t) - T_{\mathrm{HHEX}}) dt \simeq \frac{\tau_{\mathrm{CB}} \overline{\dot{m}}_{\mathrm{CB}} c_{\mathrm{p,f}}}{\tau} \overline{\Delta T}_{\mathrm{H}} \tag{3}$$

where τ is the AMR cycle period, and τ_{HB} and τ_{CB} are the hot and cold blow periods, respectively. T_{C} is the temperature of fluid exiting the regenerator bed at the cold end during the hot blow and T_{H} is the temperature of the fluid exiting the regenerator bed at the hot end during the cold blow.

The definition on the right side of Eqs. (2) and (3) is obtained by considering an average mass flow rate on the blow ($\overline{\dot{m}}_{\mathrm{B}}$) and a constant specific heat capacity of the fluid ($c_{\mathrm{p,f}}$). The temperature difference used on the right is defined by the difference of the temperature of the fluid exiting the porous media during the hot and the cold blows, and the average values of T_{HHEX} and T_{CHEX}, respectively, as:

$$\overline{\Delta T}_{\mathrm{C}} = \overline{T}_{\mathrm{C}} - \overline{T}_{\mathrm{CHEX}} \tag{4}$$

$$\overline{\Delta T}_{\rm H} = \overline{T}_{\rm H} - \overline{T}_{\rm HHEX} \tag{5}$$

where $\overline{T}_{\rm HHEX}$ and $\overline{T}_{\rm CHEX}$ are the average temperatures of the fluid entering the regenerator at the hot and cold ends, respectively.

Blow mass imbalances were eliminated by small blow period adjustments, as shown in Table 2. The blow time variation was set by the control system, which allowed the operator to set the better balance condition as defined by recent studies (Nakashima *et al.*, 2017), where mass imbalance was corrected via mass flow rate adjustment. However, in the present study it is easier to adjust this parameter by changing the time duration of the blows as the control system developed by Hoffmann *et al.* (2017) allows for a refined variation of the opening and closing times of the pairs of solenoid valves.

Table 2. Experimental adjusts of time blow correction.

$F_{\rm B}$ (%)	Value (CB & HB)	Unit
50	-15 & +15	ms
35	-10 & +10	ms

The cycle average total power ($\overline{\dot{W}}$) can be broken down in three components: the power consumption to actuate the solenoid valves ($\overline{\dot{W}}_{\rm V}$), the fluid pumping power ($\overline{\dot{W}}_{\rm P}$) and the power required to rotate the HCC magnetic circuit by the step motor ($\overline{\dot{W}}_{\rm M}$), as shown below:

$$\overline{\dot{W}} = \overline{\dot{W}}_{\rm V} + \overline{\dot{W}}_{\rm P} + \overline{\dot{W}}_{\rm M} = F_{\rm B} n_{\rm valve} \overline{\dot{W}}_{\rm valve} + \dot{V}\overline{\Delta P_{\rm p}} + 2\pi f_{\rm M}\overline{\Gamma} \tag{6}$$

where $\overline{\dot{W}}_{\rm valve}$ is the cycle average power consumption of one solenoid valve, which is calculated by the product of the average voltage and current applied per AMR cycle, and $n_{\rm valve}$ is the number of electric valves. The pumping power is defined as de product of $\dot{V}$ which is the system volumetric flow rate and $\overline{\Delta P_{\rm p}}$ which is the time average pressure difference between the inlet and the outlet of the gear pump. And the motor power is calculated by $f_{\rm M}$ and $\overline{\Gamma}$ which are the rotary frequency of the step motor and the cycle average total torque, respectively.

The coefficient of performance, COP, is defined by the ratio of the average cooling capacity and the average total power in the AMR cycle, as:

$$COP = \frac{\dot{Q}_{\rm C}}{\overline{\dot{W}}} \tag{7}$$

COP is an important parameter in refrigeration systems analysis and it is desired to be as large as possible. Increments on the COP result in an increase on the second-law efficiency ($\eta_{\rm 2nd}$), thus, the refrigeration system would be operating closer to the Carnot cycle. Second-law efficiency and the ideal, or Carnot, coefficient of performance ($COP_{\rm id}$) are defined as follows:

$$\eta_{\rm 2nd} = \frac{COP}{COP_{\rm id}} \tag{8}$$

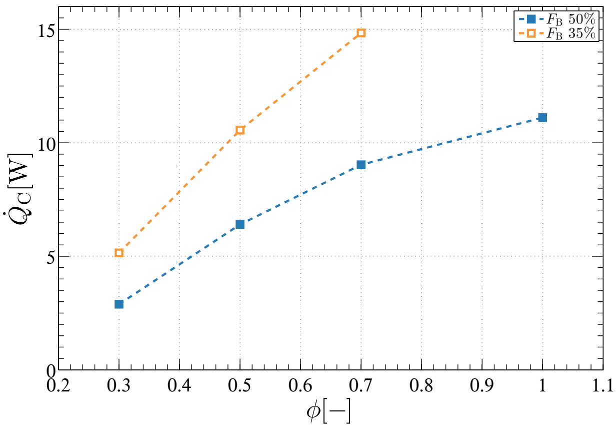
$$COP_{\rm id} = \frac{T_{\rm C}}{T_{\rm H} - T_{\rm C}} \tag{9}$$

5. RESULTS AND DISCUSSION

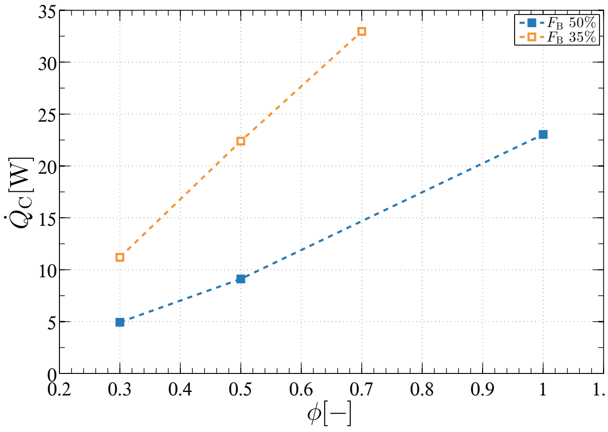
The performance analysis of the AMR apparatus carried out in this work was based on the behavior of $\dot{Q}_{\rm C}$ and COP as a function of different values of ϕ and $F_{\rm B}$ for a $\Delta T_{\rm span} = 10$ K.

Figures 3(a) and (b) show the cooling capacity as a function of the utilization factor for two different blow time fractions of 35 and 50% for operating frequencies of 0.25 and 0.5 Hz, respectively. A satisfactory agreement was found between the results obtained in this work and those found in literature and in the previous versions of this AMR apparatus for different frequencies and blow time fractions (Trevizoli, 2015; Nakashima, 2017). Moreover, a better system behavior at steady state conditions was achieved in comparison to previous versions of the apparatus (Nakashima, 2017), as a result of versatility of the electronic control system of solenoid valves.

The power consumption of solenoid valves depends on the average voltage and current in time required to open and keep the orifice accessible during the blow. Therefore, the lower voltage and current needed to activate and control the electronic valves, the lower will be the energy consumed by the flow distribution system. The solenoid valves used in this work do not have an excellent performance, they have a comparable power consumption with the rotary valves (see Tabs. 3 and 4). However, the electric valves have the advantage that the power consumption remains approximately constant with



(a)



(b)

Figure 3. Cooling capacity as a function of the utilization factor for two different blow time fractions, ΔT_{span} = 10 K and operating frequencies of (a) f = 0.25 Hz and (b) f = 0.5 Hz.

the increment of the AMR frequency, as suggested by Cardoso *et al.* (2016) (see Fig. 1). Moreover, the versatility and practicality are great features of such flow distribution system, as it is possible to control the set of valves in real time, changing the opening and closing times as well as managing the mass imbalance easier. More details about the control system can be found in Hoffmann *et al.* (2017).

Tables 3 and 4 show the power consumption breakdown as specified in Eq. (6) as a function of the utilization factor and blow time fraction and for operating frequencies of 0.25 and 0.5 Hz, respectively. It is possible to verify that the flow distribution system, still with the use of electric valves, is the largest contributor for the total power consumption. It can be seen that this power decreases with the blow time fraction. In all the experiments carried out in this study, the pumping power and the motor power consumption is less than 54% (being maximum at 0.5 Hz, F_{B} = 35% and ϕ = 0.7) of the total power required in the apparatus. Moreover, it is observable that the pumping power increases with the increment of the utilization factor, since this is proportional to the mass flow rate and the system drop pressure. The reduction of the blow fraction, for a fixed utilization factor, also increases the mass flow rate. The motor power consumption is more dependent on the AMR operating frequency and the utilization factor, but it does not have a dependence on the blow time fraction as the rotary valve has.

Table 3. Power consumption of system components as a function of ϕ and F_{B} for $f = 0.25$ Hz.

	F_{B} = 50%				F_{B} = 35%			
ϕ [-]	$\overline{\dot{W}}_{\mathrm{M}}$ [W]	$\overline{\dot{W}}_{\mathrm{P}}$ [W]	$\overline{\dot{W}}_{\mathrm{V}}$ [W]	$\overline{\dot{W}}$ [W]	$\overline{\dot{W}}_{\mathrm{M}}$ [W]	$\overline{\dot{W}}_{\mathrm{P}}$ [W]	$\overline{\dot{W}}_{\mathrm{V}}$ [W]	$\overline{\dot{W}}$ [W]
0.3	1.01	0.11	8.83	9.95	1.05	0.04	5.76	6.85
0.5	1.11	0.24	8.83	10.18	1.13	0.47	5.90	7.50
0.7	1.15	0.40	8.86	10.41	1.22	0.89	5.80	7.91
1.0	1.21	0.74	8.84	10.79	-	-	-	-

Table 4. Power consumption of system components as a function of ϕ and F_{B} for $f = 0.5$ Hz.

	F_{B} = 50%				F_{B} = 35%			
ϕ [-]	$\overline{\dot{W}}_{\mathrm{M}}$ [W]	$\overline{\dot{W}}_{\mathrm{P}}$ [W]	$\overline{\dot{W}}_{\mathrm{V}}$ [W]	$\overline{\dot{W}}$ [W]	$\overline{\dot{W}}_{\mathrm{M}}$ [W]	$\overline{\dot{W}}_{\mathrm{P}}$ [W]	$\overline{\dot{W}}_{\mathrm{V}}$ [W]	$\overline{\dot{W}}$ [W]
0.3	1.95	0.31	8.62	10.88	1.98	0.74	5.48	8.20
0.5	2.04	0.77	8.63	11.44	2.19	2.05	5.56	9.80
0.7	-	-	-	-	2.36	4.12	5.52	12.00
1.0	2.35	2.99	8.64	13.98	-	-	-	-

Figures 4 and 5 show the COP as a function of the utilization factor and for two different blow time fractions (35 and 50%) for operating frequencies of 0.25 and 0.5 Hz, respectively. These curves shows the influence of the solenoid valves in the COP behavior. Three different scenarios were assumed in this analysis: (i) considering ideal solenoid valves ($\overline{\dot{W}}_{\mathrm{V}} = 0$), so only pumping and HCC power consumption were accounted, (ii) considering the actual solenoid valves ($\dot{W}_{\mathrm{v}} \sim 4$ W and $\overline{\dot{W}}_{\mathrm{V}}$ as in Tables 3 and 4), and (iii) considering the use of solenoid valves with lower energy consumption

($\overline{\dot{W}}_{\mathrm{v}} \sim 1.5$ W), since recent studies carried out at POLO/UFSC indicated the possibility to substitute the actual solenoid valves for an alternative commercial set of solenoid valves (refered as BSV) with such nominal power consumption. The goal of this analysis is to demonstrate the impact that the solenoid valves have in the overall performance of the AMR apparatus.

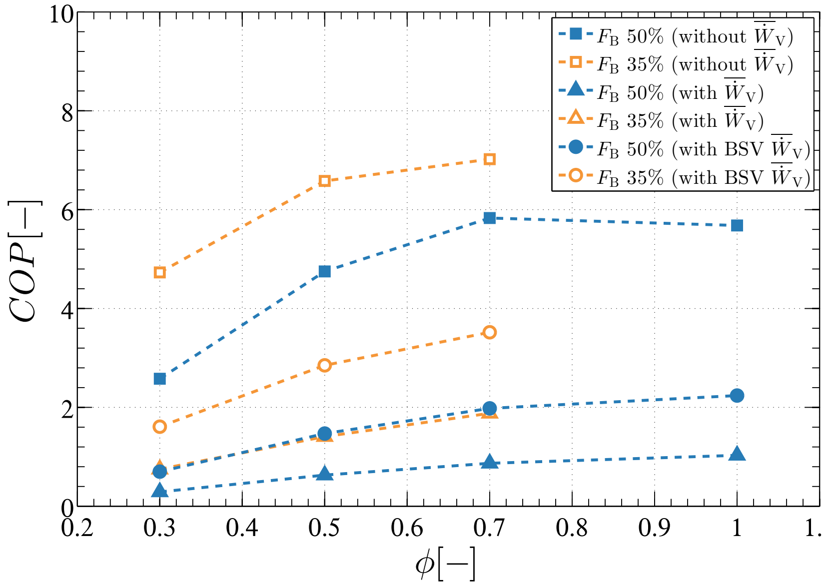


Figure 4. COP as a function of the utilization factor, for f = 0.25 Hz and ΔT_{Hex} = 10 K.

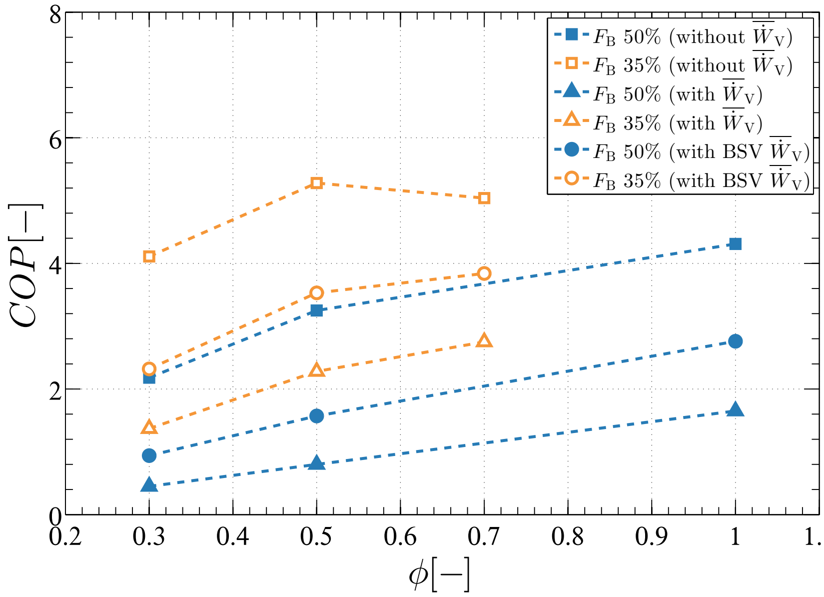


Figure 5. COP as a function of the utilization factor, for f = 0.5 Hz and ΔT_{Hex} = 10 K.

As can be seen, improving the nominal power consumption per valve and reducing the blow time fraction, the overall performance of the AMR can be improved. The behavior change of COP curves when the solenoid valves losses are accounted, as for 0.25 Hz and F_{B} = 50% where the curve reveals a peak when the solenoid valves are not considered is due to the fact that the pumping power increases with utilization but the cooling capacity increases only until a certain point. This effect is more pronounced at $f = 0.25$ Hz and $F_{\mathrm{B}} = 50\%$. When the solenoid valves are taken into account, the peak vanishes as the solenoid valves consumption is dominant in the total power consumption, and the curves monotonically grow. The best results in performance obtained in this work were found for a frequency of 0.5 Hz, blow time fraction of 35% and utilization factor of 0.7, where the cooling capacity was of 33 W, the COP of 2.75 and second-law efficiency of 0.095. Instead, if the set of solenoid valves were updated to that with lower nominal consumption, the system would have a COP of 3.90 and second-law efficiency of 0.134, in the case of using ideal solenoid valves (or no valves) the COP would be approximately of 5.10 and second-law efficiency of 0.176.

6. CONCLUSIONS

An AMR apparatus was successfully updated to operate with a novel flow distribution system based on a set of electric valves. An experimental evaluation of the updated AMR has been carried out for different operating conditions of utilization factors and blow time fractions. The control system was developed and presents a satisfactory behavior. The application of solenoid valves in the AMR systems has the potential to improve the global system performance. Moreover, the solenoid valves and the electronic control system allow a refined adjust of the blow imbalance. However, future applications of solenoid valves in AMR apparatus will depend on the use of valves with a lower power consumption. The solenoid valves solution as a flow distribution system presents a potential to enable better system thermodynamic performance, in terms of COP and second-law efficiency.

7. ACKNOWLEDGEMENTS

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9. RESPONSIBILITY NOTICE