TRANSIENT MODEL AND ENERGY ASSESSMENT OF A DIGITAL SOLENOID VALVE SYSTEM FOR A MAGNETIC REFRIGERATOR

Pedro Oliveira Cardoso, cardoso@polo.ufsc.br
Mário Cesar Destro, mario.destro@polo.ufsc.br
Manoel Guidi Alvares, manoel.alvares@polo.ufsc.br
Jaime Andrés Lozano Cadena, jaime@polo.ufsc.br
Jader Riso Barbosa Jr., jrb@polo.ufsc.br
POLO – Research Laboratories for Emerging Technologies in Cooling and Thermophysics, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, SC, 88040-900, Brazil

Victor Juliano De Negri, victor.de.negri@ufsc.br
LASHIP – Laboratory of Hydraulic and Pneumatic Systems, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, SC, 88040-900, Brazil

Abstract. *Of the rotary magnetic refrigerators developed up to date few have evaluated the impact of the flow distribution system on the overall power consumption. Most magnetic refrigerators employ rotary valves (face-to-face sealing), which, as recent works suggest, can be the major energy-consuming component of the transmission system. Bearing that in mind, a novel digital hydraulic system for a magnetic refrigerator was proposed and its mathematical model developed. In this work, the model was evaluated in terms of the operating conditions of the magnetic refrigerator developed at POLO/UFSC. The transient response of the hydraulic system was analyzed regarding the absolute pressure and the volumetric fluid flow rate at any point of the circuit, taking in consideration not only the valves but also the fluid dynamics. A transient approach was also used to evaluate the overall power consumption and it was demonstrated that under certain conditions it is independent of the operating frequency. The overall energy consumption was then determined in terms of the duty cycle of the valves and a qualitative analysis was carried out to compare the new solenoid valves system to that of a face-to-face flow management system.*

Keywords: Magnetic refrigeration, digital hydraulic system, solenoid valve.

1. NOMENCLATURE

Roman
A Cross-sectional area
Cl Pressure loss coefficient
C_V Flow rate coefficient
D Duty cycle
d_p Spheres diameter
d_0 Magnetic circuit gap (m)
e Euler's constant
f Arbitrary function
g Arbitrary function
i Electric current (A)
K_{SS} Steady state gain
K_V Flow rate coefficient
L Length (m)
L^V Inductance (H)
N Quantity
p Pressure (bar)
R Resistance (Ω)
t Time (s)
T Period (Hz)
U Voltage (V)

u_d Darcy velocity
V Volume
$\dot{V}$ Volumetric flow rate
W_n Nominal power (W)
x Displacement (m)
$\dot{x}$ Velocity (m/s)

Greek
α Tuning factor
β Bulk modulus
Δ Change
μ Viscosity
ε Porosity
ρ Density
ζ Damping ratio
τ Time constant (s)

Superscript
H   Hoses
R   Regenerator
S   System
V   Valve

Subscript
avg   Average
bed   Porous bed
B   Blow
c   Command
e   Effective
F   Forced
f   Fluid
h   Hold
in   Internal
N   Nominal
n   Natural
S   Supply
SS   Steady state
T   Tank (Reservoir)
VS   Valve system
1,2   Line 1,2

2. INTRODUCTION

Refrigeration is a technology with vast applications, present not only in domestic but also in industrial solicitations, thereby related to both research and wellbeing. Food conservation, air conditioning, process control are just a few of many other examples of its necessity. Although its good efficiency and temperature variation, steam compressors uses CFC gases, which are harmful for the environment, and even with great effort in research and development of better compressors and more efficient systems, nowadays exists a strong appeal for newer alternatives for steam compression, which should use non-harmful means to provide refrigeration.

Newer technologies have been studied since decades, and a special highlight is given to magnetic refrigeration in ambient temperature, which is based on the magnetocaloric effect. A magnetocaloric material is a magnetic refrigerant, which reversibly changes its thermic state when subjected to a variable magnetic field. Through this effect, it is possible to develop the concept of a magnetic refrigerator which uses heat transfer fluids instead of gases to perform the thermodynamic cycle. Such cycle is controlled by valves that dictate the direction and timing of the fluid flow in the regenerators. Although every experimental device and prototypes developed so far employed heat transfer fluids, only few have done systematic evaluations of losses considering those in the hydraulic system

There are two main types of hydraulic systems: power systems and fluid transport systems. While the aim of the former is to perform mechanical work, the latter are responsible for transferring fluid from one location to another to achieve a certain practical purpose which, in the case of magnetic refrigeration, is to perform the heat transfer to and from the magnetocaloric material. Because of friction and other effects, operation of hydraulic systems involves some degree of energy dissipation. In fact, friction is the one of the dominant loss mechanisms in magnetic cooling systems, which makes the hydraulic system responsible for a significant share of the energy losses. Eventually, a fraction of the heat generated by friction is transferred to the fluid in the regenerator, thus reducing the efficiency of the device (Linsingen, 2008).

In magnetic refrigerators, a synchronization is required between the magnetic and the hydraulic systems. Most prototypes reported in the literature employ rotary valves with a face-to-face sealing as flow management systems, taking advantage of the torque transmitted to the magnet (or to the regenerator) to drive the hydraulic system (Kitanovski et al, 2015). Designers recognize the simplicity of face-to-face sealing, where the opening time is controlled by oblongs and orifices. However, it has been demonstrated that the friction generated by these types of seals (Lozano, 2013; Capovilla, 2016) and the redundant constraints commonly found in these projects compromise the overall performance of magnetic refrigerators. (Eriksen et al., 2015) obtained the best performance to date of a magnetic refrigerator using a camshaft-driven valve system. However, the camshaft system impedes the adjustment of different pressure drops through the regenerator beds, requiring manual equalization of the pressure drop according to the regenerator with the lowest fluid flow rate.

Due to the aforementioned challenges, a novel digital hydraulic system based on solenoid valves is proposed in this work as a flow management system for a magnetic refrigerator. The choice for a solenoid valves system relies on two premises. The first is the lower power consumption compared to the face-to-face system. Nowadays the market offers customizable, low power valve solutions, which can be adapted to any fluid operation. Solenoid valves for water operations ranges their power consumption from 2 W to 15 W, depending on the working pressure, flow and application. As shown by (Lozano, 2013), typical working pressure ranges from 2 to 10 bar, requiring less robust valves, ergo lesser power consumption, justifying the choice for such valve system.

The other considered aspect is flexibility. By having an electric actuator, those valves require only a power source and an electronic circuit, the latter controlled by an external digital or analog control system, which feeds power to the system, in order to regulate the opening and closing time of blows. This render the solenoid valve system attractive, not only on a research point-of-view, since a finer and faster adjustment of blows to an experimental device or prototype can be easily done, in lieu of changing a physical aspect of the prototype, but also a in commercial way, as the development of specific valve systems, which could lead to a more expensive production, won't be necessary. Such flexibility also paves way to the development of newer control strategies, as the closing and opening blows' angles can be re-arranged during operation thanks to a closed-loop strategy, this way regulating, for example, differences in pressure drop between regenerators, thus maximizing the heat flow. Through the mathematical model presented in this paper, a selection map for solenoid valves with different flow rate coefficients has been developed as a function of the supply pressure of the system (volumetric flow rate). This aids the designer to define the pressure drop given by each valve. Also, a comparison of the flow management system developed by Lozano et al. (2013) with face-to-face rotary valves, and that of a novel digital hydraulic system has been evaluated the use of solenoid valves already available in the market.

3. MATHEMATICAL MODEL

3.1 Hydraulic system

A non-linear model of a hydraulic digital system for a simplified magnetic refrigerator was developed in this work, as shown schematically in Fig. 1. The system is composed of the following components: a hydraulic power unit (HPU), four digital (solenoid) valves (V), two regenerators (R), two heat exchangers (Z), two flowmeters (S), and hoses (H). The model assumes the simplest configuration of an AMR with continuous pumping and unidirectional fluid flow in the heat exchangers. The HPU is responsible for supplying and receiving the fluid to and from the hydraulic system. It acts simultaneously as a pump and as a reservoir with constant pump and discharge pressures, respectively. These are considered as the input parameters of this model. The reservoir absolute pressure (p_T) was assumed always at 1.0 bar. The flow rate provided by the HPU is independent of the internal system flow rate, since a directional valve, which acts

as a by-pass, controls it to maintain a constant supply (pump) absolute pressure (p_S). The latter is an idealization of the proposed layout by Ebel *et al.* (2016) to avoid water hammers and smooth the flow.
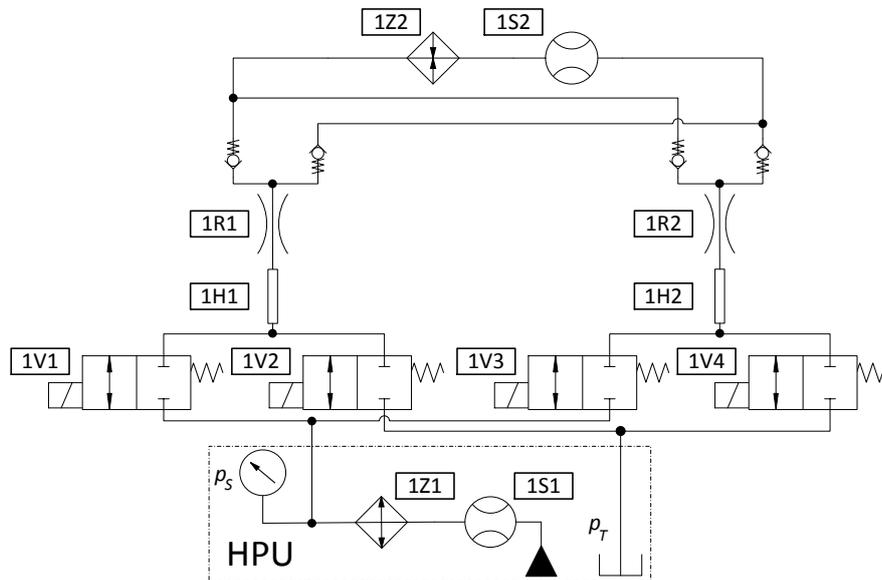

Figure 1. Digital electro-hydraulic circuit for a magnetic refrigeration system.

The mathematical model describing the directional valve is composed of two parts. First, described by the first-order equation, is the dynamic valve opening (Eq. (1)). The second part is described by the flow rate equation, being the valves 1V1, 1V2, 1V3 and 1V4 modeled by Eq. (2) and Eq. (3).

$$K_{SS} . U = \tau \cdot \frac{dx^V}{dt} + x^V \tag{1}$$

For 1V1(+) and 1V3(-):

$$\dot{V}^V = \mp \left(K_V . \frac{U}{U_N} \right) . \sqrt{p_S - p^V} \tag{2}$$

For 1V2(-) and 1V4(+):

$$\dot{V}^V = \mp \left(K_V . \frac{|U|}{U_N} \right) . \sqrt{p^V - p_T} \tag{3}$$

Also part of the hydraulic system model has the hoses, represented by the continuity equation (Eq. (4)) in the lines between the valves and the regenerator. Furthermore, it was considered the apparent fluid mass using the acceleration equation of fluid (Eq. (5)). Featuring the measuring elements, such as flow transducers and heat exchangers, was used the pressure drop equation for laminar flow (Eq. (6)).

$$\dot{V}^V = q_{v_{in}}^H + \frac{V^V}{\beta_e^H} \cdot \frac{dp^V}{dt} \tag{4}$$

$$\rho \cdot \frac{V_{in}^H}{A_{in}^H} \cdot \frac{d\dot{V}_{in}^H}{dt} + \frac{A_{in}^H}{Cl} \cdot q_{v_{in}}^H = A_{in}^H . (p^V - p_{in}^H) \tag{5}$$

$$Cl = \frac{\dot{V}_{in}^H}{\Delta p_{in}^H} \tag{6}$$

The regenerators are modeled as packed-sphere beds, for which the pressure drop can be determined by the correlation proposed by Ergun (1952) and given by:

$$\frac{\Delta p_{\text{bed}}}{L_{\text{bed}}} = 150 \cdot \frac{(1-\varepsilon)^2}{\varepsilon^3} \cdot \frac{\mu_f u_d}{d_p^2} + 1.75 \cdot \frac{(1-\varepsilon)}{\varepsilon^3} \cdot \frac{\rho_f u_d^2}{d_p} \qquad (7)$$

3.2 Power consumption

The power consumption of an electronic device is measured based on its input voltage and current. As shown by (Wang *et al*, 1993), the dynamic state equation of the current on a solenoid valve can be described as:

$$L^{\text{V}}(x, i^{\text{V}}) \frac{di^{\text{V}}}{dt} = U - Ri^{\text{V}} - \frac{\alpha L^{\text{V}}(x, i^{\text{V}})}{x + d_0} i^{\text{V}} \dot{x} \qquad (8)$$

where $L^{\text{V}}(x, i^{\text{V}})$ is the coil inductance at a certain coil current i^{V} and spool displacement x, U is the terminal voltage at the solenoid, R is circuit resistance, α is a tuning factor for the effect of spool velocity $\dot{x}$ and d_0 is the magnetic circuit gap.

The current has the same behavior of a resistor-inductor series circuit (RL-series circuit) $L^{\text{V}} di/dt = U - Ri$ with a disturbance caused by a contrary magnetic force generated by the spool displacement, represented in Eq. (8) by the fourth term, which develops while both velocity and current are non-zero. Due to friction, when a current is applied to the valve, initially the spool stays stationary until the magnetic force generated by the solenoids is strong enough to break the friction. When the displacement starts, so starts the disturbance generated by the change in inductance caused by the displacement itself. When the valve is fully open, the disturbance ceases and the current grows again, eventually reaching its steady state level. A visual representation of this behavior is shown by Fig. 2 where the behavior of the current modeled by Eq. (8) is compared to a RL-series circuit.
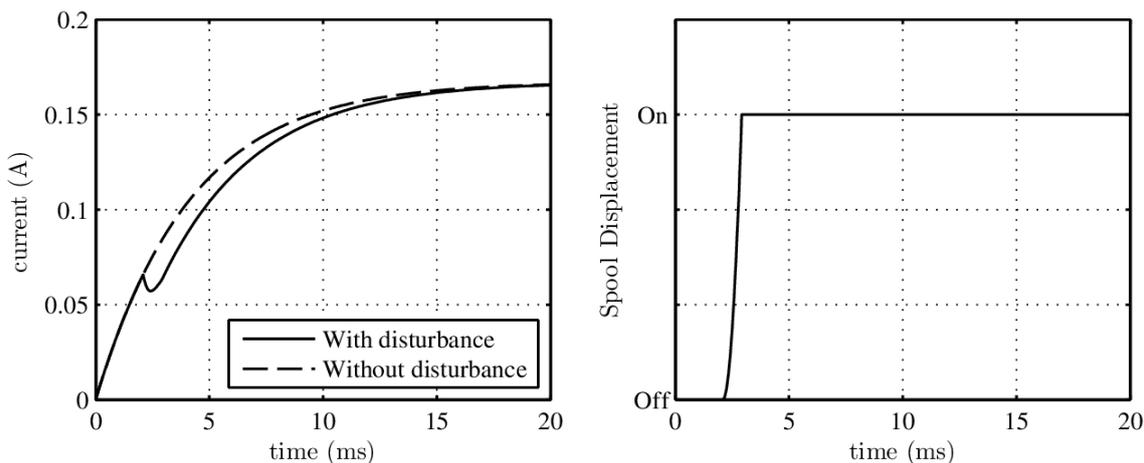

Figure 2. Current and displacemente behavior of a solenoid valve.

Since the operation of the magnetic refrigerator is periodic, and so is the behavior of the solenoid valves, during part of the period T the valve is active and the rest of the time the valve is off. Hence, the behavior of the current of the valve can be analyzed as a periodic signal, and the average consumption power $\dot{W}_{\text{avg}}$ can be calculated as:

$$\dot{W}_{\text{avg}} = \frac{1}{T} \int_0^T Ui \, dt \qquad (9)$$

Considering the source voltage constant, in order to simplify the calculation a good approximation would be to assume the current behavior of the valve as a RL-series circuit, which overestimates the power consumed. The solution of a first order ordinary differential equation (ODE) $\dot{f}(t) + af(t) = g(t)$, $a \in \mathbb{R}$, can be decomposed in two parts: its natural (f_n) and forced (f_F) response. The former is related to a null input $g(t) = 0$, where the dynamic is dictated by the initial condition of the ODE $f(0)$. The latter is related to the response given by the system due to a forced entrance $g(t)$ with initial condition $f(0)$. As an ODE is a linear system, time dynamic is given as the combination of both natural and forced response. The natural response of a RL-series circuit current (i_n^{V}) is given by Eq. (10) and its forced response current i_F^{V} by Eq. (11).

$$i_n^{\text{V}}(t) = \frac{U}{R} \left(e^{-(R/L^{\text{V}}) t} \right) \qquad (10)$$

$$i_F^V(t) = \frac{U}{R}\left(1 - e^{-(R/L^V)\,t}\right) \tag{11}$$

Where R is the total resistance of the system and L^V is the inductance of the solenoid valve. The natural response's initial condition $i_n^V(0)$ is the same as the forced response's final current, i.e., $i_n^V(0) = \lim_{t\to\infty} i_F^V(t) = U/R = i_h$ which is the hold current of a solenoid valve. This is the necessary current needed in order to keep the valve open in steady-state and $\tau = L^V/R$ is the time constant of the valve's current, related to the dynamic of the valve.

A periodic signal is defined as a signal that repeats its behavior after a period T, i.e. $x(t) = x(t+T)$. This way a periodic RL-series current can be defined firstly as the forced current i_F^V generated by a given potential U to the valve at instant $t = 0 \pm nT, n \in \mathbb{Z}$, in order to break the static friction and to displace the spool. Considering T_B the blow period, $\tau \ll T_B$ and $\lim_{t\to T_B} i_F^V \cong i_h, 0 < T_B < T$, the forced current i_F can be described as Eq. (12):

$$i_F^V(t) = i_h^V\left(1 - e^{-t/\tau}\right) \tag{12}$$

After this period the voltage drops and the spool closes due to a spring attached to the other end of the spool. On that moment, the current drops rapidly and considering $\tau \ll T - T_B$, $\lim_{t\to(T-T_B)} i_n \cong 0$ Eq. (14) describes its dynamics.

$$i_n^V(t) = i_h^V e^{-(t-T_B)/\tau} \tag{13}$$

Both considerations of $\tau \ll T$ and $\tau \ll T - T_B$ are necessary in order to simplify the calculations of the resultant current. When one or both conditions are not met, there may be not enough time to the natural and/or forced currents to reach their steady state values, rendering misleading the calculation, as the limits calculated above are not coherent. By merging both equations the behavior of the periodic RL-Series' current of the solenoid valve is given by (15).

$$i^V(t) = \begin{cases} i_h^V\left(1 - e^{-t/\tau}\right), & 0 \le t < T_B \\ i_h^V e^{-(t-T_B)/\tau}, & T_B < t \le T \end{cases} \tag{14}$$

Where T is the operation period of the magnetic refrigerator. Considering constant the nominal voltage, as its value is regulated by the power source, merging the average power integral Eq. (9) with Eq. (15):

$$\dot{W}_{avg} = \frac{U i_h^V}{T}\int_0^{T_B}\left(1 - e^{-t/\tau}\right)dt + \frac{U}{T}\int_{T_B}^{T} e^{-(t-T_B)/\tau}dt \tag{15}$$

Solving Eq. (15) yields:

$$\dot{W}_{avg} = \frac{U i_h^V}{T}\left(T_B - e^{-\frac{T_B}{\tau}} + e^{-\frac{T-T_B}{\tau}}\right) \tag{16}$$

Since $\tau \ll T_B$ and $\tau \ll T - T_B$, Eq. (17) yields,

$$\dot{W}_{avg} = W_n D \tag{17}$$

Where W_n is the nominal power $W_n = U i_h^V$ and $D = T_B/T$ the duty cycle. Since $0 \le D \le 1$ the maximum average power consumed is the holding power itself, meaning that, by choosing a low power solenoid valve one can estipulate that the maximum consumed power is its own nominal holding power. Simultaneously power is consumed by the actuation system. Considering that relays are used, where one relay activates two solenoid valves, and considering that its behavior is also defined by a RL series circuit, the total average power $\dot{W}_{VS}$ consumed by the solenoid valve system of a magnetic refrigerator, is defined by Eq. (18):

$$\dot{W}_{VS} = N_V D\left(W_n + 0.5\dot{W}_c\right) \tag{18}$$

Where $\dot{W}_c$ is the power consumed by each relay and N_V is the number of valves.

4. RESULTS AND DISCUSSIONS

As seen in Fig. 1, the modeled hydraulic system has a serially circuit characteristic, having the same flow rate in all the items. Furthermore, there is a pressure loss (pressure drop) associated with each component in the system, similar to a voltage divider circuit. Such pressure drop is due to the resistive property of the hydraulic components, directly influencing the system flow rate ($\dot{V}^S$). This resistive property is associated with the flow rate coefficient of the valves (K_V) and the pressure loss coefficient of the other components (Cl). The values of supply and reservoir pressures also influence the system flow rate and the pressure loss at the components of the hydraulic system.

On circuits that work with water as cooling fluid, it is common to use the flow coefficient C_V, and the relationship of $K_V = (7.6 \times 10^{-7})C_V$. This ratio is merely units conversion, since this C_V is associated with a flow rate in L/h and pressure drop in bar while K_V works in the international system units.

Figure 3 and 4 show the model simulations with the digital hydraulic system working at one AMR cycle with $C_V = 0.10$ and $C_V = 0.40$. Other parameters such as length of hoses and supply pressure and reservoir pressure were kept to have a comparison of the changing effects of valves flow rate coefficient in the system flow rate and pressure loss in components ($L^H = 0.50$ m, $p_S = 5$ bar and $p_T = 1$ bar). In both simulations, it can be seen that the flow rates through the items are equal. Furthermore, there is a pressure drop associated with each component, proving to be an equivalent serially hydraulic circuit as the electrical circuits. To the system working with the valve of $C_V = 0.10$ (Fig. 3), the steady state values of system flow rate and valve pressure drop are $\dot{V}^S = 75.70$ L/h and $\Delta p^V = 0.77$ bar. Already with the system using the valve with $C_V = 0.40$ (Fig. 4), these values are $\dot{V}^S = 104.50$ L/h and $\Delta p^V = 0.09$ bar. Based on this steady state values, it is can be concluded that the higher the value of C_V, the lower is the pressure drop across the valve Δp^V and the greater is the flow rates of the system $\dot{V}^S$.

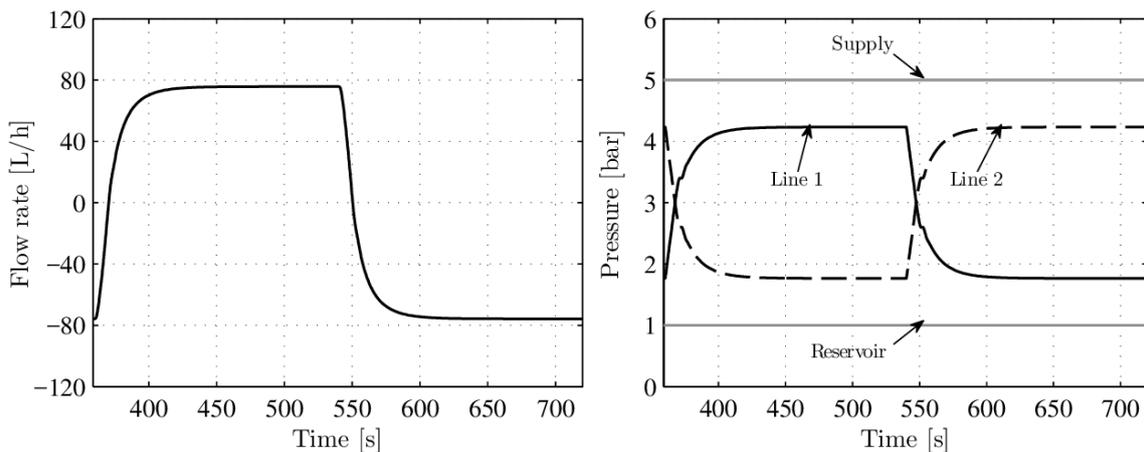Figure 3. Transient behavior of the volumetric flow rate and the fluid pressure of a digital electro-hydraulic circuit with $C_V = 0.10$.
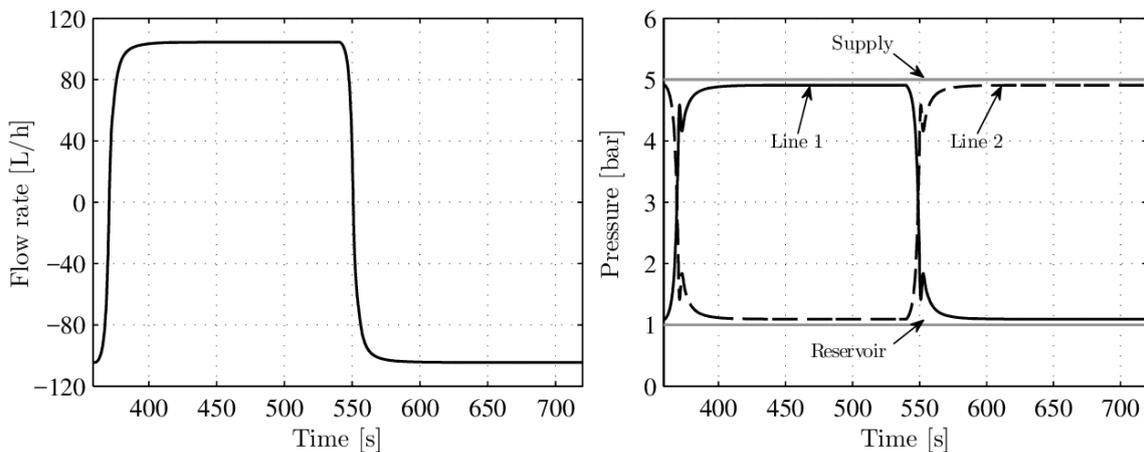

Figure 4. Transient behavior of the volumetric flow rate and the fluid pressure of a digital electro-hydraulic circuit with $C_V = 0.40$.

Figure 5 shows the model simulation to the system working with the same parameters of the latter simulation discussed above, but with 10 bar of pressure supply. The steady state values of system flow rate and valve pressure drop are $\dot{V}^S = 180.40$ L/h and $\Delta p^V = 0.27$ bar. Based on this steady state values, it is can be concluded that the higher the value of p_S, greater is the flow rates of the system $\dot{V}^S$.

The steady state values of the volumetric flow rate and the valve pressure drop from the dynamic model simulations are shown in Figs. 6(a) and (b), respectively, for several combinations of valve flow rate coefficient C_V and supply pressure p_S. As seen in Fig. 6(a), the volumetric flow rate has a larger dependence on the supply pressure than in the valve C_V. This is because in most of the analyzed cases, the valve pressure drop (Fig. 6(b)) represents a small proportion of the system pressure drop, especially for $C_V > 0.3$ L/(h$\sqrt{\text{bar}}$), since the valve pressure drop has an inversely proportional relation with the volumetric flow rate. In a first approach, these two maps can aid the designer of a magnetic refrigerator to select solenoid valves using the valve C_V for certain system p_S, resulting in a volumetric flow rate through the AMRs. These plots were built by the simplified model presented in Fig. 1, but they can be extended to relate the volumetric flow rate in the case of multiple regenerator beds operating with parallel blows.
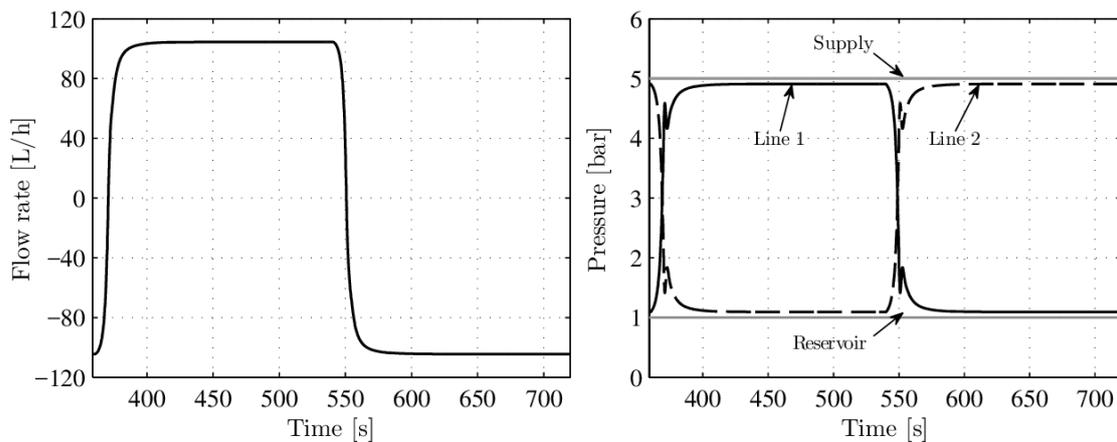

Figure 5. Flow rate and pressure drop of digital electro-hydraulic circuit with $C_V = 0.40$ and $p_S = 10$ bar.
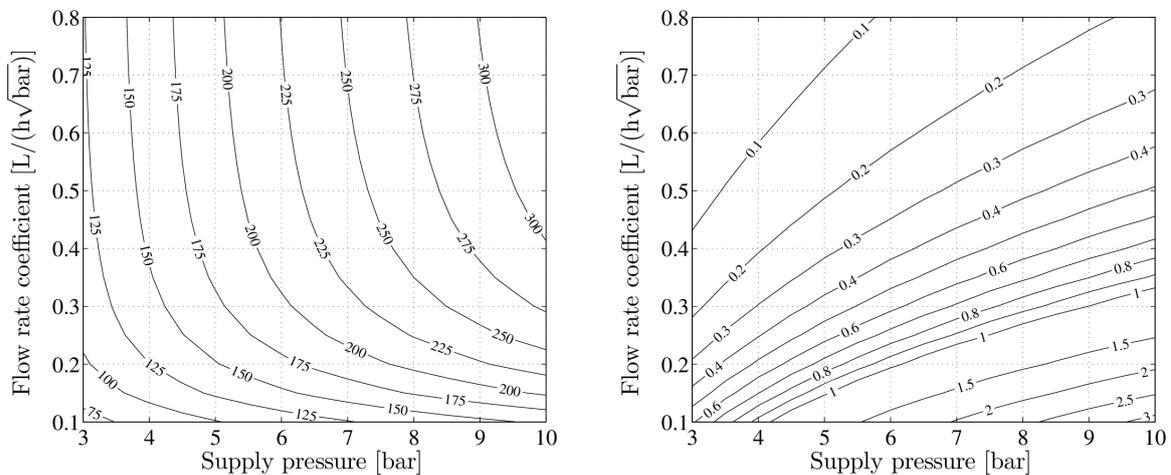

Figure 6. Steady-state dependence of the volumetric flow rate and valve pressure drop from the dynamic model simulations for several combinations of C_V and p_S.

As a case study, consider a desired $\dot{V} = 175$ L/h at a pump $p_S = 5$ bar. In principle, it would be possible, using Fig. 6(a), to select valves with C_V between 0.3 and 0.4 L/(h $\cdot \sqrt{\text{bar}}$). Each of those valves at the previous desired operating conditions would have, by looking at Fig. 6(b), a pressure drop of around 0.3 to 0.4 bar. On the other hand, by having a valve with $C_V = 0.5$ L/(h $\cdot \sqrt{\text{bar}}$) and the desire volumetric flow rate of 250 L/h, according to Fig. 5(a), the device would have to operate with a supply pressure of about 7.3 bar. In this case, by checking in Fig. 6(b), the valve pressure drop would be approximately 0.34 bar.

The total power consumption of a solenoid valve system), as proposed in this work, is calculated by Eq. (18). Also, the magnetic refrigerator developed by Lozano et al. (2016) is considered, which is composed of 8 regenerator pairs operated by a face-to-face rotary valve system that would be equivalent to a digital hydraulic system of 16 solenoid valves, each with $D = 0.25$. For typical values of power consumption of 4 W, 24 V DC voltage source and settling time of 50 ms, together with the average power consumption of an 8-relay drive system, a comparison between the

power consumption of the rotary valves evaluated experimentally by Capovilla *et al.* (2016) and the proposed solenoid valve system is shown in Fig. 7. A relevant conclusion is that due to the cyclic behavior of the system and the fast time response of the solenoid valves, the power consumption of the solenoid valve system is independent of the operating AMR frequency, and this is case would be approximately 16.7 W.
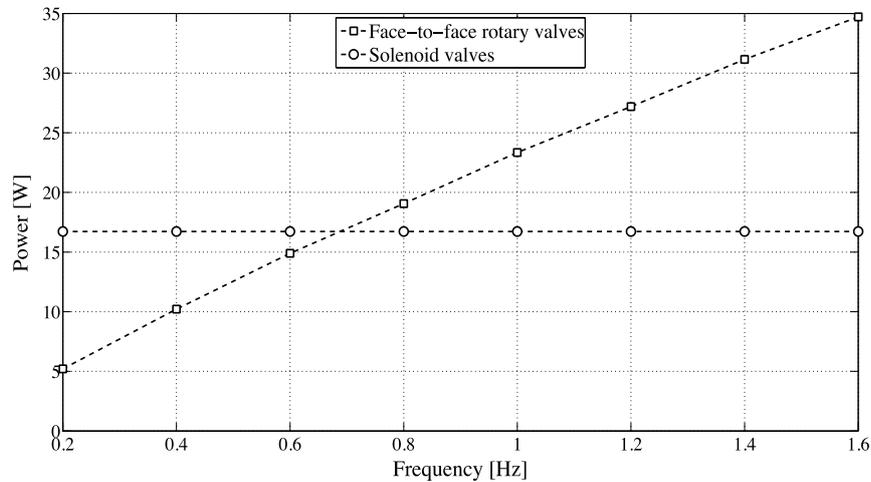

Figure 7. Comparison of the power consumption of the hydraulic system operated by rotary and solenoid valves.

5. CONCLUSIONS

In this work, a mathematical model of a novel digital hydraulic system has been developed. The dynamic behavior of the hydraulic system was evaluated at different operating conditions and two designer maps were generated as a function of the volumetric flow rate and the valve pressure drop. Even though the studied case refers to a unitary magnetic refrigerator, its operation can be extended for larger refrigerators. The power consumption of the novel valve system that would substitute the rotary valve system of an actual magnetic refrigerator has demonstrated a frequency-independent behavior and low power consumption, improving not only the efficiency but also its operational versatility. Based on the theoretical analysis presented on this paper, further work on digital-driven solenoid valves are expected to achieve a higher performance on power consumption than of the actual systems, leading to newer and more efficient systems.

5. ACKNOWLEDGEMENTS

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