

PARAMETER INFLUENCE ON THE DESIGN OF HYDRAULIC POSITIONING SYSTEMS

Irving Muraro, irving_muraro@yahoo.com.br
Victor Juliano De Negri, victor@emc.ufsc.br
Federal University of Santa Catarina, Mechanical Engineering Department, LASHIP – Laboratory of Hydraulic and Pneumatic Systems, 88040-900, Florianópolis, SC, Brazil

Henri Carlo Belan, henri@ifsc.edu.br
Federal Institute of Santa Catarina, IF-SC, 89813-000, Chapecó, SC, Brazil

José Roberto Branco Ramos Filho, robertobrancofilho@gmail.com
Esperança Institute of Higher Education, IESPES, 68040-100, Santarém, PA, Brazil

Abstract. *Position control using hydraulic systems is widely applied in several engineering fields including industry, aerospace, vehicles and energy generation. However, their design is not a simple task since it is necessary to observe their behavior according to control theory. Therefore, each application has specific static and dynamic requirements that must be fulfilled under loading conditions that are not completely known by the designer. These difficulties have led to the reuse of previous designs for the construction of new equipment, which requires changes or adjustments on some components in order to achieve the desired behavior. To overcome these constraints, a comprehensive view of the design problem is necessary in order to assure the correct sizing of the hydraulic components, giving the designer complete domain over the technical decisions during the design process. In this regard, this paper presents a method developed to organize the design effort in three main steps: static and dynamic sizing of the system, data conversion of cataloged data and dynamic behavior study. Based on that method, this paper presents a detailed study of the valve main characteristics, such as natural frequency and flow coefficient. The actuator and system natural frequencies are also analyzed, whereas it changes according to the application and operation conditions of the electro-hydraulic system. The variation and influence of these parameters in the closed loop hydraulic control system behavior using a proportional controller are evaluated by mathematical models implemented using Matlab/Simulink software. The model validation and the practical verification of the simulation results are accomplished using a workbench.*

Keywords*: Electro-hydraulic system, position control, proportional hydraulic valves, design of hydraulic systems*

1. INTRODUCTION

The electro-hydraulic position control systems have large applicability in several fields of engineering, being used mainly to drive and control high power devices with the required reliability, speed, and accuracy.

One of the most important components in hydraulic positioning systems is the proportional directional control valve, which is responsible for the accurate flow control and thereby the control of the force, position, speed and acceleration of the actuator (Kalyoncu and Haydim, 2009).

However, the dynamic behavior of hydraulic systems is highly nonlinear due to the phenomena such as valve flow-pressure characteristic and the variation of control volumes and hydraulic stiffness. These characteristics, in addition to cause difficulties in the control of such systems, also create a lot of uncertainties in the parameterization, design and modeling of the electro-hydraulic systems (Guan and Pan, 2008).

Even with the continuous technological improvement, problems concerning the selection, sizing and control of the actuation system (valve and cylinder) are still not sufficiently solved to ensure certainty to designers on the procedure to be adopted, especially considering the several dynamic and static behavioral characteristics of the valves and its effects on electro-hydraulic positioning systems (Furst, 2001).

Johnson (1995) reports the great number of hydraulic equipment with these unwanted behaviors, such as oscillations, noise and inadequate responses, resulting in an increasing negative image of such systems. The author also argues that a well-designed hydraulic systems can be consistent, reliable, durable and with smooth and silent dynamic. Therefore, a carefully design must be applied aiming the component sizing and selection.

However, the design and assembly of these systems are not trivial due of the necessity to observe their behavior according to the control theory. Each application has specific static and dynamic requirements that must be performed under loading conditions which are not always well known by the designer (De Negri *et al.*, 2008).

In order to overcome these constraints, a comprehensive view of the design problem is necessary to create procedures which assure the correct hydraulic component sizing, giving the designer complete domain over the technical decisions during the design process.

In this regard, this paper presents a method developed at the Laboratory of Hydraulic and Pneumatic Systems (LASHIP/EMC/UFSC) that organizes the design effort into three main steps: static and dynamic sizing of the system, conversion of cataloged data and dynamic behavior study. Considering this method, a study of the parameter influence of the valve and cylinder is carried out.

The influence of the natural frequency and flow coefficient of valves is discussed as well as the actuator natural frequencies. The variation and influence of these parameters in the closed loop hydraulic control system behavior using a proportional controller are evaluated by mathematical models implemented using Matlab/Simulink software. The model validation and the practical verification of the simulation results are accomplished with the use of "Proportional Hydraulic Platform" workbench.

The case studied in this paper employs a four-way spool valve and a symmetric cylinder. However, the results can be extended for a four-way spool valve and a differential cylinder set or a three-way differential spool valve and an asymmetric cylinder set.

2. THE DESIGN METHOD

As discussed in De Negri *et al.* (2008), the design method presented in this paper is part of the embodiment design phase of a system, and is performed according to Fig. 1. Before this method to be implemented, both clarification of the task and conceptual design phases must be carried out in order to generate the first version of the design requirements and the system concept, which are necessary for the execution of the embodiment activities. This information might come from professionals in other areas, who understand the process or the equipment where the system will be installed.

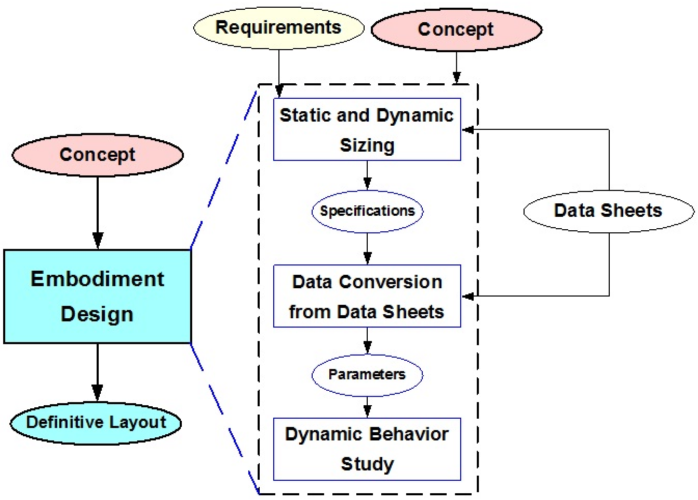

Figure 1. Embodiment design phase for a hydraulic positioning system

Considering that the aim of this method is to manage the hydraulic positioning system design, its conception depends on the type of hydraulic circuit chosen to solve a specific problem. One can observe that there are three basic configurations that have been widely applied in industry: (a) Symmetric four-way proportional valve and symmetric double effect cylinder (SV-SC); (b) Asymmetric four-way proportional valve and asymmetric double effect cylinder (AV-AC); and (c) Three-way proportional valve and asymmetric single effect cylinder (3wayV-AC). Figure 2 shows these hydraulic circuits including the position transducer and the controller that are necessary to build a hydraulic position control system.

In the following sections the three steps of the embodiment design are detailed for configuration (a), that is, a symmetric four-way valve with a symmetric cylinder. Results related to the other two arrangements can be found in Furst (2001).

2.1. Step 1 – Static and dynamic sizing

This step considers the dynamic and static requirements of the system, such as maximum displacement, settling time, forces, overshoot, among others, and uses the mathematical expressions derived from the dynamic model of the system. As a result, specifications for a valve and a cylinder are achieved. These components must be selected simultaneously, since their parameters are interdependent. The activities can be executed in the sequence shown in Fig. 3.

<u>Activity 1: System undamped natural frequency</u> – This activity begins by analyzing the global system behavior, instead of each component that comprises it. If the system can afford overshoot, it will be treated as a second-order

system with a damping ratio coefficient (ζ^{SYS}) equal to 0.7. If having an overshoot is not feasible, a second-order system with $\zeta^{SYS} = 1$ will be used. With this information and the desired system settling time (t_s^{SYS}), it is possible to calculate the system undamped natural frequency (ω_n^{SYS}) as represented in Fig. 3.

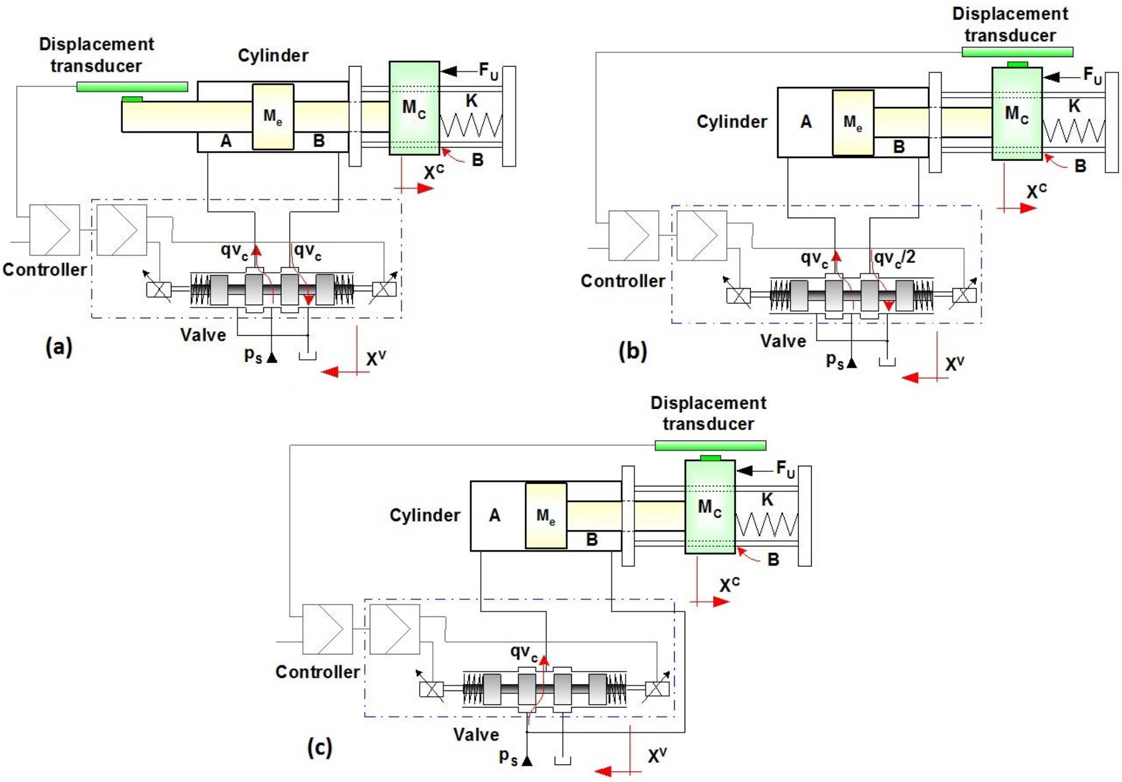

Figure 2. Three common configurations for hydraulic positioning systems

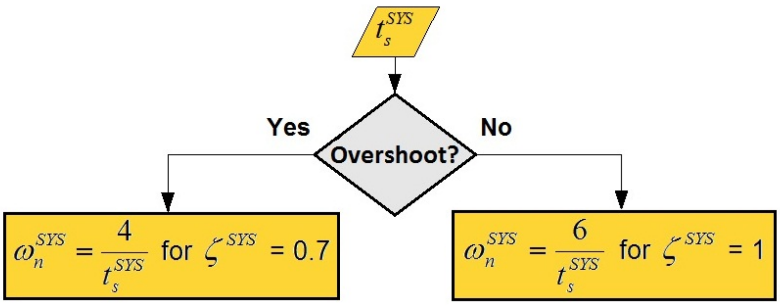

Figure 3. Activity 1 flowchart

Activity 2: Cylinder speed and acceleration - Based on the information obtained from activity 1 and on the desired steady state displacement for the positioning system ($x_{\max}^C$), the maximum speed ($v_{\max}^C$) and the maximum negative acceleration ($a_{\max n}^C$) can be calculated using equations shown in Fig. 4.

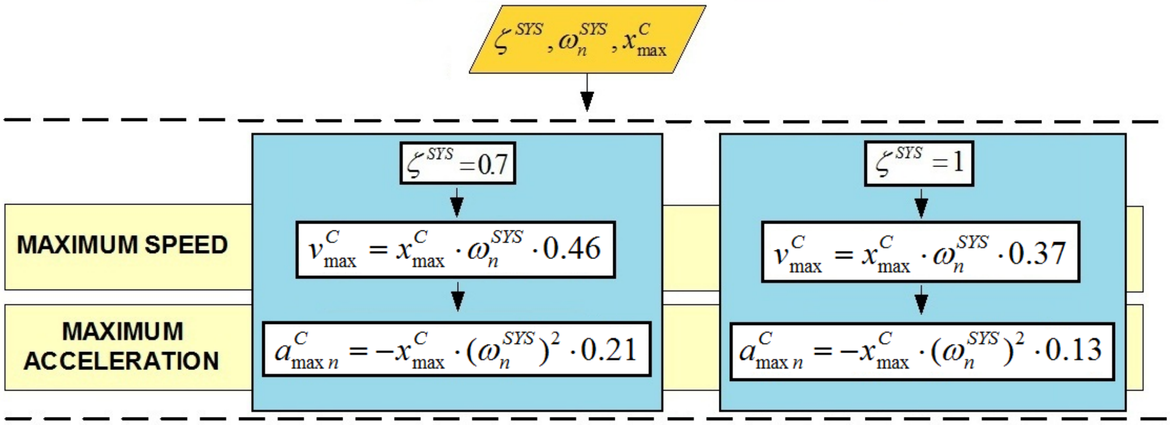

Figure 4. Activity 2 flowchart

Activity 3 and 4: Load pressure and cylinder area - For a system composed of a symmetric proportional four-way valve and a symmetric cylinder, the load pressure (p_C) is defined as the difference between the pressure in the cylinder chambers (p_A and p_B) according to the following equation:

$$p_C = p_A - p_B \tag{1}$$

Merrit (1967) *apud* Furst (2001) shows that the maximum output power for a SV-SC system occurs at the load pressure ($p_{CP\max}$) calculated by:

$$p_{CP\max} = (2/3)\cdot p_S \tag{2}$$

where the supply pressure (p_S) is a design requirement defined in a previous phase.

It is important to notice that the expected load pressure (p_C) should not be higher than the load pressure at the maximum output power ($p_{CP\max}$). If the load is increased above this limit, the pressure drop across the valve approaches zero, and the hydraulic actuator tends to stall. Considering this, the cylinder area and the load pressure are determined as shown in Fig. 5.

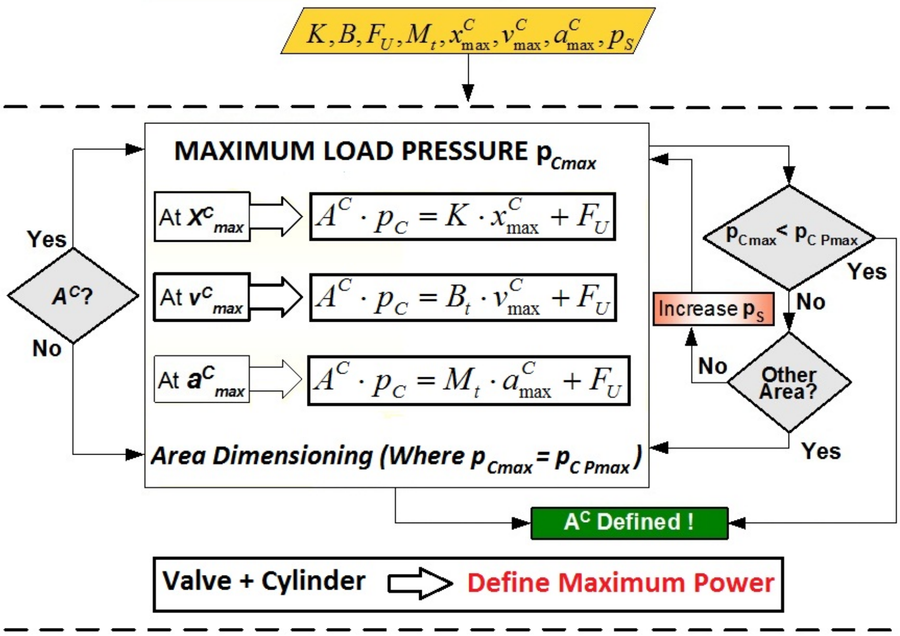

Figure 5. Activity 3 and 4 flowchart

Considering that the cylinder area (A^C) is the annulus area, F_U is the useful force applied to the load, K is the load stiffness, B_t is the viscous friction coefficient, and M_t is the total mass accelerated by the actuator, the maximum load pressure must be calculated under three operational conditions:

- When the cylinder is positioned at the maximum desired position ($x^C_{\max}$);
- When the cylinder is at its top speed ($v^C_{\max}$);
- When the cylinder is at its maximum negative acceleration ($a^C_{\max n}$).

Due to the hydraulic system nonlinearities, it is possible to define the cylinder annulus area using the negative maximum acceleration instead of the positive one. This subject is more detailed and explained in Muraro (2010).

After the required area has been defined, a commercially available cylinder must be chosen from cataloged data with an area equal to or greater than the area calculated in this activity. On the other hand, if a commercial cylinder is already available, the equations above are solved not to define the area, but to find $p_{C\max}$, in order to verify whether the supply pressure is high enough so that the 2/3 value is not exceeded. If the supply pressure is too low, the cylinder area must be increased, and a new commercially available cylinder has to be selected, or the supply pressure p_s can be redefined at this stage to make the use of the available cylinder feasible.

Activity 5: Cylinder maximum flow rate - Subsequently, when the cylinder area is defined, the designer is able to determine the maximum flow rate of the cylinder ($q_{VC\max}$) based on its maximum speed using the equation below:

$$q_{VC\max} = A^C \cdot v^C_{\max} \tag{3}$$

Activity 6: Cylinder undamped natural frequency - The next activity is to calculate the natural frequency of the cylinder (ω^C_n) when associated with the load. For a four-way valve commanding a symmetrical cylinder, this value can be evaluated using the following equation:

$$\omega^C_n = \left(\frac{4 \cdot \beta e \cdot A^{C^2}}{M_t \cdot V^C_t} \right) \tag{4}$$

where β_e is the effective fluid Bulk modulus and V_t is the total volume between valve and cylinder.

As discussed in Section 3.3 below, this frequency should be at least five times greater than the undamped natural frequency required for the system (Activity 1). This is a physical limit that determines whether the cylinder and its load have the necessary bandwidth to yield the desired time response.

Activity 7: Valve undamped natural frequency - The valve natural frequency (ω^V_n) must be at least five times the system natural frequency to avoid undesired responses of the system. This subject is explained in Section 3.2 below.

2.2. Step 2 – Data conversion from datasheets and catalogs

A valve with similar characteristics to the one specified in Step 1 must be found among those available on the market. However, each manufacturer presents the technical data in specific ways that require careful analysis by the user. To overcome this problem, the flow rate coefficient (Kv) as defined in Johnson (1995) is used, which can be easily calculated from the catalogs. This single parameter allows the designer to compare the flow rate capability of the valves and their suitability according to the specifications obtained in Step 1.

Activity 1: Pressure drop at valve at maximum flow rate – Using the maximum load pressure at the maximum speed obtained in Step 1 – Activity 4, one can calculate the total pressure drop at the valve (Δp_t) as:

$$\Delta p_t = p_S - \left| p_{C\max} \right| \tag{5}$$

Activity 2: Flow rate coefficient – Considering the maximum flow rate obtained in Step 1 – Activity 5 and the total pressure drop above, it is possible to determine the flow rate coefficient through the following equation:

$$Kv = \frac{q_{VC\max}}{\sqrt{\Delta p_t}} \tag{6}$$

Theoretically, this equation establishes the minimum value of Kv that will guarantee the condition of maximum cylinder speed. This topic is discussed in Section 3.1 below.

The other activities of this step, such as how to calculate the commercial valves flow coefficient and the effective maximum flow rate, can be found in Furst (2001) and De Negri *et al.* (2008).

2.3 Step 3 – Dynamic behavior study

After finding a commercially available cylinder and proportional valve, a simulation is recommended in order to validate the design. A reliable computer model must be available to carry out a design validation simulation. The model used in this paper was developed using the Simulink® Software available in MathWorks MATLAB®.

The model should simulate the behavior of components in an environment as similar as possible to that where the system will operate. The controller, position transducer, hydraulic power unit, hydraulic fluid, load and other relevant environmental details must be considered in order to evaluate the system behavior under such conditions. However, the model must be validated before being used to make sure the results are reliable. More details of this step can be found in Furst (2001) and De Negri *et al.* (2008).

3. PARAMETER INFLUENCE

In order to analyze the parameter influence on the design of electro-hydraulic position control systems with the method presented in this paper, different configurations of hydraulic systems were implemented using the Proportional Hydraulic Platform (PHP) shown in Fig. 6.

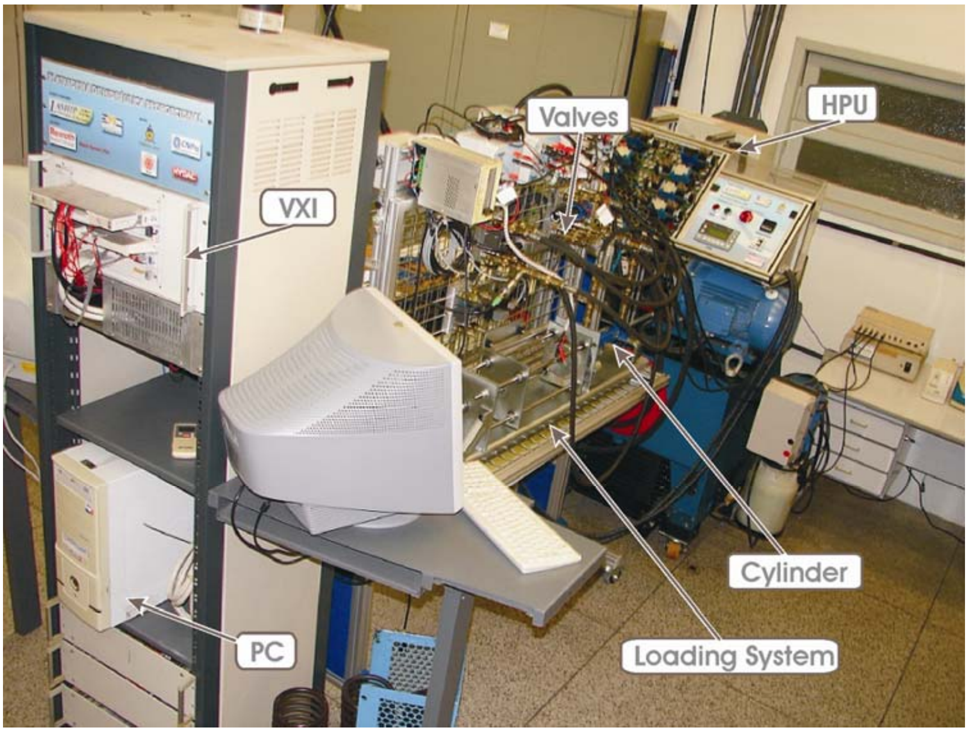

Figure 6. Proportional Hydraulic Platform

The PHP includes a hydraulic power unit (HPU) controlled by a programmable logic controller (PLC), VXI data acquisition system, pressure and displacement transducers, PC running LabVIEW® and MATLAB®, workstation, loading system, pressure reducing and proportional directional valves, cylinders, and other components to emulate the environment where the system under test will be used.

In this paper, the parameter influence is analyzed considering the system requirements presented in Table 1. The results obtained from the Step 1 of the design method are shown in Table 2. During this step, a Bosch Rexroth cylinder with 18 mm rod diameter and 25 mm bore diameter was selected. The annulus area is 2.364 cm², which is close to that calculated by the method. It was also defined the directional proportional valve that provides a desired flow (2x10^{-4} m^3/s) at the pressure drop of 23.3x10^5 Pa. Therefore, the Bosch Rexroth valve 0 811 404 038 is the closest to that resulting by the method with the required coefficient flow Kv =2.48 $L/min.bar^{1/2}$ (1.31x10^{-7} $m^3/s.Pa^{1/2}$).

Table 1. Requirements for the positioning system.

t_s^{SYS} [s]	$x_{\max}^C$ [m]	Overshoot?	p_S [Pa]	M_t [Kg]	K [N/m]
0.13	0.05	no	70x10^5	76	2618.4

Table 2. Results from Step 1.

ω_n^{SYS} [rad/s]	$v_{\max}^C$ [m/s]	$a_{\max n}^C$ [m/s^2]	$p_{C\max}$ [Pa]	A^C [m^2]	$qv_{\max}$ [m^3/s]
46.15	0.849	14.41	46.7x10^5	2.41x10^{-4}	2x10^{-4}

Before to assembly the system, a nonlinear model representing the positioning system with a PI controller and loading system was built in MATLAB®'s Simulink®. This model is comprised of blocks based on transfer functions that represent each of the components of the system assembled in a closed loop. The model uses parameters available in the manufacturers' catalogs for each component as well as in the data generated by the design method. Each block in Fig. 7 stands for a sub-model that represents a component and the complete model is presented in Muraro (2010).

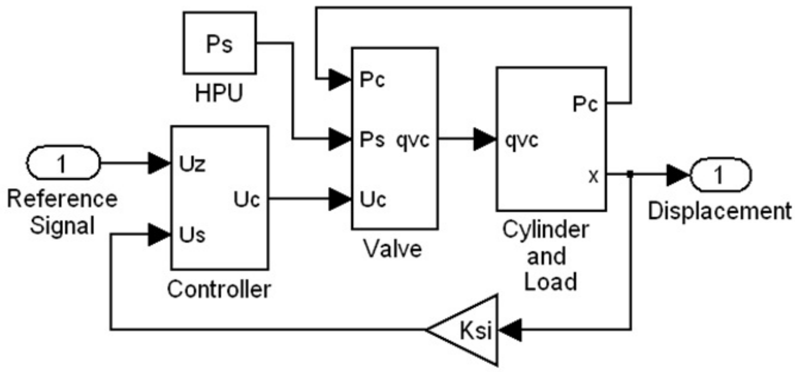

Figure 7. Simulink® model for the positioning system

The load and the cylinder blocks were designed according to experiments executed on the PHP in order to evaluate the friction forces involved. The controller implemented in MATLAB is the same P (Proportional) controller implemented by the LabVIEW® in the PHP. Furthermore, this model also represents the load loss, mass fluid acceleration and fluid compressibility in the tubes.

Just in order to validate the nonlinear model, a spring with elastic coefficient (K) of 2618.4 N/m was used as loading system and also adjusted a preload of 440 N. In this experiment, it was not coupled a mass of 76 Kg to the cylinder. The viscous friction force behavior in the system was verified for different cylinder speeds and is presented in Fig. 8, from where an average viscous friction coefficient (B_t) of 400 $N.s/m$ was considered at the design method.

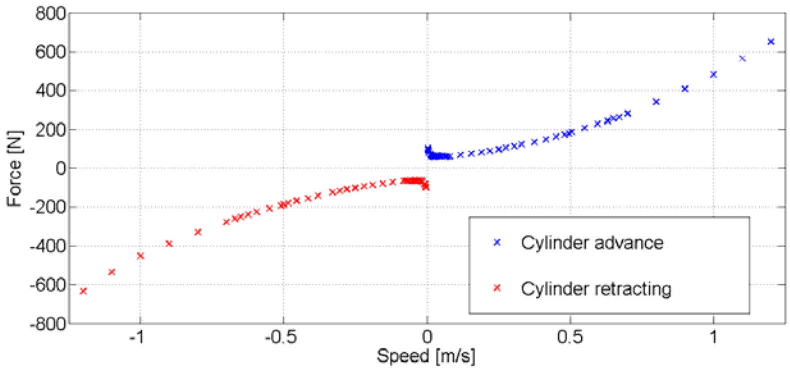

Figure 8. Viscous friction force versus cylinder speed

The components selected by the method resulted on a positioning system satisfying all established requirements. The controller (K_p) was tuned at 07 and then the positioning system was assembled and tested under the same simulated conditions. The theoretical and experimental results are shown Fig. 9.

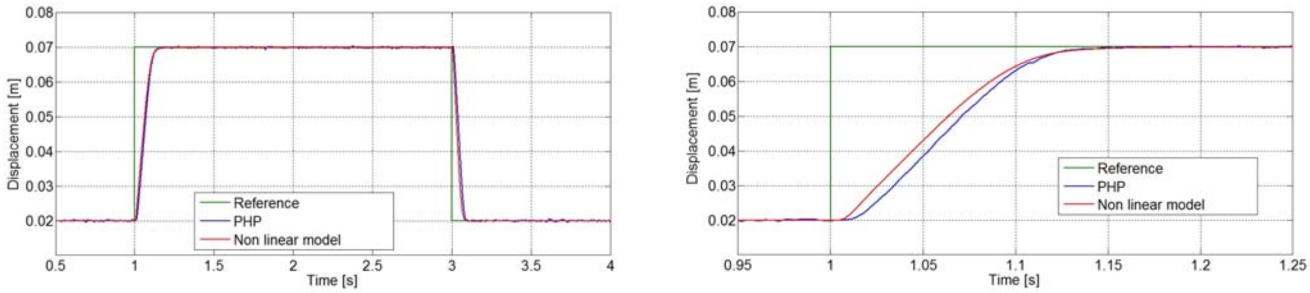

Figure 9. Step response of the positioning system: a) Advance and retracting; b) Zoom view of extending movement.

3.1. Valve flow coefficient

Applying the design method presented in this paper, a suitable value of the flow coefficient is defined for the proportional valve selection. However, not always valves with characteristics and parameters close to that suggested by the method are available on the market.

In order to determine the possible range of the flow coefficient (Kv) values and the resulting consequences on the position control system, some simulations and experiments were accomplished using valves with different coefficient values.

The flow coefficient value of the selected valve is 2.48 L*/min.bar*$^{1/2}$ (1.31x10^{-7} *m*^{3}*/s.Pa*$^{1/2}$). Figure 10 shows the simulated results of the system, composed by cylinder and load (76 *Kg*), being controlled by valves with the following coefficient values:

- 1.00 L*/min.bar*$^{1/2}$ (0.52x10^{-7} *m*^{3}*/s.Pa*$^{1/2}$) – 40% of the selected flow coefficient value;
- 1.24 L*/min.bar*$^{1/2}$ (0.65x10^{-7} *m*^{3}*/s.Pa*$^{1/2}$) – 50% of the selected flow coefficient value;
- 2.48 L*/min.bar*$^{1/2}$ (1.31x10^{-7} *m*^{3}*/s.Pa*$^{1/2}$) – selected flow coefficient value;
- 4.96 L*/min.bar*$^{1/2}$ (2.62x10^{-7} *m*^{3}*/s.Pa*$^{1/2}$) – 200% of the selected flow coefficient value.

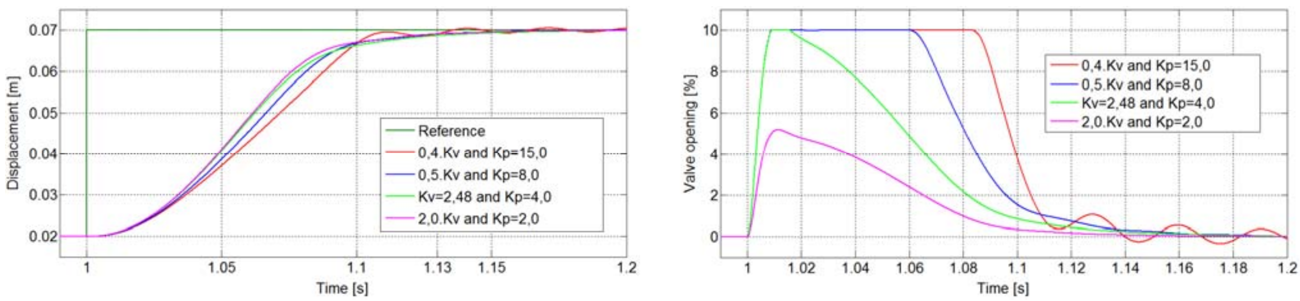

Figure 10. Simulated step response with different flow coefficient valves

The system met all specifications once the controller is tuned to obtain the best response. However, the smaller valve that can be used has a flow coefficient twice lower than the specified value (2.48 *L/min.bar*$^{1/2}$). A flow coeffcient lower than 1.24 *L/min.bar*$^{1/2}$ implies selecting valves that need high controller proportional gains, resulting in great periods of valve saturation (completely open) and tendency of oscillations on the actuator movement.

The use of a valve with flow coefficient value twice greater (4.96 *L/min.bar*$^{1/2}$) than the specified value results in smaller proportional gains and no valve saturation. Larger valves than that do not show performance improvements and are, in practice, more slow and expensive.

3.2. Valve natural frequency

The activity 7 of the static and dynamic sizing consists in determining the minimum value for the valve natural frequency that control the electro-hydraulic system designed. The Bosch Rexroth valve selected has ω_n^V nearly 440 *rad/s*, 9.5 times greater than the system natural frequency (ω_n^{SYS} = 46.15 *rad/s*).

Aiming to evaluate experimentally the system behavior using valves with different ω_n^V and being impossible to change in practice the valve dynamic performance, an emulation block was used in the closed control loop implemented in Simulink®. Inserted between the control signal and the command signal, this block emulates the performance of slower spools and represents valves with natural frequencies lower than that selected. Figure 11 shows the system experimental results using different values for the valve natural frequency.

The experiments were done without load and spring coupled to ensure a higher actuator natural frequency and, consequently, the cylinder dynamic does not interfere on system performance. Therefore, only the valve dynamic behavior influences on the system and can be analyzed on Fig. 11.

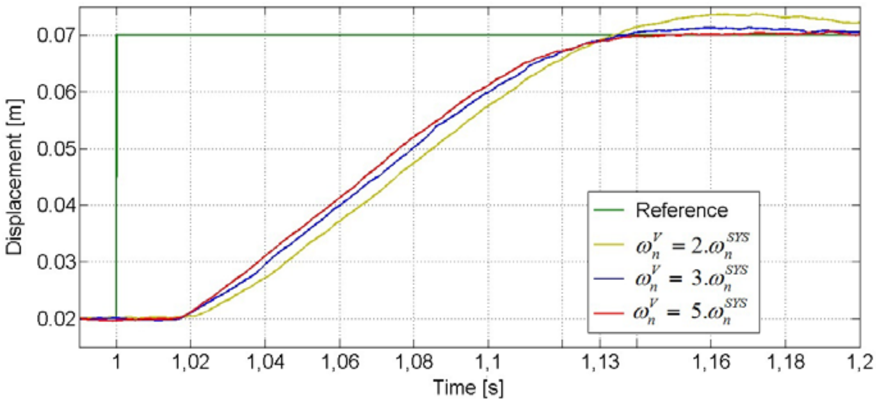

Figure 11. Experimental step response with different natural frequency valves

Only the positioning system with the valve of $\omega_n^V = 5.\omega_n^{SYS}$ presents a response without overshoot as required. Therefore, valves with natural frequency values greater than five times the system natural frequency ensure a good quality of the cylinder movement.

3.3 Actuator natural frequency

The minimum natural frequency of the selected cylinder with a load of 76 *Kg* coupled is 220 *rad/s*. This value is approximately 4.8 times greater than the system natural frequency (46.15 *rad/s*).

According to Eq. 5, the variation of actuator natural frequency value depends in practice of the coupled mass and of the fluid volume confined between the valve and cylinder. Therefore, aiming to analyze the behavior and application limits of the hydraulic system when the cylinder dynamic response gets closer of the system dynamic response, pipes were used to increase the fluid volume between valve and cylinder.

However, the use of pipelines is impractical in PHP considering the necessity to install very long pipes with low pressure drop to achieve meaningful results. Thus, this dynamic behavior analysis are made only via simulation using values of actuator natural frequency near to the system natural frequency, as shown in Fig. 12. In order to avoid that the valve dynamic interferes on the system response, the model was configured with an instantaneous response valve.

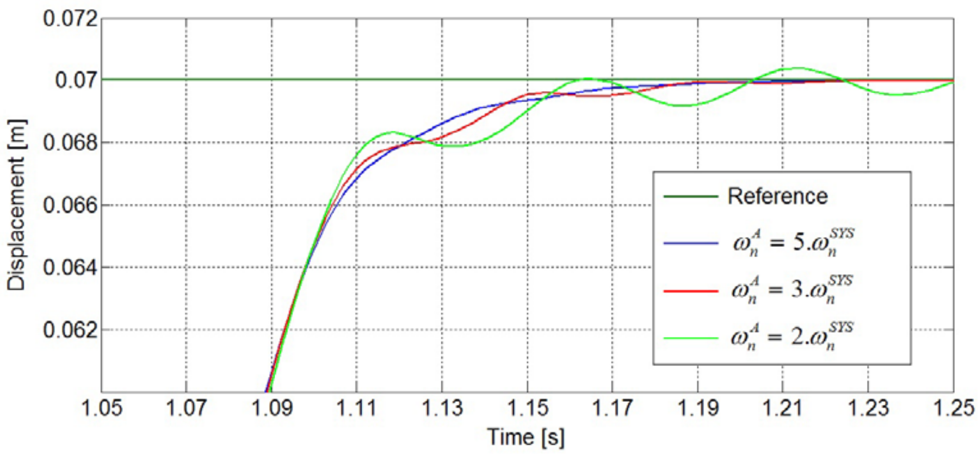

Figure 12. Simulated step response with different natural frequency actuators

The behavior of the systems with the ratios $\omega_n^A = 2.\omega_n^{SYS}$ and $\omega_n^A = 3.\omega_n^{SYS}$ has some oscillations at the end of the movement and is related to pressure fluctuations in the cylinder chambers. This phenomenon arises from the combination of the mass inertial effects and the fluid compressibility. In some cases, it may causes cavitation and, consequently, several equipment damages (Szpak et al., 2009).

4. CONCLUSION

The results demonstrated that the method is a useful tool for the hydraulic positioning system design where the selected components were adequate for the task as expected. Moreover, the simulation successfully aided in tuning the controller and achieving the required performance of the several components before the system was assembled, with no risks to equipment or personnel.

Even though the same performance can be obtained with different sets of components just through the controller adjustment, the hydraulic positioning system has dynamic limits that should be taken into account. Several performance problems cannot be solved only tuning the controller, also requiring of the designer a deep knowledge about the parameter influence on designing an electro-hydraulic system.

As discussed in this paper, the directional proportional valve to be used in a hydraulic positioning system must have at least a flow coefficient two times lower ($Kv_{\min} = 0.5 \cdot Kv$) and maximal two times greater ($Kv_{\max} = 2.0 \cdot Kv$) than the optimal flow coefficient. The choice of valves with coefficients outside this limit range results in unsatisfactory performance and the system does not meet the desired specifications.

In addition, the analysis presented in this paper point out the importance to establish a relationship between the natural frequencies of the valve, actuator, and the system. Aiming an optimal system performance, it is necessary to size a valve with a natural frequency of at least 5 times the system natural frequency ($\omega_n^V = 5 \cdot \omega_n^{SYS}$) and design a hydraulic system that ensures that the actuator natural frequency is also at least 5 times the system natural frequency ($\omega_n^A = 5 \cdot \omega_n^{SYS}$). If the system presents an actuator natural frequency lower than the established relationship it is necessary to change the configuration of the system (load, cylinder, and/or pipes) or to increase the specified settling time.

5. ACKNOWLEDGEMENTS

The authors thank CNPq (the Brazilian National Council for Scientific and Technological Development) for the funding and support provided for this project.

6. REFERENCES

De Negri, V.J., Ramos Filho, J.R.B. and Souza, A.D.C.. "A design method for hydraulic positioning systems". 51th National Conference on Fluid Power (NCFP) in conjunction with IFPE, Las Vegas, USA. Proceedings of USA: NFPA, 2008, pp. 669-679.

Furst, F.L., 2001, "Sistematização do projeto preliminar de circuitos hidráulicos de controle de posição". Master's Thesis (POSMEC), UFSC, Florianópolis, Brazil, 149 p.

Guan, C. and Pan, S., 2008, "Adaptive sliding mode control of electro-hydraulic system with nonlinear unknown parameters". Control Engineering Practice, Cambridge, v. 16, n. 11, Nov. 2008, pp. 1275-1284.

Johnson, J.L., 1995, "Design of electrohydraulic systems for industrial motion control". Ed. Parker Hannifin Corporation, Milwaukee, USA, 386 p.

Kalyoncu, M. and Haydim, M., 2009, "Mathematical modeling and fuzzy logic based position control of an electrohydraulic servosystem with internal leakage". Mechatronics, Cambridge, v. 19, n. 6, Sep. 2009, pp. 847-858.

Muraro, I., 2010, "Estudo das características comportamentais de válvulas proporcionais e seus efeitos nos posicionadores eletro-hidráulicos". Master's Thesis (POSMEC), UFSC, Florianópolis, Brazil, 202 p.

Szpak, R., Ramos Filho, J.R.B., Belan, H., De Negri, V.J.. "Theoretical and experimental study of the pressure behavior on hydraulic positioning systems". 20th International Congress of Mechanical Engineering (COBEM 2009), Gramado. Proceedings of Rio de Janeiro: ABCM, 2009.

7. RESPONSIBILITY NOTICE

The authors Muraro, I., De Negri, V.J., Belan, H.C. and Ramos Filho, J.R.B. are the only responsible for the printed material included in this paper.