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EFFECT OF PROPORTIONAL VALVES AND CYLINDERS ON THE BEHAVIOR OF HYDRAULIC POSITIONING SYSTEMS

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ABSTRACT

Hydraulic positioning control systems are widely employed in several engineering fields such as industry, aerospace, vehicles, and electrical power plants. However, their design is not a straightforward engineering task because several configurations and sizes of valves and cylinders are available, the system components exhibit nonlinear behavior, and the aspects of both fluid mechanics and control theory need to be included for achieving a suitable design. Furthermore, each application has static and dynamic requirements that need to be fulfilled under uncertain loading conditions. Dynamic simulation is an important tool for the analysis and design of hydraulic positioning systems; however, the main characteristics of the components should be known beforehand so that the parameters and model structure can be defined. To overcome these constraints, comprehensive knowledge of the design problem is necessary to ensure appropriate sizing of the hydraulic components. In this regard, this paper presents a detailed study of the influences of the natural frequency and flow coefficient of the valves. The actuator natural frequency is also analyzed, and its modification according to the system requirements is described. The influence of these parameters on the behavior of a closed-loop hydraulic control system with a proportional controller is evaluated using a detailed mathematical model implemented in MATLAB®/Simulink®. Model validation is accomplished using a workbench.

INTRODUCTION

Electro-hydraulic positioning control systems are mainly used to drive and control higher loads with reliability, speed, and accuracy. These characteristics promote their prevalence in hydraulic systems encompassing several fields of engineering.

A proportional directional control valve or servovalve is one of the most important components in hydraulic positioning control systems. This valve is responsible for accurate flow control, and therefore, controls the force, position, speed, and acceleration of the actuator [1].

Meanwhile, due to phenomena such as the pressure dependence of the valve flow and variations in the control volume and hydraulic stiffness, hydraulic systems exhibit substantial nonlinear dynamic behavior. This important characteristic results in difficulties in control and induces uncertainties in the design, modeling, and parameterization of such systems [2].

Continuous technological progress has been insufficient in resolving issues regarding selection, sizing, and control of the actuation system (comprising valves and cylinders) such that a guaranteed procedure can be adopted by designers. This fact is particularly true for the several dynamic and static behavioral characteristics of valves and their effects on the electro-hydraulic positioning systems.

Oscillations, noise, and inadequate responses are unwanted behavior in several hydraulic equipment, as reported by Johnson [3]; these characteristics are detrimental to the applicability of such systems. That author suggests that a well-designed hydraulic system should be consistent, reliable, durable, smooth, and silent. Therefore, a careful design incorporating effective sizing and selection of components is required.

However, the design and assembly of such systems are not trivial due to the necessity of including the aspects of both fluid mechanics and control theory. In addition, the static and dynamic requirements for a specific application depend considerably on the loading conditions, which are often poorly understood by designers [4].

Procedures involving a comprehensive view of the design problem are necessary to provide designers with complete domain knowledge about the necessary technical decisions. This process should ensure the appropriate sizing of the hydraulic components including the overall requirements.

In this regard, this study discusses the influence of the significant parameters of valves and cylinders on the behavior of the positioning system, including the natural frequency and flow coefficient of the valves and the natural frequency of the cylinder and load.

The determination of the initial values of the parameters is carried out by using the design method developed at the Laboratory of Hydraulic and Pneumatic Systems (LASHIP/EMC/UFSC); the design effort is categorized into three main steps: static and dynamic sizing of the system, conversion of the cataloged data, and dynamic behavior study [4] [5]. In this design process, the industrial components are selected such that the values of their static and dynamic parameters are conventionally different from their calculated values. This paper contributes toward the understanding of parametric tolerance when selecting the hydraulic components, thereby ensuring that the predicted static and dynamic behaviors are achieved.

Furthermore, a new approach that can be used to calculate the maximum load of a system considering the maximum negative acceleration instead of the maximum positive value is discussed. As shown later, despite the fact that the theoretical maximum acceleration is not achieved by an actual positioning system, it is possible to achieve the required settling time.

The influence of natural frequency and flow coefficient of valves as well as the actuator natural frequency are evaluated when considering a closed-loop hydraulic control system using a proportional (P) controller. Theoretical and experimental results are obtained after implementing the system model in MATLAB®/Simulink® and validated using a workbench.

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

WORKBENCH AND SIMULATION MODEL

The experimental tests were carried out using a proportional hydraulic platform (PHP), as shown in Figure 1.

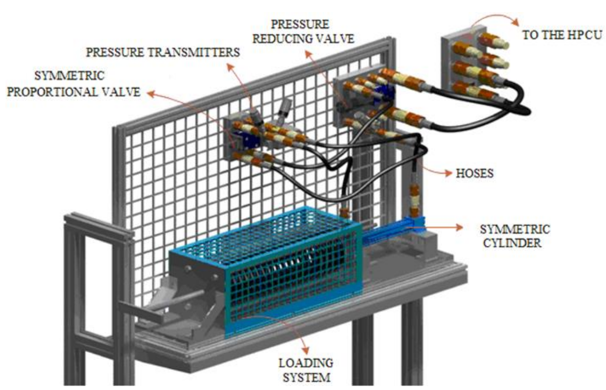

Figure 1 - PHP workstation [6]

The PHP workstation includes a hydraulic power conditioning unit (HPCU) controlled by a programmable logic controller (PLC), VXI data acquisition system, pressure and displacement transducers, PC running LabVIEW® and MATLAB®, and two workstations. The loading system, pressure-reducing and proportional directional valves, cylinders, and other components are assembled in the workstation to emulate the actual conditions in which the system under test will be used.

A nonlinear model representing the positioning system with a P controller and loading system was implemented in MATLAB®/Simulink®. This model comprises blocks based on the transfer functions that represent each component of the assembled closed-loop system. 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 Figure 2 represents the sub-model of a component: the complete model is described in Muraro [7].

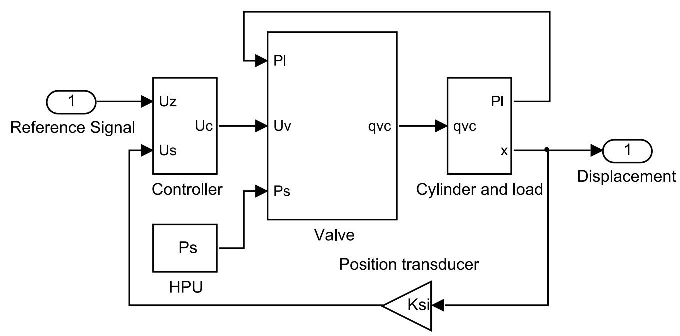

Figure 2 - Simulink® model of the positioning system [8]

The cylinder and load block was designed according to the experiments executed on the PHP to evaluate the involved friction forces. The controller implemented in MATLAB is the same proportional controller deployed in the PHP by LabVIEW®. Furthermore, this model also represents the load loss, fluid mass acceleration, and fluid compressibility in the tubes.

DESIGN METHOD OF HYDRAULIC POSITIONING SYSTEMS

As discussed in De Negri et al. [4], the design method employed in this paper is a part of the embodiment design phase of a system. Before this method can be implemented, clarifications of both the task and conceptual design phases need to be carried out to generate the initial version of the design requirements and the system concept; these are necessary for the execution of the embodiment phase. These pieces of information can be obtained from professionals in other areas who understand the process or the equipment where the system is going to be installed. Figure 3 shows a brief representation of the design method as a flowchart.

In Annex A, the first and second steps in the case of a symmetric four-way valve with symmetric cylinder sets are described. The descriptions of the methods for the asymmetric four-way valves with asymmetric cylinder sets and three-way valves with single-effect cylinder sets are described in De Negri et al. [4]. The first step yields the desired characteristics of the valve and cylinder, namely, the nominal flow rate, cylinder area, natural frequencies of both valve and cylinder, valve pressure drop under different load conditions, and selected supply pressure.

By means of the second step, commercial hydraulic components are selected, where the component parameters will not match exactly with those in the first step. The topics discussed in the present paper aim to provide the guidelines for the evaluation of a suitable. range of valve and cylinder parameters and to understand their influence on the system response.

The third and final step, as shown in Figure 3, involves studying the dynamic behavior using simulation programs, e.g., MATLAB®, Simulink®, AMESim®, or HOPSAN®. The influences of long pipes, nonlinear friction, variable loads, valve dead zone, hysteresis, etc. are exhaustively analyzed in this step.

An alternative procedure to compute the valve and cylinders parameters obtained in first and second step would be the application of the method presented in Johnson [3].

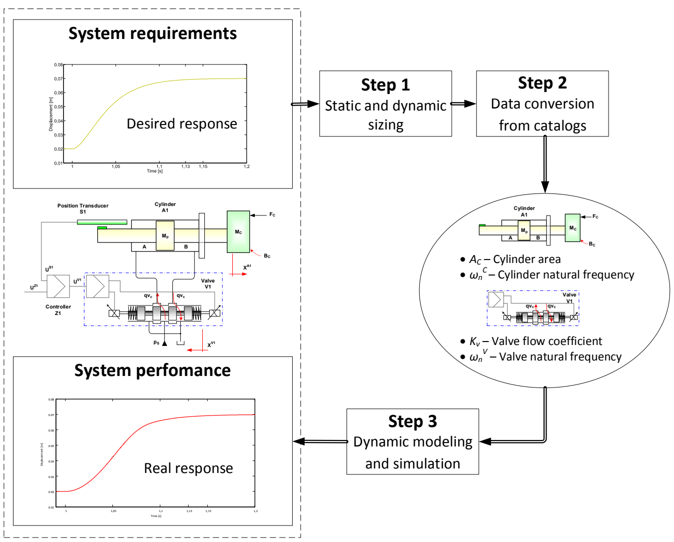

Figure 3 – Flowchart of processes in the design method

PARAMETERS AND OPERATIONAL CONDITIONS

The closed-loop system requirements are listed in Table 1; based on these values, the positioning system parameters are calculated, as well as the effect of parameter variations can be determined.

The parameters listed in Table 2 were obtained by employing the static and dynamic sizing procedures presented in Annex A.

A Bosch Rexroth CGT3MS2 25/18/200 with a rod diameter of 18 mm and bore of 25 mm was used as the hydraulic cylinder. The annulus area of this rod is 2.364 cm^2, which is close to that calculated using the proposed method. It was also defined a directional proportional valve that yielded the desired flow (2×10^{-4} m^3/s) at a pressure drop of 2.33 MPa. Therefore, based on the required flow coefficient of Kv = 2.48 L/min.bar$^{1/2}$ (1.31×10^{-7} m^3/s.Pa$^{1/2}$) obtained using the design method, the Bosch Rexroth valve 0 811 404 038 is the closest fit. Table 3 lists the cylinder and valve parameters.

Table 1 - Requirements of the positioning system

Requirement	*Value*
t_s^{SYS} [*s*]	0.13
$x_{\max}^C$ [*m*]	0.05
Overshoot?	No
p_S [*Pa*]	7×10^6
M_t [*kg*]	10.0
K [*N/m*]	2618.4

Table 2 - Parameters obtained from the design method

Parameter	*Value*
ω_n^{SYS} [rad/s]	46.15
$v_{\max}^C$ [m/s]	0.849
$a_{\max,p}^C$ [m/s^2]	106.5
$p_{LP\max}$ [Pa]	4.67×10^6
A^C [m^2]	2.28×10^{-4}
$qv_{\max}$ [m^3/s]	2×10^{-4}

The closed-loop system analyzed in the next section uses a proportional controller: due to the stroke cushioning at the cylinder end, the step signal used in the tests starts at 20 mm and goes up to 70 mm during a specified time interval of 130 ms.

Table 3 - Component data of the elementary system

Parameter	*Value*
Cylinder rod diameter [m]	18×10^{-3}
Cylinder bore diameter [m]	25×10^{-3}
Cylinder annulus area [m^2]	2.364×10^{-4}
Valve flow coefficient [m^3/s.Pa$^{1/2}$]	1.31×10^{-7}

MODEL VALIDATION

The hydraulic positioning system model proposed in this paper was validated by comparing the results obtained from the simulation with the experimental responses obtained using the abovementioned PHP.

The closed-loop hydraulic system under analysis comprised the hydraulic components mentioned earlier and the following setup:

- Reference signal: step of 50 mm;
- Forward displacement: from 20 to 70 mm;
- Return displacement: from 70 to 20 mm;
- Controller proportional gain: $K_p = 7$;
- Spring pre-load: 440 N.

Figure 4 shows the theoretical and experimental results, which show reasonable agreement.

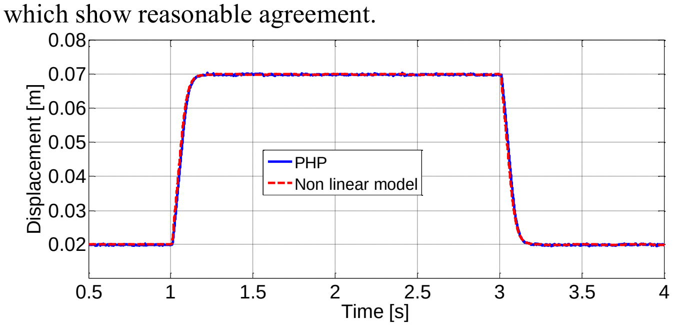

Figure 4 – Cylinder position using a valve with K_v = 2.68 and spring load

Furthermore, Figure 5 illustrates the effectiveness of the model to represent the pressure dynamics during cylinder movement. Despite the presence of high-frequency experimental signals induced by electrical noise, the transient response is effectively described. Further validation results are available in Muraro [7].

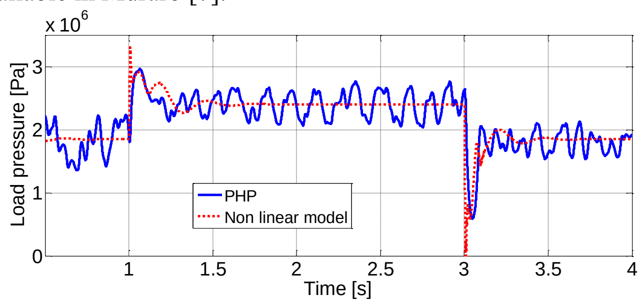

Figure 5 – Load pressure using valve with K_v = 2.68 and spring load

MAXIMUM ACCELERATION

The methods of designing positioning hydraulic systems described in Furst [5] and Furst et al. [4], summarized in Annex A, require the calculation of the maximum load pressure or the cylinder area while considering the maximum positive acceleration that occurs when a step input is applied, i.e., at t = 0 s.

Therefore, as calculated in the earlier section, when Eq. (1) is used, the cylinder area needed to move a mass of 10 kg is at least 2.28 cm^2 (Table 2) for achieving the system parametric requirements specified in Table 1.

$$a^C_{\max,p} = x^C_{\max} \cdot \omega_n^{SYS\,2} \tag{1}$$

However, as discussed in Muraro [7], due to nonlinearities inherent in the hydraulic system, it is possible to define the cylinder annulus area using the negative maximum acceleration as follows:

$$a^C_{\max,n} = -x^C_{\max} \cdot \omega_n^{SYS\,2} \cdot e^{-2} \tag{2}$$

As evident from Eq. (3) and shown in Figure 14 (Annex A), while maintaining the hydraulic force and reducing the maximum acceleration, a greater value of maximum mass that the system can position is identified as follows:

$$A^C . p_L = M_t . a^C_{\max} \tag{3}$$

Therefore, using the maximum negative acceleration and the actual cylinder area of 2.36 × 10^{-4} m^2 (Table 3), the maximum mass that the system can move is calculated as 76.5 kg instead of 10.3 kg.

To verify whether the requirement of settling time is still met in the presence of the increased mass load, the system responses using the validated model and the different masses are analyzed while using the same controller proportional gain. As shown in Figure 6, despite the decreased rise time with the increased mass, the settling time of 130 ms is achieved under both the conditions.

Figure 7 shows the cylinder acceleration under different loads. The actuator with lower mass accelerates for 18 ms and has the maximum positive acceleration of 122 m/s^2. Then, the cylinder starts decelerating until the movement ends at the maximum negative value of 26 m/s^2. On other hand, the movement with higher mass has a longer period (55 ms) with positive acceleration and has the same absolute value of 20 m/s^2 for the maximum positive and negative accelerations.

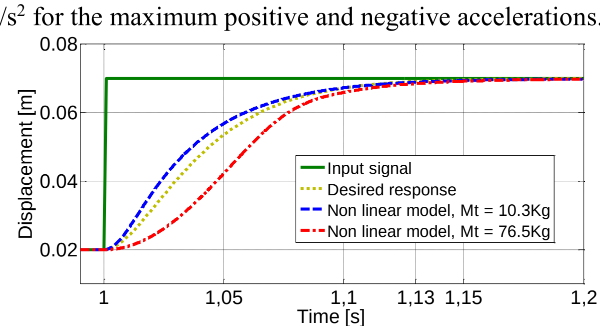

Figure 6 – Cylinder position with load masses of 10.3 and 76.5 kg

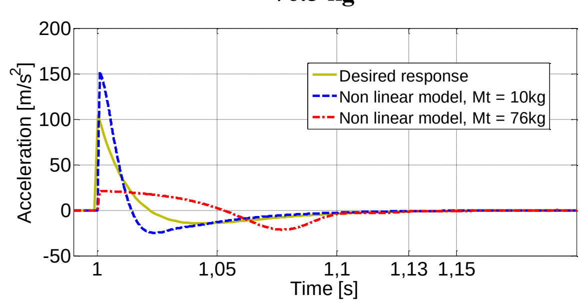

Figure 7 – Cylinder acceleration with load masses of 10.3 and 76.5 kg

These figures reveal the similarities between the reference model (desired response) and the system response under the load of 10.3 kg with reference to the acceleration and velocity referred in the design method. On the other hand, the simulation response with a mass of 76.5 kg differs from the other cases since the maximum value of positive acceleration is not achieved.

It also was noted that the valve does not saturate with lower load mass, reiterating the fact that this condition is independent of this nonlinearity. However, a higher load results in the maximum control signal for a much longer duration, compensating the lower positive acceleration; therefore, the design requirements are met [7].

Hence, it can be concluded that the maximum negative acceleration can be used in the system design. To highlight the difference between the cylinder areas that are attributable to these distinct approaches, it is considered designing a positioning system with a load mass of 76.5 kg and the other requirements presented in Table 1; however, the maximum positive acceleration was used. As a result, a cylinder area of at least 17.47 × 10^{-4} m^2 would be needed, which is considerably greater than that used throughout this section.

VALVE FLOW COEFFICIENT

As previously mentioned, because of the static and dynamic sizing methods, a suitable flow coefficient is calculated for the proportional valve selection. However, it is not always possible to procure commercially available valves having characteristics and parameters identic to those recommended by the proposed method.

To determine the possible range of flow coefficient (K_v) values and the resulting responses of the positioning control system, a few simulations and experiments were undertaken using valves having different sizes.

As shown in Table 3, the flow coefficient of the selected valve is 2.48 L/min.bar$^{1/2}$ (1.31 × 10^{-7} m^3/s.Pa$^{1/2}$). Figure 8 shows the simulated results of the system comprising a cylinder and a load (76.5 kg) controlled by valves with different coefficients (listed in Table 4).

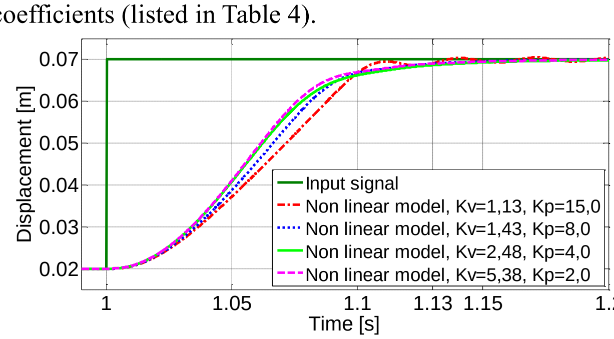

Figure 8 - Simulated displacement responses with different flow coefficients and total mass of 76.5 kg

The proposed system met all the specifications as soon as the controller proportional gain was tuned to obtain the required settling time. However, the smaller valves that could be used in the system had flow coefficients that were two times lower than the specified values. A flow coefficient lower than 1.24 L/min.bar$^{1/2}$ yields valves that need higher controller proportional gains, resulting in larger periods of valve saturation (completely open) and tendency of oscillations in the actuator movement (Figure 9).

Table 4 – Valve flow coefficients

Flow Coefficient (Kv)		Relative to selected value
L/min.bar$^{1/2}$	×10^{-7} m^3/s.Pa$^{1/2}$	%
1.13	0.60	~ 40
1.43	0.76	~ 50
2.48	1.31	Specified value
5.38	2.84	~ 200

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

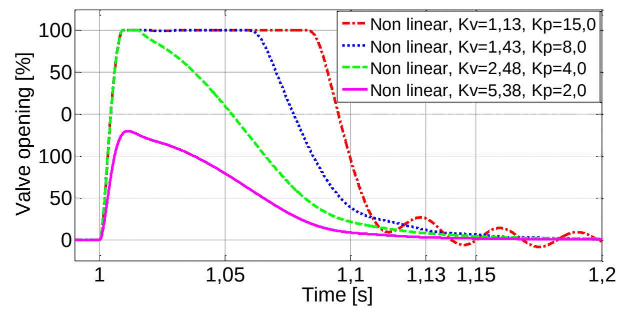

Figure 9 - Simulated valve-opening responses with different flow coefficients

These conclusions are confirmed from the simulation results, where the system response for a range of flow coefficients is analyzed in the absence of load (Figure 10). The calculated flow coefficient using the design method is 1.67 L/min.bar$^{1/2}$, where the behavior is analyzed by considering coefficients that are either half or double the above value.

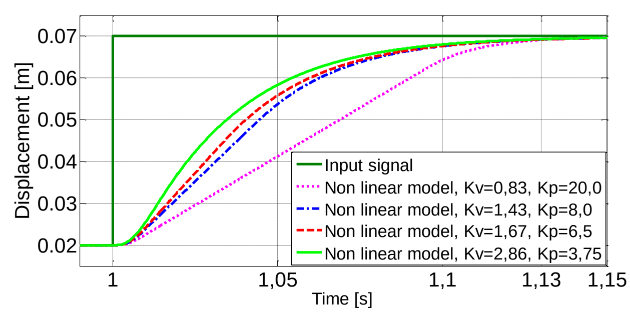

Figure 10 - Simulated displacement response with different flow coefficients and no load

VALVE NATURAL FREQUENCY

The design method presented in Annex A determines the minimum value of the valve natural frequency that controls the designed electro-hydraulic system. The selected Bosch Rexroth valve has a natural frequency of approximately 440 rad/s, which is almost 9.5 times greater than the system natural frequency (46.15 rad/s).

For experimentally evaluating the system behavior using valves having different natural frequencies (the valve dynamics cannot be modified in practice), an emulation block of the closed control loop was implemented in Simulink®. Inserted between the control and command signals, this block emulates valves with natural frequencies lower than the selected values. Figure 11 shows the experimental results described.

The experiments were undertaken without the load mass as well as the coupled spring to ensure a higher natural frequency of the actuator such that the cylinder dynamics do not interfere with the system performance.

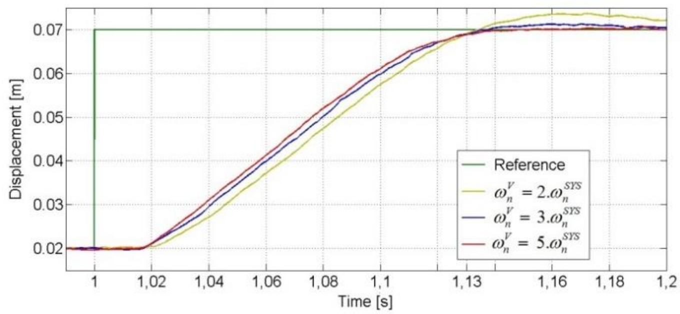

Figure 11 - Experimental step response for different valve natural frequencies

It can be observed that only the positioning system with the valve having a natural frequency of $\omega_n^V = 5.\omega_n^{SYS}$ exhibits the desired response without overshoot. Therefore, valves having natural frequencies greater than five times the system natural frequency ensure the desired response of the positioning system. However, values between three and five times the system natural frequency can yield an acceptable system response in several positioning applications.

ACTUATOR NATURAL FREQUENCY

The minimum natural frequency of the selected cylinder with a coupled load of 10 kg is 607 rad/s. This value is approximately 13.1 times greater than the system natural frequency (46.15 rad/s). Nevertheless, with the proposed new load of 76 kg, the minimum natural frequency of the cylinder is 220 rad/s or almost 4.8 times the system natural frequency (46.15 rad/s).

According to Eq. (15), variations in the actuator natural frequency, in practice, depend on the coupled mass and fluid volume confined between the valve and the cylinder. Therefore, to analyze the behavior and application limits of the hydraulic system when the cylinder dynamic response gets closer to the system dynamic response, extended pipes were considered for increasing the fluid volume between the valve and the cylinder.

This dynamic analysis was made via simulation using the values of the actuator natural frequency close to the system natural frequency, as shown in Figure 12. To avoid the influence of the valve dynamics and controller tuning on the system response, the model was configured with an instantaneous response valve and fixed proportional gain of 8.

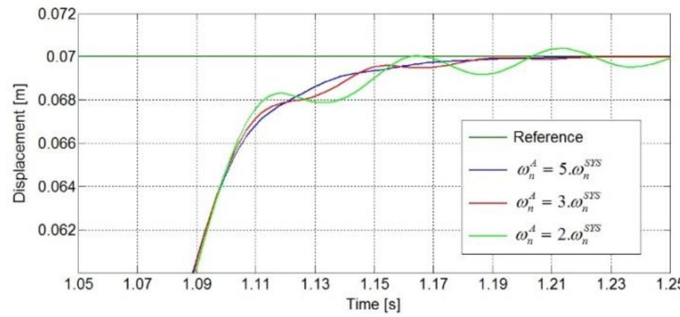

Figure 12 - Simulated step response for different actuator natural frequencies

The behavior of the systems with the ratios $\omega_n^A = 2.\omega_n^{SYS}$ and $\omega_n^A = 3.\omega_n^{SYS}$ exhibited certain oscillations at the end of the movement and can be attributed to the pressure fluctuations in the cylinder chambers. This phenomenon arises from the combination of the inertial effect and fluid compressibility. In some cases, it can cause cavitation and, consequently, equipment damage [9].

CONCLUSION

This paper analyze the influence of valve and cylinder parameters as well as the maximum acceleration that should be taken into account during the design of a hydraulic positioning system. The study background is a design method, briefly discussed in this paper, which the optimal valve flow coefficient, cylinder area, and natural frequencies of the valve and cylinder are determined.

Even though the same performance can be obtained using different sets of components by only making controller adjustments, the dynamic limits of the hydraulic positioning system need to be taken into account. Several performance problems cannot be resolved only by tuning the controller; problem resolution also requires designers to have suitable acumen regarding the influences of different parameters on the electro-hydraulic system design.

As discussed in this paper, the directional proportional valve used in a hydraulic positioning system should have a flow coefficient at least two times lower ($Kv_{min} = 0.5\,.Kv_{opt}$) and at the most two times greater ($Kv_{max} = 2\,.Kv_{opt}$) than the optimal flow coefficient. The selection of valves with coefficients outside this limited range results in unsatisfactory performance, and the system does not meet the desired specifications.

In addition, the analysis presented in this paper indicates the importance of the correlation between the natural frequencies of the valve, actuator, and system. To achieve the optimal system performance, it is necessary to select an appropriately sized valve with a natural frequency of at least five times the system natural frequency ($\omega_n^V = 5\,.\omega_n^{SYS}$) and design a hydraulic system that ensures that the actuator natural frequency is also at least five times the system natural frequency ($\omega_n^A = 5\,.\omega_n^{SYS}$). If the system has an actuator natural frequency lower than the established relationship, it is necessary to modify the system configuration (load, cylinder, and/or pipes) or increase the specified settling time.

This paper also describes the manner in which the system design can be carried out considering the maximum negative acceleration instead of the maximum positive acceleration in the system step response. Since the deceleration is lower than the acceleration, it results in the reduction of the cylinder size.

NOMENCLATURE

$a^C_{\max}$	Cylinder maximum acceleration
$a^C_{\max,n}$	Cylinder maximum negative acceleration
$a^C_{\max,p}$	Cylinder maximum positive acceleration
A^C	Cylinder area
B_t	Viscous friction coefficient
F_u	Useful force
K_L	Load stiffness
K_p	Proportional gain
Kv	Flow coefficient
M_t	Total mass
p_A	Pressure in chamber A
p_B	Pressure in chamber B
p_L	Load pressure
$p_{L\max}$	Maximum load pressure
$p_{LP\max}$	Load pressure at maximum output power
p_S	Supply pressure
$qv^C_{\max}$	Cylinder maximum flow rate
t_s^{SYS}	System settling time
t_s^V	Valve settling time
v^C	Cylinder speed
$v^C_{\max}$	Cylinder maximum speed
V_{t}	Total volume between valve and cylinder
$x^C_{\max}$	Maximum displacement (desired)
β_{e}	Effective fluid bulk modulus
Δp_t	Total pressure drop
ω_n^A	Actuator natural frequency
ω_n^C	Cylinder natural frequency
ω_n^V	Valve natural frequency
ω_n^{SYS}	System natural frequency
ζ^{SYS}	System damping ratio

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ANNEX A

DESIGN METHOD FOR HYDRAULIC POSITIONING SYSTEMS

In the following sections, two out of the three steps in the design method discussed in De Negri et al. [4] and Furst [5] are described in the case of a symmetric four-way valve with symmetric cylinder sets.

Step 1 – Static and dynamic sizing

This step considers the dynamic and static requirements of the closed-loop system, comprising the proportional valve, cylinder, load, and controller. The parametric data include the maximum displacement, settling time, forces, overshoot, and so on; the mathematical expressions derived from the dynamic model of the system are used.

As a result, the valve and cylinder specifications are achieved. These components should be simultaneously selected since their parameters are interdependent. The activities can be executed according to the flowchart shown in Figure 13.

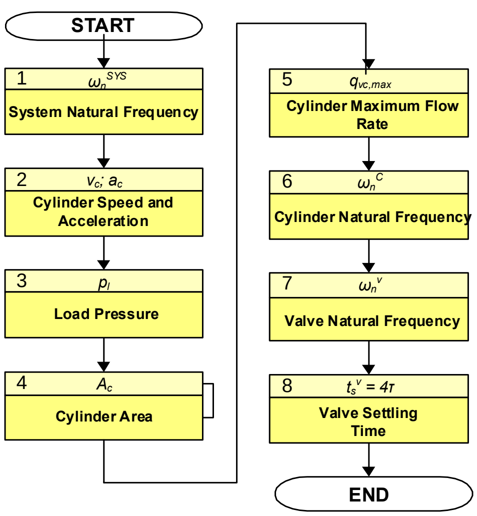

Figure 13 – Flowchart of static and dynamic sizing

Activity 1: Undamped system natural frequency – This activity begins by analyzing the global system behavior, instead of each of its constituent components. If the system can afford overshoot, it will be treated as a second-order system with a damping ratio (ζ^{SYS}) equal to 0.7. If the overshoot is unfeasible, a second-order system with $\zeta^{SYS} = 1$ needs to be employed. Based on this information and the desired system settling time (t_s^{SYS}), it is possible to calculate the undamped system natural frequency (ω_n^{SYS}) as follows:

For an under-damped system,

$$\omega_n^{SYS} = \frac{4}{\zeta^{SYS} t_s^{SYS}}, \text{ with } \zeta^{SYS} = 0.7 \tag{4}$$

For a critically damped system,

$$\omega_n^{SYS} = \frac{6}{t_s^{SYS}} \tag{5}$$

Activity 2: Cylinder speed and acceleration – Based on the information obtained from activity 1 and using the desired steady-state displacement for the positioning system (x^C_{max}), the maximum speed (v^C_{max}), maximum positive acceleration ($a^C_{max,p}$), and maximum negative acceleration ($a^C_{max,n}$) can be calculated as follows.

For an under-damped system,

$$v^C_{\max} = x^C_{\max} \cdot \omega_n^{SYS} \cdot e^{-\zeta^{SYS}\omega_n^{SYS} t} \text{ with } t = \frac{1}{\omega_n^{SYS}\sqrt{1-\zeta^{SYS2}}} \tan^{-1}\frac{\sqrt{1-\zeta^{SYS2}}}{\zeta^{SYS}} \tag{6}$$

$$a^C_{\max,p} = x^C_{\max} \cdot \omega_n^{SYS2} \tag{7}$$

$$a^C_{\max,n} = -x^C_{\max} \cdot \omega_n^{SYS2} \cdot e^{-2\frac{\zeta^{SYS}}{\sqrt{1-\zeta^{SYS2}}}\tan^{-1}\frac{\sqrt{1-\zeta^{SYS2}}}{\zeta^{SYS}}} \tag{8}$$

For a critically damped system,

$$v^C_{\max,} = x^C_{\max} \cdot \frac{\omega_n^{SYS}}{e} \tag{9}$$

$$a^C_{\max,p} = x^C_{\max} \cdot \omega_n^{SYS2} \tag{10}$$

$$a^C_{\max,n} = -x^C_{\max} \cdot \omega_n^{SYS2} \cdot e^{-2} \tag{11}$$

Activities 3 and 4: Load pressure and cylinder area – For a system constituting a symmetric proportional four-way valve and a symmetric cylinder (SV–SC), the load pressure (p_L) is defined as

$$p_L = p_A - p_B \tag{12}$$

The load pressure at the maximum output power (p_{LPmax}) for a SV–SC system is described in [10]

$$p_{LP\max} = \frac{2}{3} p_S \tag{13}$$

where the supply pressure (p_S) is a design requirement defined earlier.

It is important to note that the expected load pressure (p_L) should not be higher than the load pressure at the maximum output power (p_{LPmax}).

Considering that the cylinder area (A^C) is the annulus area, F_U is the useful force applied to the load, K_L 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

(p_{Lmax}) should be calculated under the following three operational conditions:

- When the cylinder is 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 acceleration (a^C_{max}).

After the required area has been defined, a commercially available cylinder should be selected from the catalog data with an area equal to or greater than the area calculated using this activity. On the other hand, if a commercial cylinder is already available, the equations above are solved not with regard to the area, but with regard to the determination of p_{Lmax} to verify if the supply pressure is sufficiently high such that the pressure does not exceed two-thirds of the maximum value. If the supply pressure is too low, the cylinder area should 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 reuse the available cylinder. Figure 14 describes the method to determine the load pressure and cylinder area.

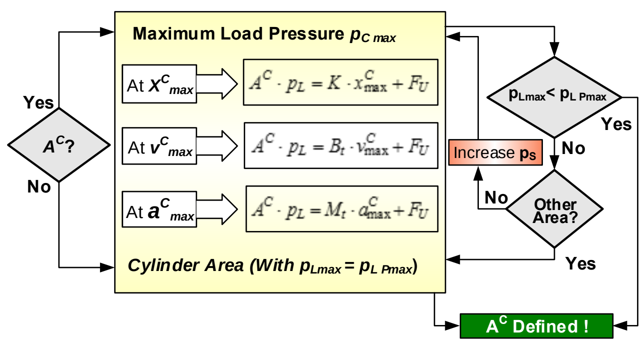

Figure 14 - Flowchart of Activities 3 and 4

Activity 5: Cylinder maximum flow rate – Subsequently, when the cylinder area is defined, designers can determine the maximum flow rate for the cylinder ($qv^C_{\max}$) based on its maximum speed, that is:

$$q^C_{V\max} = A^C \cdot v^C_{\max} \tag{14}$$

Activity 6: Undamped cylinder natural frequency – The next activity is to calculate the natural frequency of the cylinder associated with the load (ω^C_n). For a symmetrical cylinder comprising a symmetrical four-way valve, this value can be evaluated as

$$\omega^C_{n\min} = \left(\frac{4\beta e \cdot A^{C^2}}{M_t \cdot V_t} \right)^{1/2} \tag{15}$$

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 or not.

Activity 7: Undamped valve natural frequency – The valve natural frequency (ω^V_n) should be at least five times the system or cylinder natural frequency (whichever has the lowest value) to avoid undesired system responses.

Step 2 – Data conversion from catalogs

A valve with similar characteristics to the one specified in Step 1 should be found among those available in 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 coefficient (Kv) as defined in Johnson [3] is used, which can be easily calculated from the catalogs.

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, the total pressure drop at the valve (Δp_t) can be calculated as

$$\Delta p_t = p_S - |p_{L\max}| \tag{16}$$

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

$$Kv = \frac{qv^C_{\max}}{\sqrt{\Delta p_t}} \tag{17}$$