A Simplified Hydraulic Capacity-Sensitive Fluid Dynamics Numerical Model for Monitoring Aerospace Electro-Hydraulic Actuators

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Abstract. Detailed models are necessary to analyze individual components or subsystems when designing modern flight control systems. However, simpler models with sufficient accuracy are needed for preliminary layout, monitoring, diagnostics, or prognostic issues. Various simplified numerical solutions are available in the literature to simulate the fluid dynamic behaviors of a given valve geometry. These models typically calculate the differential pressure the valve regulates based on its spool opening and flow rate. In some specific applications, these models are unsuitable, requiring new simplified fluid dynamic models that calculate the flow rate delivered by the valve based on the spool displacement and differential pressure. This paper introduces a new synthetic fluid-dynamic valve model that considers the effects of spool position, hydraulic capacity, variable supply pressure, and leakage between the output ports that connect the valve to the motor element. Its advantages and disadvantages are evaluated by comparing it with other simplified numerical algorithms available in the literature, analyzing the corresponding fluid-dynamic characteristics, and comparing the behaviors simulating a typical flight control servomechanism.

Keywords: Aerospace systems, EHA, Digital twin, Hydraulic capacity, Lumped parameters, Monitoring, Non-linear modelling, Servovalve, Simplified fluid dynamic numerical model.

1 Introduction

Most commercial and military aircraft use electrohydraulic actuation systems (EHAs) in their powered flight controls. They represent a well-established technology and have a very high power density, allowing designers to create lightweight designs that can fit into the small places inside the aerodynamic surfaces of the aircraft. To meet the stringent safety standards of aviation regulations, proper redundancies and real-time health monitoring strategies [1-3] are required since flight controls are one of the most safety-critical systems of aircraft [4]. In particular, considering its nonnegligible failure rates and the potentially catastrophic criticalities deriving from these failures [5], the electrohydraulic servovalves (EHSVs) equipping these actuators typically require very effective and reliable monitoring algorithms. The level of detail provided by the various simulation models is inevitably correlated to multiple factors, among which the field of application of a given algorithm and the related performance requirements stand out. Therefore, the literature proposes several numerical models able to simulate the fluid dynamic behaviors of a given valve geometry with different levels of accuracy and fidelity. The design and development phases of EHA often requires accurate and high-fidelity simulations (especially regarding fluid-dynamics behaviors) capable of evaluating the system performance within its operating range [5]. In this regard, Urata has conducted extensive research on various fault modes, including internal valve leakages [6], eddy currents [7-9], asymmetric torque motor airgaps [10-12], and fringing [13]. In-depth simulations of the hydraulic and electrical performance of electrohydraulic servovalves (EHSVs) using finite electromagnetic elements and computational fluid-dynamics are proposed by Chen et al. [14], Henninger et al. [15] and Yang et al.[16]. These approaches, nevertheless, are typically expensive in terms of processor workload and computing time, making them inappropriate for some activities whose execution is time- or resource-constrained. Indeed, in activities such as the preliminary design of the actuation systems or the conception of real-time algorithms (e.g. monitoring and diagnostic routines), the main difficulty is maintaining a reasonable accuracy level while minimizing the corresponding computing load. In these cases, it is necessary to develop EHSVs simplified models designed and suitably set up for specified operations, e.g. preliminary design optimization or diagnostic/prognostic techniques, to be used as a digital mirror of the corresponding physical components. In general, these simplified models calculate the differential pressure regulated by the valve as a function of its spool opening and the flow rate disposed of by the valve itself [17-18]. However, models with differential pressure output are inadequate in some specific applications (e.g. asymmetric hydraulic jacks, regenerative actuators, or hydraulic transmissions in which the effect of fluid compressibility is not negligible). In these cases, it is necessary to adopt new simplified fluid dynamic models that, starting from the same hypotheses and data, calculate the flow rate delivered by the valve as a function of the spool displacement and the differential pressure acting downstream of the valve (i.e. on the output ports connecting it with the hydraulic actuator). Thus, this paper proposes a new synthetic fluid-dynamic valve model (a lumped parameters model with a semi-empirical formulation) accounting for the effects of spool position, hydraulic capacity, variable supply pressure and leakage between the output ports that connect the valve to the motor element.

2 Considered electro-hydraulic actuator

The numerical models proposed in this work have been developed referring to a typical electrohydraulic servoactuator architecture typically adopted in aerospace systems: i.e. a four-way control valve (supply port S, return port R, control port 1, control port 2) coupled with a symmetrical linear jack, as shown in Figure 1.

More specifically, we aim to create a computationally light model for prognostic applications [2], focusing on the second-stage sliding spool valve. This is because its highly nonlinear behavior is responsible for most of the computational burden in high-fidelity, CFD-based models [19-21]. In addition, it has been shown that strong linearity assumptions on the spool operation may noticeably degrade the accuracy of the whole servoactuator model in some operating conditions [22].

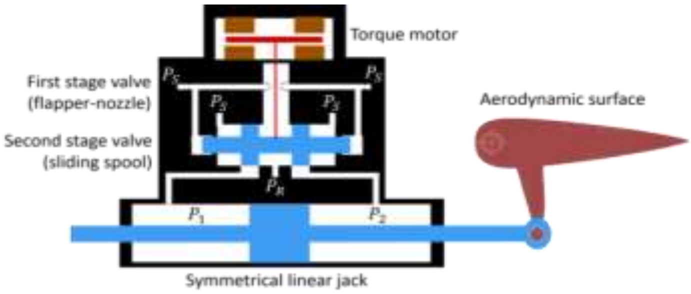

Fig. 1. Schematic of the considered electrohydraulic actuator

As shown in the schematic of Figure 1, the valve spool displacement x_S controls the areas of the four passageways, each characterized by its overlaps or underlap, to connect each control port either to the supply or return port. This allows modulation of the hydraulic power regulated by the aforementioned piloting edges, expressed in terms of flow and absolute pressure for each control port (P1 and P2), for specific characteristics of the oil and operational conditions [23-24]. The corresponding differential pressure, regulated between the two control ports is $P_{12} = P_1 - P_2$.

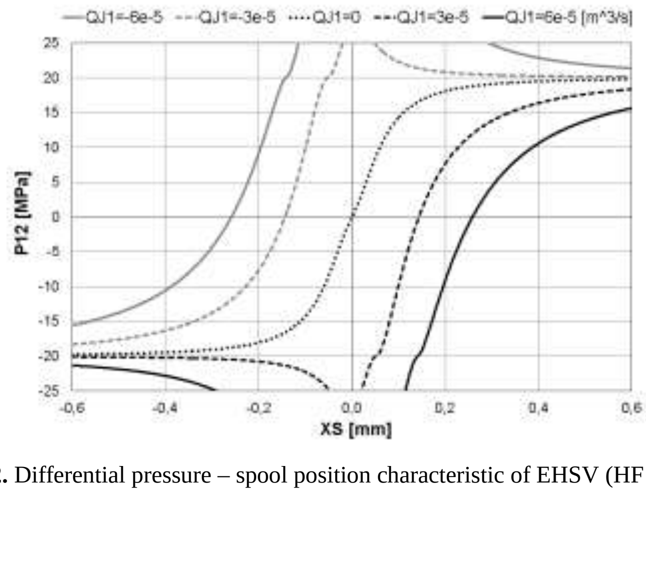

Fig. 2. Differential pressure – spool position characteristic of EHSV (HF model)

In zero-flow conditions, each control port absolute pressure is close to the supply or return pressure when the corresponding passageway is fully opened. When the spool is in an intermediate position, the control port pressures have a progressive evolution between return pressure (P_R) and supply pressure (P_S) values, as it can be seen in the valve characteristic P_{12} - x_S of Figure 2.

3 EHSV fluid-dynamic HF model

As reported in previous chapter, in this work, a high-fidelity fluid dynamic model is being used as a reference to develop lower-fidelity emulators. The high-fidelity (HF) model calculates the pressure drops across each valve passageway to determine the flow rates and pressures crossing the valve for a given spool position. This information is essential in understanding the behavior of the system. The model computes the map shown in Figure 2: for a null flow rate (i.e. $Q_J = 0$) and small spool displacements, the differential pressure $P_{12} = P_1 - P_2$ varies approximately linearly with the spool position x_S. Large values of x_S result instead in the saturation of the differential pressure to $P_{SR} = P_S - P_R$, where P_S is the supply pressure and P_R is the return line pressure. The presence of a flow rate $Q_J \neq 0$ offsets the response of the valve as an additional pressure drop is caused by the restricted flow through the small passageways of the valve. The combination of large flow rates and small spool displacements can result in a differential pressure higher than P_{SR}. The water hammer effect is a phenomenon that significantly impacts the behaviour of servovalves.

However, the HF model is too expensive to use in real-time dynamical simulations because it requires an iterative process to solve pressure drops at each time-step. Moreover, it heavily relies on various parameters, such as the geometry of the SV and the physical properties of the hydraulic fluid. All those variables, needed to correctly set the model for the simulation of a physical system, often are not directly available cannot be measured with sufficient accuracy. For example, the model is highly sensitive to the exact value of the internal clearances between the spool and sleeve, which can be affected by manufacturing tolerances and even by elastic deformations due to pressurization of the system. A full description of the HF model is available in [5, 25].

4 EHSV fluid-dynamic LF models available in literature

As previously reported, the HF model is generally computationally heavy and time-consuming, and is sensitive to several parameters, related to the SV geometry and the physical characteristics of the hydraulic fluid, that often are not directly available or cannot be measured with proper accuracy. Therefore, in a simpler, lighter and quicker approach, only the controlled differential pressure between the two control ports P_{12} and a single flow value Q_J (common to both control ports) are computed with linearized models [26]. When representing the hydraulic characteristic of servovalves in Low Fidelity (LF), it's common to use a local linearization of the flow-pressure-displacement map. This is usually done near the zero flow and closed valve condition.

By doing this, the operation of an electrohydraulic actuator can be described in the regulating regime, far from the effects of saturations and other nonlinear phenomena. To achieve this, a two-gains linear form is often employed:

$$\frac{P_{12}}{G_P} + \frac{Q_J}{G_Q} - x_S = 0 \tag{1}$$

where G_P is the pressure gain and G_Q is the flow gain. Fundamentally, two alternative formulations can be considered: one solves for differential pressure P_{12} with flow rate Q_J as feedback (pressure formulation), while the other solves for flow rate Q_J with differential pressure P_{12} as feedback (flow rate formulation).

Often, this former simplified model is expressed as:

$$P_{12} = G_P \cdot \left(x_S - \frac{Q_J}{G_Q}\right) \tag{2}$$

The spool displacement produces a proportional differential pressure value, which acts on the motor element (i.e. the linear jack). This displacement is reduced by the pressure loss caused by the controlled flow passing through the control passageways; this effect is accounted for with the flow gain.

It should be noted that, in general, the linear models used to describe hydraulic actuators, cannot capture several phenomena that can significantly affect their operation, such as leakages, limited supply pressure, and water hammer. For this reason, several simplified models have been proposed in the literature to extend the linear formulation and overcome its limitations.

The following is a brief overview of some simplified models previously developed by the authors, while Figure 3 [23] summarizes the block diagram representations of these models.

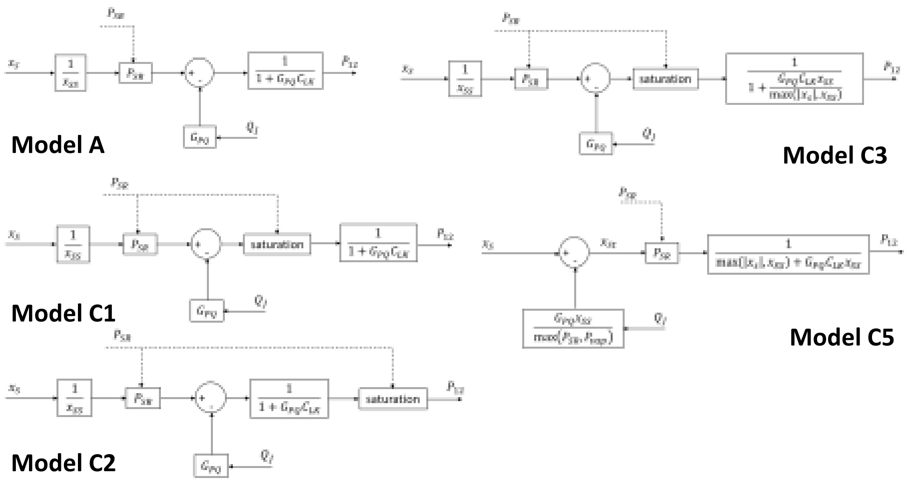

Fig. 3. Block diagrams of models A, C1, C2, C3 and C5

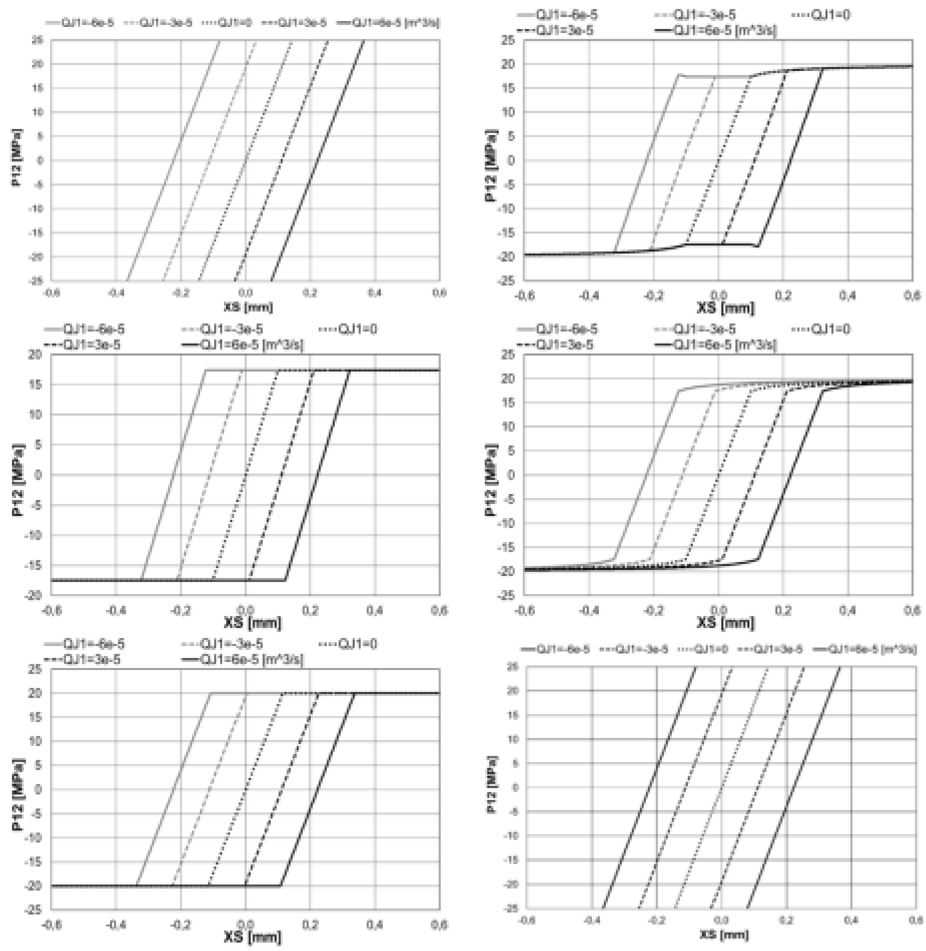

Fig. 4. Fluid-dynamic characteristic of Model A (top left), Model C1 (middle left), Model C2 (bottom left), Model C3 (top right), Model C5 (middle right) and new LF model (bottom right)

Model A [26] accounts for variable differential supply pressure P_{SR} and leakage flow:

$$P_{12} = \left(\frac{x_S}{x_{SS}} P_{SR} - Q_J G_{PQ}\right)\frac{1}{1+G_{PQ}C_{lk}} \tag{3}$$

where $G_{PQ} = G_P/G_Q$. The formulation is still fully linear, although the coefficients are not constant and are corrected with respect to Equation (2).

Model C1 [17, 25] extends the applicability of Model A to large spool displacements, by introducing a saturation of the differential pressure to the supply. The saturation is placed upstream the leakage estimation:

$$P_{12} = \frac{1}{1+G_{PQ}C_{lk}}\left[sgn\left(\frac{x_S}{x_{SS}} P_{SR} - Q_J G_{PQ}\right) max\left(P_{SR}, \left|\frac{x_S}{x_{SS}} P_{SR} - Q_J G_{PQ}\right|\right)\right] \tag{4}$$

In **Model C2** [26] the pressure saturation is moved downstream the leakage flow estimation: this is to correct the underestimation of maximum differential pressure observed in Model C1. The formulation of Model C2 is modified as follows:

$$P_{12} = \mathrm{sgn}\left[\frac{1}{1+G_{PQ}C_{lk}}\left(\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right)\right]\max\left(P_{SR}, \left|\frac{1}{1+G_{PQ}C_{lk}}\left(\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right)\right|\right) \quad (5)$$

Model C3 [27] is an alternative approach to correct the behaviour of Model C1: the pressure saturation is accounted for within the leakage computation block:

$$P_{12} = \frac{1}{1+\frac{G_{PQ}C_{lk}\,x_{SS}}{\max(|x_S|,x_{SS})}}\left[\mathrm{sgn}\left(\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right)\max\left(P_{SR}, \left|\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right|\right)\right] \quad (6)$$

Model C5 [28] introduces an equivalent spool-position $\boldsymbol{x_{St}}$ taking into account the effects of variable P_{SR} and oil flow Q_J drained across the valve:

$$x_{st} = x_S - \frac{Q_J(G_{PQ}x_{SS})}{\max(P_{SR},P_{vap})} \quad (7)$$

$$P_{12} = x_{St}P_{SR}\,\frac{1}{\max(|x_S|,x_{SS}) + G_{PQ}C_{lk}\,x_{SS}} \quad (8)$$

Figure 4 shows the characteristic maps of the models reported above (representing the P_{12} regulated as a function of the spool displacement $\boldsymbol{x_{St}}$ and parameterized according to the discharged flow rate Q_J). For brevity, the authors omitted the detailed description of these maps. For more details, the readers should refer to the related works reported in the literature.

5 Proposed hydraulic capacity-sensitive LF model

All the models shown in the previous chapter calculate the differential pressure the valve regulates based on its spool opening and flow rate. In some specific applications, these models are unsuitable, requiring new simplified fluid dynamic models that calculate the flow rate delivered by the valve based on the spool displacement and differential pressure. Therefore, this section introduces a new synthetic fluid-dynamic valve model that considers the effects of spool position, hydraulic capacity, variable supply pressure, and leakage between the output ports that connect the valve to the motor element. As understandable by the schematic shown in Fig. 5, this model derives directly from Eq. (1), expressed as a function of the instantaneous flow rate Q_J.
Similarly to what has already been seen regarding Model A, a linear formulation is adopted (i.e. no saturations on the maximum value of the P_{12} pressure regulated downstream of the spool) that is based on the already known valve gains (in particular $GPQ = GP/GQ$) and it is sensitive to the hydraulic line differential pressure P_{SR}. It should be noted that unlike models that have differential pressure as the output variable (see Figure 4), the fluid control valve followed by a hydraulic capacity can be modeled without problems when a leak acts between the control ports, connecting the valve the same as the driving element, is present.

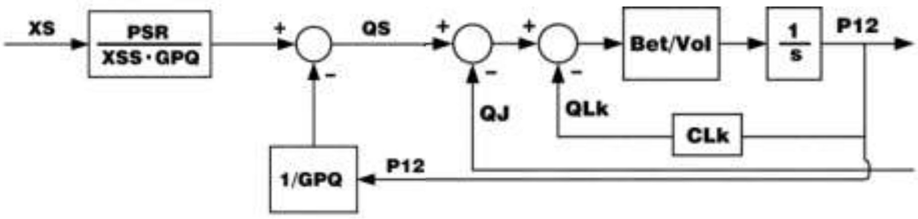

Fig. 5. Block diagram of the proposed hydraulic capacity-sensitive LF model (Model Aq)

The leakage feedback loop cannot be instantaneous because the hydraulic capacity introduces a first-order lock within the loop, as in the right part of Figure 5, to prevent numerical instabilities. The proposed computational algorithm considers all these factors and simulates effects due to variations in flow rates, pressure gains, and supply pressure through a simplified approach. It is used to compute the fluid-dynamic characteristic of the EHSV and simulate the dynamic behavior of a typical onboard EHA. The algorithm's abilities have been verified by comparing the results with those obtained from the previously published HF model by the authors.

6 Numerical results of the EHA test bench

The proposed simplified model has been integrated into a virtual test bench, which simulates a full position control electro-hydraulic actuator (EHA) consistent with this described in Sec. 2. Its behaviors have been compared with a corresponding numerical model equipped with the HF fluid dynamic model introduced in Sec. 3.

This detailed model considers the key electrical, hydraulic, and mechanical features of all relevant system components, which include inertia, the hydraulic piston dry and viscous friction, and a third-order electromechanical model of the first and second-stage dynamics of the EHSV. Figures 6 and 7 show the simulated dynamic responses of the EHA models under a proper combination of position commands (Com), external loads (FR), and changes in the hydraulic supply pressure (PSR).

The input sequence has been defined to highlight the performance of the fluid-dynamic models and their impact on the behavior of the simulated EHA [18].

Figure 6 shows the dynamic response of the full actuator implementing the HF fluid dynamic model of the valve. It can be used as a reference for comparison with the corresponding response of the proposed simplified model.

On the other hand, Figure 7 illustrates the response of the proposed model (Aq): it can simulate with acceptable accuracy the behaviours of the reference model for the conditions of unloaded actuation (from 0 to 0.2 s) and the aiding-load condition (at 0.75 s) despite a slightly higher starting acceleration and a lower stopping deceleration (related to the model inability to properly account for the water hammer effect).

In the event of opposing load actuation at 0.3 seconds, the Aq model underestimates the impact of the external force FR. Therefore, when the supply pressure PSR decreases (at 0.35 seconds), the back movement of the jack is also underestimated.

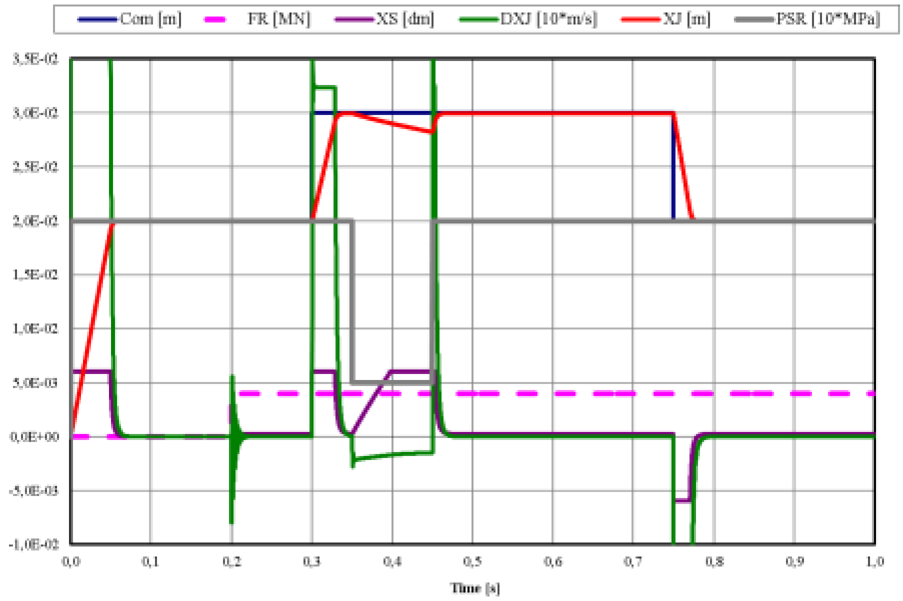

Fig. 6. Test bench simulation with the HF model; Com - position command, FR - external load, XS - spool position, DXJ - jack speed, XJ - jack position, PSR - supply pressure

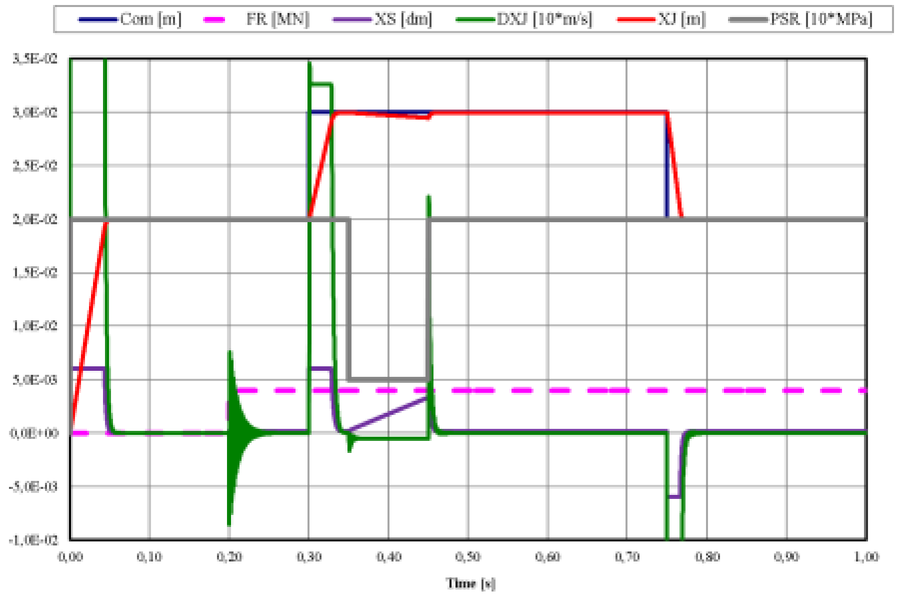

Fig. 7. Test bench simulation with the Aq model; Com - position command, FR - external load, XS - spool position, DXJ - jack speed, XJ - jack position, PSR - supply pressure

7 Conclusions

In this paper, a new simplified fluid dynamic model of EHSV sensitive to the compressibility of the hydraulic fluid was developed and tested. The model could overcome most of the problems associated with previous low-fidelity emulators and approximate the behavior of an HF simulation with reasonable accuracy (including pressure oscillations, fluid compliance, and the effects of leakage losses). At the same time, compared to the HF reference or well-known CFD codes, the computational time/effort is relatively reduced and, therefore, promising for real-time applications (especially preliminary design and diagnostic monitoring) since the formulation only requires the calculation of simple algebraic and trigonometric functions, without the need for iterative solvers. However, given that the starting model (Model A) is highly simplified, the accuracy and reliability of the proposed model are not yet compatible with applications such as model-based prognostics of hydraulic systems. Therefore, the authors aim to continue this research by implementing new simplified fluid dynamic models based on the much more performing (but complex) type C models.

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