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A NEW SERVOVALVE SIMPLIFIED FLUID DYNAMIC MODEL SENSITIVE TO SYSTEM HYDRAULIC CAPACITIES

Alessandro Aimasso* Parid Alimhillaj** Matteo Bertone* Carlo G. Ferro* Matteo D. L. Dalla Vedova*

* Department of mechanical and aerospace engineering, Politecnico di Torino, Italy

** Energy Department, Faculty of Mechanical Engineering, Polytechnic University of Tirana, Albania

ABSTRACT

Designing modern flight control systems requires highly detailed models to analyse individual components or subsystems. However, for tasks like preliminary design, monitoring, diagnostics, or prognostics, simpler yet sufficiently accurate models are necessary. Literature offers various simplified numerical solutions that can simulate the fluid dynamic behaviours of specific valve geometries with varying levels of detail and accuracy. These simplified models usually calculate the differential pressure regulated by the valve based on its spool opening and the flow rate it manages. In certain applications, such as asymmetric hydraulic jacks, regenerative actuators, or systems where fluid compressibility is significant, models focused solely on differential pressure are insufficient. Instead, new simplified fluid dynamic models are needed to determine the flow rate delivered by the valve as a function of spool displacement and differential pressure. This paper introduces a new synthetic fluid-dynamic valve model - a lumped parameters model with a semi-empirical formulation - that considers spool position, hydraulic capacity, variable supply pressure, and leakage between the output ports connecting the valve to the motor element.

Keywords: Flight control systems, Servovalve, Simplified numerical model, Synthetic fluid-dynamic valve model

1 INTRODUCTION

Electrohydraulic actuation systems (EHAs) are commonly used in commercial and military aircraft for powered flight controls. These systems are a mature and proven technology known for their high-power density, allowing designers to develop lightweight components that blend perfectly into the tight spaces within an aircraft's aerodynamic surfaces. To achieve the stringent safety standards imposed by aviation legislation, it is vital to build robust redundancies and real-time health monitoring procedures [1-3], as flight controls are among the most safety-critical systems [4]. Given the high failure rates and potentially catastrophic consequences of these failures, the Electrohydraulic Servovalves (EHSVs) utilized in these actuators require extremely effective and dependable monitoring algorithms [5].

Contact author: Matteo Dalla Vedova

Corso Duca Degli Abruzzi 24, 10129, Torino, Italy.
E-mail: matteo.dallavedova@polito.it

The level of detail supplied by these monitoring methods is fundamentally tied to a variety of criteria, the most significant of which are the specific application field and associated performance constraints. As a result, the extant literature includes many numerical models that recreate the fluid dynamic behaviours of certain valve designs, with varying degrees of precision and faithfulness. During the design and development phases of electrohydraulic actuators (EHAs), very realistic and faithful simulations are frequently required, particularly when measuring the system's performance over its operational range. To answer this demand, some researchers have conducted considerable study on various fault types, including internal valve leakages [6], eddy currents [7-9], hydraulic asymmetries. Furthermore, the literature [10-16] has thorough simulations of the hydraulic and electrical performance of EHSVs, which use finite electromagnetic elements and computational fluid dynamics. However, these approaches are costly in terms of processor usage and processing time, making them unsuitable for applications with strict time or resource constraints. Preliminary actuation system design and the creation of real-time algorithms, such as monitoring and diagnostic routines, are particularly challenging in terms of ensuring a sufficient level of accuracy while minimizing associated processing demands.

In such cases, it is required to develop simplified EHSVs models that are explicitly specialized for certain operations, such as early design optimization or diagnostic and prognostic procedures. To meet this need, some researchers have extensively explored various fault modes, including issues such as internal valve leakages [6], eddy currents [7-9], asymmetric torque motor airgaps [10-12], and fringing [13]. These are digital representations of the physical components that calculate the controlled differential pressure based on the spool opening and valve flow rate [17-18]. However, models with differential pressure output are insufficient for certain applications, such as asymmetric hydraulic jacks, regenerative actuators, and hydraulic transmissions, where fluid compressibility must be neglected. In these cases, using novel simplified fluid dynamics models is critical. Such models, which are based on the same underlying assumptions and data, calculate the flow rate delivered by the valve as a function of spool displacement and differential pressure acting downstream of the valve, specifically on the output ports that connect it to the hydraulic actuator.

This work presents a new fluid-dynamic valve lumped parameter numerical model based on a semi-empirical formulation that accounts for spool position, hydraulic capacity, changing supply pressure, and internal leakage.

2 ELECTRO-HYDRAULIC ACTUATORS FOR AIRCRAFT FLIGHT CONTROL SYSTEM

The numerical models reported in this paper were developed using a conventional EHA setup commonly seen in aerospace applications. Figure 1 shows a four-way control valve with supply port S, return port R, control port 1, and control port 2, all connected to a symmetrical linear jack. The authors intend to provide a computationally efficient model designed for diagnostic/prognostic applications [2], with a focus on the fluid-dynamic properties of the second-stage sliding spool valve. This is due to its highly nonlinear properties, which add significantly to the computational complexity of high-fidelity, CFD-based models [19-21]. It should be noted that adopting high linearity assumptions for the valve might frequently impair the overall accuracy of the EHA model [22].

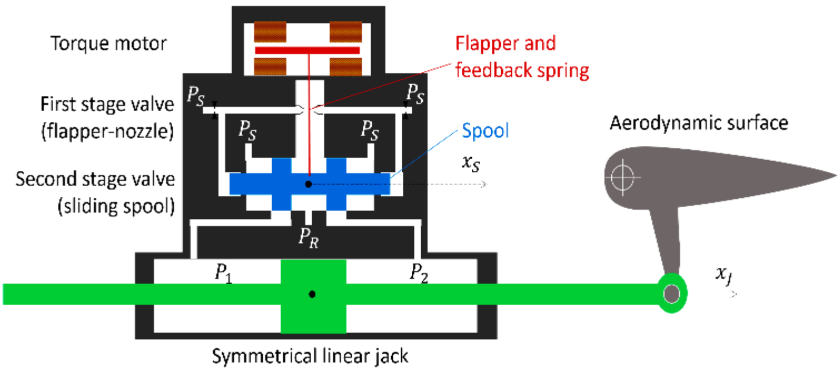

Figure 1 Schematic of the simulated electrohydraulic actuator.

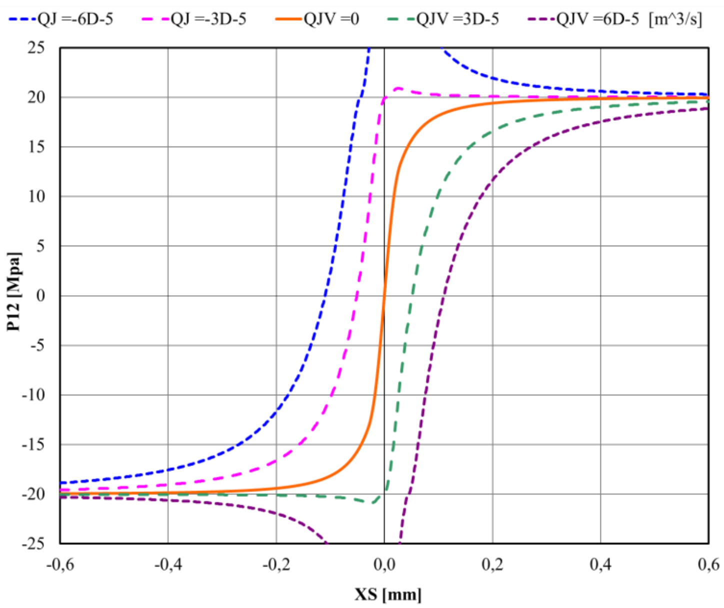

Figure 2 Differential Pressure (P12) vs Spool Position (XS) characteristic of EHSV (HF model).

According to the design in Figure 1, the movement of the valve spool, denoted as XS, determines the size of the four paths and their corresponding overlaps or underlaps, resulting in connections between each control port and either the supply or return port. This mechanism allows for the adjustment of hydraulic power, which is regulated by the aforementioned piloting edges and characterized in terms of flow and absolute pressure for each control port (P1 and P2) based on specific oil qualities and operational conditions [23-24]. The resulting differential pressure between the two control ports is indicated as P12 and calculated as P1 minus P2. Under zero-flow conditions, the absolute pressure at each control port nearly resembles the supply or return pressure when the tunnel is completely open. As the spool moves into an intermediate position, the pressures at the control ports gradually transition between return pressure (PR) and supply pressure (PS), as shown in Figure 2 with the valve's characteristic curve P12 vs XS.

3 ELECTROHYDRAULIC SERVOVALVE FLUID-DYNAMIC MODELS

3.1 HF MODEL

As previously stated, this study uses a high-fidelity fluid dynamic model as a baseline to create lower-fidelity emulators. The high-fidelity (HF) model calculates pressure differentials across each valve channel to establish flow rates and pressure values for a particular spool position, providing crucial information about system behaviour. The model produces the graph shown in Figure 2, in which, for insignificant flow (QJ = 0) and slight spool displacements, the differential pressure P12 = P1-P2 has an essentially linear relationship with the spool position XS. However, high XS values cause the differential pressure to saturate at PSR=PS-PR, where PS is the supply pressure and PR is the return line pressure. Adding a non-zero flow rate (QJ ≠ 0) changes the valve's reaction, as restricted flow via the narrow passageways causes an additional pressure drop. This, in turn, can provide a water-hammer event with differential pressure greater than PSR, especially when high flow rates are combined with minor spool displacements. Nonetheless, using the HF model in real-time dynamic simulations is problematic due to its high computational cost, necessitating an iterative strategy to resolve pressure drops at each time step. Furthermore, it is strongly dependent on a number of elements, including the physical properties of the hydraulic fluid and the geometry of the SV (servo valve). Many of the critical variables needed to effectively configure the model to replicate a physical system are frequently unavailable or cannot be determined with enough precision. For example, the model is more sensitive to the precise values of internal clearances between the spool and sleeve, which might be modified by manufacturing tolerances or even elastic deformations caused by system pressurization. The full description of the HF model can be found in [25].

3.2 SIMPLIFIED EHSV FLUID-DYNAMIC MODELS IN LITERATURE

The HF model is often computationally complex and time-consuming, and it is sensitive to a variety of parameters related to SV geometry and hydraulic fluid properties. Often, these factors are unavailable or difficult to assess accurately. To achieve a simpler, more efficient, and faster approach, linearized models are used to compute only the controlled differential pressure between the two control ports (P12) and a single flow value (QJ) that is the same via both control ports [26]. To replicate the hydraulic characteristics of servovalves in Low Fidelity (LF) models, local linearization of the flow-pressure-displacement relationship is frequently used, particularly in near zero-flow and closed-valve conditions.

The dynamic response of an electrohydraulic actuator in regulation (i.e., away from saturations and other nonlinear phenomena) can be reproduced using the following two-gains linear form:

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

where GP represents the valve's pressure gain [Pa/m] and GQ represents its flow gain [m^2/s]. There are two possible formulations to consider: one involves solving for differential pressure P12 with flow rate QJ as feedback (also known as pressure formulation), and the other involves solving for flow rate QJ with differential pressure P12 as feedback (also known as flow rate formulation). The first (expressed in pressure) is commonly represented as:

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

The spool displacement generates a proportional differential pressure value, which influences the motor element dynamics. However, the pressure loss caused by the controlled flow traveling via the control passageways reduces this displacement (flow gain compensates).

Linear models that characterize hydraulic actuators frequently fail to capture with sufficient fidelity phenomena such as leakages, supply pressure fluctuations, and water hammers, resulting in much poorer overall performance and generality. As a result, numerous simplified models have been proposed to build on the linear formulation and overcome its constraints. Five of them (A, C1, C2, C3, and C5) are briefly described in the next section, with block diagrams summarized in Figure 3.

Model A [26] can account for variable differential supply pressure PSR and valve internal leakages (by means of the coefficient CLk):

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

The formulation remains entirely linear, but the related coefficients are not constant and are rectified in accordance with Eq. (2).

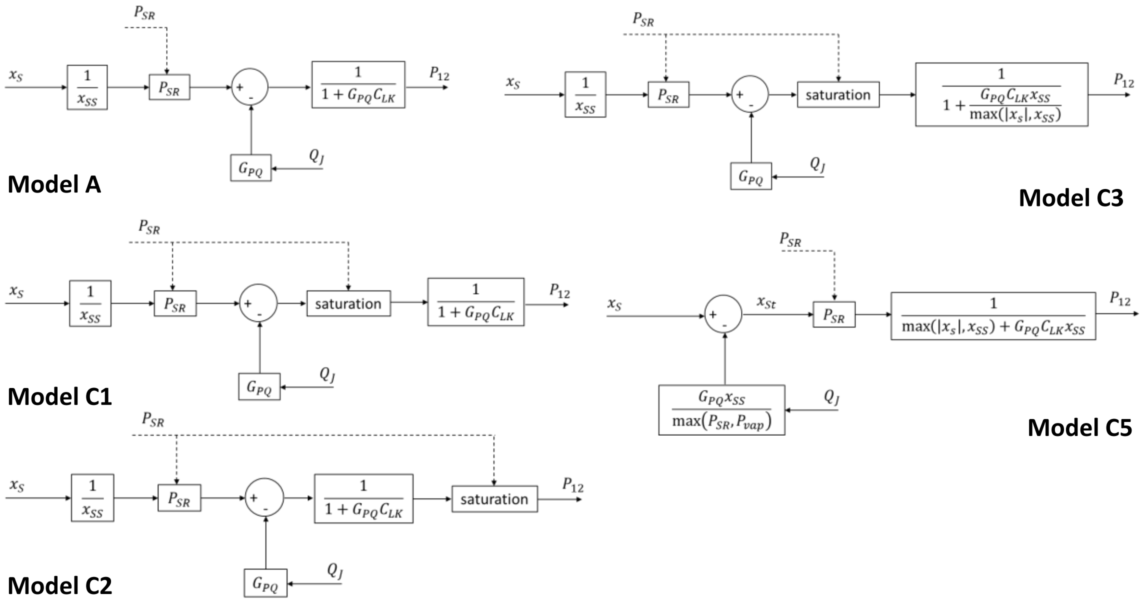

Figure 3 Block diagrams of models A, C1, C2, C3 and C5.

Model C1 [25] expands the Model A scope to include considerable spool displacements by restricting the controlled differential pressure to PSR. As shown in Fig 3, the saturation block is located upstream of the leakage block.

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

In Model C2, the pressure saturation is shifted to a downstream position in the leakage flow calculation [26]. This method rectifies the underestimation of differential pressure observed in Model C1

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

Model C3 provides an alternative model architecture for addressing the issues seen in Model C1.
It incorporates the pressure saturation directly into the block that calculates the internal leakage [27].

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

Model C5 overcomes the previous algorithms by introducing a nonlinear architecture that factors in the influences of variable PSR and the oil flow QJ introducing an equivalent spool position XSt [28]:

$$P_{12} = x_{St}P_{SR}\frac{1}{\max(|x_s|, x_{SS}) + G_{PQ}\,C_{lk}\,x_{SS}} \qquad (7)$$

$$x_{st} = x_S - \frac{Q_J(G_{PQ}x_{SS})}{\max(P_{SR}, P_{vap})} \qquad (8)$$

For brevity, the authors have omitted in this work the detailed description of the genesis of these models and their peculiar characteristics. For further details, the reader is referred to related literature.

4 NEW HYDRAULIC-CAPACITY SENSITIVE LF EHSV FLUID DYNAMIC MODEL

All the models outlined in the previous section calculate the valve's controlled differential pressure based on its spool opening and flow rate. However, in some applications, these models may not be appropriate. In such cases, simplified fluid dynamic models are required, which determine the flow rate provided by the valve using spool displacement and differential pressure. This section provides a novel synthetic fluid dynamic valve model that takes into consideration spool position, hydraulic capacity, changeable supply pressure, and leakage between the valve's output ports and the motor element. In Figure 4 is reported This new algorithm, known as Model C5q, was created by rewriting Eq. (1) as a function of the regulated flow rate QJ and incorporating the non-linear architecture and measures introduced with Model C5 (i.e., saturation of regulated pressure P12, plus counter-reaction in flow rate, leakage, and valve gains depending on operating conditions). This method is based on the valve gains (GPQ = GP/GQ) and responds to the hydraulic line differential pressure PSR.

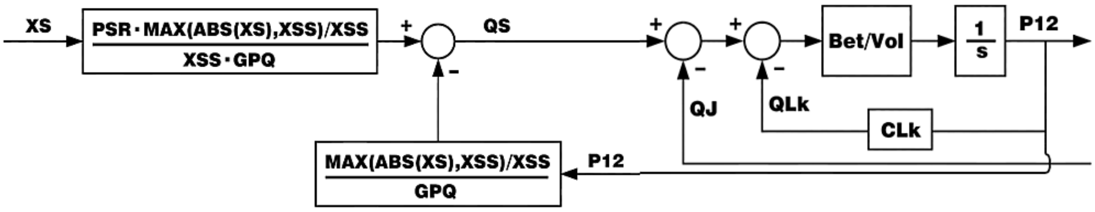

Figure 4 Block diagram of the proposed hydraulic capacity-sensitive LF model (Model C5q).

Unlike the models discussed in the previous chapter, C5q allows for the simulation of dynamic interactions between various system components, such as valves, connecting pipes, and final actuators, in terms of regulated hydraulic flow rate, leakage losses, system volume, and hydraulic fluid compressibility. The hydraulic capacity causes a first-order lag in the leakage feedback loop, as show in Figure 4. This lag is critical for avoiding numerical instability. To evaluate its performance, the scientists compared the results to those of a previously published HF model.

4.1 ABOUT EHSV P12-XS CHARACTERISTCS

The peculiar characteristics of the different models can be analysed based on the dynamic response they induce when implemented within a numerical model simulating an electro-hydraulic actuator or, alternatively, by examining the corresponding comma fluid dynamic characteristic, i.e., the family of curves that define the relationship between spool displacement XS and differential pressure P12 adjusted for a given discharged flow rate QJ. The next chapter will widely analyse the dynamic responses produced by different models. However, in this paragraph, we will examine the fluid dynamic characteristics of models A, C2, C5, and C5q in case of CLk = 0 [m3/s/Pa] and PSR = 20 [MPa]. According to Eq. (3), model A (Fig. 5) cannot accurately consider pressure saturations and tends to overestimate the stiffness of simulated EHAs (making them relatively insensitive to the effects of external loads or PSR supply pressure drops).

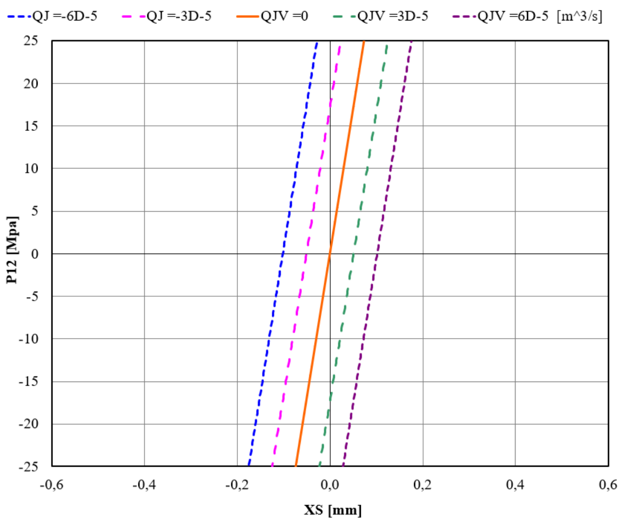

Figure 5 EHSV characteristic P12-XS - Model A [17].

Several non-linear models have been proposed in the literature that implement (albeit with different algorithms) pressure saturation for high XS spool displacements to overcome this drawback. These models (e.g., C1, C2, and C3) are thus closer to reality for conditions of large valve opening. Still, they cannot correctly simulate phenomena such as water hammer or counter-reaction in flow (see Fig 6 relating to the fluid dynamic characteristic of the Model C2).

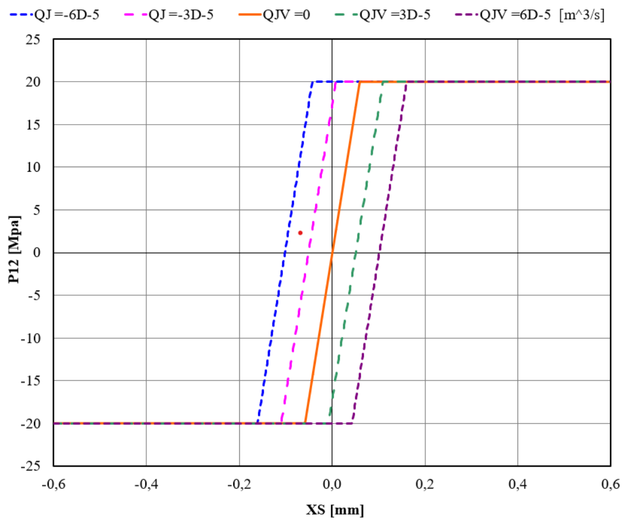

Figure 6 EHSV characteristic P12-XS - Model C2 [17].

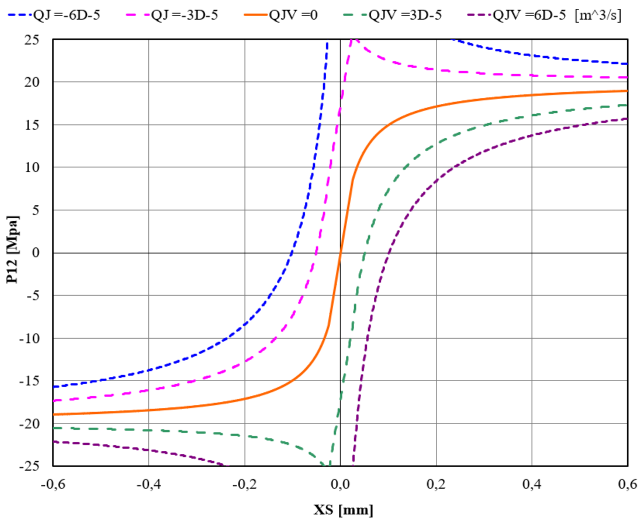

Figure 7 EHSV characteristic P12-XS - Model C5 [18].

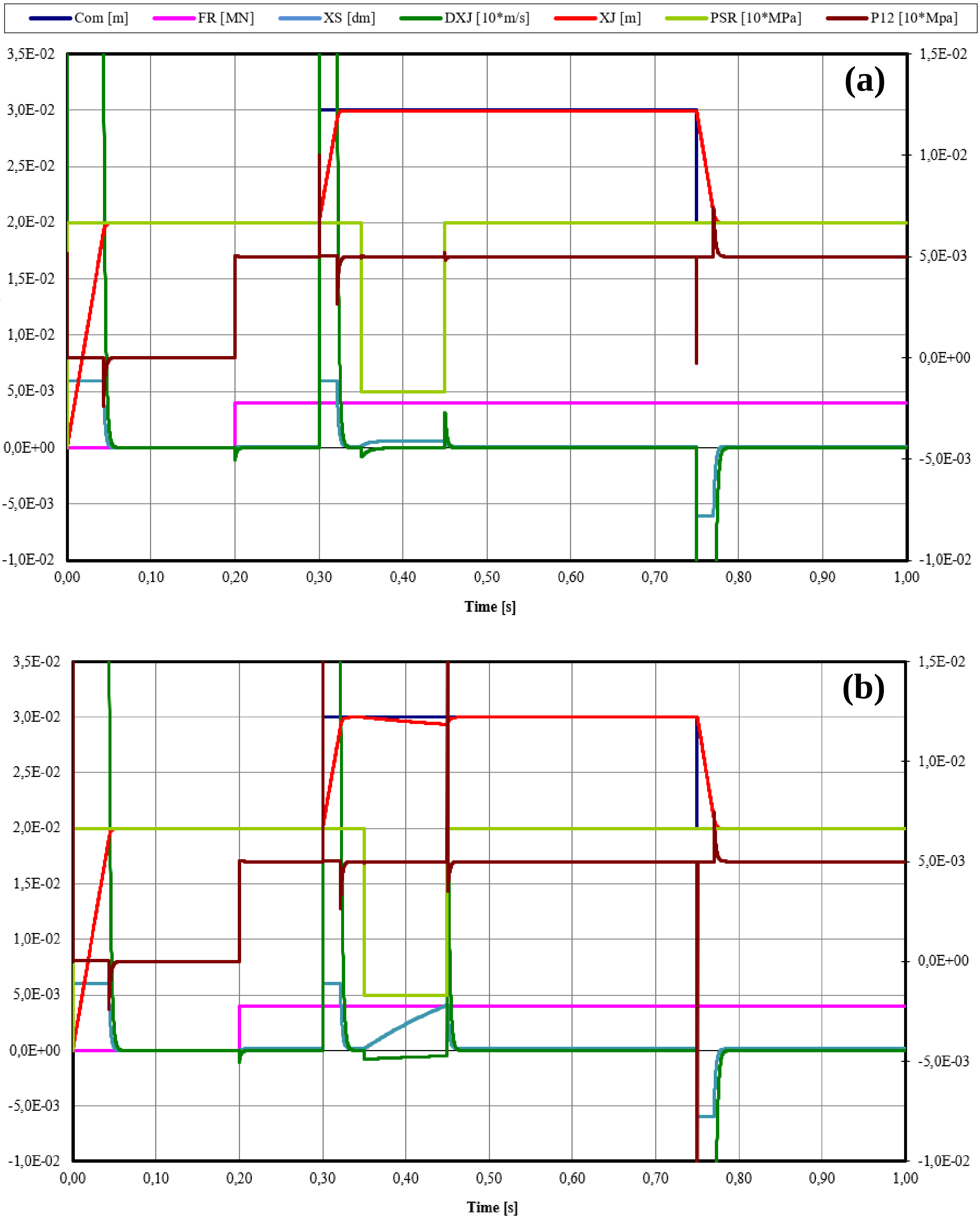

Figure 8 Test bench simulation with A model **(a)** and C2 model **(b)**; Com - position command: FR - external load, XS - spool position, DXJ - jack speed, XJ - jack position, PSR - supply pressure, MJ – jack inertia = 10 [kg]

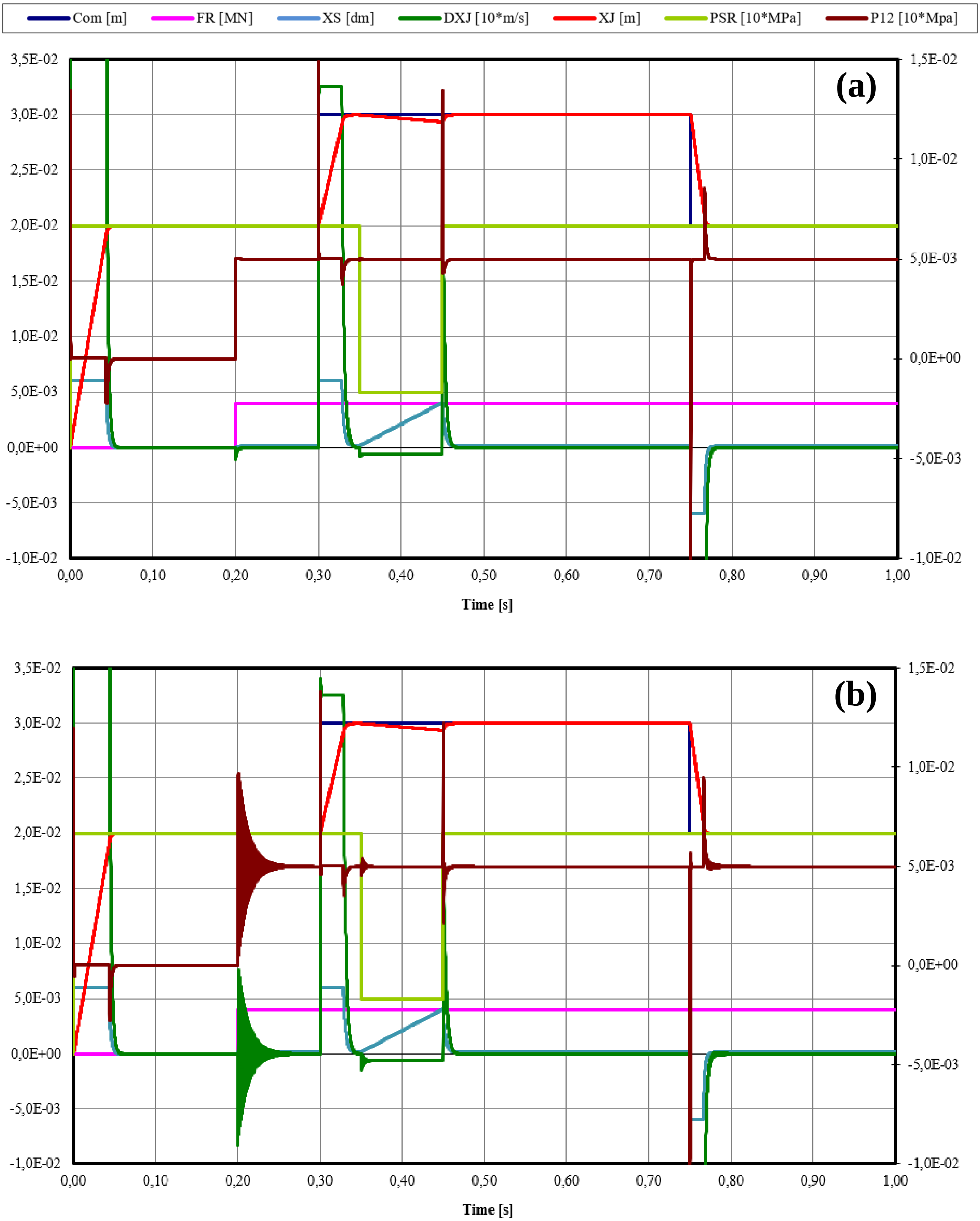

Figure 9 Test bench simulation with C5 model **(a)** and C5q model **(b)**; Com - position command: FR - external load, XS - spool position, DXJ - jack speed, XJ - jack position, PSR - supply pressure, MJ – jack inertia = 10 [kg]

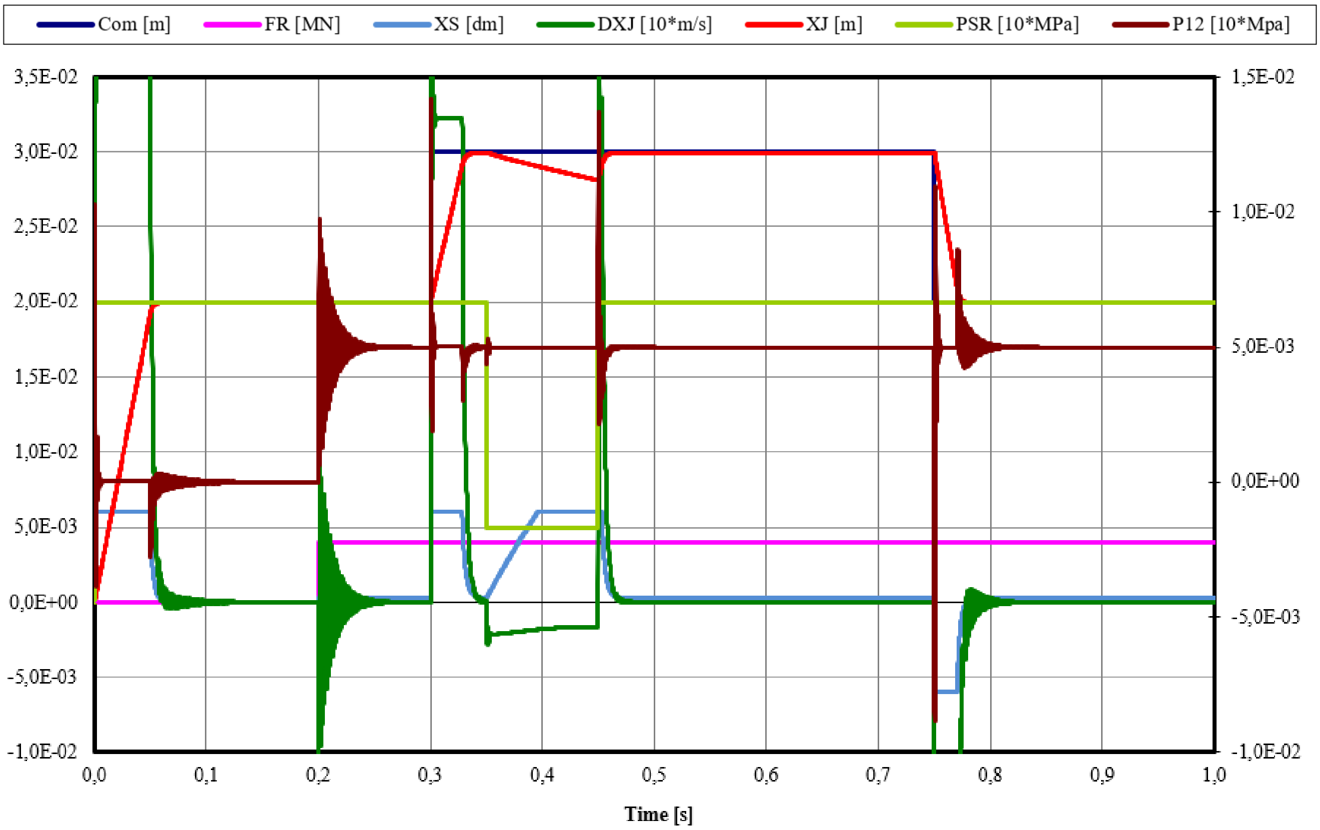

Figure 10 Test bench simulation with HF model; Com - position command: FR - external load, XS - spool position, DXJ - jack speed, XJ - jack position, PSR - supply pressure, MJ – jack inertia = 10 [kg]

To overcome these drawbacks, the authors have previously proposed some simplified numerical models capable of simulating the effects mentioned above, albeit in an approximated way. In particular, in [18, 22], the authors proposed an extremely compact algorithm called Model C5, based on linear gain modeling (GP and GQ), capable of simulating the fluid dynamic performance of the valve while simultaneously taking into account water hammer, counter-reaction in flow rate, and of the saturation of the regulated delivery pressure for high spool openings.
Model C5, shown in Figure 7, effectively replicates the performance of Model HF under zero-flow conditions and can also assess its performance under non-zero-flow conditions with appropriate accuracy. However, under non-zero-flow conditions, Figure 7 shows some discrepancies (because of the linearized approach) in the maximum pressure P12 amplitudes and the corresponding spool displacements. Despite having a different mathematical formulation, the P12-XS characteristic of the new Model C5q (reported in Figure 11) is identical to that of the above Model C5. Infact, the hydraulic pressure P12, flow rate QJ, and the speed at which the jack operates DXJ are all influenced by the combined effect of the hydraulic fluid compressibility and the circuit hydraulic capacity (related to pipes and actuator chambers).

Consequently, the circuit hydraulic compliance could generate oscillations that are not always negligible during the actuation transients. Still, stationary conditions represented by the points of the P12-XS fluid characteristic, representative of stationary equilibrium conditions, do not generate appreciable effects.

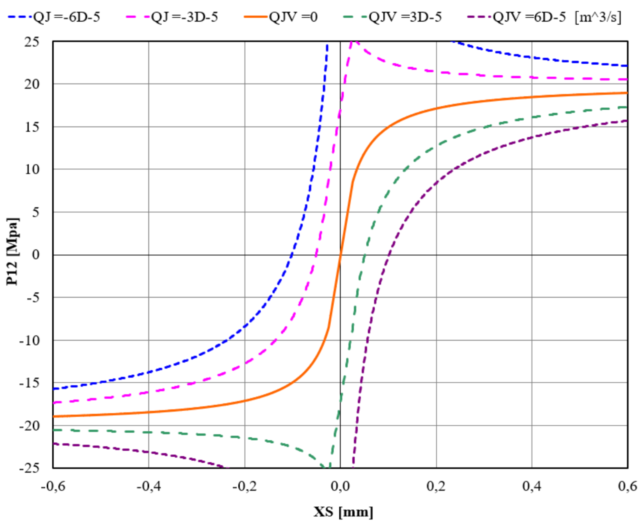

Figure 11 EHSV characteristic P12-XS - Model C5q.

5 DISCUSSIONS OF THE PRELIMINARY RESULTS

Model C5q has been integrated into a virtual testing setup replicating the position-control electro-hydraulic actuator (EHA) detailed in Chapter 2. It was compared against a numerical model incorporating the HF fluid dynamic model from Chapter 3 to evaluate its performance. This EHA model includes critical electrical, hydraulic, and mechanical properties of all system components, such as inertia, dry and viscous friction in the hydraulic piston, and a third-order electromechanical model of the dynamics in the first and second stages of the EHSV.

Figures 8, 9 and 10 present the time-domain dynamic responses of the EHA models in simulated tests with various combinations of position commands (Com), external loads (FR), and changes in hydraulic supply pressure (PSR). The input sequence was carefully designed to showcase the fluid-dynamic models' capabilities and their impact on the EHA's behaviour [18]. These figures compare the dynamic responses of the simulated EHA implementing the different numerical fluid dynamic simplified model (FDSM) reported in Chapters 3 and 4.

Figure 10 displays the dynamic response of the EHA incorporating the Model HF. This response served as a baseline to compare the dynamic behaviour with the different LF models already described.

Figure 8 illustrates the dynamic response of the EHA system equipped with the fluid dynamic model A. The linear relationship between spool displacement XS and regulated differential pressure P12 (due to the lack of saturations on the maximum pressure supplied by the EHSV) makes the system almost insensitive to external loads and temporary system depressurization phenomena (light green curve in the range from 0.35 s to 0.45 s).

Models C2 (Fig. 8b) and C5 (Fig. 9a) overcome the Model A limits by adopting non-linear formulations that aim to approximate the reference response with greater accuracy.

Model C2 overestimates the actuation speed DXJ for actuation under opposing load (0.3 s) as it cannot simulate the related pressure peaks, resulting in the same calculated as unloaded actuation at the beginning of the simulation.

On the other hand, the C5 Model accurately simulates actuation conditions with an opposing load, estimating a DXJ equal to that predicted by the HF Model.

However, based on the gain modelling introduced in [29-31], both tend to underestimate the actuator backward displacement following the partial depressurization of the system under opposing load FR (among 0.35 s and 0.45 s). Both models describe the relationship between differential pressure and flow rate regulated by the EHSV metering edges through a linearized formulation that tends to underestimate this backward displacement.

While guaranteeing a greater degree of fidelity (compared to the simplified models mentioned previously), these models are incapable of considering the hydraulic fluid's compressibility and the pressure (and flow) transients that derive from it. Therefore, they are unsuitable for simulating systems characterized by high hydraulic capacity values, flexible hoses, fluid with high gaseous contamination, or asymmetric actuators.

Figure 9b illustrates the response of the proposed fluid dynamic model (C5q). The comparison with Figure 10 reveals that the proposed algorithm accurately simulates the main dynamic characteristics of the HF system. It should be noted that, if compared to other algorithms in the literature, the proposed fluid dynamic model C5q significantly reduces errors under nominal conditions and during operation under partial depressurization (from 0.35 to 0.45 seconds) or when subjected to external loads (opposing from 0.2 to 0.75 seconds and aiding after that). Furthermore, in the case of hydraulic transmissions characterized by non-negligible compressibility, the C5q Model can satisfactorily simulate the pressure transients generated within the circuit following commands or disturbances. As an example, the underdamped oscillatory transients relating to the P12 delivery pressure are shown below, simulated by the C5q (Fig. 12) and HF (Fig. 13) models downstream of the instant time = 0.2 s in response upon application of an external load FR = 4000 N.

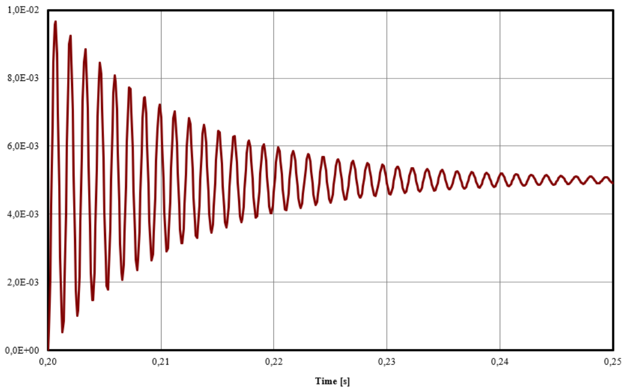

Figure 12 P12 response of the EHA with C5q Model Transient due to a load step FR = 4000 N (time = 0.2 sec).

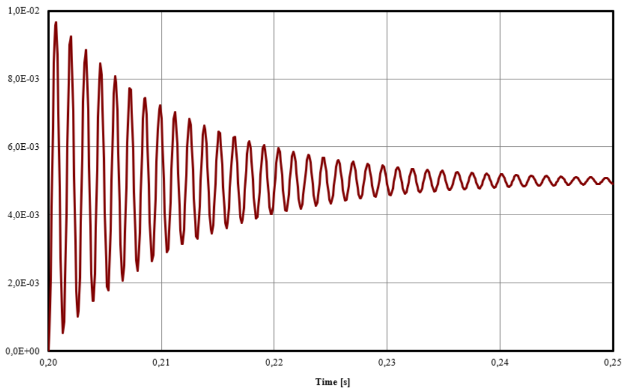

Figure 13 P12 response of the EHA with HF Model Transient due to a load step FR = 4000 N (time = 0.2 sec).

In conclusion, although with some evident differences (e.g. the behavior in the actuation condition with an opposing external load and temporary partial depressurization of the hydraulic system, which shows evident deviations attributable to the linearized gain modeling of the SV fluid dynamics), the proposed C5q model allows to simulate the response of the HF reference with a degree of fidelity much higher than previous FDSMs and, above all, it will enable simulating with satisfactory accuracy the pressure transients that arise in the hydraulic circuit following an external command or disturbance.

6 CONCLUSIONS

The paper introduces a new, simplified fluid dynamic model for an EHSV that considers the compressibility of hydraulic fluid. This model addresses limitations found in previous low-fidelity emulators and aims to closely mimic the behaviour of a high-fidelity (HF) simulation, including pressure oscillations, fluid compliance, and leakage losses. Furthermore, the model offers significantly reduced computational time and effort compared to HF reference models or conventional CFD codes. This feature especially appeals to real-time applications, such as initial design and diagnostic monitoring. Nevertheless, it is essential to acknowledge that Model C5q, as a simplified model, has certain limitations regarding accuracy and reliability. These limitations, discussed in the previous chapter, may make it less suitable for applications such as model-based prognostics of hydraulic systems. These shortcomings could be overcome, at least partially, by developing new simplified non-linear fluid dynamic models based, for example, on data-driven approaches implemented using ANNs, DoE, or AI algorithms, capable of combining the typical reduced computational efforts and high execution speed of the models presented here with the high level of fidelity and robustness of the HF.

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