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A review of simplified servovalve models for digital twins of electrohydraulic actuators

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Abstract. The development and detail design of complex electrohydraulic actuators for aircraft flight controls require the use of accurate, high fidelity fluid-dynamic simulations in order to predict the behaviour of the system within its whole operating envelope. However, those simulations are usually computationally expensive, and simplified models are useful for the preliminary design phases and real-time health monitoring. Within this context, this work presents a review of low fidelity models for the fluid-dynamic behaviour of an electrohydraulic servovalve. Those are intended to run in real time as digital twins of the physical system, in order to enable the execution of diagnostic and prognostic algorithms. The accuracy of the simulations is assessed by comparing their results against a detailed, physics-based high fidelity model, which computes the response of the equipment accounting for the pressure-flow characteristics across all the internal passageways of the valve.

1. Introduction

Electro Hydraulic (EH) servomechanisms are widely employed within current generation aircraft flight control systems, since they provide power density unmatched by alternative technologies and a very high reliability. Among the most common applications in aerospace technology, EH actuators are used for powering fly-by-wire aerodynamic surfaces, landing gear retraction, steering and braking, as well as several secondary users. However, to match the safety requirements for the use in commercial and military aviation, redundancies and health monitoring strategies are usually needed [1,2]. Urata proposes in [3-9] several studies on the effect on the behavior of hydraulic servovalves of various fault modes, namely leakage flux, fringing, eddie currents, and asymmetry of the air gap in the torque motor. Detailed, high fidelity analysis of servovalve behavior are available in [10, 11], leveraging Computational Fluid Dynamics (CFD) and electromagnetic Finite Elements (FE). However, those high fidelity simulations are usually too computationally expensive for onboard time-constrained applications, such as monitoring and diagnostic routines. Within this context, accurate yet simple models are needed to simulate the behavior of the system in real time. The most basic architecture of an electrohydraulic actuator includes a control electronics module, which compares the current position with the setpoint to compute the current command to a servovalve. The valve, usually with a two-stage, flapper-nozzle design, is the regulating element that routes the hydraulic fluid to the control ports of the motor element, typically a linear jack or a rotary hydraulic

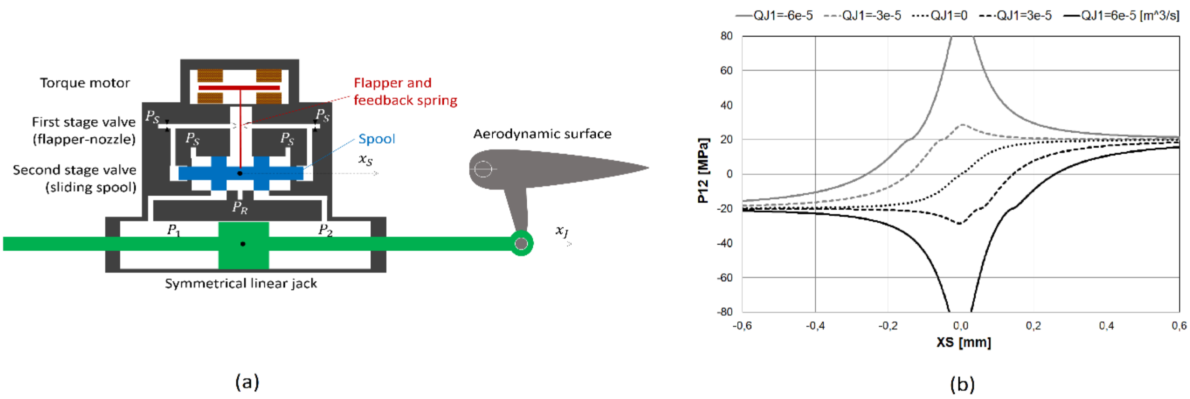

Figure 1. (a) Typical design of a hydraulic actuator and its flapper-nozzle servovalve. (b) Pressure-flow rate-spool position characteristic of the servovalve, obtained through the High Fidelity model

motor, which converts hydraulic power into mechanical power in order to move the user (e.g. a control surface of the aircraft). The servovalve is the most complex element of the actuator. The typical design for a two-stage, four-way, flapper-nozzle servovalve is represented in Figure 1 (a). The input current flows through the torque motor and moves the flapper towards one of the nozzles. As a result, the pressure on the two sides of the spool becomes unbalanced and causes the spool itself to move from its null position, opening or closing the fluid passageways on the sleeve. Eventually, the spool displacement, through the feedback spring, draws the flapper back to an equilibrium position, equalizing the pressures on the spool.

In this work we propose a review of simplified models for the fluid-dynamic behavior of a servovalve, which is the component of an EH actuator characterized by the most complex and nonlinear behavior; accurately modelling its behavior is an essential step to obtain reliable real-time emulators of the operation of the servosystem. Those models are intended to run with low computational resources within digital twins and monitoring models for electrohydraulic actuators. The models are evaluated and compared with a high fidelity, physics-based simulation.

2. High fidelity model

We employ a High Fidelity (HF) model as a simulated test bench for the simplified representation of the servovalve [12]. The HF model evaluates the flow-pressure characteristic of each passageway, to determine the pressure drop across the valve. As shown in Figure 1 (b), the differential pressure to the control ports P_{12} varies approximately linearly with spool position x_S and flow rate Q_J, when the spool itself is close to the null position. As the spool moves further away from its center position, P_{12} increases up to the supply pressure differential (P_{SR}). When the valve is closed (i.e. $x_S \approx 0$), the differential pressure can grow higher than P_{SR}, if the hydraulic circuit downstream the servovalve imposes a flow rate $Q_J \neq 0$: this effect is known as water hammer.

The HF model is computationally expensive and not suitable for real time evaluation. Additionally, it depends on a number of parameters which are not available from the datasheet of a servovalve, such as information related to the geometry of the fluid passageways and the clearances between spool and sleeve. Hence, low fidelity simulations usually rely on linearized servovalve models, which only depend on parameters that are easily measurable considering the valve as a black-box. One of the most common linearized formulations can be expressed as:

$$P_{12} = G_P(x_S - Q_J/G_Q)$$

where G_P is the pressure gain (i.e. the ratio between differential pressure and spool position at zero flow) and G_Q is the flow gain (i.e. the ratio between flow rate and spool position at zero differential pressure). This linearized model has a significant flaw that lies in its inability to account for the water hammer effect, nor for pressure saturation and leakage through the valve clearances.

3. Simplified models

The limitations of the linearized models usually make them unsuitable for real-time health monitoring tasks, since the discrepancies between the model and physical system can trigger false positive fault detections. Hence, the following paragraphs provide a review of more elaborate yet computationally light servovalve models, intended to overcome some of the issues associated with the linearized ones.

3.1. Model A

Model A was firstly introduced in [13] (issued in Italian language). It is intended to account for a variable supply differential pressure P_{SR}, as well as for the leakage across the clearance between the spool and sleeve of the valve, through the leakage coefficient C_{LK}. As shown in Figure 2 (a). The variable P_{SR} is considered by replacing the pressure gain G_P with P_{SR}/x_{SS}, where x_{SS} is the spool displacement needed saturate the control pressure, so that $P_{12} = P_{SR}$. To account for leakage, the model assumes that an additional flow rate $Q_{LK} = P_{12}C_{LK}$ passes through the valve causing a pressure drop, quantified as $Q_{LK}G_{PQ}$, where $G_{PQ} = G_P/G_Q$.

3.2. Model C1

Model C1, introduced in [14], is a modification of model A intended to add a sensitivity to the pressure saturation. Excluding the water hammer effect, a valve can normally provide a maximum differential pressure equal to the supply differential pressure of the hydraulic circuit. Model C1, with a layout similar to model A, has a saturation block upstream the leakage evaluation, as shown in the block diagram of Figure 2 (b).

3.3. Model C2

Model C2, proposed in [14] is described by the block diagram shown in Figure 2 (c). It corrects model C1 by placing the saturation downstream the leakage evaluation block. This way, the maximum differential pressure is allowed to rise up to the supply differential pressure, as opposed to model C1 where $|P_{12}| \leq P_{SR}/(1 + G_{PQ}C_{LK})$.

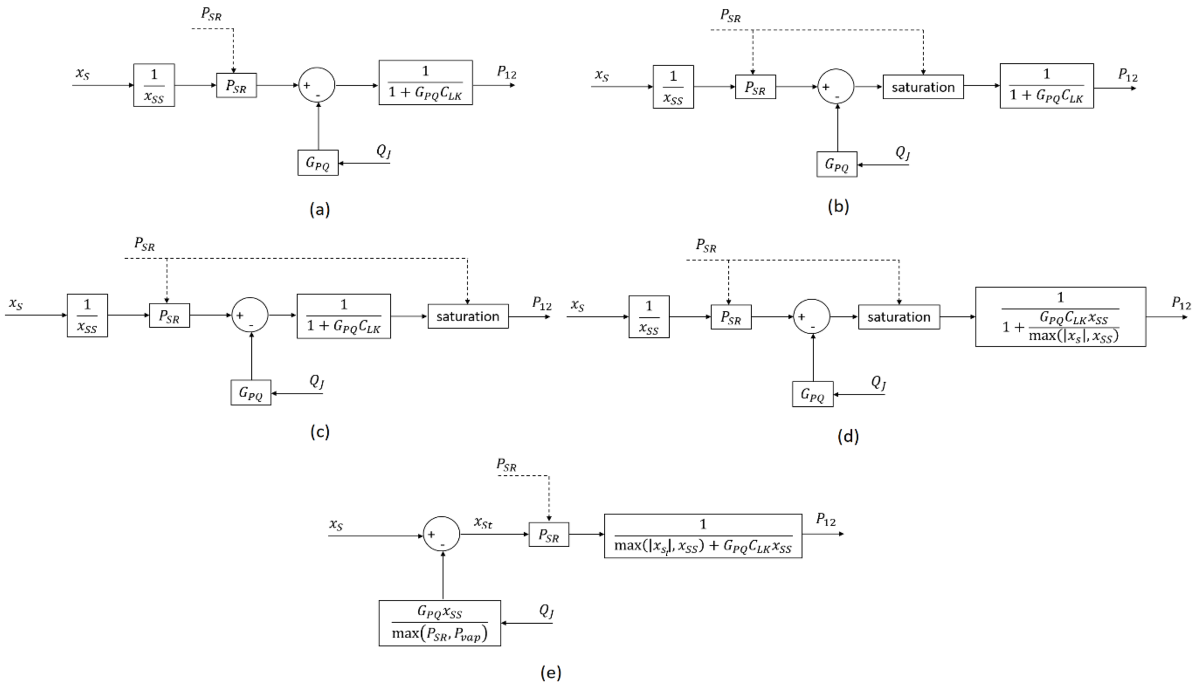

Figure 2. (a) Block diagram of Model A. (b) Block diagram of Model C1. (c) Block diagram of Model C2. (d) Block diagram of Model C3. (e) Block diagram of Model C5.

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3.4. Model C3

Model C3, introduced by [15] is an alternative correction of model C1. The saturation is kept upstream of the leakage evaluation; this leakage evaluation block has a modified transfer function intended to account internally for the pressure saturation, through the limitation of the effective spool displacement x_S to the saturation value x_{SS}. The block diagram of Model C3 is shown in Figure 2 (d).

3.5. Model C5

Model C5 was initially proposed in [16]. It is intended to evaluate the interaction between pressure saturation and leakage evaluation in a more in a more realistic way. To do so, an equivalent spool position x_{St} is introduced that accounts for the fluid flow passing through the valve and a potentially variable supply pressure:

$$x_{St} = x_S - \frac{Q_J G_{PQ} x_{SS}}{\max\left(P_{SR}, P_{vap}\right)}$$

that is equivalent to reduce the effect of the flow feedback with P_{SR}, down to the vapor pressure P_{vap}. The effective spool position is multiplied by a variable pressure gain, to compute the differential pressure and account for saturation and leakage flow:

$$P_{12} = x_{St} \frac{P_{SR}}{\max(|x_{St}|, x_{SS}) + G_{PQ} C_{LK} x_{SS}}$$

As a result, the effect of the leakage flow is evaluated in a manner similar to Model C3, by considering the interaction with the saturation of the differential control pressure. The model is shown in the block diagram of Figure 2 (e).

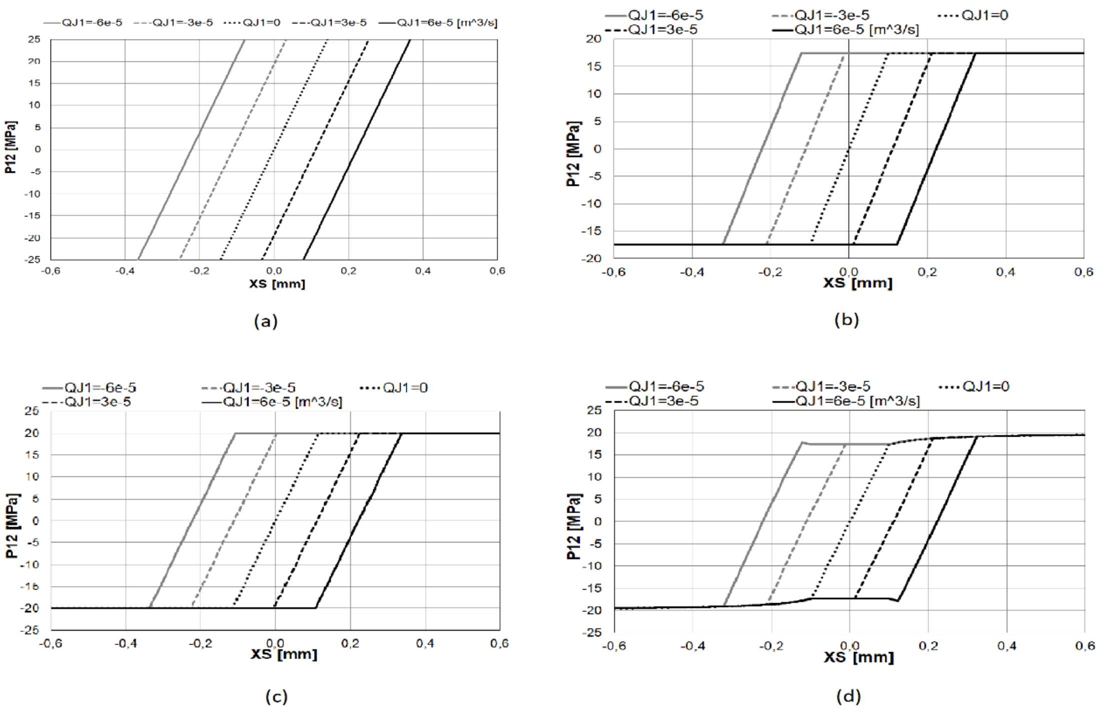
4. Results

The models are assessed by computing their pressure-spool position characteristic for multiple values of the flow rate Q_J. This results in the maps shown in Figure 8, that shall be compared to that of the HF model (Figure 2). From the comparison with the HF map, clearly none of the simplified models is able to correctly simulate the water hammer effect. Although this condition rarely happens in the normal operation of a hydraulic system, it can have a significant impact on the behavior of the actuator in case of a pressure drop in the hydraulic supply.

As expected, Model A is not able to account for the pressure saturation at $\pm P_{SR} = 20 MPa$. The saturation is correctly estimated by models C2 and C5, while Model C1 underestimates the maximum differential pressure value, being the leakage evaluation downstream the pressure saturation. Model C3 has an anomalous behavior near saturation for small spool displacements, meaning that the proposed technique to consider the interaction between pressure saturation and leakage flow is not completely satisfying. This behavior is corrected by the modification introduced with Model C5.

5. Conclusions

Five simplified models for the fluid dynamic behavior of a hydraulic servovalve were reviewed and compared with a high fidelity simulation. The results showed that none of those simplified emulators is able to simulate the behavior of the valve in its whole operating envelope. Specifically, the simplified models are not suitable for evaluating the water hammer effect. With this regard, further development is required in future works. However, some of those simplified models (in particular models C2 and C5) can be successfully employed, with different levels of accuracy, provided a limited operating envelope is considered.



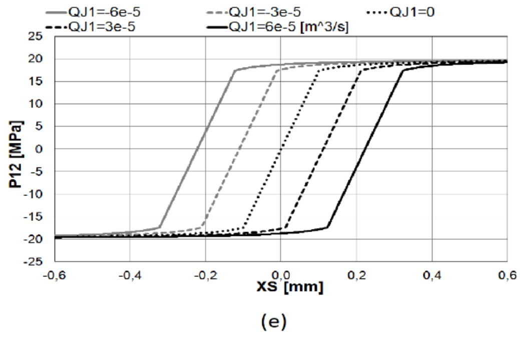


Figure 3. Pressure-flow-spool position maps of the simplified models. (a) Model A; (b) Model C1; (c) Model C2; (d) Model C3; (e) Model C5

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