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# Digital twins for prognostics of electro-hydraulic actuators: novel simplified fluid dynamic models for aerospace valves

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**Abstract**. In the design and development phases of electro-hydraulic actuators (EHAs) used for aircraft flight controls, it is often necessary to carry out accurate and high-fidelity fluid dynamics simulations to evaluate the system behaviour within its entire operating range and, if necessary, investigate its most critical issues. These high-fidelity simulations (nowadays achievable with different techniques and commercial software) generally become pretty expensive from a computational perspective. Therefore, especially in the preliminary design phases or implementing system health monitoring algorithms (in real-time), the need to adopt simplified models emerges definitely (albeit capable of guaranteeing the appropriate level of detail and accuracy). These simplified models are also essential for developing effective and reliable model-based prognostic strategies capable of performing early health assessments of EHA valves. This work proposes a new lumped-parameters simplified numerical model, which, despite having a very compact formulation and reduced computational costs, simulates the internal fluid dynamics of the valve, overcoming some critical issues typical of other models available in the literature. It evaluates valve performance as a function of spool position and environmental conditions (e.g. supply pressure), better-assessing flow rate feedback, internal leakages, and other operating conditions (e.g. spool fine adjustment, pressure supply variable, overpressure, or water hammer). The performance of this numerical model is evaluated comparing with other simplified models published in the literature. Moreover, it is validated with a high-fidelity digital twin that simulates the behaviour of the valve, taking into account the geometry of the spool, the properties of the hydraulic fluid, and the local internal fluid-dynamics (laminar or turbulent regime, cavitation, etc.).

**Keywords**: Aerospace systems, EHA, Digital twin, Flight commands, Lumped parameters, Monitoring, Non-linear modelling, Servovalve, Simplified fluid dynamic numerical models.

## 1. Introduction

Electrohydraulic actuation systems (EHAs) are commonly employed in powered flight controls of most commercial and military aircraft. They represent a well proven technology and feature a very high power density, enabling to design lightweight systems fitting in the tight spaces available inside the aerodynamic surfaces of the aircraft.

As the flight controls are among the most safety critical system of airborne vehicles, redundancies and real-time health monitoring [1-3] are needed to match the strict safety requirements of commercial aviation. Therefore, this study focuses on the simplified modelling of electrohydraulic systems, and specifically of the behavior of the servovalve: faults of this component indeed have a significant impact on the performances of the entire actuator. Different fault modes and their effect on the servovalve's response are studied in detail by Urata [4-10], including fringing, torque motor airgap asymmetry, eddy currents, and leakage flow. Henninger et al. [11] and Chen et al. [12] propose detailed simulations of the hydraulic and electric performance of EH servovalves combining electromagnetic finite elements and Computational Fluid Dynamics (CFD).

Most electrohydraulic actuators for flight controls are based on a flapper-nozzle, four ways servovalve, as shown in Figure 1. The electric torque motor offsets the position of the flapper, partially closing one of the nozzles and bringing a pressure unbalance on the two sides of the sliding spool (depicted in blue in Figure 1). The spool moves form the center position under the effect of pressure, until a feedback spring draws the flapper back to the center. The position of the spool rules the opening and closing of the valve passageways, controlling pressures $P_1$ and $P_2$ and flow rate $Q_J$ to the two chambers of the hydraulic jack (green in Figure 1) and moving the aerodynamic control surface.

This paper proposes a new simplified fluid-dynamic model of the servovalve, intended to run in real time and to require very limited computational resources: possible applications include real-time health monitoring and preliminary system design. The accuracy of the model is validated against a high fidelity CFD simulation and compared with other low fidelity models available in literature.

## 2. HF model

A High Fidelity (HF) fluid dynamic model is employed in this work as a reference for the development of the lower fidelity emulators. The HF model solves the pressure drops across each passageway of the valve in order to determine the pressures and flow rates crossing the valve for a given spool position. 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 $P_{SR}$. This phenomenon is known as the water hammer effect, and has a significant impact on the servovalve behavior.

The HF model is too computationally expensive to be included in a real-time dynamical simulation, as an iterative process is needed to solve the pressure drops at each timestep. In addition, to correctly set the model for the simulation of a physical system, several parameters are needed, some of which may be difficult to obtain either experimentally or from detail design data. 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 detailed description of the HF model is available in [13, 14].

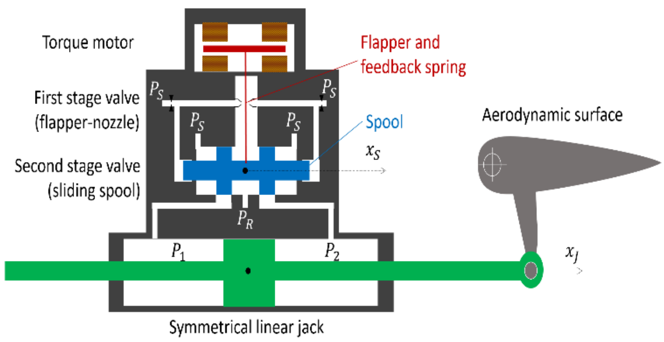


Figure 1: Simplified architecture of an electrohydraulic flight control actuator

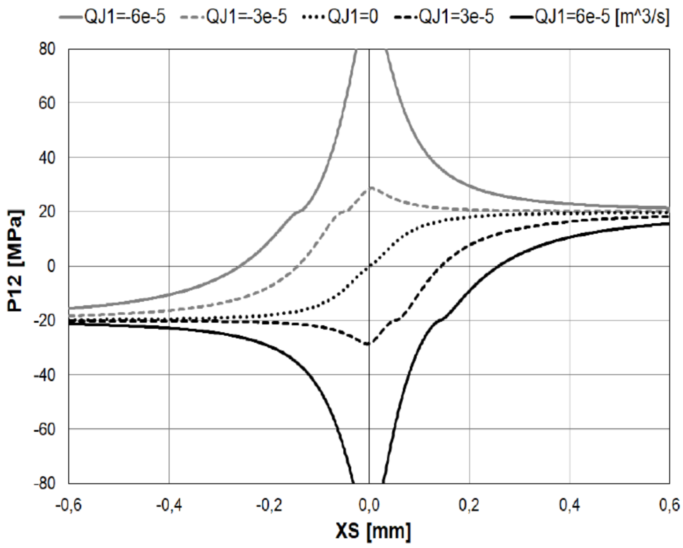


Figure 2: spool position – pressure – flow rate map of the servovalve computed by the HF model

## 3. LF models

Common Low Fidelity (LF) representations of the hydraulic characteristic of servovalves rely on the local linearization of the flow-pressure-displacement map, usually near the zero flow, closed valve condition. This allows to describe the operation of an electrohydraulic actuator in the regulating regime, far from the effects of saturations and other non-linear phenomena. 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. Two alternative formulations are possible, either considering flow rate $Q_J$ as a feedback variable and solving for the differential pressure $P_{12}$ (pressure formulation) or the opposite (flow rate formulation).

These linear models are not able to capture several phenomena that significantly affect the operation of hydraulic actuators, such as leakages, limited supply pressure, and water hammer.

### *3.1. Previous LF models*

Several simplified models available in literature attempt to extend the linear formulation in order to overcome its limitations. The next sections present an overview of some simplified models previously developed by the authors, while Figure 3 summarizes and compares the block diagram representations of the models.

*3.1.1. Model A.* Model A [15] accounts for the variable differential supply pressure $P_{SR}$ and for the leakage flow:

$$P_{12} = \left(\frac{x_S}{x_{SS}} P_{SR} - Q_J G_{PQ}\right) \frac{1}{1 + G_{PQ} C_{lk}} \tag{2}$$

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 (1).

*3.1.2. Model C1.* Model C1 [14, 16] extends the applicability of Model A to large spool displacements, by introducing a saturation of the differential pressure to the supply pressure. The saturation is placed upstream the leakage estimation:

$$\boldsymbol{P_{12} = \frac{1}{1 + G_{PQ}C_{lk}}\left[\mathrm{sgn}\left(\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right)\max\left(P_{SR}, \left|\frac{x_S}{x_{SS}}P_{SR} - Q_J G_{PQ}\right|\right)\right]} \tag{3}$$

*3.1.3. Model C2.* In Model C2 [14, 16] the pressure saturation is moved downstream the leakage flow estimation: this is to correct the underestimation of maximum differential pressure observed in Model C1 (see Section 4). The formulation of Model C2 is modified as follows:

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

*3.1.4. Model C3.* Model C3 [17] is an alternative approach to correct the behavior of Model C1: the pressure saturation is accounted for within the leakage computation block:

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

*3.1.5. Model C5.* Model C5 [17, 18] introduces an equivalent spool-position $x_{St}$ taking into account the effects of variable $P_{SR}$ and oil flow $Q_J$ drained across the valve:

$$x_{st} = x_S - \frac{Q_J(G_{PQ}x_{SS})}{\max(P_{SR}, P_{vap})} \tag{6}$$

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

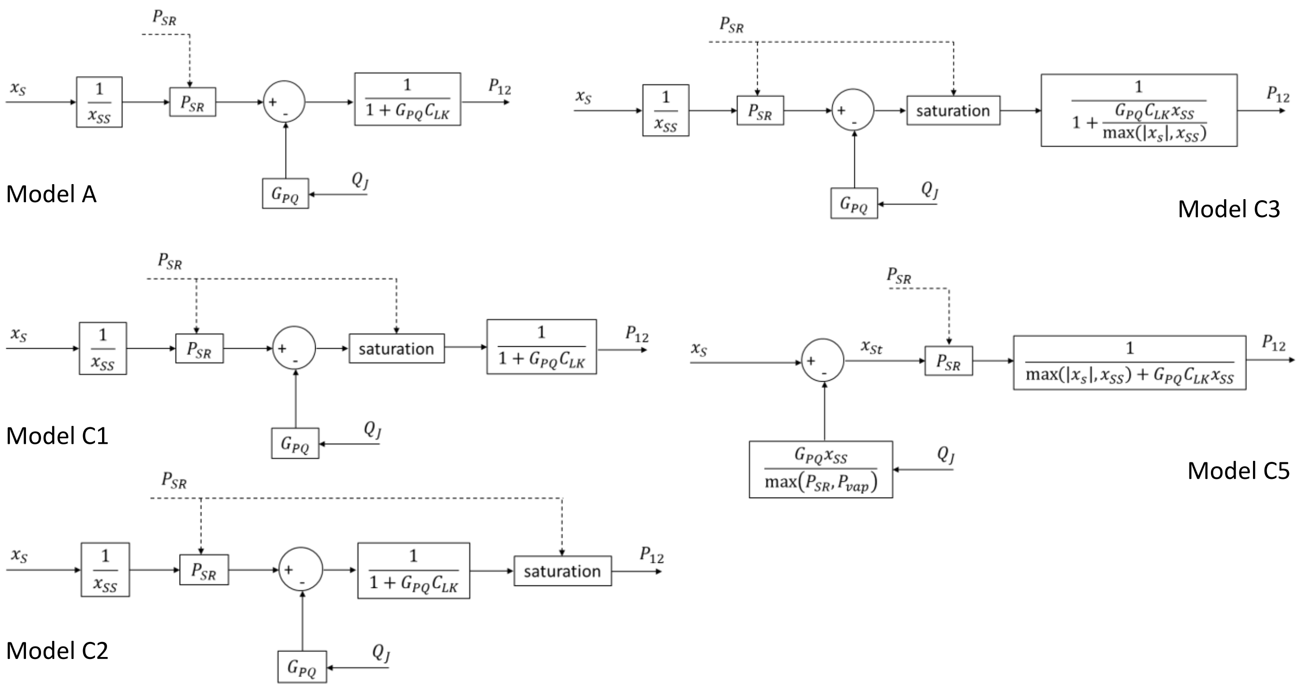


Figure 3: Block diagrams of models A, C1, C2, C3 and C5

### *3.2. Model E*

Model E is intended to map the fluid-dynamic behavior of the servovalve overcoming the limitations of the previous, linear-based emulators; the downside is a higher abstraction from the underlying physical phenomena, as Model E employs shape functions that are not derived from physics, but aim at reproducing the output of higher fidelity simulations.

The differential pressure versus spool position characteristic of the valve, for a zero flow condition, is approximated with a hyperbolic tangent function, scaled with the supply pressure $P_{SR}$ and the saturation displacement $x_{SS}$:

$$P_{12P} = P_{SR} \cdot \tanh\left(\frac{x_S}{x_{SS}}\right) \tag{8}$$

A dynamical pressure drop proportional to flow rate $Q_J$ is scaled with the normalized inverse square of the spool position $x_{SS}^2/x_S^2$: indeed, a constant flow rate will produce a larger pressure drop when crossing a valve that is near to the closed position. This formulation allows to inherently account for the water hammer effect:

$$P_{12} = P_{SR} \cdot \tanh\left(\frac{x_S}{x_{SS}}\right) - Q_J \cdot \frac{G_P}{G_Q} \cdot \left(\frac{x_{SS}}{x_S}\right)^2 \tag{9}$$

In the ideal case of zero leakage, a non-null flow crossing a completely closed valve (i.e. $x_S = 0$) results in an infinite pressure drop. Clearly, this cannot hold true for a physical servovalve, and shall be prevented in the model as well to avoid numerical singularities. A leakage coefficient $C_{lk}$ is introduced for the purpose, defined as:

$$C_{lk} = \frac{Q_J}{P_{12}} \qquad \text{for } x_S = 0 \tag{10}$$

This is added to the spool position contribution to dynamical pressure drops, effectively introducing a lower bound to the minimum passageway of the valve. This reflects the behavior of the small clearance designed to allow relative sliding between the spool and sleeve:

$$P_{12} = P_{SR} \cdot \tanh\left(\frac{x_S}{x_{SS}}\right) - Q_J \cdot \frac{1}{\left(\frac{x_S}{x_{SS}}\right)^2 \frac{G_Q}{G_P} + C_{lk}} \tag{11}$$

The complete Model E, expressed by Equation (11), is schematically represented in the block diagram of Figure 4.

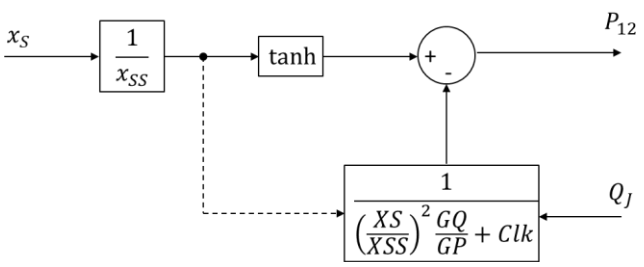


Figure 4: block diagram of Model E

## 4. Results and discussion

The models are compared and validated against the response of the HF simulation. For each model, the differential pressure – spool position – flow rate characteristic is computed, to be compared with the HF map of Figure 2. The maps of Models A, C1, C2, C3 and C5 are shown in Figure 5. We can observe that none of these simplified emulators is able to approximate the behavior of the HF model in its whole domain. In particular, none of the models accounts for water hammer effects, while Model A is a linear representation unable to account for pressure saturation. Models C1 and C3 fail to predict the pressure saturation correctly, while models C2 and C5 can be still employed if the dynamics of the actuator can exclude the operation in conditions of high flow and small spool displacement.

The response of Model E is shown in Figure 6 and compared to the high fidelity one. Clearly, this emulator is able to mimic the HF model with high accuracy in its whole operational envelope, including the effects of water hammer, saturations and leakage.

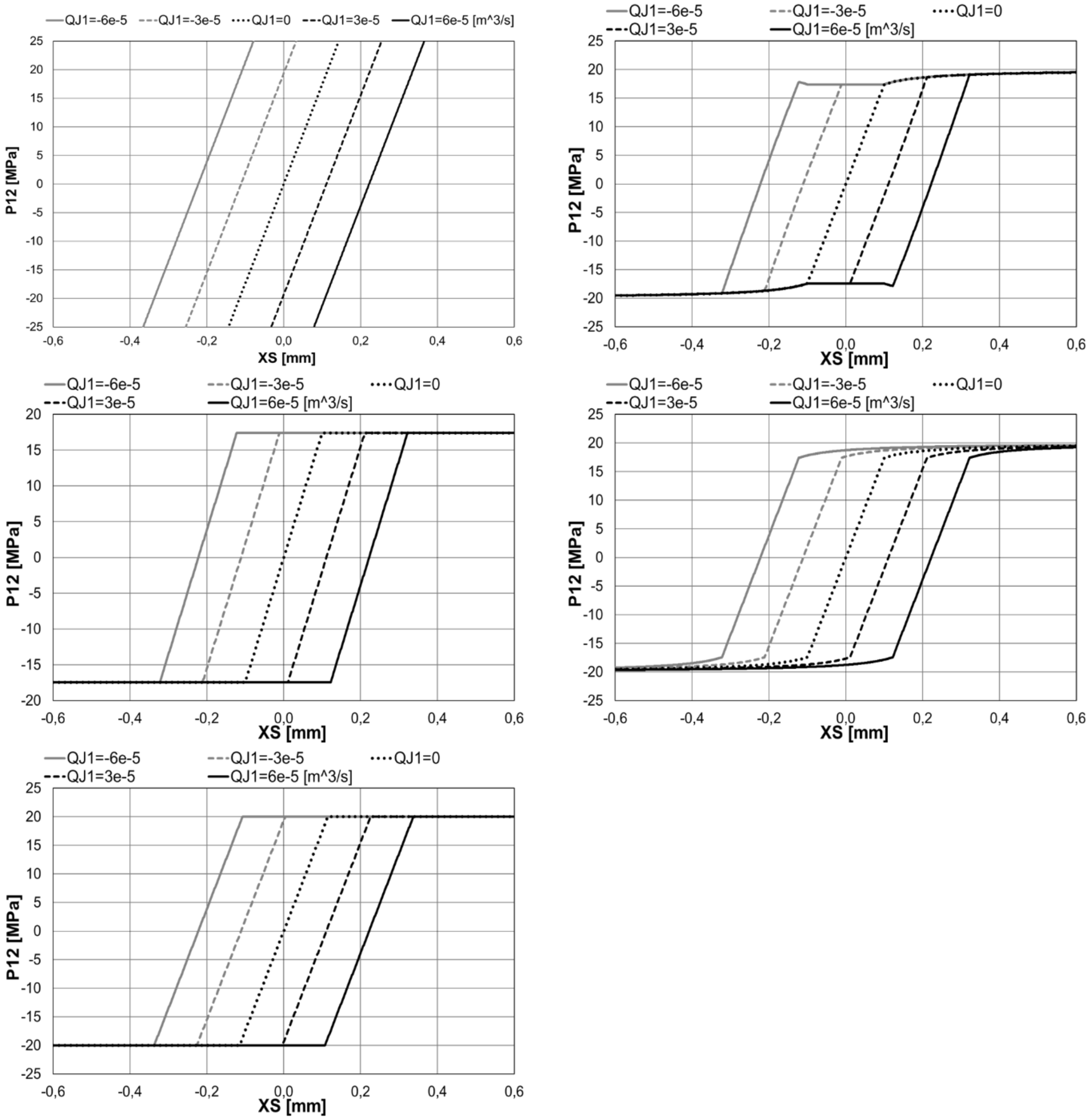


Figure 5: fluid-dynamic characteristic of Model A (top left), Model C1 (middle left), Model C2 (bottom left), Model C3 (top right), and Model C5 (middle right).

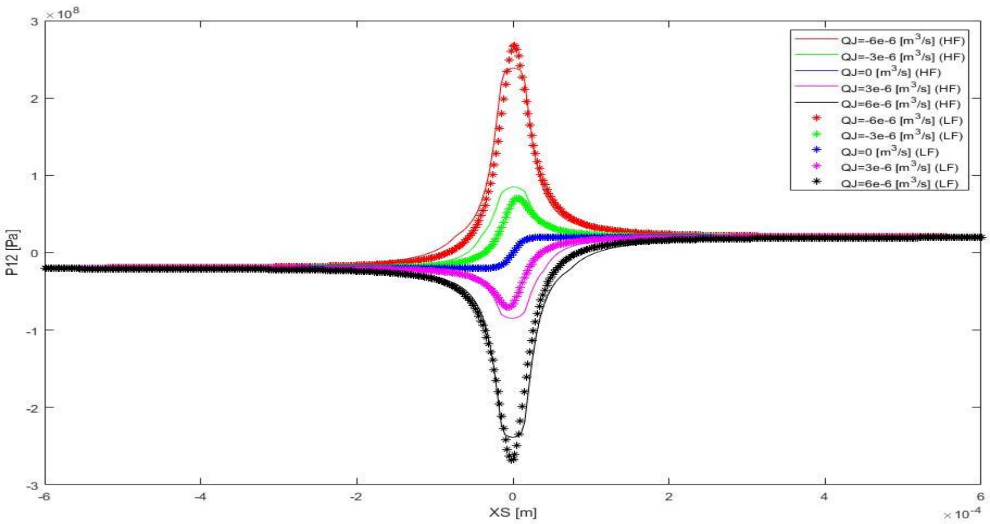


Figure 6: fluid dynamic characteristic of Model E compared with the HF model.

## 5. Concluding remarks

A simplified fluid dynamic model of an electrohydraulic servovalve was developed and tested. The model was able to overcome most of the issues associated with previous low fidelity emulators and approximate the behavior of a CFD simulation with good accuracy, including the water hammer, pressure saturation and leakage effects. At the same time, the computational time is low and suitable for real-time evaluation, as the formulation only requires to compute simple algebraic and trigonometric functions, without the need for iterative solvers.

Future work will focus on including the servovalve model into a complete dynamical simulation of an electrohydraulic actuator to compare its response with higher fidelity models and experimental data.

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