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NUMERICAL MODEL FOR HYDRAULIC SERVO-VALVES

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Abstract: *Hydraulic actuation systems remain essential in applications requiring high force density, robust operation and precise motion control. Recent advancements in electro-hydraulic architectures, intelligent sensing and energy-efficient components have significantly improved performance, reliability and sustainability. This study presents a coupled electro–mechanical–hydraulic model of a servo-valve in order to analyze its dynamic behavior under varying operating conditions. The model integrates electrical control signals, mechanical armature motion, and hydraulic flow to capture the system's transient response accurately. A numerical approach is employed to simulate key performance metrics, allowing evaluation of system stability during operation. The frequency analysis performed was intended to evaluate the dynamic response of the servo-valve at different excitation frequencies. The obtained results for the coupled electro–mechanical–hydraulic servo-valve model provide insight into the system's frequency response. The numerical results from the Bode analysis confirm the accuracy of the coupled model and its suitability for predicting real-world servo-valve performance. The results demonstrate that the coupled model provides a robust framework for design optimization, control strategy development and fault analysis in precision hydraulic applications.*

Keywords: *Fluid power, hydraulic actuation, servo valves, numerical model*

1. Introduction

Hydraulic actuation systems play a central role in industrial equipment, heavy machinery, aerospace structures and robotic systems where the key requirements include high power, reduced weight ratio, smooth motion control and reliability under harsh operating conditions.
Despite the emergence of high-power electric drives, hydraulics remains unmatched in many high-load applications due to the compressibility characteristics of fluids, high stiffness levels and the ability to generate large forces with compact actuators.
The last decade has seen significant innovation in hydraulic power control, particularly through electro-hydraulic interfaces, intelligent sensors and advanced control electronics, while all of this are allowing the improved efficiency, closed-loop accuracy and reduced environmental impact.
This paper investigates current trends in the design and optimization of hydraulic actuation systems, discussing both theoretical and practical aspects. Among all circuit elements servo-valves are critical components in precision hydraulic systems, translating electrical control signals into precise mechanical motion and hydraulic flow. Their performance depends on the complex interactions between electrical, mechanical, and hydraulic subsystems. Modeling the coupled electro–mechanical–hydraulic dynamics of servovalves provides a deeper understanding of their transient behavior, enabling improved design, control, and reliability in high-performance applications such as aerospace, robotics, and industrial automation.

2. Main principles of Fluid Power

Hydraulic actuation is based on transmitting power via pressurized fluid, while according to Pascal's principle, pressure value applied to a confined fluid is distributed uniformly.
The resulting force is given by 0, 0:

$$F = p \cdot A$$

where
F – output force,
p – system pressure,
A – piston surface area.
This fundamental relationship is central to the design of cylinders, motors and valves.
Regarding the system architecture a classical hydraulic actuation system includes a power source (pump, electric motor), a fluid reservoir, control valves (which can be directional, pressure, proportional, servo), actuators (linear or rotary), filtration and cooling subsystems, sensors for pressure, temperature and position.
Modern architectures integrate digital control, compact power units and modular valve manifolds 0, 0, 0.

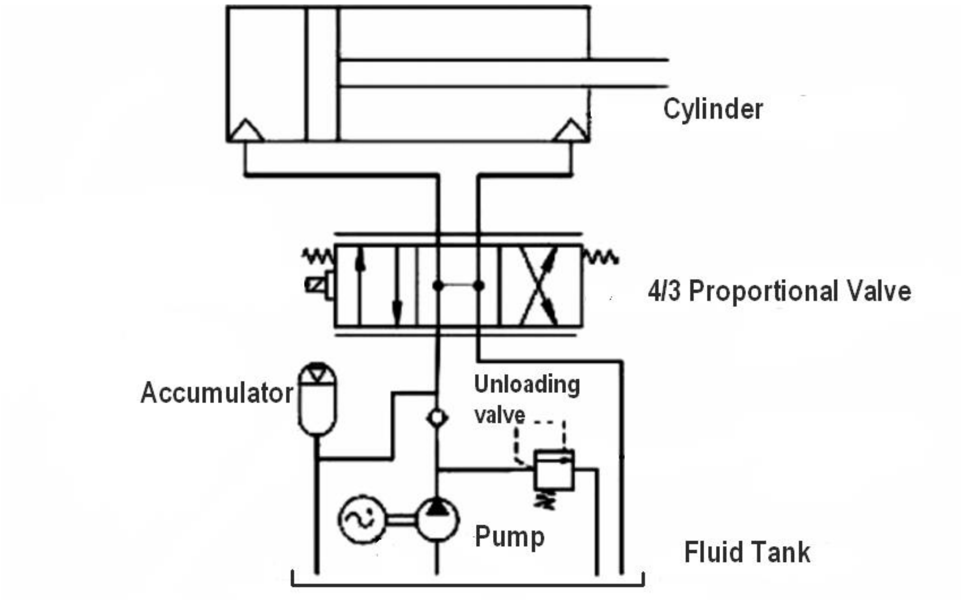


Fig. 1. Hydraulic circuit schematically representation

The physical and mathematical model for a hydraulic drive system includes fluid dynamic equations, cylinder mechanical equations, control valve modelling, linearization and transfer function.
Hydraulic valves are mechanical or electro-hydraulic elements that regulate or limit the pressure within the hydraulic and control circuits of hydraulic control, control and drive systems.

3. Hydraulic servo-valve numerical model

Typically, a hydraulic valve can be modeled on three levels represented by an electrical-mechanical model for servo-valves, a mechanical-hydraulic model for proportional valves and servo-valves and the fully nonlinear electro-mechanical-hydraulic model that is used in advanced control and precise simulations.
The electrical-mechanical model for servo valves includes the dynamics of the spool and the moving armature, since the spool produces electromagnetic force, then the armature that moves the valve element, as well as other assembly elements represented by an elastic spring-type element that provides damping and mechanical resistance.
The mechanical-hydraulic model for proportional valves and servo-valves has the ability to connect the spool position with the flow through the orifices, which means the variation of the orifice section that determines the volumetric flow rate, then the pressure differences that influence the flow rate and the return force of the spool.

The fully nonlinear electro-mechanical-hydraulic model combines electromechanical dynamics with fluid dynamics, including nonlinear effects.
The stages of modeling hydraulic valves represented by servo valves and proportional valves used in automation, industrial hydraulics and simulations of the operation of these components are presented.
The flow rate model through orifice (nonlinear model) for a proportional opening valve can be described as follows 0, 0:

$$Q = C_d A(x)\sqrt{\frac{2}{\rho}(p_1 - p_2)} \tag{1}$$

where:
$A(x)$ – nonlinear function of drawer position;
C_d - discharge coefficient;
ρ - fluid density.
The pressure model in actuator chambers can be described as 0, 0:

$$\begin{aligned} p_1 &= \frac{\beta}{V_1}(Q_1 - A_1\dot{y}) \\ p_2 &= \frac{\beta}{V_2}(Q_2 - A_2\dot{y}) \end{aligned} \tag{2}$$

where:
y - cylinder rod position;
β - compressibility modulus;
Q_1, Q_2 - the circulated flow rates through the valve.

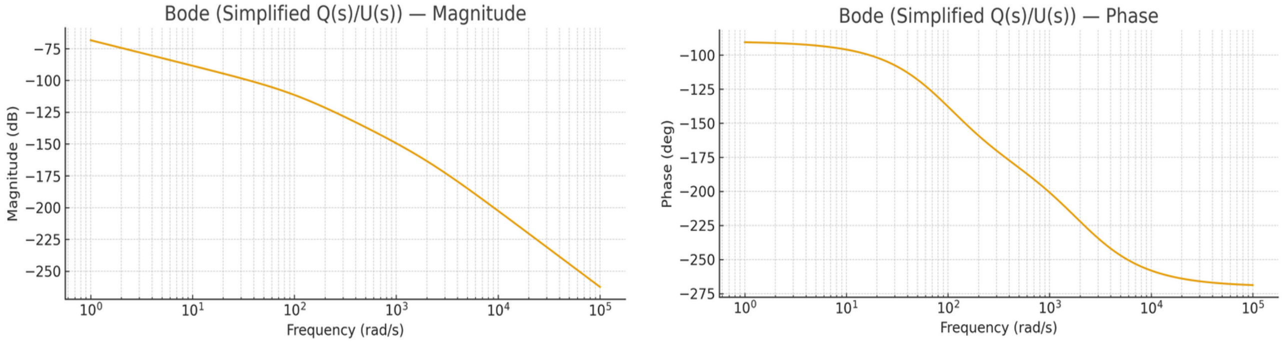
For a numerical analysis the servo-valve is represented using a coupled electro–mechanical–hydraulic model that captures the essential dynamic behavior of the device.
The electrical subsystem describes the current through the coil using a first-order differential equation based on the coil resistance and inductance.
The resulting current produces an electromagnetic force proportional to Kii, which drives the spool in the mechanical subsystem.
The spool motion is modeled as a second-order system including inertia, viscous damping, and optionally stiffness.
The hydraulic subsystem relates the spool displacement to the flow through the orifice, which follows a nonlinear square-root law with respect to the pressure drop. Around a chosen operating point, the flow equation is linearized, leading to the sensitivity coefficients *Gx* and *Gp* that describe the small variations in spool position and pressure which are able to affect the fluid flow.
When combined, these three subsystems form a transfer function from input voltage to output flow that exhibits distinct electrical and mechanical poles and captures the dominant bandwidth limitations of the servo-valve.
The main parameters included in the numerical analysis are presented in table 1.

Table 1: Model parameters used for numerical analysis

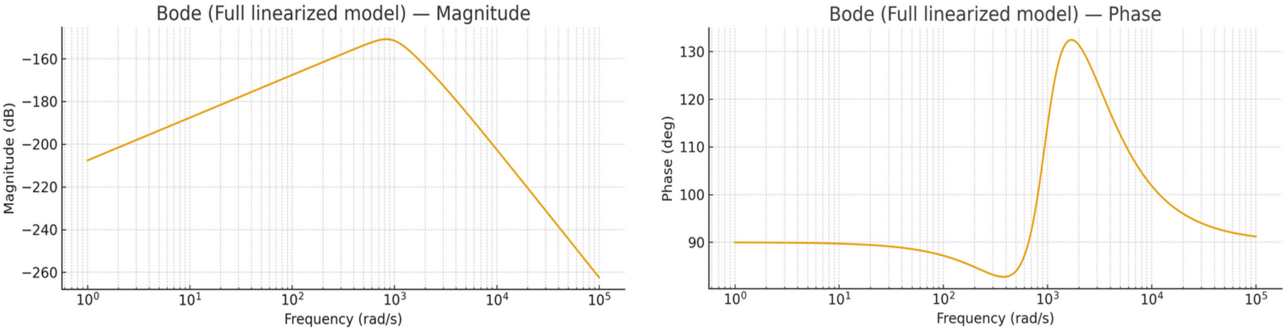
Symbol	Description	Value	Units	Physical Role in the Model
L	Coil inductance	0.001	H	Governs current dynamics; part of the electrical pole *(L s + R)*
R	Coil resistance	2	Ω	Defines electrical response speed; sets pole at −R/L
m	Spool mass	0.01	kg	Mechanical inertia; part of 2nd-order mechanical dynamics
b	Mechanical damping	1	N·s/m	Controls resonance damping; affects stability and bandwidth
K_i	Force–current coefficient	0.1	N/A	Converts coil current into force acting on the spool
Cd	Discharge coefficient	0.7	—	Describes turbulent flow efficiency through the orifice
w	Orifice width	1×10^{-4}	m	Geometric parameter of the valve slot
ρ	Fluid density	850	kg/m³	Affects flow rate value
x_0	Spool displacement at operating point	1×10^{-4}	m	Linearization point for flow derivative
Δp_0	Pressure drop at operating point	5×10^{6}	Pa	Sets steady-state flow and G_x,
G_x	∂Q/∂x (flow sensitivity to displacement)	0.00759	m³/(s·m)	Main flow gain; scales system magnitude
G_p	∂Q/∂Δp (flow sensitivity to pressure)	7.59×10^{-14}	m³/(s·Pa)	Determines hydraulic feedback strength
β	Bulk modulus of hydraulic fluid	$1.5–2\times10^{9}$	Pa	Determines pressure dynamics (dp/dt)

4. Results and discussion

The purpose of the present model was to analyze the dynamic behavior of the servo-valve and identify the factors that limit the frequency band and the stability of the hydraulic system.
By separating and coupling the electrical, mechanical and hydraulic subsystems, the model allows the evaluation of the influence of each component on the frequency response and on the sensitivity of the flow to voltage, position and pressure variations.
The linearized model was used in order to obtain Bode diagram plots and to determine the pole-zero structure, highlighting dominant poles, operating regimes and the effect of pressure on local gains.



a) Bode diagrams for magnitude and phase –simplified model



b) Bode diagrams for full linearized model

Fig. 2. The obtained results from the servo-valve model numerical analysis

The objective was to obtain a simplified but representative tool that allows the prediction of servo-valve performances and to support the design and optimization of hydraulic actuator systems.
The Bode diagrams obtained from the simplified transfer function highlight the dominant dynamic contributions of the electrical circuit and spool mechanics. The magnitude plot shows a clear low-frequency gain followed by two characteristic roll-off regions associated with the coil electrical pole and the second-order mechanical subsystem, while the phase curve confirms the corresponding phase lag accumulation.
In contrast, the Bode plots of the full linearized model exhibit an extended low-frequency dynamics due to hydraulic compressibility and flow-pressure coupling. This is reflected by an additional pole at low frequencies and a slight modification of the high-frequency slope, making the full model more representative of real servo-valve behavior.
The schematic diagram summarizes the coupling between electrical, mechanical, and hydraulic domains and visually supports the structure captured in the mathematical model.
The complete model, which includes hydraulic pressure dynamics and fluid compressibility, produces an additional low-frequency pole. This makes the phase curve more negative at very low frequencies and slightly reduces the system gain. However, the dominant bandwidth is still determined by the mechanical subsystem, with the hydraulic and electrical subsystems contributing secondary effects.
Overall, the results confirm that the servo-valve behaves like a cascaded mechanical–electrical system with strong low-frequency gain, limited bandwidth, and smooth roll-off.
The model captures the essential dynamic behavior and provides a more accurate representation of the low-frequency hydraulic effects.

5. Conclusions

The coupled electro–mechanical–hydraulic model for servo-valves provides a comprehensive framework for analyzing and predicting system behavior by integrating the electrical control, mechanical actuation, and hydraulic response.
The model captures the dynamic interactions between the electrical input, mechanical armature motion and hydraulic fluid flow, enabling more accurate predictions of valve response under varying load conditions.
It allows assessment of transient behavior such as overshoot, settling time and bandwidth, which are critical for precision control applications.
Coupled modelling criteria helps in optimizing parameters like armature mass, spring stiffness, damping, and spool geometry to achieve desired performance, minimize overshoot and enhance stability.
Integrating the multi-domain interactions makes it easier to detect and analyze anomalies, such as spurious oscillations due to mechanical resonance or hydraulic delay.
It provides a foundation for advanced control strategies, such as PID tuning, feedforward compensation, or adaptive control, improving system responsiveness and robustness.

In essence, the coupled electro–mechanical–hydraulic approach provides a holistic understanding of servo-valve behavior, which is essential for high-precision hydraulic control applications in aerospace, robotics and industrial automation, while the approach emphasizes the importance of multi-domain modelling in order to accurately capture real-world performance, enabling better design, control and reliability.

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