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A seventh-order model for dynamic response of an electro-hydraulic servo valve

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Abstract In this paper, taking two degrees of freedom on the armature–flapper assembly into account, a seventh-order model is deduced and proposed for the dynamic response of a two-stage electro-hydraulic servo valve from nonlinear equations. These deductions are based on fundamental laws of electromagnetism, fluid, and general mechanics. The coefficients of the proposed seventh-order model are derived in terms of servo valve physical parameters and fluid properties explicitly. For validating the results of the proposed model, an AMESim simulation model based on physical laws and the existing low-order models validated by other researchers through experiments are used to compare with the seventh-order model. The results show that the seventh-order model can reflect the physical behavior of the servo valve more explicitly than the existing low-order models and it could provide guidance more easily for a linear control design approach and sensitivity analysis than the AMESim simulation model.

1. Introduction

Two-stage electro-hydraulic servo valves are playing a significant role in high-precision electro-hydraulic servo-systems in which accurate position control is required and have a significant influence on the performance of the whole electro-hydraulic servo-systems. They are capable of converting low electrical signals for precise movements of spools to control high-power on low-speed hydraulic actuators.[1,2]

Many attempts for modeling and studying the dynamic characteristics of electro-hydraulic servo valves and their components have been carried out. For this research area, Merritt[1] has done a lot of useful work to explain the working principle of electro-hydraulic servo valves and proposed an ideal mathematical model for electro-hydraulic servo valves which has been widely distributed and followed by many authors of books and research papers. In later studies, considering the effects of some non-linearities in the form of transfer functions or state-space equations, many other researchers[3,4] established higher-order models which gave a more realistic explanation of the behavior of servo valves. Gordic et al.[5,6] investigated the effects of the variations of torque motor parameters on servo valve performance and also proposed a comprehensive mathematical model of spool position feedback servo valves.

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Focusing on the performance of a servo valve torque motor, Li et al.[7–9] studied the influence of magnetic fluids on the dynamic characteristics of the torque motor and observed that magnetic fluids could increase the stability of the torque motor. Urata[10–14] has done much useful and significant research on the effects of unequal air gap, magnetic reluctance and leakage flux of permanent magnets, and magnetic stiffness of torque motors. In recent years, Dasgupta and Murrenhoff[15] gave a comprehensive model of a closed-loop servo valve-controlled hydromotor drive system by using the bondgraph simulation technique. Mu and Li[16] deduced a non-linear model to improve the dynamic response of the main spool. Jiao et al.[17,18] investigated the model and matching design of electro-hydraulic load simulator control by a closed-loop servo valve. The cavitation phenomenon in the flapper-nozzle pilot stage of an electro-hydraulic servo valve with an innovative flapper shape was understood in the research of Li et al.[19,20]

However, in previous research, all have supposed an armature–flapper assembly with one degree of freedom. Merritt[1] proposed a third-order model and Kim[3,4] proposed a fifth-order model. The difference between Merritt's model and Kim's model is that the spool valve resonance and pressure feedback on the flapper are ignored in Merritt's model.

In this work, considering an armature–flapper assembly with two degrees of freedom, i.e., a rotation degree provided by electromagnetic torque and a translation degree provided by flow force, a new mathematical model for the dynamic response of two-stage electro-hydraulic servo valves is deduced. State equations are used during the deduction. While the nature of electro-hydraulic servo-systems is nonlinear, it is often desirable to have a linear model for a linear control design approach and sensitivity analysis. Therefore, an accurate linear model for electro-hydraulic servo-systems would be useful for valve design in tailoring the valve dynamics from a control standpoint as well as for high-performance control system design. This new mathematical model offers a seventh-order linear model for the dynamic response of two-stage electro-hydraulic servo valves. At the end of this paper, in order to verify the proposed seventh-order model, an AMESim simulation model based on physical laws and the existing low-order models validated by other researchers through experiments are used to compare with this seventh-order model. The results demonstrate that the seventh-order model can reflect the physical behavior of a servo valve more explicitly than the existing low-order models and it could provide guidance more easily for a linear control design approach and sensitivity analysis than the AMESim simulation model.

2. Working principle of electro-hydraulic servo valves

The schematic diagram of an electro-hydraulic servo valve is illustrated in Fig. 1. This type of electro-hydraulic servo valve consists of two stages[1,2]: the first stage is a double nozzle–flapper valve which consists of a toque motor, a flapper, two nozzles, and a feedback spring, and the second stage is a precision ground four-way control spool.

The function of the nozzle–flapper valve driven by the torque motor through electrical signals is like a hydraulic amplifier putting out a large hydraulic signal to control the position of the spool. Two variable throttle orifices are formed by the annular area between the nozzles and the flapper when the flapper moves between the two nozzles. Here, i_1 and i_2 are the input electrical signal applied to the coil, p_a and p_b are the inlet or outlet pressure of the load. When there is no electrical signal applied to the coil, the armature stays in the middle between the yokes. This leads the flapper be kept in the middle of the two nozzles, so the pressures at the ends of the spool are equal, and the force balance created by equal pressure in both end chambers holds the spool in a stationary position. When there is an electrical signal applied to the coil, it generates an electromagnetic torque on the armature ends to deflect the armature–flapper assembly from the neutral position. The flapper moves closer to one nozzle decreasing the flow area through this nozzle and letting that of the other nozzle increase. Hydraulic oil is jetted out from both of the nozzles to the flapper, in relation to the two fixed orifices on both sides, which therefore results in a differential pressure over the spool. This differential pressure drives the spool to slide, and the displacement of the spool valve will be fed back to the flapper by the feedback spring. The spool continues to move until the equilibrium between the pressure difference over the spool, the flow forces, and the feedback spring force reaches, and then the servo valve will deliver an output flow proportional to the input current.

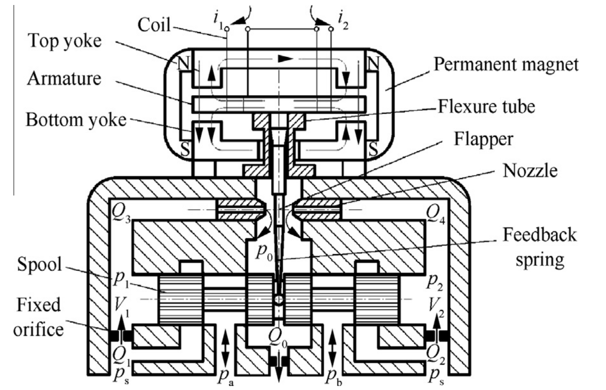

Fig. 1 Schematic diagram of a two-stage electro-hydraulic servo valve.

3. Mathematical model for dynamic response of servo valve

Fig. 2 shows the schematic diagram of the armature–flapper assembly. Considering the flow force working on the flapper,

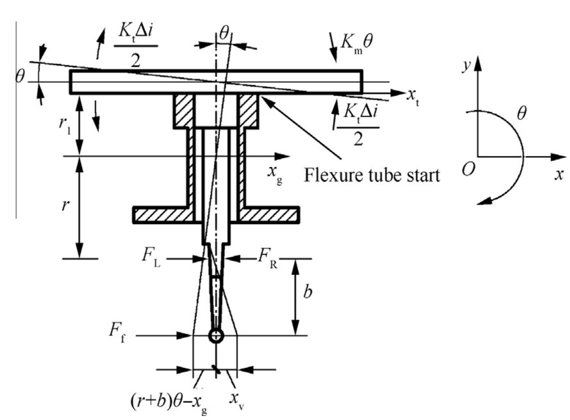

Fig. 2 Schematic diagram of the armature–flapper assembly.