Modeling and Parameter Sensitivity Analysis of Valve-controlled Helical Hydraulic Rotary Actuator System

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Title page

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ORIGINAL ARTICLE

Modeling and parameter sensitivity analysis of valve-controlled helical hydraulic rotary actuator system

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Abstract: As a kind of hydraulic rotary actuator, helical hydraulic rotary actuator has the excellent characteristics of large angle, high torque and compact structure, which has been widely used in various fields. However, the core technology is in the hands of several companies and has not been disclosed, and the relevant reports are mostly limited to the component level. From the perspective of designing the driving system, the dynamic characteristics of the output when the helical rotary actuator is applied to the closed-loop system are explored. There are two main problems to be studied: one is to establish a reliable mathematical model, and the other is to consider the influence of system parameter perturbation on the output in practice. In this paper, firstly, the dynamic model of valve-controlled helical rotary actuator angle closed-loop system is derived in detail, which has never been reported in the existing literature. Then, the sensitivity analysis of 23 main parameters in the model with perturbation of 10% is carried out under nine working conditions. Finally, the system dynamics model and the sensitivity analysis results are verified by the prototype experiment and co-simulation, which shows the reliability of the theoretical results in this paper.

Keywords: Helical hydraulic rotary actuator • Dynamic model, Closed-loop system • Sensitivity analysis • Prototype experiment • Co-simulation

1 Introduction

Helical hydraulic rotary actuator (HHRA) is a kind of hydraulic actuator that converts hydraulic energy into mechanical energy of rotary motion [1]. Compared with

the traditional rotary hydraulic actuator, such as the vane type, rack and gear type, helical rotary actuator has the advantages of large rotation angle, high output torque, compact structure and high volumetric efficiency due to its unique double helical pair structure [2,3]. The rotation angle range of the helical actuator can be designed to exceed 360°, and for the same rotation angle range, the helical actuator usually has smaller weight and volume, which makes the structure installed with the equipment more compact [4]. Another obvious advantage is that the helical actuator can achieve higher actuating pressure. It can still maintain high efficiency at 21MPa, which can not be realized for other forms of hydraulic rotary actuator [5]. Because of these outstanding characteristics, helical rotary actuator has been widely used in some special occasions that require large torque, large rotation angle and limited installation space, including agriculture machinery, construction machinery, mining machinery and so on [6,7]. However, only a few companies can produce such actuators at present [6-9], and they do not disclose core technologies such as design and manufacturing. The reports on HHRA, in addition to some patents [10-12], related research mainly focuses on the component level, including the research of structural design, dynamic model, flow field analysis, transmission efficiency and sealing [13-16].

As driving unit, the performance of valve-controlled actuator system directly affects the control performance of the whole machine [17]. Such systems are normally designed by mathematical models derived from constant coefficient ordinary differential equations [18]. But it faces two problems: one is that there are few articles on the research of helical rotary actuator driving system. Furthermore, the dynamic model of the rotary actuator is relatively simple in the existing literature [2,3], which may

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not be accurate enough when modeling analysis is needed. Therefore, there is a lack of reliable mathematical model of the hydraulic system for design and analysis. The other problem is that the parameters in the model are usually regarded as constants [19], but these parameters may not be consistent with the parameters in practical application, because the system parameters are actually time-varying [20]. For example, structural wear leads to parameter changes, inaccurate valve test parameters, pressure fluctuation of the hydraulic system, etc. To obtain the valve-controlled actuator system that meets the design requirements, the influence of parameter perturbation on the dynamic characteristics of the system is worth studying, so the sensitivity analysis of system parameters is involved.

There are many methods for sensitivity analysis, which can be classified into sensitivity function in time domain, frequency domain and performance-index sensitivity, and each sensitivity analysis method has its own advantages and application range [21]. Sensitivity analysis theory is widely used in mechanical design, power system, water conservancy system and so on [22-24]. In the application of hydraulic system, trajectory sensitivity analysis method in time domain is the most commonly used, which can be used for nonlinear systems with time-varying parameters. Vilenius et al. [20] introduced and applied a sensitivity model suitable for deadband type nonlinearities to a typical servo system in the sawmill industry. They simplified the valve-controlled actuator system into the fifth-order model and derived the first-order trajectory sensitivity mathematical expression of the system. Under a single working condition, the sensitivity of system parameters is given when the 10 main parameters change by 1%. Kim et al. [25] used the fourth-order state equation to analyze the nonlinear dynamics of the axial piston pump system, and analyzed the sensitivity of the parameters in the model by the first-order sensitivity equation. Pietola et al. [26,27] studied the transient-state and steady-state parameter sensitivity of a heavily loaded flexible electro-hydraulic position servo system, and compared three control methods of P-control, model reference control (MRC) and P + PID/x'-control. Farahat et al. [21] built the fourth-order linear mathematical model of electro-hydraulic servo position control system. The sensitivity of position control of valve-controlled actuator system to 18 main parameters changed by 1% is studied, and four sensitivity indexes are given. Kong et al. [28,29] established a sixth-order nonlinear model of the valve-controlled actuator system based on position control, and the influence degree and rules of 10% parameters change with respect to

displacement output response are obtained by the first-order trajectory sensitivity method. Then they deduced the method of second-order trajectory sensitivity and analyzed the effect on system displacement step response with parameter variation (10% and 20%) under nine working conditions. Ba et al. [30-32] proposed the method of first-order sensitivity matrix and applied it to the dynamic sensitivity analysis of the valve-controlled actuator system of legged robot. Huang et al. [33] used the first-order trajectory sensitivity method to find out the dominant parameters of the electro-hydraulic actuator in legged robot, and then estimated them online in real time.

The main difference between the above articles mainly lies in the degree of simplification of the hydraulic system model. For sensitivity analysis methods, they are similar, which are essentially based on the trajectory sensitivity. However, the hydraulic actuators of the system are all linear actuators in the existing research, and the sensitivity analysis of the helical rotary actuator driven system has not been reported. Compared with linear actuator, the structure of helical rotary actuator is complex and the model parameters are numerous, so the results of parameter sensitivity analysis may be different.

Starting from the purpose of designing and analyzing the dynamic characteristics of valve-controlled helical rotary actuator system, in this paper, a relatively accurate dynamic model of the actuator was established first, in which the mechanical analysis on a designed helical rotary actuator was carried out, and the friction between different parts and different load types was considered. Then, considering the nonlinear characteristics of pressure and flow at the valve port and the dynamic characteristics of servo valve, a sixth-order nonlinear state equation was established to describe the angular displacement closed-loop hydraulic system. Under different step input and load conditions, the sensitivity of 23 parameters in the closed-loop system model was solved by the first-order trajectory sensitivity analysis method. Finally, the dynamic model of actuator and sensitivity analysis results of the system are verified by experiments and co-simulation respectively, and a sensitivity index is used to explain the sensitivity results in detail.

This paper is organized as follows. In Section 2, the dynamic model of helical rotary actuator is deduced and the mathematical model of closed-loop system is established. The parameter sensitivity equation of the system is solved in Section 3. The experiments and co-simulation verification are given in Section 4. Section 5 summarizes the article.

2 Nonlinear dynamic model of system

The angular displacement closed-loop system of valve-controlled HHRA is the research object of this paper. In the following, the dynamic model of the rotary actuator will be established first, and then the servo valve and proportional controller will be introduced to establish the model of whole hydraulic system.

2.1 Basic working principle of HHRA

The basic structure of the designed HHRA is shown in Figure 1. It is mainly composed of a housing with an integral ring gear and two moving parts: the output shaft and the annular piston sleeve. A_1 and B_1 are the oil inlet and outlet. The helical spline teeth on the outside diameter of the piston engage matching the splines of the housing's ring gear (helical pair G_1), and the inside diameter of the piston carries a second set of splines, of opposite hand, which engage the matching splines of the output shaft (helical pair G_2). The seal ring on the piston divides the housing into two independent chambers. As hydraulic pressure is applied, the piston moves axially relative to the housing, similar to a hydraulic linear actuator. Simultaneously, due to the role of G_1 , the piston rotates and finally behaves as helical motion. When the piston is in helical motion, it drives the output shaft to rotate under the action of G_2 . Therefore, the rotation direction of the output shaft can be controlled by filling pressure oil into different oil ports (A_1 or B_1).

Figure 1 Structural diagram of helical hydraulic rotary actuator

2.2 Dynamic model of HHRA

According to the working principle described in the previous section, the structure sketch of HHRA is obtained (Figure 2), which is composed of piston (label 1), output shaft (label 2), housing (label 3) and bearings (label b1 and b2). The housing is fixed, and the axial displacement of the output shaft is limited by the bearings. The relative motion

parts are mainly G_1 and G_2 , that is, the helical motion of piston and the rotation of output shaft. For the convenience of analysis, it is assumed that the pressure oil is introduced into the left chamber of the actuator and the right chamber of the actuator returns oil. In this case, the motion state of the piston and shaft has been marked in Figure 2.

Figure 2 The structure sketch of HHRA when working

Kinematic analysis should be carried out first before dynamic modeling. Because HHRA belongs to a single degree of freedom system, once the displacement or rotation angle of the piston is determined, the motion state of the whole system is also determined. The state variables have the following kinematic relationship:

$$\varphi_1 = \frac{2\pi x_{a1}}{S_1} \quad (1)$$

$$\varphi_2^1 = \frac{2\pi x_{a1}}{S_2} \quad (2)$$

$$\varphi_2 = \varphi_1 + \varphi_2^1 = \frac{2\pi x_{a1}(S_1 + S_2)}{S_1 S_2} \quad (3)$$

where φ_1 is the rotation angle of the piston relative to the housing, φ_2^1 is the angle of the output shaft relative to the piston, φ_2 is the angle of the output shaft relative to the housing, x_{a1} is the axial displacement of the piston, S_1 and S_2 are the leads of G_1 and G_2 , respectively. After obtaining the kinematic relationship, the mechanical analysis of the piston and output shaft is carried out respectively, as shown in Figure 3.

The forces and moments on the piston are shown in Figure 3(a). Take the coordinate system direction in the figure as the positive direction, and the dynamic equation of piston can be expressed as follows:

Figure 3 Mechanical analysis of piston and output shaft

$$F_{hyd} - f_{1,3} - F_{1,3} - F_{1,2} = m_1 \ddot{x}_{a1} \quad (4)$$

$$T_{1,3} - T_{1,2} - T_{f1,3} = J_1 \ddot{\phi}_1 \quad (5)$$

where F_{hyd} is the resultant force of the oil pressure in the two chambers of actuator, m_1 is the mass of the piston, J_1 is the moment of inertia of the piston, $F_{1,3}$ and $F_{1,2}$ are the axial forces generated by G_1 and G_2 respectively, $T_{1,3}$ and $T_{1,2}$ are the torques generated by G_1 and G_2 respectively, $f_{1,3}$ and $T_{f1,3}$ are the force and torque caused by friction between the seal ring on the piston and the housing. The moments on the output shaft (since there is no movement in the axial direction of the output shaft, only the moments are drawn here) are shown in Figure 3(b). Considering the load characteristics, its dynamic equation is expressed as follows:

$$T_{2,1} - T_{2,b1} - T_{2,b2} = J_2 \ddot{\phi}_2 + B_l \dot{\phi}_2 + K_l \phi_2 + T_l \quad (6)$$

where $T_{2,1}$ is the torque generated by G_2 and $T_{2,1} = T_{1,2}$, $T_{2,b1}$ and $T_{2,b2}$ are the torques generated by bearing friction, J_2 is the moment of inertia of the output shaft and load, B_l is the load damping coefficient, K_l is the load stiffness coefficient, T_l is the external load torque, the force $F_{1,3}$, $F_{1,2}$ and torque $T_{1,3}$, $T_{1,2}$ caused by G_1 and G_2 can be obtained through the mechanical analysis of helical pair, as shown in Figure 4.

Since the influence of piston gravity on the helical pair is related to the installation position, to simplify the analysis, it is assumed that the output shaft is installed horizontally, so the influence of gravity can be ignored. As shown in the Figure 4(a) and Figure 4(b), $F_{1,3}$, $T_{1,3}$, $F_{1,2}$ and $T_{1,2}$ are calculated as follows:

$$F_{1,3} = F_{x'1,3} = N_{1,3} (\cos \lambda_1 + \mu_1 \sin \Psi_1) \quad (7)$$

$$T_{1,3} = \frac{F_{y'1,3} d_1}{2} = \frac{N_{1,3} (\sin \lambda_1 - \mu_1 \cos \Psi_1) d_1}{2} \quad (8)$$

$$F_{1,2} = F_{x'1,2} = N_{1,2} (\cos \lambda_2 + \mu_2 \sin \Psi_2) \quad (9)$$

$$T_{1,2} = \frac{F_{y'1,2} d_2}{2} = \frac{N_{1,2} (\sin \lambda_2 - \mu_2 \cos \Psi_2) d_2}{2} \quad (10)$$

where $N_{1,3}$ and $N_{1,2}$ are the normal forces produced by G_1 and G_2 respectively, μ_1 and μ_2 are the dynamic friction coefficients of G_1 and G_2 , Ψ_1 and Ψ_2 are the helix angle of G_1 and G_2 , d_1 and d_2 are the pitch diameters of G_1 and G_2 . The oil pressure F_{hyd} can be expressed as:

$$F_{hyd} = (p_1 - p_2) A_p \quad (11)$$

where p_1 is the oil pressure in the left chamber of the actuator and p_2 is the pressure in the right chamber, A_p is the effective action area of piston. According to equations (1) ~ (11), the dynamic equation of HHRA can be expressed as follows:

$$(p_1 - p_2) A_p = A \ddot{\phi}_2 + B \dot{\phi}_2 + C \phi_2 + D \quad (12)$$

where

$$A = Q_1 \cdot \left(\frac{J_1 S_2}{S_1 + S_2} + J_2 \right) + Q_2 \cdot J_2 + \frac{m_1 S_1 S_2}{2\pi(S_1 + S_2)},$$

$$B = (Q_1 + Q_2) \cdot B_l, \quad C = (Q_1 + Q_2) \cdot K_l,$$

$$D = Q_1 \cdot (T_{f1,3} + T_{2,b1} + T_{2,b2} + T_l) +$$

$$Q_2 \cdot (T_{2,b1} + T_{2,b2} + T_l) + f_{1,3},$$

$$Q_1 = \frac{2(\cos \lambda_1 + \mu_1 \sin \Psi_1)}{d_1(\sin \lambda_1 - \mu_1 \cos \Psi_1)}, \quad Q_2 = \frac{2(\cos \lambda_2 + \mu_2 \sin \Psi_2)}{d_2(\sin \lambda_2 - \mu_2 \cos \Psi_2)}$$

Figure 4 Mechanical analysis of S1 and S2 helical pairs

2.3 Dynamic model of valve-controlled HHRA system

This paper studies the angle closed-loop system using servo valve, and the schematic diagram of valve-controlled actuator system is shown in the Figure 5. The flow at the valve port can be expressed as follows:

$$q_1 = \begin{cases} K_d x_v \sqrt{p_s - p_1} & \text{for } x_v \geq 0 \\ K_d x_v \sqrt{p_1 - p_0} & \text{for } x_v < 0 \end{cases} \quad (13)$$

$$q_2 = \begin{cases} K_d x_v \sqrt{p_2 - p_0} & \text{for } x_v \geq 0 \\ K_d x_v \sqrt{p_s - p_2} & \text{for } x_v < 0 \end{cases} \quad (14)$$

where x_v is the valve spool displacement, p_s is the supply pressure, p_0 is the return oil pressure, K_d is the converted flow coefficient. Taking into account the influence of oil leakage and oil compressibility, the flow continuity equation of HHRA can be expressed as follows:

$$q_1 = A_p \dot{x}_{a1} + \frac{(V_{g1} + A_p L_0 + A_p x_{a1})}{\beta_e} \dot{p}_1 + C_{ip}(p_1 - p_2) + C_{ep} p_1 \quad (15)$$

$$q_2 = A_p \dot{x}_{a1} - \frac{(V_{g2} + A_p(L - L_0) - A_p x_{a1})}{\beta_e} \dot{p}_2 + C_{ip}(p_1 - p_2) - C_{ep} p_2 \quad (16)$$

where β_e is the effective bulk modulus of system, V_{g1} and V_{g2} are the volume of oil inlet channel and oil outlet channel connecting valve and actuator, L is the maximum axial stroke of piston, and L_0 is the initial axial position of piston, C_{ip} is the internal leakage

coefficient of actuator, and C_{ep} is the external leakage coefficient.

Figure 5 Schematic diagram of valve-controlled actuator system

The second order differential equation is used to approximate the dynamics of the servo valve:

$$\ddot{x}_v + 2\zeta\omega\dot{x}_v + \omega^2 x_v = K_{avv}\omega^2 I \quad (17)$$

where ζ is the damping ratio of the valve, ω is the natural frequency of the valve, K_{avv} is the gain of the valve and I is the input signal of the valve. And the characteristics of angle closed-loop control can be described by the equation:

$$I = K_p(u - \varphi_2) \quad (18)$$

where K_p is the gain of P-controller and u is the input signal. The block diagram of valve-controlled HHRA system based on angular displacement closed-loop control

is established by equations (12) ~ (18), as shown in Figure 6.

Figure 6 Block diagram of angle closed-loop system

The state space equation is used to describe the sixth-order system, and the selection of state variables is as follows:

$$\mathbf{x} = [x_1, x_2, x_3, x_4, x_5, x_6]^T = [\varphi_2, \dot{\varphi}_2, x_v, \dot{x}_v, p_1, p_2]^T$$

The state equation of valve-controlled HHRA system is obtained:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \end{bmatrix} = \begin{bmatrix} x_2 \\ -\frac{B}{A}x_2 - \frac{C}{A}x_1 + \frac{A_p}{A}x_5 - \frac{A_p}{A}x_6 - \frac{D}{A} \\ x_4 \\ \omega^2 K_p K_{axv} u - \omega^2 K_p K_{axv} x_1 - 2\zeta\omega x_4 - \omega^2 x_3 \\ \frac{\beta_e}{V_{g1} + \Gamma(\varphi_{20} + x_1)} [K_d x_3 \sqrt{A_1} - (C_{ip} + C_{ep})x_5 + C_{ip}x_6 - \Gamma x_2] \\ \frac{\beta_e}{V_{g2} + \Gamma(\varphi_{2max} - \varphi_{20} - x_1)} [-K_d x_3 \sqrt{A_2} - (C_{ip} + C_{ep})x_6 + C_{ip}x_5 + \Gamma x_2] \end{bmatrix} \quad (19)$$

where $\Gamma = \frac{A_p S_1 S_2}{2\pi(S_1 + S_2)}$,

$$A_1 = \frac{[1 + \text{sgn}(x_v)]p_s}{2} + \frac{[-1 + \text{sgn}(x_v)]p_0}{2} - \text{sgn}(x_v)p_1,$$

$$A_2 = \frac{[1 - \text{sgn}(x_v)]p_s}{2} + \frac{[-1 - \text{sgn}(x_v)]p_0}{2} + \text{sgn}(x_v)p_2,$$

φ_{20} is the initial angle of output shaft, φ_{2max} is the maximum angle of output shaft.

3 Sensitivity model and analysis

The actual system parameters always have varying degrees of perturbation. When the system is sensitive to some parameters, this perturbation will make the actual output very different from the mathematical model, which may deviate from the design requirements. Analyzing parameter sensitivity can help us find these key parameters, so as to take response measures to reduce parameter sensitivity or optimize parameters. In this paper, the first-order sensitivity method is selected to analyze the influence of 10% perturbation from the nominal parameter value on the rotation angle of output shaft φ_2 , because this method is accurate enough and easier to calculate than the high-order

sensitivity method [29].

3.1 The first-order trajectory sensitivity equation of system

There are many different types of parameters in the system dynamics model according to (19), including the structural parameters of the HHRA, the working parameters, the parameters of servo valve and the control parameters. The fluctuations of different types of parameters have different time scales. This paper selects 23 main parameters in the system model, which cover all types, as shown below:

$$\alpha = [J_1, J_2, S_1, S_2, \Psi_1, \Psi_2, \mu_1, \mu_2, d_1, d_2, m_1, A_p, \varphi_{2max}, \varphi_{20}, \beta_e, C_{ip}, K_d, p_s, p_0, \omega, \zeta, K_{axv}, K_p]$$

For the actual structural parameters of the HHRA ($\alpha_1 \sim \alpha_{13}$), they are related to the manufacturing errors and the wear degree of the corresponding parts. Therefore, they will not change in the short term, but may change after the system runs for a long time due to wear. Affected by change of gas content, oil temperature and load conditions, the working parameters ($\alpha_{14} \sim \alpha_{19}$) are not constant, but in

a state of rapid dynamic fluctuation. Because the nonlinear high-order dynamic characteristics of servo valve are simplified into second-order differential equation, its real dynamic characteristics cannot be fully described. Therefore, the parameters of servo valve ($\alpha_{20} \sim \alpha_{22}$) are not fixed, which are affected by the input signal and working parameters. For the control parameter α_{23} , it needs to be dynamically adjusted with the change of system conditions to maintain the robustness of the system performance. As the above parameters may have different degrees of perturbation, it is necessary to analyze the influence of the above parameters variation on the output characteristics of the system to improve the performance of system.

The dynamics equations and sensitivity equations of the closed-loop system can be expressed as follows [20]:

$$\begin{cases} \dot{\mathbf{x}} = \mathbf{f}(\mathbf{x}, \mathbf{u}, \mathbf{a}, t) & (a) \\ \dot{\lambda}^i = \left(\frac{\partial \mathbf{f}}{\partial \mathbf{x}} \right)_n \lambda^i + \left(\frac{\partial \mathbf{f}}{\partial \mathbf{a}_i} \right)_n & (b) \end{cases} \quad (20)$$

where $\lambda^i = \left(\frac{\partial \mathbf{x}}{\partial \mathbf{a}_i} \right)_n$ is the sensitivity function of parameter $\mathbf{a}_i$, $i = 1 \dots 23$.

Denote the vector as follows:

$$\mathbf{x} = [x_1, x_2, x_3, x_4, x_5, x_6]^T, \quad \mathbf{u} = [u, T_l]^T$$

Assuming that the initial state is zero state, then the initial conditions for (20) are:

$$\begin{cases} \mathbf{x}_0 = \mathbf{0} \\ \lambda_0^i = \left(\frac{\partial \mathbf{x}_0}{\partial \mathbf{a}_i} \right)_n = \mathbf{0} \end{cases} \quad (21)$$

Once the sensitivity functions are obtained, the variation of state caused by parameter perturbation can be determined by the following equation (using Taylor theorem and ignoring higher-order terms):

$$\delta \mathbf{x} = \left(\frac{\partial \mathbf{x}}{\partial \mathbf{a}} \right)_n \delta \mathbf{a} \quad (22)$$

3.2 The first-order trajectory sensitivity equation of system

To solve equations (20), firstly, the values of nominal parameters are determined, as shown in Table 1. The structural parameters of HHRA are consistent with those in

the 3D model. The parameters of servo valve are got from dynamic characteristic curve in the stylebook. The values of working parameters, control parameters or other parameters that are difficult to determine are selected based on the application requirements and engineering experience.

Table 1 Nominal value of parameters

Parameter	Description	Value	Unit
$\alpha_{1n} = J_1$	Moment of inertia of the piston	8.368×10^{-4}	$\text{kg} \cdot \text{m}^2$
$\alpha_{2n} = J_2$	Moment of inertia of the output shaft	1.057×10^{-4}	$\text{kg} \cdot \text{m}^2$
$\alpha_{3n} = S_1$	Lead of G1	0.12	m
$\alpha_{4n} = S_2$	Lead of G2	0.09	m
$\alpha_{5n} = \psi_1$	Helix angle of G1	29.324	°
$\alpha_{6n} = \psi_2$	Helix angle of G2	40.962	°
$\alpha_{7n} = \mu_1$	Dynamic friction coefficients of G1	0.005	-
$\alpha_{8n} = \mu_2$	Dynamic friction coefficients of G2	0.005	-
$\alpha_{9n} = d_1$	Pitch diameter of G1	0.068	m
$\alpha_{10n} = d_2$	Pitch diameter of G2	0.033	m
$\alpha_{11n} = m_1$	Mass of piston	0.969	kg
$\alpha_{12n} = A_p$	Effective action area of piston	3.951×10^{-3}	m^2
$\alpha_{13n} = \phi_{2\max}$	Maximum angle of output shaft	178.5	°
$\alpha_{14n} = \phi_{20}$	Initial angle of output shaft	14	°
$\alpha_{15n} = \beta_e$	Effective bulk modulus of system	6.85×10^8	N/m^2
$\alpha_{16n} = C_{ip}$	Internal leakage coefficient of actuator	3×10^{-14}	$\text{m}^3 / (\text{s} \cdot \text{Pa})$
$\alpha_{17n} = K_d$	Converted flow coefficient	4.95×10^{-4}	$\text{m}^2 / (\text{s} \sqrt{\text{Pa}})$
$\alpha_{18n} = p_s$	Supply pressure	18×10^6	Pa
$\alpha_{19n} = p_0$	Return oil pressure	0.1×10^6	Pa
$\alpha_{20n} = \omega$	Natural frequency of the valve	134	rad/s
$\alpha_{21n} = \zeta$	Damping ratio of the valve	1.1	-
$\alpha_{22n} = K_{axv}$	Gain of the valve	0.433	m/A
$\alpha_{23n} = K_p$	Gain of P controller	0.001	-

Secondly, the state equations of (20a) are established and solved based on MATLAB SIMULINK to get the state $\mathbf{x}$ of every moment. It is generally believed that the step input is the most severe working state of the system, so the step signal is used as the input signal to evaluate the dynamic performance of the system. The dynamic response characteristics of the system are different under different given values of angular displacement and load conditions. Therefore, in order to cover various working conditions as much as possible, the step signals of 45°, 90° and 135° are used as the inputs and the external load torques are taken as no-load, 100N·m and 200N·m respectively. The step response of angular displacement closed-loop system is solved under nine working conditions, and all working

conditions are shown in Table 2.

Table 2 Working conditions

Condition	Amplitude (°)	Load torque (N·m)
1	45	0
2	90	0
3	135	0
4	45	100
5	90	100
6	135	100
7	45	200
8	90	200
9	135	200

After solving the state equations, the state value of each

moment within the calculation duration is obtained. The results are brought into the coefficient matrix and free matrix of equations (20b). Programming in MATLAB, the fourth-order Runge-Kutta method is used to solve the first-order trajectory sensitivity equations. Finally, the sensitivity functions of angular displacement step response to 23 parameters are obtained under nine different conditions. Because the sensitivity function curves of some parameters have different orders of magnitude under no-load and load conditions, for brevity, only the sensitivity function solutions of different step signals under no-load conditions are given here, as shown in Figure 7.

Figure 7 Sensitivity function of each parameter

The curve in Figure 7 shows the dynamic sensitivity of each parameter to the rotation angle of the output shaft within 0.3 seconds. Due to the influence of different ranges of nominal values, the sensitivity functions of different parameters may have different orders of magnitude, which makes it hard to intuitively compare the sensitivity of various parameters from the figure. To make the results

clearer, equation (22) was used, and the variation of angle is obtained according to the sensitivity function of different parameters, so as to uniformly characterize the influence of each parameter. Here, the influence of 10% fluctuation of each parameter on x_1 under the above nine conditions is given, as shown in the Figure 8.

Figure 8 Change of rotation angle corresponding to each parameter under nine conditions

In Figure 8, it is obvious that under the no-load conditions, the parameters more sensitive to the rotation angle are $\alpha_{12}, \alpha_{17}, \alpha_{22}, \alpha_{23}$, in which the curves corresponding to α_{17}, α_{22} and α_{23} are exactly the same. The parameters $\alpha_3, \alpha_4, \alpha_{18}, \alpha_{20}, \alpha_{21}$ take the second place, and other parameters have little effect on the rotation angle. However, with the increase of load, the parameters $\alpha_5, \alpha_6, \alpha_9, \alpha_{10}$ gradually become sensitive, and their influence on the rotation angle can not be ignored.

Meanwhile, some parameters will fluctuate in the initial stage. The amplitude of fluctuation increases with the increase of load, but it will decay rapidly. Only qualitative results can be obtained here, and a more detailed quantitative analysis of the sensitivity of each parameter will be carried out in Section 4.3.

4 Experiment and co-simulation verification

A valve-controlled actuator test platform was built to verify the accuracy of the system dynamics model in this paper and the theoretical calculation results of parameter sensitivity are verified by co-simulation. For validation of sensitivity theoretical results, the impact of parameter changes on the system are needed to study, which are very small changes. However, small changes in some parameters are difficult to measure accurately in the prototype experiment. For example, the system pressure, there is pressure pulsation in itself and the pressure sensor may also be affected by noise, so it is difficult to accurately increase or decrease the pressure by about 10%. What's more, structural parameters or some working parameters can not be changed in the prototype experiment. For these reasons, the verification of the results of sensitivity analysis was completed by co-simulation of MATLAB and

Adams in this paper.

4.1 The first-order trajectory sensitivity equation of system

The valve-controlled HHRA test platform and system schematic diagram are shown in Figure 9. The closed-loop system mainly includes a rotary actuator, a servo valve, a valve controller, an encoder, a diaphragm coupling, a magnetic powder brake and some auxiliary connections. One end of the output shaft of the actuator is equipped with the encoder, and the other end is connected with the brake through a coupling. The encoder feeds back the angle signal to the valve controller, which processes the desired signal and feedback signal and outputs the control signal to the servo valve. The brake is used for loading and the load torque can be set by its proprietary controller.

Figure 9 Test platform and schematic diagram for valve-controlled HHRA system

The setting of experimental parameters is basically consistent with the nominal values in Table 1, and step response experiments are carried out under the above nine working conditions. The data of rotation angle signal fed back by the encoder is written into the computer through the acquisition card and the acquisition frequency is 1000Hz. After filtering the data, the step curves under all working conditions are obtained, as shown in the Figure 10(a). Take the rise time as the evaluation index of the system dynamic performance (i.e. the time when the initial value reaches 90% of the steady-state value), and compare the experimental results with the calculation results of the theoretical model, as shown in Figure 10(b). From the

comparison results, it can be known that under various working conditions, the maximum error between the theoretical rise time and the experimental value is no more than 7%, which verifies the accuracy of the system dynamics model established in this paper.

4.2 Verification of sensitivity solution results

As mentioned above, this section will complete the verification of sensitivity solution through co-simulation. Firstly, the co-simulation model is established based on MATLAB and Adams platform, and then four parameters with fast change in time scale are selected for sensitivity verification.

Figure 10 Step response test curve and rise time under nine working conditions

4.2.1 Co-simulation model

First, the 3D model is imported into Adams and constraints and contacts are added. For moving parts, a revolute joint is added to the output shaft, and a cylindrical joint is added to the piston. For other parts such as housing and bearing, fixed pairs are added. Contacts are added between the piston and output shaft as well as the piston and housing. Hertz contact theory is used to calculate the contact force, and the corresponding contact parameters are shown in Table 3.

Table 3 Setting of contact parameters

No.	Description	Value	Unit
1	Stiffness	1×10^8	N/m
2	Force exponent	1.5	-
3	Damping	5×10^4	N·s/m
4	Penetration depth	1×10^{-4}	m
5	Dynamic coefficient	0.005	-

Then export the Adams controls plant. The hydraulic thrust F_{hyd} is used as the input signal of the system, and the output signal is the rotation angle and angular velocity of the output shaft. Finally, combined with equation (13) ~ (18) and the Adams controls plant, the co-simulation model for valve-controlled HHRA system was established in MATLAB SIMULINK, as shown in the Figure 11. Next, the parameter sensitivity problem will be solved on the co-simulation model and compared with the theoretical calculation results to complete the verification.

4.2.2 Verification of parameter sensitivity

For the sensitivity solution on the co-simulation model, the method adopted is to change a certain parameter by 10%,

keep the other parameters at the nominal value, and conduct simulation to obtain the step response curve of the output shaft. Then, by making a difference between the obtained curve and the curve before the parameter changes, the influence of the parameter change on the dynamic curve of the rotation angle can be obtained, which will be compared with the results in Figure 8.

As mentioned in Section 3.2, parameters $\alpha_3, \alpha_4, \alpha_5, \alpha_6, \alpha_9, \alpha_{10}, \alpha_{12}, \alpha_{17}, \alpha_{18}, \alpha_{20}, \alpha_{21}, \alpha_{22}, \alpha_{23}$ are more sensitive to step response than other parameters, so these parameters need more attention. Parameters $\alpha_3, \alpha_4, \alpha_5, \alpha_6, \alpha_9, \alpha_{10}, \alpha_{12}$ are the structural parameters of the actuator, which will not change in a short time under normal working conditions, and the sensitivity curves of parameters $\alpha_{17}, \alpha_{22}, \alpha_{23}$ in Figure 8 are exactly the same. Considering parameters that change rapidly over time and for the convenience of expression, only parameters $\alpha_{18}, \alpha_{20}, \alpha_{21}, \alpha_{23}$ were selected for verification. The comparison between the theoretical results and co-simulation results is shown in Figures 12~14.

From the curve, although the shape of the theoretical and experimental curves is not completely consistent due to the influence of contact parameters or simulation step size or some other factors, the trend and range of the curves are almost the same for each parameter under different input conditions. This consistency proves the accuracy of the solution results of parameter sensitivity in the previous section, and also shows the effectiveness of using first-order trajectory sensitivity analysis method to analyze the parameter sensitivity of the system. Since the sensitivity analysis is used to evaluate influence degree of parameters variation on system output, the small error between the theoretical and experimental dynamic curves has little effect on the analysis results. Therefore, the

method in this paper can be safely used to analyze similar parameter sensitivity problems.

Figure 11 Co-simulation model in MATLAB SIMULILNK

Figure 12 Comparison of theoretical and co-simulation values: the change of rotation angle corresponding to each parameter under no-load conditions

Figure 13 Comparison of theoretical and co-simulation values: the change of rotation angle corresponding to each parameter under 100N·m conditions

Figure 14 Comparison of theoretical and co-simulation values: the change of rotation angle corresponding to each parameter under 200N·m conditions

4.3 Results analysis

In this section, the sensitivity solution will be analyzed in more detail. From Figure 8, it can be found that the sensitivity function of each parameter changes with time. To quantify this dynamic process, the sensitivity index needs to be determined first. Farahat gave four sensitivity indexes in his article, and proposed that Entirety-Index is the best method because it comprehensively considers all the performance characteristics. In this paper, the same method is used as the sensitivity index, and its expression is [21]:

$$I_E = \int_0^T |\lambda_i^j(t)| \delta \alpha_j dt \quad (23)$$

According to equation (23), the curves in Figure 8 are integrated to obtain the sensitivity index values of the 23 parameters under nine working conditions. The results are shown in Figures 15~17, where only the parameters with large sensitivity index are given. For parameters not shown in the figure, the sensitivity index values are very small, and their influence can be ignored. The height of the column in the figure represents the integration of the absolute change of the output shaft rotation angle within the simulation time (0.3s). Generally speaking, the larger the value, the more sensitive the output is to parameters.

It can be seen from Figures 15~17 that when the system is under the same load conditions, the larger the step signal, the greater the sensitivity index for a parameter. Moreover, the sensitivity index ranking of parameters is the same when different step input signals are given. For no-load conditions, the order from high to low according to the sensitivity index of parameters is:

$$(I_E^0) \alpha_{12} = \alpha_{17} = \alpha_{22} = \alpha_{23} > \alpha_4 > \alpha_{18} > \alpha_3 > \alpha_{20} > \alpha_{21} > \alpha_{others} \quad (24)$$

The corresponding quantitative relationship is:

$$\begin{aligned} (I_E^0) \alpha_{12} = \alpha_{17} = \alpha_{22} = \alpha_{23} &= 1.75 \alpha_4 \\ &= 1.95 \alpha_{18} = 2.33 \alpha_3 = 3.06 \alpha_{20} \\ &= 3.11 \alpha_{21} > \alpha_{others} \end{aligned} \quad (25)$$

For 100N. m load conditions, there are:

$$\begin{aligned} (I_E^{100}) \alpha_{17} = \alpha_{22} = \alpha_{23} &> \alpha_{12} > \alpha_{18} > \alpha_4 \\ &> \alpha_3 > \alpha_{20} > \alpha_{21} > \alpha_6 > \alpha_{10} \\ &> \alpha_5 > \alpha_9 > \alpha_{others} \end{aligned} \quad (26)$$

$$\begin{aligned} (I_E^{100}) \alpha_{17} = \alpha_{22} = \alpha_{23} &= 1.12 \alpha_{12} = 1.62 \alpha_{18} \\ &= 1.75 \alpha_4 = 2.33 \alpha_3 = 3.45 \alpha_{20} \\ &= 3.5 \alpha_{21} = 11.2 \alpha_6 = 16.1 \alpha_{10} \\ &= 17.5 \alpha_5 = 21.2 \alpha_9 > \alpha_{others} \end{aligned} \quad (27)$$

And for 200N. m load conditions, there are:

$$\begin{aligned} (I_E^{200}) \alpha_{17} = \alpha_{22} = \alpha_{23} &> \alpha_{18} > \alpha_{12} > \alpha_4 \\ &> \alpha_3 > \alpha_{20} > \alpha_{21} > \alpha_6 > \alpha_{10} \\ &> \alpha_5 > \alpha_9 > \alpha_{others} \end{aligned} \quad (28)$$

$$\begin{aligned} (I_E^{200}) \alpha_{17} = \alpha_{22} = \alpha_{23} &= 1.28 \alpha_{18} = 1.37 \alpha_{12} \\ &= 1.75 \alpha_4 = 2.33 \alpha_3 = 4 \alpha_{20} \\ &= 4.06 \alpha_{21} = 4.42 \alpha_6 = 6.39 \alpha_{10} \\ &= 7 \alpha_5 = 8.43 \alpha_9 > \alpha_{others} \end{aligned} \quad (29)$$

Figure 15 Histogram of sensitivity index under different step input and no-load conditions

Figure 16 Histogram of sensitivity index under different step input and 100 N · m conditions

Figure 17 Histogram of sensitivity index under different step input and 200N·m conditions

It can be seen from expressions (24) ~ (29) that when the system is under different load conditions, the sensitivity ranking of parameters and the numerical relationship between parameters change. In order to

Figure 18 Histogram of sensitivity index corresponding to parameters under nine working conditions

5 Conclusion and future work

In this paper, a complete mathematical model of valve-controlled HHRA system with angular displacement closed-loop is established. Then the influence degree and law of 10% parameter change on the angular displacement step response are quantitatively obtained by using the first-order trajectory sensitivity analysis method. Finally, the prototype experiment and co-simulation are used to verify the accuracy of the system dynamics model and sensitivity results respectively. The good agreement between the theoretical value and the experimental (or

observe the changes of various parameters with different working conditions, the sensitivity index of parameters under nine working conditions are summarized, as shown in Figure 18.

It can be clearly seen from Figure 18 that the sensitivity of each parameter increases with the increase of step signal. However, with the increase of system load, the sensitivity of some parameters (including S_1 , S_2 , Ψ_1 , Ψ_2 , d_1 , d_2 , K_d , p_s , K_{axv} , K_p) increases, while the sensitivity of others (including A_p , ω , ζ) decreases. And the sensitivity of the former increases significantly, while the sensitivity of the latter decreases relatively little. Especially for parameters Ψ_1 , Ψ_2 , d_1 , d_2 , the order of magnitude of sensitivity index is only 10^{-8} under no-load conditions, but when with 200N·m load, their sensitivity index values become tens of thousands of times that with no-load.

co-simulation) value shows that the nonlinear system model established in this paper can represent the dynamic characteristics of valve-controlled HHRA closed-loop system and the sensitivity equations are accurate. Therefore, a series of analysis results in this paper are reliable, and can provide a basis for structural parameter optimization or control compensation of similar systems. The conclusions of sensitivity analysis are as follows:

(1) By analyzing the sensitivity of the angular displacement closed-loop system when the parameters change by 10%, the sensitivity ranking and value of parameters are different under different working conditions.

In general, under the nine working conditions of different step inputs and loads described in this paper, the perturbation of structural parameters S_1 , S_2 , Ψ_1 , Ψ_2 , d_1 , d_2 , A_p of the HHRA, working parameters K_d , p_s servo valve parameters ω , ζ , K_{axv} , and the control parameter K_p has a greater impact on the dynamic characteristics of the system. Moreover, the sensitivity of these parameters will increase significantly with the increase of step input signal. While for other parameters, the impact caused by small-scale perturbation can be ignored.

(2) For the structural parameters with high sensitivity, the sensitivity can be reduced by carefully selecting parameters or optimizing the structure. For the working parameters with high sensitivity and fast perturbation, it is the key to take measures to reduce their fluctuation in designing the system. The parameters of the servo valve are highly sensitive, so the values of the valve parameters in modeling should be the same as the actual system as much as possible, because a wrong value may cause great deviation between the actual situation and design requirements. For the control parameters, it is foreseeable that they have a great impact on the system output, because the dynamic characteristics of system are often changed by adjusting the controller parameters in practice.

In the follow-up work, the valve-controlled HHRA system designed in this paper will be used for the driving of legged robot, because the characteristics of helical hydraulic cylinder are very suitable for high-speed robot, which needs more compact structure and high explosive force. And the sensitivity analysis of system parameters with different control algorithms will be carried out to adapt to systems with different control performance requirements.

6 Declaration

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Availability of data and materials

The datasets used or analysed during the current study are available from the corresponding author on reasonable request.

Authors' contributions

The author's contributions are as follows: ZJH and XB was in charge of the whole trial; ZK was a major contributor in writing the manuscript; GMY, ZHZ, WXM, HHP and SQ assist in procurement, experiment, etc.

Competing interests

The authors declare that they have no competing interests.

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