doi:10. 3772/j. issn. 1006-6748. 2023. 02. 006

Output feedback control and parameters influence analysis of active suspension electro-hydraulic servo actuator①

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Abstract

The hydraulic servo actuator of heavy vehicle active suspension is investigated to clarify the correlation between system parameters and the control characteristics of active suspension hydraulic servo system. Accordingly, a nonlinear physical model of electro-hydraulic servo active suspension system is built. Compared with the conventional nonlinear modeling, the model in this study considers the asymmetry of working areas caused by single rod hydraulic cylinder in the suspension system. In accordance with the model, a nonlinear output feedback controller based on backstepping is designed, and the effectiveness of the controller is proved based on the experimental platform. The dynamic response curve of the electro-hydraulic servo control system under the change of parameters is generated based on the simulation model. The sensitivity of electro-hydraulic servo control performance to the change of system physical parameters is investigated, and two evaluation indexes are proposed to quantify and compare the effect of all physical parameter changes on position control system. As revealed by the results, the position control characteristics of suspension actuator are more sensitive to the changes of flow gain of the servo valve, system supply oil pressure and effective working areas of cylinder, and the two evaluation indexes are over 10 times higher than other physical parameters.

Key words: active suspension, nonlinear, hydraulic servo actuator, output feedback

0 Introduction

Active suspension technology is a method to enhance suspension performance, and it is capable of actively controlling the energy increase or decrease of suspension system through actuator[1-2]. With the accelerated integration of electronics, information, materials and other technologies with the vehicle industry, automotive products are developing to intelligence and high technology[3], which also brings more possibilities to automotive active suspension technology. For instance, using active suspension to control body attitude can provide a stable body platform for special vehicles to perform tasks[4-5], attitude control of turning vehicles to improve vehicle safety[6-7]. In accordance with the road profile estimation technology, the vehicle body height can be adjusted in advance to increase the trafficability[8].

To ensure the attitude control of the vehicle, the suspension actuator is required to achieve the displacement required by the target attitude quickly and accurately. The electro-hydraulic servo actuator is applied to the active suspension system of heavy vehicles due to its high power weight ratio and fast response speed[9]. However, numerous nonlinear factors exist in the hydraulic system, consisting of the strong nonlinearity of the servo valve flow output and the uncertainty of system physical parameters[10-13]. To solve the above problems, some control methods have been applied, including adaptive backstepping control[14-16], auto disturbance rejection control[17], sliding model[18-20], etc. These methods employ model feedforward, disturbance observation, parameter estimation and robust control to solve the nonlinear problems of the system[21-22], while effectively enhancing the tracking performance of the electro-hydraulic servo system. Most of the above studies aim at the electro-hydraulic servo system with symmetrical hydraulic cylinder, and the controller should

① Supported by the National Natural Science Foundation of China (No. U20A20332, 52175063) and Hebei Province PhD Graduate Innovation Funding Project (No. CXZZBS2021121).

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Received on Jane. 1, 2022

measure the signals such as displacement, speed, and pressure. However, the electro-hydraulic servo suspension system has a compact structure, so it can only adopt a single rod hydraulic cylinder. Furthermore, due to the limitation of mechanism and volume, it can only measure the position signal, thus further increasing the design difficulty of the controller.

In addition, less literature has paid attention to the sensitivity of different nonlinear factors to the position control performance of electro-hydraulic servo actuator. Different nonlinear links have different effects on the position control performance of hydraulic servo actuator. The effect of parameter change on control performance was investigated in Ref. [23] through sensitivity analysis. However, the analysis in the paper is based on PI controllers, which is difficult to apply with complex mathematical models and high order controllers due to the difficult derivation process of the sensitivity equation. In Ref. [24], a feedback linearized controller was designed for a hydraulic servo system, and the effect of four parameter changes on the control performance of the system was explored, whereas it did not set the parameter changes to the same value. Thus, the effect of the four parameters on the control performance cannot be compared horizontally. No control method can compensate for all nonlinear factors. Quantitative analysis of the effect of the physical parameters of the system on the control performance, targeted optimization controller design takes on a critical significance in enhancing the position control performance of the suspension hydraulic system.

To solve the above problems, the nonlinear physical model of the electro-hydraulic servo system of the active suspension is built, an output feedback controller is designed based on backstepping, and experimental verification is conducted. The correlation between the change of suspension actuator system parameters and system control characteristics is studied based on the nonlinear physical model and nonlinear output feedback controller. The evaluation index is set to quantitatively analyze the sensitivity of physical parameters to displacement tracking performance. This study provides a reference for the controller design and parameter matching of active suspension electro-hydraulic servo system.

1 Dynamic model

To explore the suspension system of vehicle, a lumped parameter mathematical model is built to express the dynamics. Fig. 1 illustrates the suspension system configuration with unsymmetrical hydraulic cylinder. The displacement refers to the only measurable signal of the suspension system.

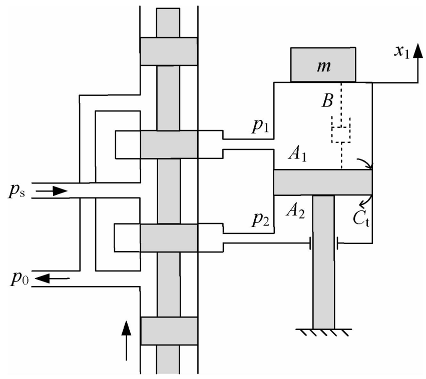

Fig. 1 Schematic of the suspension actuator

The dynamics equation for this cylinder is written as

$$m\ddot{y} = P_1A_1 - P_2A_2 - B\dot{y} + f(t) \tag{1}$$

where, m and y denote vehicle body weight and the displacement of the load; P_1 and P_2 represent the pressures in cylinder rodless cavity and rod cavity; A_1 and A_2 are the equivalent working areas of cylinder without rodless cavity and with rod cavity respectively; B denotes the viscous friction coefficient of liquid medium; and $f(t)$ expresses modeling error and other disturbances.

Neglecting the external leakage, the flow continuity equation can be written as

$$q_1 = A_1\dot{y} + \frac{V_1}{\beta_e}\dot{P}_1 + C_t(P_1 - P_2) + d_1(t) \tag{2}$$

$$q_2 = A_2\dot{y} - \frac{V_2}{\beta_e}\dot{P}_2 + C_t\ (P_1 - P_2)_1 + d_2(t) \tag{3}$$

where, q_1 denotes the supplied flow rate to the rodless cavity; q_2 is the return flow rate of the rod cavity; V_1 and V_2 are the effective volume of the two cavity; β_e represents the effective oil bulk modulus; C_t expresses the internal leakage coefficient of the cylinder due to pressure; $d_1(t)$ and $d_2(t)$ represents modeling error, unmodeled pressure dynamics, other disturbances, and so on.

The correlation between flow rate and the spool valve displacement of the servovalve can be described as

$$q_1 = k_q u[s(u)\sqrt{P_s - P_1} + s(-u)\sqrt{P_1 - P_r}] \tag{4}$$

$$q_2 = k_q u[s(u)\sqrt{P_2 - P_r} + s(-u)\sqrt{P_s - P_r}] \tag{5}$$

where, k_q and u denote the flow gain and input voltage of the servo valve, P_s and P_r represent the supply pressure and return pressure of the hydraulic, and the function $s(u)$ is defined as

$$s(u)=\begin{cases}1 & u\geqslant 0\\ 0 & u<0\end{cases} \tag{6}$$

Defining the status of the suspension system as $\boldsymbol{x} = [x_1 \quad x_2 \quad x_3]^{\mathrm{T}} = [y \quad \dot{y} \quad P_{\mathrm{L}}/m]^{\mathrm{T}}$. According to Eqs(1) – (5), the nonlinear dynamic model of suspension system can be written as

$$\begin{aligned}&\dot{x}_1 = x_2\\ &\dot{x}_2 = x_3 - \frac{b}{m}x_2 + f(t)\\ &\dot{x}_3 = \delta x_2 + \varphi x_3 + g(x_3, u) + q_1 + d(t)\end{aligned} \tag{7}$$

where,

$$P_{\mathrm{L}} = P_1A_1 - P_2A_2,\ \delta = \left(-\frac{A_1^2}{V_1} - \frac{A_2^2}{V_2}\right)\frac{\beta_{\mathrm{e}}}{m},$$

$$q_l = \frac{P_s\beta_e C_t(A_1 - A_2)}{m(A_1 + A_2)},$$

$$\varphi = \frac{-2\beta_e C_t}{V_1(A_1 + A_2)} + \frac{-2\beta_e C_t A_2}{A_1 V_1(A_1 + A_2)},$$

$$g(x_3, u) = \frac{\beta_e k_q s}{m\sqrt{A_2+1}}.$$

$$[s(u)\sqrt{A_1P_s - mx_3} + s(-u)\sqrt{A_2P_s + mx_3}]$$

modeling error and other disturbances are attached to $f(t)$ and $d(t)$, respectively.

Eqs(1) – (5) suggest that the vehicle suspension system has model able nonlinearity (e. g., the flow output nonlinearity of the servo valve and the asymmetry of the hydraulic cylinder). Moreover, the vehicle suspension system also has some parameter uncertainties (e. g., β_{e} affected by temperature and dissolved gas content, as well as C_t varying with the wear degree of the seal). To facilitate the design of observer and controller, the nominal value of parameters is taken as system parameters. Interference arising from parameter changes is incorporated into $f(t)$ and $q_l + d(t)$.

2 Control strategy

2.1 ESO design

To design the observer, $q_1 + d(t)$ is extended as a novel state variable. Subsequently, Eq. (5) is written as

$$\begin{aligned}&\dot{x}_1 = x_2\\ &\dot{x}_2 = x_3 - \frac{b}{m}x_2 + f(t)\\ &\dot{x}_3 = \delta x_2 + \varphi x_3 + g(x_3, u) + x_4\\ &\dot{x}_4 = \delta(t)\end{aligned} \tag{8}$$

In accordance with the observer design method in-Refs[25]and [26], an ESO is constructed as

$$\begin{aligned}&\dot{\hat{x}}_1 = \hat{x}_2 - 4w_0(\hat{x}_1 - x_1)\\ &\dot{\hat{x}}_2 = \hat{x}_3 - \frac{b}{m}\hat{x}_2 - 6w_0^2(\hat{x}_1 - x_1)\\ &\dot{\hat{x}}_3 = \delta\hat{x}_2 + g(\hat{x}_3, u) + \varphi\hat{x}_3 + \hat{x}_4 - 4w_0^3(\hat{x}_1 - x_1)\\ &\dot{\hat{x}}_4 = -w_0^4(\hat{x}_1 - x_1)\end{aligned} \tag{9}$$

where, w_0 denotes the observer gain, which can be changed to adjust the bandwidth of one observer.

Let $\varepsilon_i = x - \hat{x}/w_0^{i-1}$ denote the scale estimation error, the dynamic of the state estimation error can be expressed as

$$\dot{\boldsymbol{\varepsilon}} = w_0\boldsymbol{B}_1\boldsymbol{\varepsilon} + \boldsymbol{B}_2\frac{\eta(\varepsilon_2)}{w_0} + \boldsymbol{B}_3\frac{\gamma(\varepsilon_2, \varepsilon_3) + g(\tilde{x}_3, u)}{\boldsymbol{w}_0^2} + \boldsymbol{B}_4\frac{\lambda(t)}{\boldsymbol{w}_0^3} \tag{10}$$

where,

$$\boldsymbol{B}_1 = \begin{bmatrix}-4 & 1 & 9 & 0\\ -6 & 0 & 1 & 0\\ -4 & 0 & 0 & 1\\ -1 & 0 & 0 & 0\end{bmatrix},\ \boldsymbol{B}_2 = \begin{bmatrix}0\\1\\0\\0\end{bmatrix},\ \boldsymbol{B}_3 = \begin{bmatrix}0\\0\\1\\0\end{bmatrix},\ \boldsymbol{B}_4 = \begin{bmatrix}0\\0\\0\\1\end{bmatrix}$$

If $\boldsymbol{B}_1$ denotes Hurwitz, there exists a positive definite matrix $\boldsymbol{P}$, so it yields:

$$\boldsymbol{B}_1^{\mathrm{T}}\boldsymbol{P} + \boldsymbol{P}\boldsymbol{B}_1 = -2\mathbf{I} \tag{11}$$

The Lyapunov function is selected as

$$V_1 = \frac{1}{2}\boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{\varepsilon} \tag{12}$$

According to Refs[25] and [26], the derivative of Lyapunov function can be written as

$$\begin{aligned}\dot{V}_1 = {}& w_0\boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{\varepsilon} + \boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{P}\boldsymbol{B}_2\left(\frac{f(t)}{w_0} - \frac{b\varepsilon_2}{w_0 m}\right) + \boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{P}\boldsymbol{B}_4\frac{\lambda(t)}{w_0^3}\\ &+ \boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{P}\boldsymbol{B}_3\left(\frac{\delta w_0\varepsilon_2 + \varphi w_0^2\varepsilon_3 + g(\tilde{x}_3, u)}{w_0^2}\right)\end{aligned} \tag{13}$$

2.2 Backstepping controller design

The task of the command filter is to solve the differential explosion problem of the backstepping method and reduce the computational effort. Design error variable $e_1 = x_1 - x_d, e_2 = x_2 - \alpha'_1, e_3 = x_3 - \alpha'_2$, where x_d is the desired displacement. α'_1 and α'_2 are virtual control quantities, obtained by means of a command filter, which is modeled as

$$\begin{aligned}&\dot{h}_1 = h_2\\ &\dot{h}_2 = -\omega_i^2 h_1 - 2\xi\omega_i(h_2 - \alpha_i)\end{aligned} \tag{14}$$

where, h_1 and h_2 denote the virtual control quantities α_i and their derivatives, respectively; ξ and ω_i represent the filter bandwidth and damping of the filter to be designed; $\alpha_i - \alpha'_i$ can be made sufficiently small by adjusting the parameters to be designed.

Define error compensation variables $z_1 = e_1 - \rho_1$, $z_2 = e_2 - \rho_2$, where $\dot{\rho}_1 = -k_1\rho_1 + \alpha_1 - \alpha_i'$, $\dot{\rho}_2 = -k_2\rho_2 + \alpha_2 - \alpha_2'$.

Set k_i as the feedback gain, and design the virtual control function based on the state estimation as follows.

$$\begin{aligned}\alpha_1 &= -k_1 e_1 + \dot{x}_d - \rho_2 \\ \alpha_2 &= -k_2(\hat{x}_2 - \rho_2) + \frac{b}{m}\hat{x}_2 - \dot{\alpha}'_1\end{aligned} \tag{15}$$

Set $z_3 = e_3$ and bring it into Eq. (8)

$$\dot{z}_3 = \delta x_2 + \varphi x_3 + g(x_3, u)u + x_4 - \dot{\alpha}_2' \tag{16}$$

Based on Eq. (9), the control law can be obtained as

$$u = \frac{1}{g(x_3, u)} \cdot [-\hat{\delta}x_2 - \hat{\varphi}x_3 - \hat{x}x_4 + \dot{\alpha}'_2 - k_3(\hat{x}_3 - \alpha'_2)] \tag{17}$$

The Lyapunov function is selected as

$$V_2 = \frac{1}{2}z^{\mathrm{T}}z \tag{18}$$

The time derivative of Eq. (18) is written as

$$\begin{aligned}\dot{V}_2 = &-k_1 z_1^2 - k_2 z_2^2 - k_3 z_3^2 + z_1 z_2 + z_2 z_3 + \tilde{g}uz_3 + \\ &w_0\delta z_3\varepsilon_2 + w_0^2(\varphi + k_3)z_3\varepsilon_3 + w_0^3 z_3\varepsilon_4 + z_2 f(t) + \\ &w_0\left(k_2 - \frac{b}{m}\right)z_2\varepsilon_2\end{aligned} \tag{19}$$

where, $\tilde{g} = g(x_3, u) - g(\hat{x}_3, u)$. According to the definition of $g(x_3, u)$, it is bounded by the left and right differentiation at $u = 0$ and differentiable everywhere in other ranges, so it can be considered to meet the Lipschitz condition. Subsequently, it can be considered as $\tilde{g} \leqslant c|\varepsilon_3|$.

The Lyapunov function is selected as

$$V = V_1 + V_2 \tag{20}$$

In accordance with Eqs(12), (13), (18) and (19), the following equations can be yielded:

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \leqslant \boldsymbol{ZAZ}^{\mathrm{T}} + \sigma \tag{21}$$

where $\boldsymbol{Z} = [|z_1|, |z_2|, |z_3|, |\varepsilon_1|, |\varepsilon_2|, |\varepsilon_3|, |\varepsilon_4|]^{\mathrm{T}}$,

$$\boldsymbol{A} = \begin{bmatrix} \boldsymbol{A}_1 & \boldsymbol{0} & \boldsymbol{A}_2 \\ \boldsymbol{0} & w_0\text{-}\gamma\text{-}1 & \boldsymbol{0} \\ \boldsymbol{0}_2^T & \boldsymbol{0} & (w_0\text{-}\gamma\text{-}1)\mathbf{I} \end{bmatrix},$$

$$\boldsymbol{A}_1 = \begin{bmatrix} k_1 & -\frac{1}{2} & 0 \\ -\frac{1}{2} & k_2 - \frac{1}{2} & 0 \\ 0 & 0 & k_3 \end{bmatrix}, \boldsymbol{A}_2 = \begin{bmatrix} 0 & -\frac{n_1}{2} & -\frac{n_3}{2} \\ 0 & 0 & -\frac{n_2}{2} \\ 0 & 0 & -\frac{n_4}{2} \end{bmatrix}$$

$n_1 = w_0\left(k_2 - \frac{b}{m}\right)w_0$, $n_2 = w_0^2(\varphi + k_3) + c|u_{\max}|$, $n_3 = w_0\delta$, $\sigma = \left(\frac{1}{2} + \frac{l_1^2}{2w_0^2}\right)f_m^2 + \frac{l_3^2}{2w_0^6}d_m^2$, $l_i = |PB_i|$, $n_4 = w_0^3$,

$$\gamma = 1 + \frac{l_2\delta}{w_0} + \frac{l_2 c\,|u|_{\max}}{w_0^2} + \frac{l_1 b}{m} + l_2\varphi$$

Adjusting the parameters so that the matrix $\boldsymbol{A}$ is positive definite and set λ as the Eigen values of the matrix, the following equation can be obtained according to Ref. [27].

$$V(t) \leqslant V(0)e^{(-\tau t)} + \frac{\sigma}{\tau}(1 - e^{(-\tau t)}) \tag{22}$$

where $\tau = 2\lambda_{\min}(\boldsymbol{A})\min(1, 1/\lambda(\boldsymbol{P}))$. This proves that the system compensation tracking error and the state estimation error are bounded, and similarly the tracking error is bounded according to the definition of compensation tracking error.

3 Simulation and experiments

An experimental platform is built to verify the performance of the designed control strategy. As depicted in Fig. 2, the experimental platform consists of a bench with guide mechanism, a hydraulic cylinder of the same specification as a certain vehicle suspension, an inertial load, a servo valve with a bandwidth over than 70 Hz, a displacement sensor, a combined power source (quantitative pump and accumulator), safety and auxiliary components, as well as a measurement and control system. The sampling period is 1 ms. The physical parameters of the experimental platform are listed in Table 1, and the complete description of the experimental platform can be found in Ref. [28].

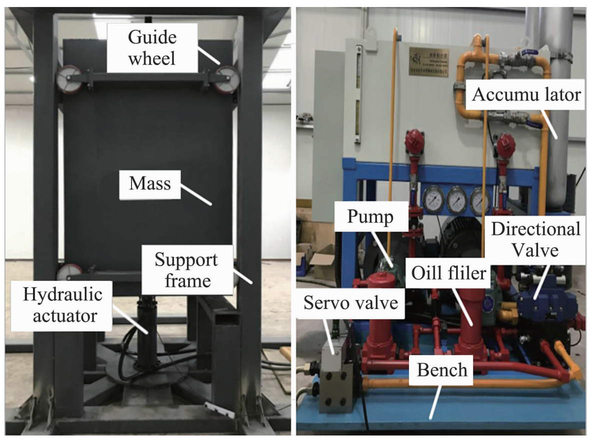

Fig. 2 Suspension electro-hydraulic servo actuator experimental platform

To be close to the engineering practice, the command signal will be designed as the unit step response signal of the first-order system with adjustable parameters.

$$x_d = x_{ds}(1 - e^{-\frac{t}{t_s}}) \tag{23}$$

where x_{ds} denotes the target value of the step signal and t_s is the time constant. According to the displacement limit of the actuator of the suspension experimental platform, take $x_{ds} = 0.1$ and set the time constant as $t_s = 0.01$. The controller parameters are set to $k_1 = 1600$, $k_2 = 2100$, $k_3 = 150$. To examine the control effect of output feedback controller (OCFC), proportional integral (PI) controller is selected for line comparison, and its parameters are set as $k_p = 300$, $k_i = 210$. As depicted in Fig. 3, PI controller and OCFC controller make the cylinder displacement track the command signal. Since the OCFC controller adopts feedforward compensation based on dynamic model and state estimation based on observer, the OCFC controller has less jitter and better dynamic performance when the oil cylinder is about to reach the target position. As depicted in Fig. 4, OCFC controller makes the cylinder reach steady state at 0.3 s, 0.19 s ahead of PI controller.

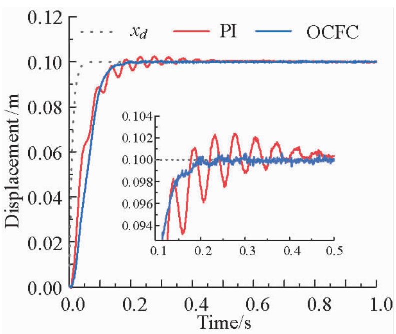

Fig. 3 Comparison of position tracking between two controllers

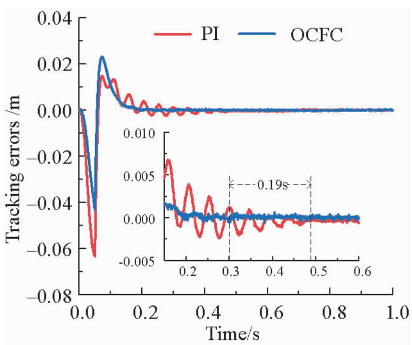

Fig. 4 Tracking error curves of two controllers

To evaluate the effect of physical parameters on the control performance of suspension actuator, this study reduces each parameter by 20% separately, and compares the change of displacement signal before and after the reduction of each parameter. Some physical parameters in the suspension test bench (e. g., β_e). B is difficult to be changed quantitatively. Accordingly, the way of digital simulation is used, and the parameters of the mathematical model are set with reference to the real value of the suspension experimental platform. Select parameters β_e, B, m, k_q, P_s, A_1, C_t and analyze the effect of parameter changes on the control effect. The selected parameters cover all physical parameters of the suspension actuator control system. Parameters A_1, A_2 and n jointly represent the equivalent working areas of cylinder, and there is a known mathematical relationship. To simplify the analysis, only parameter A_1 is selected for discussion.

To observe the hydraulic cylinder position tracking change caused by 20% change of each parameter, the motion trajectory $x_d = 0.1(1 - e^{-t/0.01})$ is used. The results are illustrated in Figs 4 and 5.

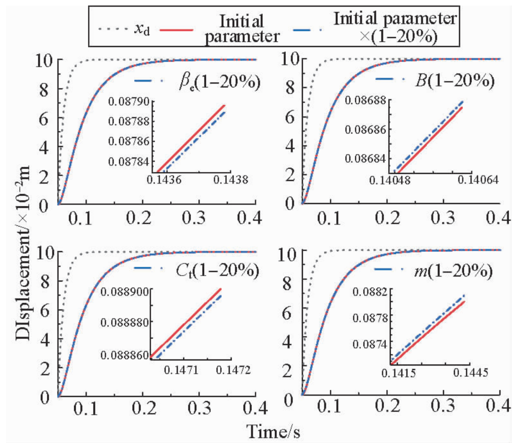

Fig. 5 Comparison of position tracking curve before and after β_e, B, m, C_t change

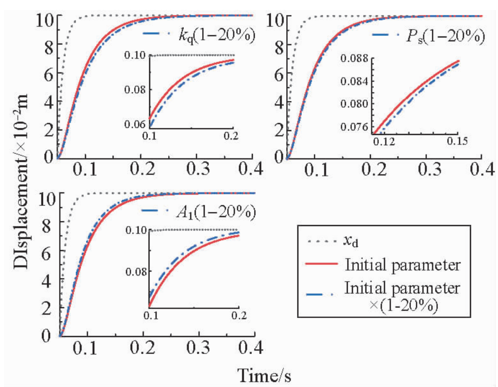

Fig. 6 Comparison of position tracking curve before and after k_q, P_s, A_1 change

As depicted in Fig. 5 and Fig. 6, when the respective parameter changes by 20%, the cylinder displacement can rapidly track the command signal, thus further verifying the robustness of the controller. As revealed in Fig. 5, after 20% of the change of parameters β_e, B, m and C_t, the displacement curve of the cylinder is consistent with that before the change of parameters. The change of the curve can be identified only after the tracking curve is locally enlarged, thus suggesting that the small-scale perturbation of the above four parameters has a slight effect on the control performance of the system. As depicted in Fig. 6, parameter k_q, P_s and A_1 significantly affect the displacement of hydraulic cylinder. The reduction of parameters k_q and P_s reduces the dynamic characteristics of the system position tracking process, while the reduction of parameter 3 enhances the dynamic characteristics of the system position tracking process. The above phenomenon is consistent with the objective law and further confirms the accuracy of the simulation analysis.

To analyze the effect of various parameters on cylinder displacement, two performance indexes are defined.

(1) The maximum value of the absolute value of the displacement difference before and after the parameter change serves as the index to evaluate the peak value of the impact on displacement tracking, which is defined as

$$M_p = \max_{i=(1\sim N)} \{ | x_1(i) - x'_1(i) | \} \tag{24}$$

where N denotes the serial number of the signal sampling point; x'_1 is the displacement of the oil cylinder after changing the parameters.

(2) The cumulative value of displacement error before and after parameter change serves as an index to evaluate the impact on the overall tracking process, which is defined as

$$M_a \sum_{i=1}^{N} | x_1(i) - x'_1(i) | \tag{25}$$

In accordance with the definition of performance index and the displacement response data of the respective parameter, the corresponding values of 1 and 2 can be obtained and represented by histogram, as presented in Fig. 7. Fig. 7 indicates that when the parameters change by 20%, the peak value and cumulative value of the influence on displacement tracking are different. k_q, P_s and A_1 have the most significantly effect on the displacement tracking process. The peak value M_p of the three parameters on displacement tracking is higher than 0.002, the cumulative value M_a is higher than 0.2, and the two evaluation indexes are over 10 times higher than other physical parameters. The M_p value of the flow coefficient k_q of the servo valve reaches 0.0041, and M_a reaches 0.4800. Other parameters have light effect on the displacement tracking process.

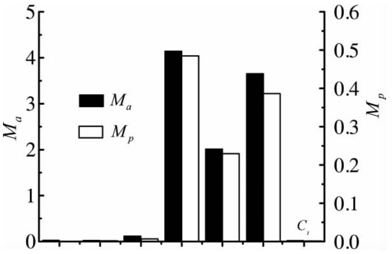

Fig. 7 Comparison of M_p and M_a values of various parameters

4 Conclusion

In this study, the nonlinear mathematical model of hydraulic suspension actuator system is built, and the asymmetry of suspension cylinder and parameter uncertainty are considered. To solve the problem of actuator position tracking of hydraulic suspension, an output feedback controller based on backstepping is designed. Command filtering is employed in the controller to solve the complex differential calculation problem of backstepping. Based on the experimental platform of suspension actuator, OCFC and PI control methods are compared to verify the effectiveness of the controller. Two performance evaluation indexes are designed, and the correlation between the change of seven physical parameters and displacement tracking performance is quantitatively analyzed. As revealed by the results, the flow gain of the servo valve, system supply oil pressure and effective working areas of cylinder have a great influence on the position control characteristics of the suspension actuator. Focusing on increasing the identification accuracy of the above three parameters can enhance the position control performance of the suspension system.

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