

Research Article

Robust Control of EHCS of Intelligent Commercial Vehicle under Load Change

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Received 27 October 2021; Accepted 29 November 2021; Published 22 December 2021

Academic Editor: Chun-xi Yang

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Electro Hydraulic Coupling Steering (EHCS) system is a new type of intelligent commercial vehicle steering system, having strong nonlinear characteristics. Besides, the change of load would cause the change of control system parameters, making not easy to establish an accurate control model of it. To realize the robustness of EHCS under the change of load, the controller based on the adaptive control method is proposed in this paper. To this end, the cosimulation model of EHCS is first established, where the constructed control model is simplified to a 2-degree-of-freedom model under reasonable simplification and assumption. Then, the steering angle controller is designed based on the model reference adaptive theory. Finally, some simulations are given to show the effectiveness of the proposed control method.

1. Introduction

As commercial vehicles pay more attention to use efficiency and cost savings, intelligence of it is more urgent than that of passenger vehicles. In addition, there is large volume, high load, and higher center of mass in commercial vehicles and the demand for driving safety is stronger [1]. The steering system is an important part of intelligent commercial vehicle, which directly affects the handling, stability, and driving safety. The steering system of commercial vehicles has gradually developed from mechanical to intelligent and electric power [2], which has gone through several stages: hydraulic power (HPS), electronically controlled hydraulic power (ECHPS), electric power (EPS), and steer-by-wire (SBW), e.g. The traditional HPS requires a lot of energy and has the disadvantage that the assist characteristics can not be adjusted [3], which will make the driver feel heavy when turning at low speed and float at high speed. ECHPS realizes the adjustable assist characteristics on the basis of HPS. EPS can only be installed on light trucks although it has a good road feel [4]. SBW has no real mechanical feedback, and safety redundancy measures are not easy to ensure safety for commercial vehicles. The electrohydraulic coupled steering (EHCS) system in this paper adds a set of power motor

unit on the basis of traditional HPS, which combines the advantages of high energy density of hydraulic system and high intelligence of electronic control system [5]. It can not only have good steering characteristics but also can be used as the hardware basis of active lateral control, matching the intelligent commercial vehicle.

EHCS is a complex nonlinear system, which has strong nonlinear characteristics such as hydraulic assistance and friction, e.g. [6]. Then, the load of the steering system will change with the vehicle speed, axle load, and other factors, and the parameters of the hydraulic system are time-varying, so it is not easy to establish an accurate control model [7–9]. Therefore, in order to control the angle of EHCS during autopilot, a simplified control model of EHCS must be established. Domestic and foreign scholars have done a lot of research on the steering system model. A 2-degree-of-freedom nonlinear hydraulic steering system model is proposed by Dell'Amico [10]. The system parameters are identified by the bench, and the model is linearized by the first-order Taylor expansion. Acarman et al. [8] approximate the hydraulic assist curve by using 8-degree polynomials and establish a 4-degree-of-freedom model of an online hydraulic steering system. Tai and Tomizuka [11] established a 2-degree-of-freedom linear model from motor torque to steering wheel

angle, steering wheel angle to wheel angle by linearizing the assist curve, and identified the unknown parameters by an open-loop frequency test. Garcia et al. [12] also established the state equation model of the self-driving steering system of commercial vehicles by using the linearized assist curve and the equivalent model of the mechanical system with 2-degree-of-freedom. Based on the above facts, the establishment of a 2-degree-of-freedom system model is sufficient for controller design.

As the low signal-to-noise ratio of the front wheel angle of the commercial vehicle, it can not be measured by the sensor, and the path tracking of the intelligent commercial vehicle is difficult to control the wheel directly. Therefore, the mainstream angle controller of EHCS uses the steering wheel angle instead of the wheel angle as the control target [13], and a large number of control methods have been proposed, such as fuzzy PID control [14], adaptive sliding control [15], and adaptive control [16, 17]. The method of a nonlinear robust controller is proposed by Tai and Tomizuka [11]; a first-order filter is introduced to reduce the inherent jitter of a nonlinear robust controller. Acarman et al. [8] designed a sliding mode controller to satisfy the robust stability for the modeling boundary and parameter uncertainty of the linear hydraulic steering system. Based on the linearized 2-degree-of-freedom model of the EHCS, Garcia et al. [12] designed the steering angle PI controller. Some simple controllers such as the PI controller are not easy to adjust the change in EHCS while artificial intelligence control with complex structure needs long adjustment time, which can not achieve an instantaneous dynamic response. Adaptive control can automatically compensate for unpredictable changes such as model parameters and input signals [18] matching the EHCS, so the model reference adaptive theory is used to design the angle controller of EHCS.

From the above discussions, the controller of EHCS based on the adaptive control method is developed. The 2-degree-of-freedom control model of EHCS is designed with reasonable assumptions and simplifies. Based on this model, the controller using model reference adaptive theory is designed by Lyapunov stability theory. Finally, the simulation results show that the system has good robustness under different load.

The main contributions of this paper are as follows. The cosimulation model and simplified 2-degree-of-freedom model are established for EHCS, which provides the basis for the development of EHCS in the future. A controller based on model reference adaptive theory is proposed, which provides a new way to angle control of EHCS, and verifies the robustness of the controller under different load changes. It makes a certain contribution to the control of a steering system in an intelligent commercial vehicle.

The rest of the paper is organized as follows: Section 2 introduces the cosimulink model and simplified model. Section 3 designs a controller based on a parameter adaptive control system. Section 4 is the simulation results and analysis. Section 5 presents some conclusions.

2. Modeling of EHCS

The structure of EHCS is shown in Figure 1. The EHCS is divided into three parts: mechanical, hydraulic, and motor,

in which the mechanical model includes a steering wheel, torsion, worm gear, and circulating ball power steering gear. Hydraulic module contacts of a rotating valve, hydraulic circuit, and hydraulic pump. The mechanical and hydraulic models are built in AMESim, and the torque control model of permanent magnet synchronous motor (PMSM) is built in Simulink.

2.1. Establish Cosimulation Model. The mechanical model and hydraulic model are established in AMESim; the mechanical part is partially simplified as follows. The damping coefficient between worm gears and worms is ignored. The intelligent commercial vehicle is driverless, and the driver input torque is zero. The torsional stiffness of the upper torsion bar is represented by the rotating spring damper. The equivalent moment of inertia between the steering string and the worm gear is represented by the upper rotating load, while the moment of inertia between the worm wheel and the lower torsion bar is represented by the lower rotating load. The lower torsion bar module is equivalent to a rotating spring with stiffness of 1.5 Nm/deg, and the torsion bar deformation angle signal is collected by the steering sensor on the both ends of the spring. Motor driving torque is the only torque input of the model, which is built in Simulink. The circulating ball power steering gear is equivalent to a two-stage transmission pair containing a screw nut transmission pair and a rack tooth fan transmission pair [19]. The angle acquisition module is used to collect the rotation angle of the rocker shaft, ignoring the transmission ratio and elastic influence of the steering rod system. The rotation angle of the rocker shaft is regarded as the wheel angle.

The main component of the hydraulic module is the rotary valve, which is equivalent to a variable throttle according to its working principle [20], and the opening area of the variable throttle hole is controlled by the torsion bar deformation angle in the mechanical module. According to the parameters of the rotary valve and the method shown in the reference [19–21], the relationship between the overflow area of rotary valve port A and the deformation angle of the torsion bar θ_{lg} is calculated as shown in Equation (1). The hydraulic pump and piston adopt the module included in AMESim.

$$\begin{cases} 7A + 85\theta_{lg} - 180 = 0, & \theta_{lg} \in [-5, -2], \\ 340A + 977\theta_{lg} - 3486 = 0, & \theta_{lg} \in (-2, 1.4], \\ 8295A - 13083\theta_{lg} - 69994 = 0, & \theta_{lg} \in (1.4, 5.35], \end{cases} \quad (1)$$

where A is the overflow area of the rotary valve and θ_{lg} is the deformation angle of the torsion bar.

The cosimulation part with Simulink is established, in which the input is the steering resistance moment of the front axle and the motor torque and the output is the rotation angle of the two wheels, the required speed of the motor and the steering wheel angle. The final model is shown in Figure 2.

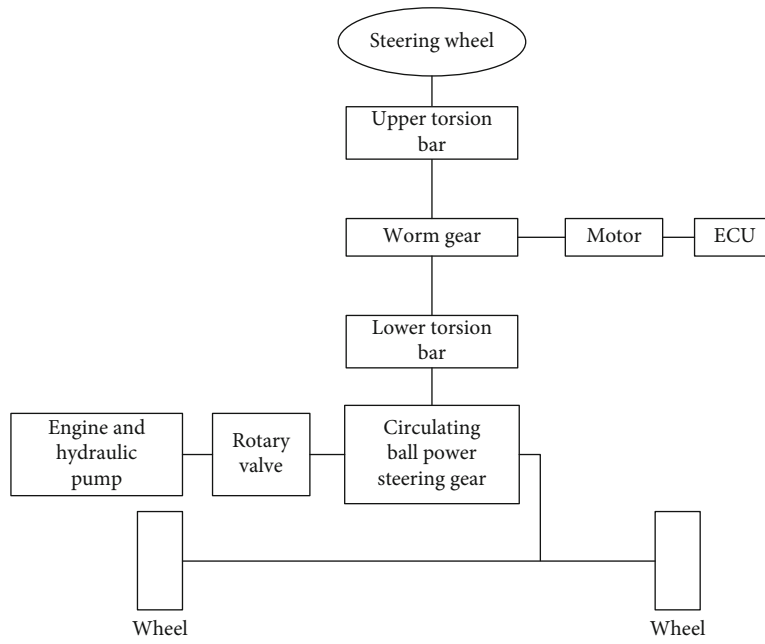


FIGURE 1: Structure diagram of EHCS.

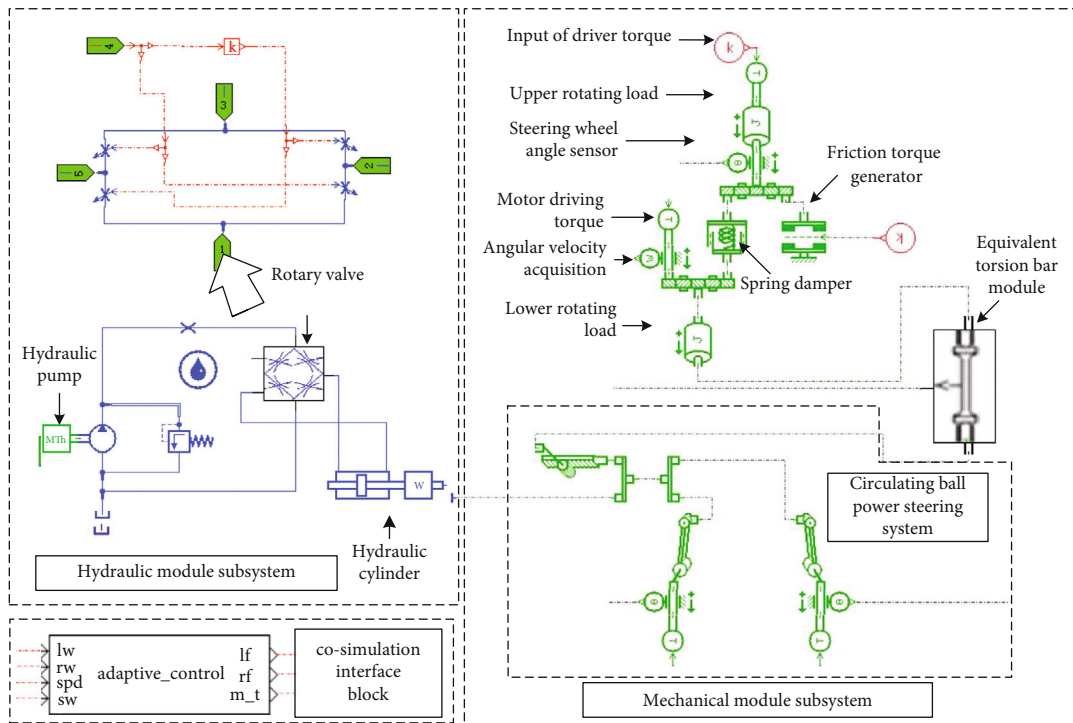


FIGURE 2: Mechanical and hydraulic system modeling in AMESim.

The dynamic model of the permanent magnet synchronous motor is shown in Equation (2). In order to facilitate control, the strong coupling current and voltage should be decoupled. If the d -axis current is controlled to zero, the motor torque is proportional to the q -axis current. As shown in Equation (3), the motor torque can be controlled by controlling the q -axis current.

$$T_e = p_n [\psi_f i_q + (L_d - L_q) i_q i_d], \quad (2)$$

$$T_e = J_e \frac{d\omega_m}{dt} + R_m \omega_m + T_L,$$

$$T_{e1} = p_n \psi_f i_q, \quad (3)$$

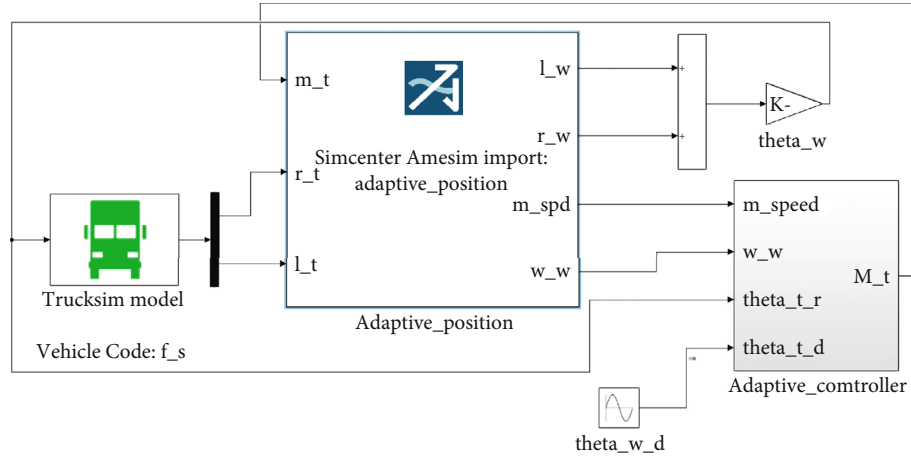


FIGURE 3: The combined simulation model.

where T_e and T_{el} are the motor electromagnetic torque, T_L is the load torque, ψ_f is the stator winding flux, p_n is the motor pole logarithm, L_q and L_d are the q - and d -axis inductance, respectively, i_q and i_d are the q - and d -axis current, respectively, J_e is the motor moment of inertia, ω_m is the rotor mechanical angular speed, and R_m is the resistance coefficient.

The space vector pulse width modulation method, the inverter in Simulink, and the motor torque controller are used to build the torque control model of the permanent magnet synchronous motor (PMSM). The input of the model is the desired speed and q -axis current of the motor, and the output is the torque of the motor. In the TruckSim, a two-axle truck is selected, which is 5.5 t at no load and 9.5 t at full load. The transmission ratio of the steering system is set to 1, and the flexibility is set to 0. At this time, the wheel angle of the model in TruckSim is basically equal to the steering wheel angle. If the steering control of TruckSim is set to full, the wheel angle calculated in Simulink is input to the steering wheel angle in TruckSim, and the output of TruckSim is the steering resistance moment of the wheels on the front axle.

The cosimulation model is established as shown in Figure 3, including TruckSim function, AMESim function, angle controller, and SVPWM motor controller.

2.2. Simplified Control Model. In order to simplify the EHCS as a 2-degree-of-freedom model, the following ignoring and assumptions are made: (1) inertia mass at the upper end of the lower torsion bar is equivalent to the worm gear, and inertia mass body at the lower end of the lower torsion bar is equivalent to the rocker shaft, dividing the whole steering system into two mass systems [20]; (2) the steering load is regarded as external interference; (3) the angle influence of the steering string is ignored; (4) the friction torque of the system is ignored; (5) the elastic influence of the steering rod system is ignored; (6) the elastic influence of the upper end of the upper torsion bar, that is, the steering string, is ignored. The force analysis of the simplified EHCS system is shown in Figure 4.

Ignoring the friction force of the system and linearizing the hydraulic assist force as $T_h = \psi_0 T_{k2}$, the dynamic model is shown in following:

$$\begin{aligned} J_u \ddot{\theta}_u + D_u \dot{\theta}_u + k_2 \theta_u &= T_u + nk_2 \theta_d, \\ J_d \ddot{\theta}_d + D_d \dot{\theta}_d + (k_{rw} + n^2 k_2 + n\psi_0 k_2) \theta_d &= (n + \psi_0) k_2 \theta_u, \end{aligned} \quad (4)$$

where J_u is the equivalent moment of inertia from the inertia mass at the top of the lower torsion bar to the worm gear, kg/m^2 ; D_u is upper mass damping coefficient, Nm/rad/s ; k_2 is the elastic stiffness of upper torsion bar, Nm/rad ; J_d is the equivalent moment of inertia from the bottom of the lower torsion bar to the tooth fan shaft, kg/m^2 ; D_d is lower mass damping coefficient, Nm/rad/s ; k_{rw} is the simplified from steering system load to the stiffness of the spring model, Nm/rad ; T_f is friction torque of lower mass body, Nm ; n is the transmission ratio from the screw to the fan shaft, about 25; and T_h is the equivalent moment of the hydraulic system acting on the lower mass body, Nm .

Based on the above assumptions and ignoring, the steering wheel angle is approximately the worm gear angle, which is given as

$$\begin{aligned} \theta_s &= \theta_u, \\ \theta_w &= \theta_d, \end{aligned} \quad (5)$$

where θ_s is the angle of the steering wheel and θ_{wh} is the angle of the wheel.

The transfer function of the controlled system can be obtained as shown as

$$G_p(S) = \frac{\theta_w(s)}{\theta_s(s)} = \frac{\theta_d(s)}{\theta_u(s)} = \frac{(n + \psi_0) k_2}{J_d s^2 + D_d s + (k_{rw} + n^2 k_2 + n\psi_0 k_2)}. \quad (6)$$

After standardizing the transfer function, it is concluded that the transfer function is in the overdamping state,

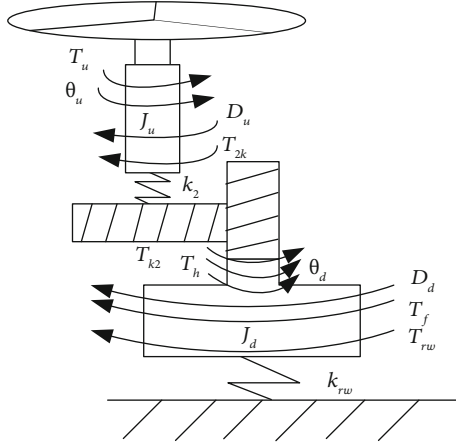


FIGURE 4: Diagram of force acting on a simplified system model.

meaning the response time is too long. In order to lay the groundwork for the model reference theory in the next chapter, an ideal transfer function model is proposed on the basis of the system transfer function. The following requirements are put forward for the ideal model: (1) It has a fast dynamic response and the rise time is less than 0.04 s. (2) The output amplitude should accurately express the input amplitude; in other words, the zero frequency amplitude should be 1. (3) The bandwidth of the system should meet the requirements of the steering system of the whole vehicle. (4) Within the required system bandwidth, the delay of the system should be less than 15 degrees as far as possible. The ideal model is a second-order model, and its transfer function is in a state of underdamping.

3. Angle Control Strategy Based on Model Reference Adaptive Theory

3.1. Design of Model Reference Adaptive Controller. Model reference adaptive control originates from the deterministic servo problem and is developed from the tracking problem. The system consists of a reference model, a controlled object, and a controller. The reference model is a given ideal model, which defines the performance indexes such as overshoot, damping performance, and adjustment time. The consistency between the controlled model and the reference model is measured by the state error vector [22]. The adaptive mechanism is designed according to the system stability criterion, and the corresponding adaptive law is designed to modify the variable parameters according to the direction of less deviation, so that the actual performance index of the system is close to the ideal performance index.

In this paper, aiming at the great variation of steering resistance moment with variable load of intelligent commercial vehicle and the nonlinear problems in the control of EHCS, a control framework is proposed based on model adaptive theory, and the control model is designed according to the ideal model as a reference model. Firstly, it is proposed that the state equation of the controlled system and the reference model with single input and single output is

shown as

$$\begin{aligned}\dot{x}_p &= A_p x_p + b_p u, \\ y_p &= h^T x_p, \\ \dot{x}_m &= A_m x_m + b_m u, \\ y_m &= h^T x_m,\end{aligned}\quad (7)$$

where x_p is state vector of the controlled system, u is controlled vector, y_p is the output of a controlled system, x_m is the state vector of the reference model, and y_m is the output of the reference model.

The transfer functions of the controlled system and the reference model can be obtained from Equation (7) as shown as

$$\begin{aligned}G_p(s) &= h^T (sI - A_p)^{-1} b_p = k_p \frac{N_p(s)}{D_p(s)}, \\ G_m(s) &= h^T (sI - A_m)^{-1} b_m = k_m \frac{N_m(s)}{D_m(s)},\end{aligned}\quad (8)$$

where k_p and k_m are gain coefficients.

The angle control of the EHCS takes the steering wheel angle as the input and the rocker shaft angle as the output. Define the error between the reference model and the actual output of the controlled system as $e_1 = y_m - y_p$, the control objectives of adaptive systems are shown as

$$\lim_{t \rightarrow \infty} |e_1| = \lim_{t \rightarrow \infty} |y_m(t) - y_p(t)| = 0. \quad (9)$$

In order to make the adaptive adjustable system match the reference model completely, the adaptive system should have enough variable parameters, and the number of variable parameters should be the same as the uncertain parameters of the system. Define four variable parameters $\sigma = [\sigma_1 \ \sigma_2 \ \sigma_3 \ \sigma_4]$ and two auxiliary signals in order to design controller, the auxiliary signal is shown as

$$\begin{aligned}\dot{\lambda}_1 &= A_f \lambda_1 + b_f u, \\ \dot{\lambda}_2 &= A_f \lambda_2 + b_f y_p,\end{aligned}\quad (10)$$

where A_f and b_f are related to the $N_m(s)$ of the transfer function of the reference model.

Define w is the variable signal of the control system, $w = [r \ -\lambda_1 \ -y_p \ -\lambda_2]$, the controller is built as shown in Figure 5.

$\hat{\sigma}$ is the variable parameter obtained in the ideal state, δ is the parameter error, and the augmented equation of state of the controlled system is obtained by substituting $\sigma = \hat{\sigma} + \delta$ into (7) as shown in (11). When $\delta = 0$, the augmented equation of state of the system in the ideal state, that is, the augmented equation of state of the reference model, is obtained as shown as

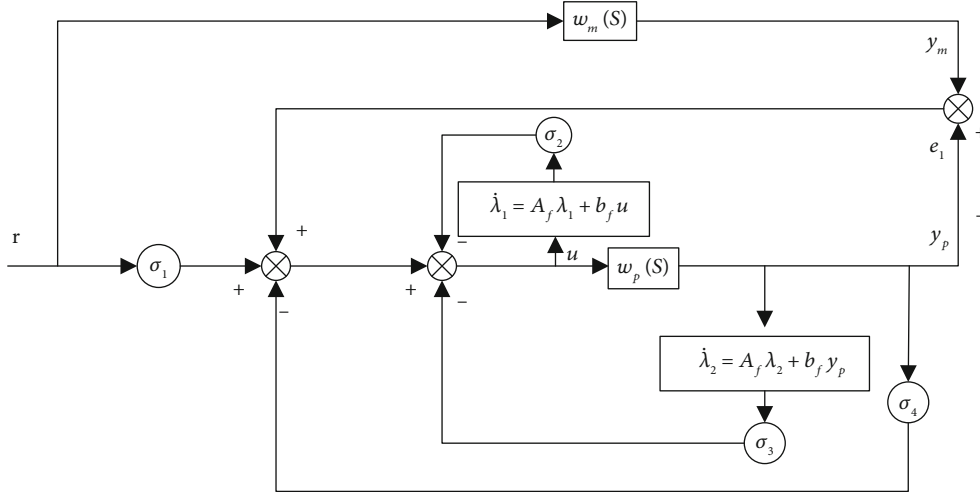
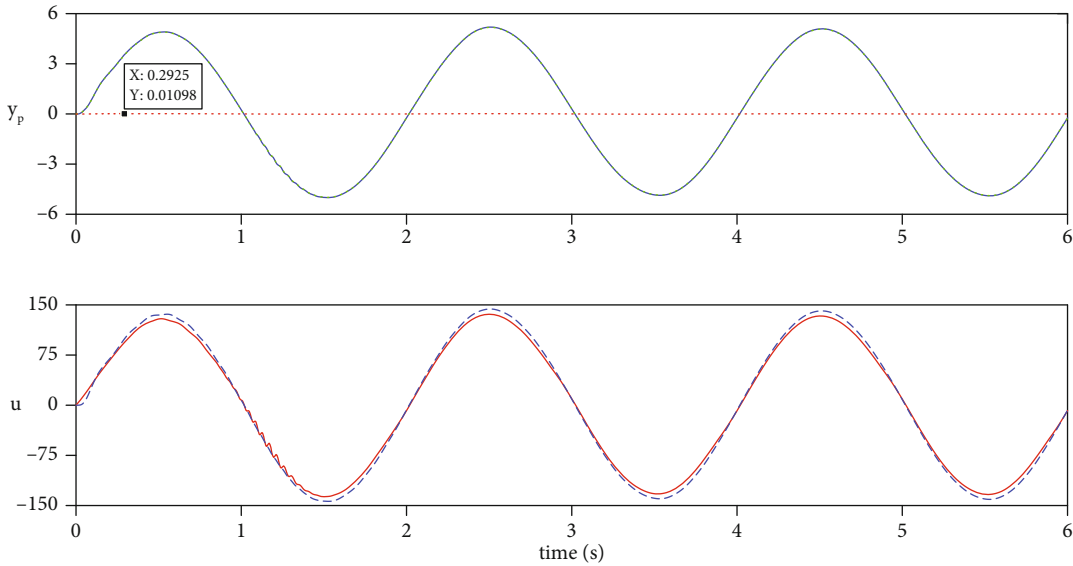


FIGURE 5: Control of model reference adaptive system.

FIGURE 6: The control action u and system output y_p .

$$\begin{aligned} \dot{x} &= A_c x + b_c (\hat{\sigma}_1 r + \delta^T w), \\ \dot{x}_{mc} &= A_{mc} x + b_{mc} \hat{\sigma}_1 r, \end{aligned} \quad (11)$$

where $x^T = [x_p^T \ \lambda_1^T \ \lambda_2^T]$ and $x_{mc}^T = [x_{mp}^T \ \lambda_{m1}^T \ \lambda_{m2}^T]$.

The augmented error state equation can be obtained by subtraction of Equation (11) as given as

$$\dot{e} = A_c e + b_c \delta^T w = A_c e + b_c u_c, \quad (12)$$

where $e = [x_p - x_m, \lambda_1 - \lambda_{m1}, \lambda_2 - \lambda_{m2}]^T$ and $u_c = \delta^T w$.

3.2. Design of Adaptive Law according to Lyapunov Stability. The adaptive law is designed by using Lyapunov stability theory, which is defined according to the Lyapunov function to satisfy the stability of the system [23], and the Lyapunov function and its derivatives are defined as shown as

$$\begin{aligned} V &= \frac{1}{2} (e^T P e + \delta^T \Gamma^{-1} \delta), \\ \dot{V} &= \frac{1}{2} e^T (P A_c + A_c^T P) e + \delta^T w b_c^T P e + \delta^T \Gamma^{-1} \dot{\delta} \\ &= \frac{1}{2} e^T (P A_c + A_c^T P) e + \delta^T (w b_c^T P e + \Gamma^{-1} \dot{\delta}) = \dot{V}_1 + \dot{V}_2. \end{aligned} \quad (13)$$

It is obvious that the function V is positive definite, but in order to make the system stable, \dot{V} should be negative definite.

According to Equation (12) and Kalman-Yakubovich theory [23], the transfer function $G_e(s)$ of e is strictly positive real if and only if there are positive definite matrices P and Q , such that Equation (14) holds. Therefore, when the transfer function of e is positive real, it can be obtained that $P A_c +$

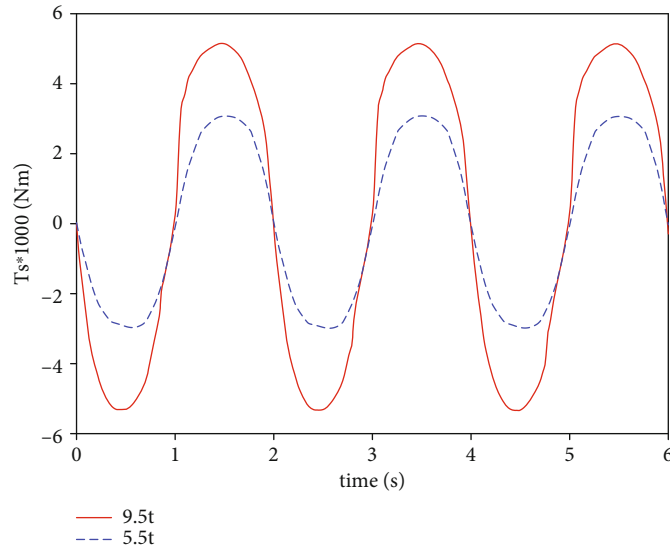


FIGURE 7: Steering resistance moment.

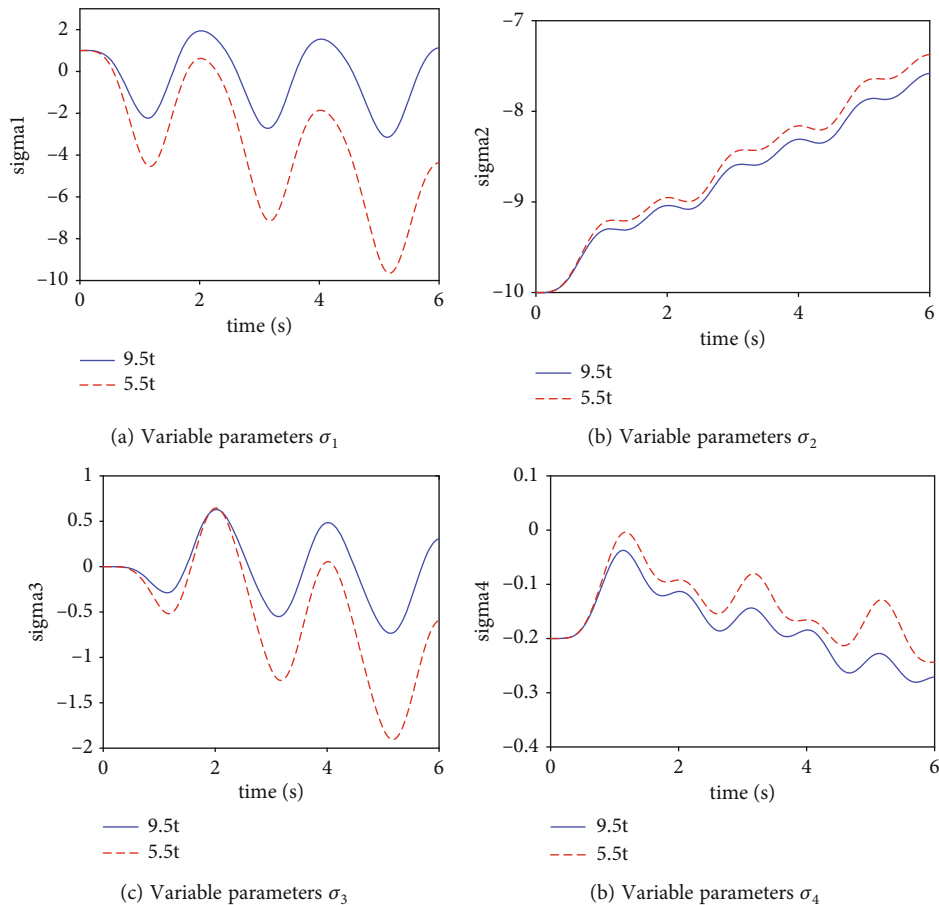


FIGURE 8: Comparison of variable parameters.

$A_c^T P = -Q$; by the words, \dot{V}_1 is negative definite.

$$A^T P + P A = -Q. \tag{14}$$

It is obviously to infer the \dot{V} will be negative definite when \dot{V}_2 is zero. Hence, by defining adaptive law $\dot{\delta} = -\Gamma w b_c^T P e$, \dot{V}_2 will be zero, which satisfies the condition that \dot{V} is negative definite.

In order to make the strictly positive realness of the error model, the auxiliary polynomials $L(s) = s + a$ are introduced, and the transfer functions of the system model, the reference model, and the error model are shown as

$$\begin{cases} w_p(S) = L(s)G_p(s), \\ w_m(S) = L(s)G_m(s), \\ w_e(S) = L(s)G_e(s). \end{cases} \quad (15)$$

Define $\zeta = L^{-1}(S)w = [r \quad -\lambda_1 \quad -y_p \quad -\lambda_2]^T / (s + a)$, the adaptive law of system variable parameters designed according to Lyapunov stability theory is shown as

$$\begin{cases} \dot{\sigma} = \dot{\hat{\sigma}} + \dot{\delta} = -\Gamma\zeta(t)e_1, \\ \dot{\zeta} = -a\zeta(t) + w(t), \end{cases} \quad (16)$$

where Γ is a positive definite symmetric matrix.

The input of the controlled system is shown as

$$u = \sigma^T w(t) + e(t)\zeta^T(t)\Gamma\zeta(t). \quad (17)$$

By introducing auxiliary polynomials $L(s) = s + 2$, it can be obviously proved that $w_p(S)$, $w_m(S)$, and $w_e(S)$ are strictly positive real. Define $\Gamma = \text{diag}\{p_i\}$, $i = 1, 2, 3, 4$, according to the adaptive law of variable parameters as shown as

$$\begin{cases} \dot{\sigma} = -\Gamma L^{-1}(S)w e_1(t), \\ \sigma = -\int_0^t \Gamma\zeta(t)e_1(t)d\tau + \sigma(0). \end{cases} \quad (18)$$

The input of the controlled object is shown as

$$\begin{aligned} u &= \sigma^T(t)w(t) + e_1(t)\zeta^T(t)\Gamma\zeta(t) \\ &= \left[-\int_0^t \sum p_i \zeta_i e_1(t) d\tau + \sigma(0) \right]^T w(t) + e_1(t) \sum p_i \zeta_i^2. \end{aligned} \quad (19)$$

The $\sigma(0)$ is the initial value of variable parameters; it can be optional in theory; however, a bad initial value may cause the system not to work normally. In addition, the selection of weighting matrix Γ has a great influence on the approximation reference model. Therefore, the selection of $\sigma(0)$ and Γ should be careful.

4. Robustness Verification of Angle Control Strategy

The cosimulation model in Section 2 is used to track the desired rotation angle of the front wheel under the load of 9.5 t and 5.5 t, respectively, and the sine waveform with a period of 2 s is selected to verify the simulation control strategy. The control action u and system output y_p under the two loads are shown in Figure 6, respectively. The steering resistance moment corresponding to the two loads is shown

in Figure 7, and the change of variable parameters during the operation of the model is shown in Figure 8.

It can be seen from the above facts that given the same desired rotation angle, the maximum difference of steering resistance moment under different loads is about 3000 Nm, but under the same expected rotation angle, the actual turning angle after the control strategy is basically the same. There is a certain oscillation in the first 0.3 s, which is caused by the same initial value of the variable parameters, and then, the system is stable after the variable parameters change with time. It is proved that the system has good robustness under the change of load.

5. Conclusion

Aiming at the EHCS with nonlinear uncertain parameters, after simplifying the control model to a 2-degree-of-freedom model, a controller based on adaptive theory is proposed by using Lyapunov stability theory. The adaptive law of variable parameters is designed, and the proposed control method can make the parameters change with time and compensate the nonlinear model disturbance. The load change is used to analyze the stability of control performance, and the results show that though the load difference is large, the operation of the EHCS can be effectively controlled and better performance can be obtained when tracking the target angle, meaning it has great robustness.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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