Research on predictive sliding mode control strategy for horizontal vibration of ultra-high-speed elevator car system based on adaptive fuzzy

Hong Wang¹, Mingqin Zhang¹, Ruijun Zhang¹ and Lixin Liu²

Abstract
In order to effectively suppress horizontal vibration of the ultra-high-speed elevator car system. Firstly, considering the nonlinearity of guide shoe, parameter uncertainties, and uncertain external disturbances of the elevator car system, a more practical active control model for horizontal vibration of the 4-DOF ultra-high-speed elevator car system is constructed and the rationality of the established model is verified by real elevator experiment. Secondly, a predictive sliding mode controller based on adaptive fuzzy (PSMC-AF) is proposed to reduce the horizontal vibration of the car system, the predictive sliding mode control law is achieved by optimizing the predictive sliding mode performance index. Simultaneously, in order to decrease the influence of uncertainty of the car system, a fuzzy logic system (FLS) is designed to approximate the compound uncertain disturbance term (CUDT) on-line. Furthermore, the continuous smooth hyperbolic tangent function (HTF) is introduced into the sliding mode switching term to compensate the fuzzy approximation error. The adaptive laws are designed to estimate the error gain and slope parameter, so as to increase the robustness of the system. Finally, numerical simulations are conducted on some representative guide rail excitations and the results are compared to the existing solution and passive system. The analysis has confirmed the effectiveness and robustness of the proposed control method.

Keywords
Horizontal vibration, ultra-high-speed elevator, predictive sliding mode control, fuzzy logic system, hyperbolic tangent function

Date received: 24 February 2021; accepted: 27 February 2021

Introduction
With the increase of high-rise buildings, the popularization of ultra-high-speed elevators has become an inevitable trend. However, the high speed of elevators makes the horizontal vibration of the car system more severe. Intense vibration will not only seriously affect the comfort of passengers, but also affect the normal operation of the instruments on the elevator car, and even lead to safety accidents. Therefore, elevator horizontal vibration suppression has become one of the key problems to be solved urgently in the field of ultra-high-speed elevator.¹² At present, control methods for elevator horizontal vibration reduction include passive vibration reduction¹ and active control.⁴⁵ Guide shoe spring and rubber block are often used to isolate the car system in passive vibration reduction. The damping force provided by this method is small, which is difficult to meet riding comfort requirements of ultra-high-speed elevator. Therefore, active control method has become the main way to solve the horizontal vibration of the ultra-high-speed elevator.

It is necessary to establish a reasonable dynamic model for the active control research on horizontal vibration of the car system. Currently, the guide shoe model is simplified as a linear system, and the active control model is built based on the constant mass and only considering the displacement excitation of the guide rail in the studies of the car system horizontal vibration control.⁶⁻⁸ Nevertheless, the guide shoe has nonlinear characteristics due to the use of rubber

¹School of Mechanical and Electrical Engineering, Shandong Jianzhu University, Jinan City, Shandong, China
²Shandong Fuji Zhiyu Elevator Co., Ltd, Dezhou City, China

Corresponding author: Mingqin Zhang, School of Mechanical and Electrical Engineering, Shandong Jianzhu University, Fengming Road, 1000 Lihecheng District, Jinan, Shandong 250101, China.
Email: zhangmq@sdjzu.edu.cn

Creative Commons CC BY: This article is distributed under the terms of the Creative Commons Attribution 4.0 License (https://creativecommons.org/licenses/by/4.0/) which permits any use, reproduction and distribution of the work without further permission provided the original work is attributed as specified on the SAGE and Open Access pages (https://us.sagepub.com/en-us/nam/open-access-at-sage).
material of the shoe lining in the rolling guide shoe. In addition, the change of passenger load and the different specifications of parts on the car frame will cause uncertainty in the mass and moment of inertia of the car and the car frame. Meanwhile, due to the influence of factors such as shaft air flow, wire rope swing and so on, there are other uncertain external disturbances in the car system. Therefore, in order to simulate a more accurate and reliable situation, it is necessary to consider the influence of the nonlinearity of guide shoe and the above-mentioned compound uncertainty on the performance of the elevator system. Sharma and Grover\textsuperscript{10} studied the nonlinear characteristics of rolling guide shoe, described it by Bouc-Wen hysteretic model, and determined the undetermined coefficients in the model through parameter identification. Mei et al.\textsuperscript{11} used Hertz contact theory to establish a three-dimensional rolling contact model of the elevator guidance system for the nonlinear problem of rolling guide shoe lining. Feng et al.\textsuperscript{12} considered the uncertainty of the external load and designed a position controller and attitude controller to reduce the horizontal vibration of the elevator car system. When the above scholars constructed the horizontal vibration model of elevator car system, they only considered the nonlinearity of guide shoe or the uncertainty of the car system. The influence of the both nonlinearity of guide shoe and uncertainty of the car system on the active control performance of elevator car system was neglected.

The traditional control method is not effective in dealing with the nonlinear and compound uncertain disturbance problems in the car system. Sliding mode control (SMC) is appropriate for various uncertain nonlinear systems.\textsuperscript{13-15} Its main advantage is that the sensitivity to system uncertainties and disturbances is low, and it has been applied in elevator related fields.\textsuperscript{16,17} However, due to the inherent discontinuous switching characteristics of SMC when dealing with uncertain problems, the system will produce serious chattering phenomenon, which will affect the control effect of the system. To overcome the chattering of SMC, various methods have been proposed in the literature, one method is to apply a saturation function to the SMC control gain when the sliding surface is located in the boundary of the sliding hyper-plane. For example, Park and Tsuji\textsuperscript{18} and Zhou et al.\textsuperscript{19} designed a predictive sliding mode controller based on switching term (PSMC-ST), and introduced the sliding surface into the quadratic performance index of nonlinear predictive control, thus reducing the chattering of SMC. This method is simple, but it can’t guarantee the output convergence. Another common method is to use a disturbance observer to estimate and compensate the external disturbance and uncertainty, so as to solve the chattering of SMC.\textsuperscript{20,21} However, it has high accuracy only for constant or slow changing disturbances. Another way to solve the chattering of SMC is to apply fuzzy logic system (FLS). Due to the universal approximation characteristic, FLS can effectively realize the adaptive approximation of complex uncertain systems and does not depend on the system model. At present, it has been widely used in various uncertain systems,\textsuperscript{22,23} which provides a new method to solve the compound uncertainty of the car system.

Based on the above analysis, aiming at the problem of the horizontal vibration of the car system and the chattering of SMC, a predictive sliding mode controller based on adaptive fuzzy (PSMC-AF) is designed in this paper, and simulation experiments are used to analyze and verify the horizontal vibration control effect of the PSMC-AF for ultra-high-speed elevator car system. This article mainly contributes are presented as follows:

1. Considering the nonlinearity of guide shoe, parameter uncertainties and uncertain external disturbances of the elevator car system, a more realistic horizontal vibration active control model of the 4-DOF ultra-high-speed elevator car system is constructed.
2. A PSMC-AF is designed to reduce the horizontal vibration of the car system. In order to reduce the influence of compound uncertainty of the car system, a FLS is established to approximate the compound uncertain disturbance term (CUDT) on-line.
3. The continuous smooth hyperbolic tangent function (HTF) is used to introduce sliding mode switching term to compensate the fuzzy approximation error. Furthermore, the adaptive laws are designed to estimate the error gain and slope parameter for improving the performance of the controller.

The rest of the article is organized as follows. In section “An active control model of the 4-DOF elevator car system,” the active control model for horizontal vibration of the car system is constructed. In section “Design of predictive sliding mode controller based on adaptive fuzzy,” the PSMC-AF is designed and the stability of the controller is proved. In section “Simulink study,” the effectiveness of the controller is verified by the simulation. Finally, conclusions are drawn in section “Conclusion.”

**An active control model of the 4-DOF elevator car system**

**Nonlinear dynamic model of car system with active control**

The elevator car system mainly includes car and car frame, which is connected by rubber block. Guide shoes are installed on the upper and lower sides of the car frame and form the guiding system of the car with the guide rails, shown in Figure 1. In order to simplify the complexity of its dynamic analysis and controller design, the following assumptions are made:
1. Because the elastic connection between the car and the car frame, the rubber block is regarded as a spring-damping system with the same structure and parameters.

2. Because the guide wheel is close to the surface of the guide rail, the mass of the guide wheel is ignored. The guide shoe is simplified as a non-linear spring-damping system with the same structure and parameters.

3. Since the car frame is not a regular cube, the deviation of the center of mass between the car and the car frame is considered.

Based on the above basic assumptions, the non-linear dynamic model of car system with active control is established as shown in Figure 2 (right is the positive direction of force). The parameters and their physical significance in the model are shown in Table 1.

According to Newton’s second law, the dynamic equation for the ultra-high-speed elevator car system dynamic model shown in Figure 2 can be modeled as

\[
\begin{align*}
    m_c \ddot{x}_c &= \sum_{i=1}^{4} (F_{ci} + F_{di}) + u_1 + u_2 \\
    m_f \ddot{y}_f &= -\sum_{i=1}^{4} (F_{ci} + F_{di}) + \sum_{i=1}^{4} (F_{ki} + F_{ei}) \\
    &- u_1 - u_2 + u_3 + u_4 \\
    J_c \ddot{\theta}_c &= -l_1 \sum_{i=1,3} (F_{ci} + F_{di}) + l_2 \sum_{i=2,4} (F_{ci} + F_{di}) \\
    &+ l_2 \cdot u_1 - l_2 \cdot u_2 \\
    J_f \ddot{\theta}_f &= l_3 \sum_{i=1,3} (F_{ci} + F_{di}) - l_6 \sum_{i=2,4} (F_{ci} + F_{di}) \\
    &- l_3 \cdot u_1 + l_6 \cdot u_2 + l_4 \cdot u_3 - l_4 \cdot u_4
\end{align*}
\]

Considering the nonlinearity of the guide shoe spring-damping system, the nonlinear spring force \( F_k \) and nonlinear damping force \( F_c \) can be described as follows: 24

\[
\begin{align*}
    F_{ki} &= k_1 \cdot \Delta z_i + k_3 \cdot \Delta z_i^3 \\
    F_{ci} &= c_1 \cdot \Delta \dot{z}_i + c_3 \cdot |\Delta \dot{z}_i| + c_2 \sqrt{\Delta \dot{z}_i^3} \text{sgn}(\Delta \dot{z}_i) (i = 1, \ldots, 4)
\end{align*}
\]
where \( k_i \) and \( c_i \) are the linear stiffness coefficient and linear damping coefficient of the guide shoe model, \( k_{sn} \) is a nonlinear stiffness coefficient, \( c_{i1} \) and \( c_{i2} \) are damping asymmetry correction coefficient and damping nonlinear correction coefficient, respectively. \( \Delta z_i (i = 1, \ldots, 4) \) are the displacements of the car frame, which obey the following relationships
\[
\Delta z_1 = x_1 + l_3 \sin \theta_f - x_f \\
\Delta z_2 = x_2 - l_4 \sin \theta_f - x_f \\
\Delta z_3 = x_3 + l_3 \sin \theta_f - x_f \\
\Delta z_4 = x_4 - l_4 \sin \theta_f - x_f
\] (3)

Besides, the spring force \( F_{hi} \) and damping force \( F_{di} \) generated by the rubber blocks are given by
\[
\begin{align*}
F_{hi} &= k_i \cdot \Delta z_i^2 \quad (i = 1, \ldots, 4) \\
F_{di} &= c_i \cdot \Delta z_i \quad (i = 1, \ldots, 4)
\end{align*}
\] (4)
in which \( k \) and \( c \) stand for the stiffness and damping coefficients of the rubber block, respectively. \( \Delta z_i (i = 1, \ldots, 4) \) are the deformations of the rubber blocks, which obey the following relationships
\[
\begin{align*}
\Delta z_1 &= x_f + l_1 \sin \theta_c - x_c \\
\Delta z_2 &= x_f - l_2 \sin \theta_c + l_6 \sin \theta_f - x_c \\
\Delta z_3 &= x_f + l_1 \sin \theta_c - l_6 \sin \theta_f - x_c \\
\Delta z_4 &= x_f - l_2 \sin \theta_c + l_6 \sin \theta_f - x_c
\end{align*}
\] (5)

**State equation of elevator car system considering nonlinearity and uncertainty**

In the ultra-high-speed elevator car system, due to the change of the number of passengers and the different specifications of the applications parts installed on the car frame, the car mass, the car moment of inertia, the car frame mass and the car frame moment of inertia vary within a certain range, which makes the horizontal vibration of the car system more complicated during operation. The uncertainties of the above four parameters can be described by the nominal value of the parameters themselves and their possible perturbation momentum, which are given by
\[
M = M + \Delta M
\] (6)

where \( M \) is the nominal value of the parameter, \( \Delta M \) denotes the perturbation momentum of the parameter.

The two main performances of the elevator car system are run stability and ride comfort, that is, the controller can guarantee the closed-loop car system in the presence of nonlinear factors and uncertain interference: the horizontal displacement and acceleration of the car system are as close to zero as possible, that is, the amplitude of \( x_c, x_f, \theta_c, \theta_f \) and \( \ddot{x}_c, \ddot{x}_f, \dot{\theta}_c, \dot{\theta}_f \) are reduced to the maximum extent. Defining the state variable as \( x = [x_{01} \ x_{02} \ x_{03} \ x_{04}]^T, x_{01} = [x_c \ x_f \ \theta_c \ \theta_f]^T, \)
\( x_{02} = [\dot{x}_c \ \dot{x}_f \ \dot{\theta}_c \ \dot{\theta}_f]^T; \) the control vector is \( u = [u_1 \ u_2 \ u_3 \ u_4]^T \) and \( |u| \leq u_{\text{max}} \); the output vector is \( y = x_{01} = c(x) \); the uncertain external disturbances except the guide rail excitation can be expressed as \( d(t) = [d_1(t) \ d_2(t) \ d_3(t) \ d_4(t)]^T \).

Consequently, the state equation of car system considering the compound uncertainty of the car system can be further expressed as follows
\[
\begin{align*}
\dot{x}_{01} &= x_{02} \\
\dot{x}_{02} &= M(A(x) + MBu + \Delta MA + \Delta MBu + (M + \Delta M)d(t)) \\
y &= c(x)
\end{align*}
\] (7)

where
\[
M = \left[ \begin{array}{c c c c}
\frac{1}{m_1} & \frac{1}{m_2} & \frac{1}{m_3} & \frac{1}{m_4} \\
\frac{1}{J_1} & \frac{1}{J_2} & \frac{1}{J_3} & \frac{1}{J_4} \\
\frac{1}{J_5} & \frac{1}{J_6} & \frac{1}{J_7} & \frac{1}{J_8} \\
\frac{1}{J_9} & \frac{1}{J_10} & \frac{1}{J_11} & \frac{1}{J_12}
\end{array} \right]^T
\]
\[
\Delta M = \left[ \begin{array}{c c c c}
\frac{\Delta m_1}{m_1} & \frac{\Delta m_2}{m_2} & \frac{\Delta m_3}{m_3} & \frac{\Delta m_4}{m_4} \\
\frac{\Delta J_1}{J_1} & \frac{\Delta J_2}{J_2} & \frac{\Delta J_3}{J_3} & \frac{\Delta J_4}{J_4} \\
\frac{\Delta J_5}{J_5} & \frac{\Delta J_6}{J_6} & \frac{\Delta J_7}{J_7} & \frac{\Delta J_8}{J_8} \\
\frac{\Delta J_9}{J_9} & \frac{\Delta J_10}{J_10} & \frac{\Delta J_11}{J_11} & \frac{\Delta J_12}{J_12}
\end{array} \right]^T
\]
\[
A(x) = \left[ \begin{array}{c c c c}
\sum_{i=1}^{4} F_{fi}(x) \\
\sum_{i=1}^{4} F_{fi}(x) \\
\sum_{i=1}^{4} F_{fi}(x) \\
\sum_{i=1}^{4} F_{fi}(x)
\end{array} \right]
\]
\[
B = \left[ \begin{array}{c c c c}
1 & 1 & 0 & 0 \\
-1 & -1 & 1 & 1 \\
-1 & -1 & 0 & 0 \\
-1 & -1 & 0 & 0
\end{array} \right]
\]

Equation (7) can be expressed as
\[
\begin{align*}
\dot{x} &= f(x) + \Delta f(x) + Gu + \Delta Gu + D \\
y &= c(x)
\end{align*}
\] (8)

where \( f(x) = [x_{02} \ \Delta MA(x)]^T \) and \( G = [0^{4 \times 4} \ MB]^T \) are the standard values without considering the compound uncertainty of the system; \( \Delta f(x) = [0^{4 \times 4} \ \Delta MA(x)]^T \) and \( \Delta G = [0^{4 \times 4} \ \Delta MB]^T \) are the uncertain quantities corresponding to \( f(x) \); \( G \), \( D = [0^{4 \times 4} \ (M + \Delta M)d(t)]^T \) is the value corresponding to other uncertain external disturbances.

Select position A on the car floor and position B on the car frame floor as the observation places, point A and point B are the center of the car and the car frame floor, respectively. Define \( x_c^* \) and \( a_c^* \) as the displacement and acceleration of the horizontal vibrations of position A, \( x_f^* \) and \( a_f^* \) as the displacement and acceleration of the horizontal vibrations of position B.

\[
\begin{align*}
x_{01}^* &= x_c + l_1 \sin \theta_c \\
x_{02}^* &= x_f + l_6 \sin \theta_f \\
a_c^* &= x_c + l_1 \left( \sin \theta_c \cdot \dot{\theta}_c + \cos \theta_c \cdot \dot{\theta}_c \right) \\
a_f^* &= x_f + l_6 \left( \sin \theta_f \cdot \dot{\theta}_f + \cos \theta_f \cdot \dot{\theta}_f \right)
\end{align*}
\] (9)

**Model experiment verification**

To verify the rationality of the model, the horizontal vibration acceleration of the real elevator was measured in the 7 m/s ultra-high-speed elevator test tower of Shandong
Table 2. Parameters and numerical values of real elevator used in simulation.

| Parameter | Value (Unit) | Parameter | Value (Unit) |
|-----------|--------------|-----------|--------------|
| $m_c$     | 1.5e3 kg     | $I_z$     | 1.17 m       |
| $m_f$     | 9.0e2 kg     | $I_z$     | 1.18 m       |
| $J_c$     | 1.8e3 kgm²   | $L$       | 1.2 m        |
| $J_f$     | 2.6e3 kgm²   | $L$       | 1.8 m        |
| $k$       | 4.0e5 N/m    | $l_6$     | 1.075 m      |
| $c$       | 1.2e2 N/s/m  | $l_6$     | 1.275 m      |
| $k_s$     | 3.0e4 N/m    | $k_n$     | 3.0e6 N/m    |
| $c_s$     | 6.0e2 N/s/m  | $c_1, c_2$| 20 N/s/m    |

The controller has a strong adaptive ability, which can effectively suppress chattering and reduce system stability errors. Taking the derivative of the sliding surface, we can obtain

$$\dot{\delta} = Y_\mu - Y^d_\mu + H_1\dot{e} + H_2e$$  \hspace{1cm} (12)
The predicted sliding mode performance index is constructed

\[ J(x, u, t) = \frac{1}{2} \dot{\delta}^T (t + \tau) Q \dot{\delta}(t + \tau) + u^T(t) R u(t) \] (13)

where \( \dot{\delta}(t + \tau) = \delta(t + \tau) + \tau \dot{\delta}(t) \) is the predicted sliding mode surface in the time period \( t + \tau \) in the future, \( \tau > 0 \) is the prediction time domain and its selection directly affects the accuracy of the prediction model, \( Q \) is a positive definite symmetric matrix, \( R \) is a positive semidefinite symmetric matrix.

When the control vector \( u \) is not weighted, that is, the weight matrix \( R = 0 \), the predictive control problem \( \min J(x, u, t) \) is solved, the predictive sliding mode control law under the nominal condition is

\[ u_{p*} = -\tau^{-1} \phi(x)^{-1} [\dot{\delta} + \tau (\epsilon(x) + H_1 \dot{\epsilon} + H_2 \epsilon - Y_\mu^d)] \] (14)

**Design of predictive sliding mode controller considering nonlinearity and uncertainty.** Based on the above, considering the uncertainty of mass, moment of inertia and external disturbances of elevator car system, (10) can be written as

\[ Y_\mu = \epsilon(x) + \Delta \epsilon(x) + (\dot{\phi}(x) + \Delta \phi(x)) u + D^* \] (15)

where \( \Delta \epsilon(x) \) and \( \Delta \phi(x) \) are the uncertainty corresponding to \( \epsilon(x) \) and \( \phi(x) \), and \( D^* \) is the uncertainty corresponding to uncertain external disturbances.

Based on (12) and (15), we can obtain

\[ \dot{\delta} = \epsilon(x) + \Delta \epsilon(x) + (\dot{\phi}(x) + \Delta \phi(x)) u + D^* - Y_\mu^d + H_1 \dot{\epsilon} + H_2 \epsilon \] (16)

And the compound uncertain disturbance term (CLUDT) is expressed as

\[ Z_D(x, u, D^*) = \Delta \epsilon(x) + \Delta \phi(x) u + D^* \] (17)

From (13), (15), (16), and (17), a predictive sliding mode control law considering nonlinearity and uncertainty is obtained

\[ u = -\tau^{-1} \phi(x)^{-1} [\dot{\delta} + \tau (\epsilon(x) + Z_D(x, u, D^*) + H_1 \dot{\epsilon} + H_2 \epsilon - Y_\mu^d)] \] (18)

Due to the changes of mass, moment of inertia and uncertain external disturbances in the car system cannot be measured, that is, \( Z_D(x, u, D^*) \) is unknown. With the increase of parameter variation and external disturbances, the performance of the control system will decline sharply, which will reduce the run stability and ride comfort of elevator. In order to improve the system performance, it is necessary to deal with \( Z_D(x, u, D^*) \).

**Design of adaptive fuzzy auxiliary controller.** An FLS is established to approximate \( Z_D(x, u, D^*) \). In order to solve the horizontal vibration caused by the compound uncertainty of the system, an adaptive fuzzy scheme is added to the conventional predictive sliding mode control. Using the approximation property of FLS, \( Z_D(x, u, D^*) \) is estimated and compensated to improve the chattering problem caused by SMC.

Based on the product reasoning method and the central average antifuzzy controller,26 the FLS of CUDT is designed as follows:

\[ \Psi(x(t)) = \sum_{q=1}^{\Gamma_{zq}^d} \frac{\Psi_q(\Pi_{q=1}^{\Gamma_{q1}^d} \partial \phi_j(x_j))}{\sum_{q=1}^{\Gamma_{q1}^d} (\Pi_{q=1}^{\Gamma_{q1}^d} \partial \phi_j(x_j))} = C_{z}^T \xi_{z}(x) \] (19)

where \( \Psi_q = \max_{\phi \in \mathbb{R}} \partial \phi_q(\Psi) \), the membership function \( \partial \phi_j(x_j) = \alpha_j^f \exp[-((x_j - b_j^f)/\epsilon_j^f)^2] \) used in this study is a Gaussian-shaped form. Let \( \xi_{zq} = \sum_{q=1}^{\Gamma_{zq}^d} \Pi_{q=1}^{\Gamma_{q1}^d} \partial \phi_j(x_j) \) be fuzzy basis functions and denote \( \xi_{z}(x) = [\xi_{z1}, \xi_{z2}, \ldots, \xi_{zQ}]^T \) and \( \Gamma_z = [\Psi_1^T, \Psi_2^T, \ldots, \Psi_P^T]^T = [\Gamma_{z1}^T, \Gamma_{z2}^T, \ldots, \Gamma_{zQ}^T]^T \).

**Lemma 1.** Let \( \alpha(x) \) be a continuous function defined on a compact set \( \Omega \). Then for any constant \( \omega > 0 \), there exists a FLS such as27

\[ \sup_{x \in \Omega} |\alpha(x) - \frac{\Gamma_z^T \xi_{z}(x)}{C_{z}}| \leq \omega \] (20)

By lemma 1, \( Z_D(x, u, D^*) \) can be expressed as

\[ Z_D(x, u, D^*) = \frac{\Gamma_z^T \xi_{z}(x) + \omega}{C_{z}} \] (21)

where \( \omega \) is the fuzzy approximation error of FLS to the CUDT, and \( |\omega| \leq \omega \). \( \omega \) is the upper bound of fuzzy approximation error, \( \Gamma_z \) is the ideal parameter vector, \( \Omega_z = \{\Gamma_z|\|\Gamma_z\| \leq M_z\} \), \( M_z \) is the design parameter, and \( M_z > 0 \).

In the car system, the perturbation momentum of parameters and uncertain external disturbances are bounded, that is, \( \Gamma_z \) is bounded, so the projection algorithm in adaptive control is introduced. The following adaptive law is used to adjust the parameter vector \( \Gamma_z \):

\[ \dot{\Gamma}_z = P_f (\Theta_T \xi_{z}(x) \dot{\delta}^T) + \Theta_T \xi_{z}(x) \dot{\delta}^T - 1 \Theta_T \hat{\delta}^T \Gamma_z \xi_{z}(x) \dot{\Gamma}_z \] (22)

where
Design of adaptive fuzzy auxiliary controller based on HTF. In order to further improve the performance of the controller and the comfort of the ultra-high-speed elevator, the sliding mode switching term is used to compensate the fuzzy approximation error \( \omega \) of FLS to the CUDT. Since the fuzzy approximation error in the car system changes continuously with time, the continuous smooth HTF is considered in this paper to replace the traditional discontinuity sign function, which can effectively reduce chattering in SMC. The HTF is defined as follows:

\[
I_z = \begin{cases} 
0, & \| \dot{\Gamma}_z \| < M_T, \text{ or } \| \ddot{\Gamma}_z \| = 0, \\
1, & \| \dot{\Gamma}_z \| = M_T, \| \ddot{\Gamma}_z \| \| \xi_z \| > 0
\end{cases}
\]  

(23)

where \( \dot{\Gamma}_z \) is the parameter estimation vector, and \( \Theta_T > 0 \) is the self-learning law of the FLS.

\[
\tanh \left( \frac{\delta}{\varphi^2} \right) = \frac{\delta_+ - \delta_-}{\delta_+ + \delta_-} + \frac{4}{\varphi^2}
\]  

(24)

where \( \varphi^2 > 0 \), the change speed of inflection point of the HTF is determined by the value of \( \varphi^2 \).

Compared with the sign function, the change rate of the HTF is more gentle as the independent variable value of \( \xi \). As can be seen from Figure 5, when the value of \( \varphi^2 \) is larger, the slope of the function is smaller, and the control effect is smoother. When the value of \( \varphi^2 \) is small, the HTF is close to the discontinuous sign function, and buffeting phenomenon will appear, which will affect the control effect. Therefore, in order to avoid chattering in the system, an appropriate value should be assigned to the slope parameter \( \varphi^2 \).

Lemma 2. For any real number \( \zeta \in R \) and non-zero real number \( \nu \), the following inequality holds.

\[
0 \leq |\zeta|(1 - \tanh(|\zeta/\nu|)) \leq \phi |\nu|
\]  

(25)

where \( \phi > 0 \), the minimum value \( \phi_{\text{min}} \) of \( \phi \) satisfies \( \phi_{\text{min}} = \zeta^*(1 - \tanh \zeta^*), \quad \zeta^* \) satisfies \( e^{-2\varphi} + 1 - 2\zeta^* = 0 \).

From (18), (21), and (22), an adaptive fuzzy auxiliary control law based on HTF is designed.

\[
u_{af} = -\varphi^{-1}(t) \left[ \Gamma_{\text{Tz}} \xi_z(x) + K_\text{t} \tanh \left( \frac{\delta}{\varphi^2} \right) \right]
\]  

(26)

To ensure that the motion of the system can reach the sliding surface, the error gain \( K_\text{t} \) must be sufficient to eliminate the effect of the fuzzy approximation error \( \omega \). Define

\[
K_\text{t} = \dot{\omega} + \eta
\]  

(27)

where \( \eta > 0 \) is a constant.

Set the adaptive laws as

\[
\dot{\varphi} = \kappa_{\text{m}} \| \delta \| \quad \dot{\zeta} = -4\gamma \dot{K}_\text{t} \varphi^2
\]  

(28)

(29)

where the constant \( \kappa_{\text{m}} \) and \( \gamma \) are constant parameters, which are very important to determine the convergence rate of the estimated parameters. The larger \( \kappa_{\text{m}} \) and \( \gamma \) can be selected to achieve the purpose of fast convergence, which means that the system needs a larger control input and the actuator control input is limited in reality. Therefore, the selection of \( \kappa_{\text{m}} \) and \( \gamma \) needs to consider the actual input and convergence rate of the system.

According to (14), (18), and (26), the predictive sliding mode control law based on adaptive fuzzy can be expressed as

\[
u = \nu_{ps} + \nu_{af}
\]  

(30)

Figure 5. HTF with different slope parameter and sign function.
\( V = \mathbf{Z}^T \mathbf{N}_i \mathbf{Z} \) \hspace{1cm} (31)

\( \mathbf{Z} = [\delta^T, \Gamma_z^T, \tilde{K}_i, \theta]^T \in \Omega_\mathbf{Z} \) \hspace{1cm} (32)

\( \mathbf{N}_i = \frac{1}{2} \text{diag} \{ I_s, \Theta_i^{-1} I_r, \kappa_i^m I_k, I_e \} \) \hspace{1cm} (33)

where \( \tilde{K}_i = K_i - \bar{K}_i \) is the gain error, and \( \bar{K}_i \) is the estimated value of the error gain \( K_i \), and \( \Gamma_z = \Gamma_z^+ - \Gamma_z^- \) is the parameter error vector. \( I_s, I_r, I_k, \) and \( I_e \) are unit matrices with proper dimensions.

The time derivative of \( V \) gives

\[ \dot{V} = \delta^T \dot{\delta} + \frac{1}{\Theta_i} \Gamma_z^T \Gamma_z - \frac{1}{\kappa_i} \tilde{K}_i \dot{\tilde{K}} + \frac{d}{dt} q^2 \] \hspace{1cm} (34)

From (16), (26), and (30), we can obtain

\[ \dot{\delta} = -\tau^{-1} \delta + \Gamma_z^T \xi(x) + \omega - \bar{K}_i \tanh(\delta/q^2) \] \hspace{1cm} (35)

Considering (27) and (28), we obtain

\[ \dot{\bar{K}}_i = \kappa_i \| \bar{\delta} \| \] \hspace{1cm} (36)

From (22), (34), (35), and (36), we have

\[ \dot{V} = \delta^T \left( -\tau^{-1} \delta + \Gamma_z^T \xi(x) + \omega - \bar{K}_i \tanh(\delta/q^2) \right) \]

\[ = \delta^T \left( -\tau^{-1} \delta + \Gamma_z^T \xi(x) + \omega - \bar{K}_i \tanh(\delta/q^2) \right) - \frac{\Gamma_z^T}{\Theta_i} \Theta_i \left[ \Theta_i \xi(x) \delta \right]^T - \bar{K}_i \| \delta \| - 4\bar{\gamma} \bar{K}_i q^2 \]

\[ = \delta^T \left( -\tau^{-1} \delta + \Gamma_z^T \xi(x) + \omega - \bar{K}_i \tanh(\delta/q^2) \right) - \frac{\Gamma_z^T}{\Theta_i} \Theta_i \left[ \Theta_i \xi(x) \delta \right] - \frac{\Gamma_z^T}{\Theta_i} \left( \Theta_i \xi(x) \delta \right) \delta^T - 1 \Theta_i \| \delta^T \Gamma_z \xi(x) \delta \| \Gamma_z \]

\[ = \dot{\tilde{K}}_i \| \delta \| - 4\gamma \tilde{K}_i q^2 \]

From inequality \( 2ab \leq a^2 + b^2 \), we can obtain

\[ \dot{V} = -\tau^{-1} \delta^T \delta < 0 \] \hspace{1cm} (43)
Since \( \tau > 0 \) in the controller, \( \dot{V} < 0 \) can be seen from (43), thus the control system is asymptotically stable. The proof is over.

**Simulation study**

For the sake of demonstrating the validity and superiority of the control method proposed in this paper, simulation analysis is carried on the 7 m/s ultra-high-speed elevator car system. In the simulation, the parameter uncertainties are described as: \( \Delta n_i = 1000 \sin t \) kg, \( \Delta \dot{n}_i = 100 \sin t \) kg, \( \Delta J_r = 1200 \sin t \) kg \cdot m^2, \( \Delta J_f = 278 \sin t \) kg \cdot m^2. The uncertain external disturbances \( d(t) \) is taken on by the random number which has a mean value of 0 and standard deviation 4. The performance of the elevator car system under passive control and under the control of PSMC-ST and PSMC-AF are compared and evaluated. The design parameters for the proposed controller are selected as \( T \) is 0.1 s, \( \Theta_F = 1 \), \( \kappa_n = 10 \), \( \gamma = 0.1 \), the initial value of \( G_z \) is 0. Take five membership functions:

\[
\begin{align*}
\vartheta_{NM}(x_i) &= \exp \left[ -\frac{(x_i - \pi/6)^2}{\pi/24} \right], \\
\vartheta_{NS}(x_i) &= \exp \left[ -\frac{(x_i - \pi/12)^2}{\pi/24} \right], \\
\vartheta_{PM}(x_i) &= \exp \left[ -\frac{(x_i - \pi/12)^2}{\pi/24} \right].
\end{align*}
\]

Then, two representative rail excitation conditions are employed and the numerical results are presented in the following.

**Impulse excitation response at guide rail joint**

Due to the influence of manufacturing, installation and other factors, impulse excitation will be generated by the guide rail joint, which will cause the horizontal vibration of the car system. The standard section length of the elevator guide is 5 m and the running speed is 7 m/s. Therefore, two pulse signals with amplitude of 5 mm and interval time of 0.71 s are set as the impulse excitation at guide rail joint after 1 s of the simulation.

Under the three control conditions, the horizontal vibration displacement and acceleration response curves of the car system excited by the guide rail joint pulse are shown in Figures 7 and 8. It can be seen that the displacement and acceleration of the two controlled active car systems are less than those of the passive car system despite the presence of nonlinear guide shoe and uncertainty of the system. Compared with PSMC-ST, PSMC-AF can make the car system have lower acceleration and displacement peak value, shorter stability time, stronger ability to approach uncertain disturbance, and it is more helpful to improve riding stability and comfort.

It is widely recognized that the control performance of the system can be quantified by reference to RMS and MAX values. The RMS and MAX values of horizontal vibration displacement and acceleration of the car system are given in Tables 3 and 4, which reflect the percentage of performance improvement achieved by the PSMC-ST and PSMC-AF with respect to the passive control. It is obvious that the run stability (displacement) is significantly improved under the active control cases, because the RMS and MAX values decrease by around 40% when using the PSMC-ST and by more than 53% when using the PSMC-AF. It is clear that ride comfort (acceleration) with the PSMC-ST (~70%) is inferior to that of with the PSMC-AF (>80%).

The comparison of the actuator output force is given in Figure 9. It is seen from the figure that the actuator 3 has the problem of output saturation under the control of PSMC-ST, so the optimal control cannot be achieved. Compared with PSMC-ST, under the control

![Figure 7. Time response of displacement under impulse excitation. (a) Comparison of car floor vibration displacement. (b) Comparison of car frame floor vibration displacement.](image-url)
of PSMC-AF, the four actuators meet the control force constraint and their output force is smaller, which can play the role of energy saving and reduce energy waste.

**Random excitation response at guide rail surface**

In order to further evaluate the performance of the proposed controller, considering the irregularity of the guide rail surface caused by the manufacturing. The surface roughness error of the guide rail approximately conforms to a normal distribution. Gauss white noise was used to simulate the random excitation of the guide rail surface, the mean value is $\mu_{dg} = 0$, and the standard deviation is $\sigma_{dg} = 0.6mm$.29

Under the random excitation of the guide rail surface and uncertain external disturbances, Figures 10 and 11 compare the horizontal displacement and acceleration of the three control cases of the car systems over time. After carefully observing the response curves, one can find that the PSMC-ST is robust against the horizontal vibration of the car system better than the passive control; however, its performance is not as good as that of PSMC-AF. This indicates the superiority of the PSMC-AF in spite of the existence of nonlinear guide shoe and uncertainty of the system. The RMS and MAX values are compared in Tables 5 and 6. It is clear that the PSMC-ST and PSMC-AF can achieve satisfactory car system performance, stabilizing the system attitude and improving ride comfort. In addition, for the proposed controller, the RMS and MAX values of the car’s vibration displacement are reduced by 85.8% and 81.1%, and the RMS and MAX values of the vibration displacement of the car frame are decreased by 82.2% and 79.3%, respectively; the RMS and MAX values of the car’s vibration acceleration are reduced by 71.7% and 66.9%, and the RMS and MAX values of the vibration acceleration of the car frame are decreased by 69.7% and 62.1%, respectively. These reductions are

---

**Table 3.** Comparison of displacement response under impulse excitation.

| Control method | Displacement response of car (m) | Displacement response of car frame (m) |
|----------------|---------------------------------|--------------------------------------|
|                | RMS   | MAX   | RMS    | MAX   |
| Passive        | 9.890e-5 | 3.196e-4 | 1.023e-4 | 3.495e-4 |
| PSMC-ST        | 5.466e-5 (44.7%) | 1.925e-4 (39.8%) | 5.527e-5 (46.0%) | 1.978e-4 (43.4%) |
| PSMC-AF        | 4.199e-5 (57.5%) | 1.493e-4 (53.3%) | 4.350e-5 (55.5%) | 1.539e-4 (56.0%) |

**Table 4.** Comparison of acceleration response under impulse excitation.

| Control method | Acceleration response of car (m/s²) | Acceleration response of car frame (m/s²) |
|----------------|------------------------------------|----------------------------------------|
|                | RMS   | MAX   | RMS    | MAX   |
| Passive        | 0.0274 | 0.1462 | 0.0722 | 0.4086 |
| PSMC-ST        | 0.0081 (70.4%) | 0.0390 (73.3%) | 0.0147 (79.6%) | 0.0855 (79.1%) |
| PSMC-AF        | 0.0055 (80.0%) | 0.0272 (81.4%) | 0.0083 (88.5%) | 0.0486 (88.1%) |

**Figure 8.** Time response of acceleration under impulse excitation. (a) Comparison of car floor vibration acceleration. (b) Comparison of car frame floor vibration acceleration.
greater than those seen for the PSMC-ST. The reason why the PSMC-AF has an advantage over the PSMC-ST is owing to its excellent capability to deal with uncertainty. These results further confirm the effectiveness of PSMC-AF.

In order to verify the control effect of the proposed controller in the frequency domain, the frequency response curves of the horizontal vibration acceleration of the car system are shown in Figure 12. It can be seen that the resonance frequencies of the response curves are concentrated in the low frequency range of 0–10 Hz. Under the active control, the amplitude of low-frequency range of car system is significantly reduced, and the amplitude of car system under the control of PSMC-AF is smaller than PSMC-ST.

Table 5. Comparison of displacement response under random excitation.

| Control method | Displacement response of car (m) | Displacement response of car frame (m) |
|----------------|---------------------------------|--------------------------------------|
|                | RMS                    | MAX                      | RMS | MAX |
| Passive        | 1.212e-4               | 2.638e-4                 | 1.231e-4 | 2.919e-4 |
| PSMC-ST        | 3.251e-5 (73.2%)       | 1.018e-4 (61.4%)         | 3.528e-5 (71.3%) | 1.160e-4 (60.3%) |
| PSMC-AF        | 1.720e-5 (85.8%)       | 4.991e-5 (81.1%)         | 2.190e-5 (82.2%) | 6.039e-5 (79.3%) |

Table 6. Comparison of acceleration response under random excitation.

| Control method | Acceleration response of car (m/s²) | Acceleration response of car frame (m/s²) |
|----------------|-------------------------------------|------------------------------------------|
|                | RMS                   | MAX                      | RMS           | MAX |
| Passive        | 0.0342                | 0.0922                    | 0.1118        | 0.2969 |
| PSMC-ST        | 0.0144 (57.9%)        | 0.0463 (49.8%)           | 0.0472 (57.8%) | 0.1317 (55.6%) |
| PSMC-AF        | 0.0108 (68.4%)        | 0.0305 (66.9%)           | 0.0339 (69.7%) | 0.1124 (62.1%) |

Figure 9. The actuator output force of the car system under pulse excitation. (a) The output force of actuator 1. (b) The output force of actuator 2. (c) The output force of actuator 3. (d) The output force of actuator 4.
Figure 10. Time response of displacement under random excitation. (a) Comparison of car floor vibration displacement. (b) Comparison of car frame floor vibration displacement.

Figure 11. Time response of acceleration under random excitation. (a) Comparison of car floor vibration acceleration. (b) Comparison of car frame floor vibration acceleration.

Figure 12. Frequency response of acceleration under random excitation. (a) Comparison of car floor vibration acceleration. (b) Comparison of car frame floor vibration acceleration.
PSMC-ST, PSMC-AF has better control effect and requires less output force, which can save energy while ensuring good control effect, verifying the effectiveness, and robustness of PSMC-AF.

Conclusions

1. In this paper, considering the nonlinearity of guide shoe, parameter uncertainties and uncertain external disturbances of the elevator car system, a more practical active control model for horizontal vibration of the 4-DOF elevator car system was derived, the correctness and rationality of the established model are verified by real elevator experiment, which lays a foundation for the research on the horizontal vibration active control method of the car system.

2. Based on the above active control model, a PSMC-AF was designed to reduce the horizontal vibration of the car system. In order to reduce the influence of compound uncertainty on the car system, a FLS was established to approximate the CUIT on-line, the continuous smooth HTF was introduced into the sliding mode switching term to compensate the fuzzy approximation error. Finally, the stability of the closed-loop system was proved by Lyapunov stability theory.

3. By MATLAB to simulate the horizontal vibration of 7 m/s elevator car system, the results show that PSMC-AF can attenuate more than 53% of the RMS and MAX of the horizontal vibration displacement of car system, and reduce more than 62% of the RMS and MAX of the horizontal vibration acceleration of car system under the different rail excitation conditions. Furthermore, the control effect of the PSMC-AF is much better than the PSMC-ST and passive control. This proves the superiority of the proposed controller, which has good robustness to the nonlinearity of guide shoe and the uncertainty of the car system, and can ensure the run stability and ride comfort of the ultra-high-speed elevator.

4. The control strategy proposed in this paper has strong robustness to the nonlinearity and uncertainty of the system, but the sliding surface require a process to converge to zero asymptotically. Shortening the time to reach the sliding surface and eliminating the chattering effect on the closed-loop system more effectively, meanwhile ensuring the control accuracy have become a problem to be further studied in the future.

Acknowledgements

The authors are grateful for the equipment support provided by Shandong Fiji Zhiyu Elevator Co., Ltd. The authors sincerely thank the editors and reviewers for their insights and comments to further improve the quality of the manuscript.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This research was funded by the Shandong Province Natural Science Foundation, China (Grant No. ZR2017ME0049) and the Key Research Development Project of Shandong Province (Grant No. 2018GSF122004).

ORCID iD

Mingjin Zhang https://orcid.org/0000-0003-2159-1041

References

1. Knezevic BZ, Blanusa B and Mercetic DP. A synergistic method for vibration suppression of an elevator mechatronic system. J Sound Vib 2017; 406: 29–50.
2. Zhang RJ, Wang C and Zhang Q. Response analysis of the composite random vibration of a high-speed elevator considering the nonlinearity of guide shoe. J Braz Soc Mech Sci Eng 2018; 40(4): 1–10.
3. Bao JH, Zhang P, Zhu CM, et al. Vibration control of high speed elevator lifting system based on tensioning device. Vib Shock 2017; 36(14): 221–226.
4. Santo D, Balthazar JM, Tusset AM, et al. On nonlinear horizontal dynamics and vibrations control for high-speed elevators. J Vib Control 2018; 24(5): 825–838.
5. Nakano K, Hayashi R, Suda Y, et al. Active vibration control of an elevator car using two rotary actuators. J Syst Des Dyn 2011; 5(1): 155–163.
6. Cao SX, He Q, Zhang RJ, et al. Active control strategy of high-speed elevator horizontal vibration based on LMI optimization. Control Eng Appl Inf 2020; 22(1): 72–83.
7. Qiu LM, Wang ZL, Zhang SY, et al. A vibration-related design parameter optimization method for high-speed elevator horizontal vibration reduction. Shock Vib 2020; 2020: 1–20.
8. Chen C, Zhang RJ, Zhang Q, et al. Mixed guaranteed cost control for high speed elevator active guide shoe with parametric uncertainties. Mech Ind 2020; 21(5): 502.
9. Zhang Q, Yang Z, Wang C, et al. Intelligent control of active shock absorber for high-speed elevator car. Proc IMechE, Part C: J Mechanical Engineering Science 2018; 233(11): 3804–3815.
10. Sharma JN and Grover D. Thermoelastic vibration analysis of MemS/Nems plate resonators with voids. Acta Mechanica 2012; 223(1): 167–187.
11. Mei DQ, Du XQ and Chen ZC. Vibration analysis of high-speed traction elevator based on rolling guide shoe–guide rail contact model. J Mech Eng 2009; 45(5): 264–270.
12. Feng YH, Zhang JW and Zhao Y. Modeling and robust control of horizontal vibrations for high-speed elevator. J Vib Control 2009; 15(9): 1375–1396.
13. Ji XH, Wang CW, Zhang ZY, et al. Nonlinear adaptive position control of hydraulic servo system based on sliding mode back-stepping design method. Proc IMechE,
14. Pai MC. Quasi-output feedback global sliding mode tracker for uncertain systems with input nonlinearity. *Nonlinear Dyn* 2016; 86(2): 1215–1225.

15. Heydari ZR and Karimaghaee P. Projective synchronization of different uncertain fractional-order multiple chaotic systems with input nonlinearity via adaptive sliding mode control. *Adv Differ Equations* 2019; 2019(4): 100–111.

16. Hu Q, Du HB and Yu DM. Fuzzy adaptive integral sliding mode control of suspension altitude for single maglev guiding system in linear elevator. In: *International conference on mechatronics and industrial informatics (ICMII 2013)*, Guangzhou, China, 13–14 March 2013, pp.1708–1711. Switzerland: Trans Tech Publications Ltd.

17. Hu Q, Hao ML and Liu H. Adaptive sliding mode control of maglev guiding system for ropeless elevator based on feedback linearization. In: *23rd Chinese control and decision conference*, Mianyang, China, 23–25 May 2011, pp.1007–1010. New York: IEEE.

18. Park KB and Tsuji T. Terminal sliding mode control of second-order nonlinear uncertain systems. *Int J Robust Nonlinear Control* 1999; 9(11): 769–780.

19. Zhou JS, Liu ZY and Pei R. A nonlinear continuous predictive variable structure control scheme. *J Electr Mach Control* 2000; 4(4): 227–229.

20. Acary V, Brogliato B and Orlov YV. Chattering-free digital sliding-mode control with state observer and disturbance rejection. *IEEE Trans Autom Control* 2012; 57(5): 1087–1101.

21. Lu XL, Du FP, Jia Q, et al. Sliding mode force control of an electrohydraulic servo system with RBF neural network compensation. *Mechanika* 2019; 25(1): 32–37.

22. Wu HZ, Liu SY, Cheng C, et al. Observer based adaptive double-layer fuzzy control for nonlinear systems with prescribed performance and unknown control direction. *Fuzzy Sets Syst* 2020; 392: 93–114.

23. Tong SC, Li YM and Sui S. Adaptive fuzzy tracking control design for SISO uncertain nonstrict feedback nonlinear systems. *IEEE Trans Fuzzy Syst* 2016; 24(6): 1441–1454.

24. Dangor M, Dahunsi OA, Pedro JO, et al. Evolutionary algorithm-based PID controller tuning for nonlinear quarter-car electrohydraulic vehicle suspensions. *Nonlinear Dyn* 2014; 78(4): 2795–2810.

25. Isidori A. *Nonlinear control systems: an introduction*. 3th ed. New York, NY: USA, 1995, p.124.

26. Zhang Q and Dong JX. Disturbance-observer-based adaptive fuzzy control for nonlinear state constrained systems with input saturation and input delay. *Fuzzy Sets Syst* 2020; 392: 77–92.

27. Diao SZ, Sun W and Yuan W. Adaptive fuzzy practical tracking control for flexible-joint robots via command filter design. *Meas Control* 2020; 53(5–6): 814–823.

28. Wang JH, Zhu HY, Zhang CL, et al. Adaptive hyperbolic tangent sliding-mode control for building structural vibration systems for uncertain earthquakes. *IEEE Access* 2018; 6: 74728–74736.

29. Yang Z, Zhang Q, Zhang RJ, et al. Transverse vibration response of a super high-speed elevator under air disturbance. *Int J Struct Stab Dyn* 2019; 19(9): 1950103.