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RESEARCH ARTICLE

Effect of the Controllers With Control Delay on Vehicle-Guideway Coupling Vibration for Maglev Train at Standing Still

KEREN WANG[1,2], **WEIHUA MA**[3], **XIAOHAO CHEN**[4,5], **AND RUIMING ZOU**[6]

[1]School of Computing and Artificial Intelligence, Southwest Jiaotong University, Chengdu 611756, China
[2]Department of Intelligent Manufacturing, Leshan Vocational and Technical College, Leshan 614000, China
[3]State Key Laboratory of Traction Power, Southwest Jiaotong University, Chengdu 610031, China
[4]Key Laboratory of Intelligent Operation and Maintenance Technology and Equipment for Urban Rail Transit of Zhejiang Province, Zhejiang Normal University, Jinhua, Zhejiang 321004, China
[5]College of Engineering, Zhejiang Normal University, Jinhua, Zhejiang 321004, China
[6]CAAC Academy, Civil Aviation Flight University of China, Guanghan 618307, China

Corresponding author: Ruiming Zou (rmzou@outlook.com)

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ABSTRACT The commercial operation of maglev trains is significantly affected by vibrations resulting from the coupling between the vehicle and track. Previous research has primarily focused on single electromagnet suspension systems, neglecting the dynamic disparities between these systems and vehicle suspension systems. Furthermore, there is a lack of comprehensive analysis regarding instability mechanisms in the suspension of maglev trains on flexible track beams. To address these gaps, a vertical dynamics model for vehicle-guideway coupling was developed with three levitation frames. Initially, this study examined the impact of controller delays on the divergence rate between the vehicle and track beam when employing a double-loop PID control algorithm. Additionally, it analyzed how implementing a redundant control mode at levitation points influences vibration responses caused by these delays. The findings indicate that control delays have a more pronounced impact on the track beam compared to the vehicle system, making it more susceptible to initial vibration divergence. For instance, instability in the suspension system occurs when the time delay of the single-point suspension controller reaches about 2.146 ms. Moreover, utilizing a redundant control method for levitation points can partially alleviate coupling vibrations resulting from controller delays. For example, the instability of the suspension system can be caused by a controller delay exceeding about 7ms. Importantly, when the first-order vibration frequency of the track beam falls within a specific range and reaches a critical threshold for controller delay, energy supplied to the track beam by the levitation system can surpass its damping dissipation capacity, leading to sustained coupling vibrations.

INDEX TERMS Maglev vehicle, vehicle-track coupling vibration, delay in levitation controller, redundancy control, analysis of energy mechanism.

I. INTRODUCTION

The operation of maglev vehicles is characterized by the absence of direct mechanical contact with the track, thereby eliminating the adhesion constraints commonly encountered in conventional wheel-rail systems. Moreover, maglev technology offers a multitude of advantages, including minimal vibration and noise levels, seamless operational performance, a compact turning radius, and impressive climbing capabilities. As a result, these vehicles have garnered significant interest from various nations and continue to be a focal point of ongoing research [1], [2], [3]. The classification of maglev vehicles can be based on their suspension principles, which include Electromagnetic Suspension (EMS),

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Electrodynamic Suspension (EDS), high-temperature superconducting suspension, and electromagnetic-permanent magnet hybrid suspension. Furthermore, they can also be categorized by speed into medium-low speed, medium speed, high speed, ultra-high speed, and aerospace speed maglev systems. In the realm of commercial operations, EMS maglev trains are notably exemplified by the Incheon Airport Line in South Korea, the Changsha Maglev Airport Line, the Beijing Maglev S1 Line, as well as the Phoenix and Qingyuan Maglev Tourist Lines [4], [5]. The cornerstone of maglev train technology lies in suspension control, which is crucial for ensuring safe and efficient operation. The suspension control system of EMS maglev trains primarily consists of suspension sensors, controllers, and electromagnets. Its essential function is to maintain a precise air gap of 8 to 10 mm between the train and the track. Inadequate control performance exhibited by the suspension controller can compromise the train's stability, leading to jolts or swaying that pose potential safety hazards. Moreover, subpar control performance can also negatively impact operational speed and passenger comfort, thereby diminishing the overall travel experience. Therefore, the stable and reliable performance of suspension controllers should be prioritized in order to ensure the safe operation of maglev trains.

To enhance the suspension performance of maglev train control, a nonlinear dynamic model utilizing state equations with magnetic flux feedback was established, and an adaptive sliding mode controller was devised within the sliding mode framework to mitigate the upper bounds of uncertainty and disturbances [6]. A sliding mode adaptive state feedback controller for the maglev system, designed using Radial Basis Function (RBF) network approximation, demonstrates superior dynamic response, robustness, and reduced overshoot compared to traditional PID and fuzzy controllers, while effectively accommodating flexible trajectories and external disturbances [7]. Moreover, a novel nonlinear controller has been developed for the suspension system subjected to external periodic disturbances, showcasing enhanced system stability and improved disturbance suppression compared to classical linearized feedback control [8]. The challenge of random noise in suspension gap signals during maglev train operation was addressed by incorporating a Tracking Differentiator (TD) at the output of the gap sensor for tracking filtering, demonstrating that this controller achieves stable suspension while effectively mitigating random noise with varying intensities in the gap sensor [9]. The suspension control problem under system parameter perturbations has been addressed by designing a reinforcement learning-based suspension controller that demonstrates a faster dynamic response and superior tracking accuracy amidst parameter fluctuations compared to traditional Proportional-Integral-Derivative (PID) control methods [10]. Additionally, a model reference adaptive self-learning suspension controller has been proposed to effectively address the challenges posed by unknown nonlinear forces and uncertainties in transfer functions due to track irregularities, with adjustable parameters meticulously fine-tuned based on the system state, error, and time to stabilize the suspension gap at a constant value and enhance the controller's adaptability [11]. Lastly, a sliding mode active disturbance rejection control scheme was implemented to enhance the bandwidth, robustness, and fault tolerance of the suspension controller, ensuring stable suspension of the maglev train even in the event of simultaneous failures of multiple sensors during operation [12].

After examining the aforementioned literature, it can be inferred that the suspension control system faces numerous challenges primarily due to its technical intricacies and complexities. Firstly, accurately detecting the suspension gap in intricate electromagnetic environments and adverse weather conditions proves to be a formidable task. Secondly, factors such as flexible track vibrations, inherent nonlinearity, dynamic eddy current effects, coupling disturbances from multiple electromagnets, significant load fluctuations, and the deterioration of suspension performance on curved tracks present a plethora of unknown risks to ensure stable operation of maglev trains in complex environments [13]. The current stage necessitates the resolution of vehicle-track coupled vibration, which has emerged as a prevalent and urgent issue in the commercial operation of maglev trains. The probability of vehicle-track coupled vibration occurring is significantly increased when maglev trains are in a static levitation state or traverse elastic bridges at relatively low speeds [14], [15]. For instance, the maglev test vehicles developed by the National University of Defense Technology and Tongji University both encountered significant vehicle-track coupled vibration phenomena during real-world testing [16], [17]. The occurrence of this phenomenon can be attributed to time delays in the control loop of the suspension controller, which are inevitable and often give rise to intricate dynamic behaviors such as system resonance and bifurcation [18], [19].

The neglect of time delay's impact on the performance of maglev controller will impede its effective application in engineering practice. In response to the vibration issues induced by controller time delays in maglev trains, numerous scholars, both domestically and internationally, have conducted extensive research. The local dynamics surrounding the trivial solution of maglev train suspension systems with time-delayed feedback signals were investigated, followed by a linear stability analysis using the characteristic root method, which concluded that both the existence of Hopf bifurcation and the amplitude of periodic solutions can be determined by adjusting the time delay and control parameters [20]. The stability of the vehicle-track coupling system was investigated through an analysis focusing on a single suspended electromagnet, revealing that minimizing time delays in the gap and speed channels of the suspension controller is crucial for maintaining stability, while introducing time delay in the current channel can be advantageous to system stability [21]. The stability and bifurcation behavior of the electromagnetic suspension system for maglev vehicles, along with the coupling system between the electromagnetic rail beam, were further investigated through theoretical and numerical analysis, establishing a relationship between the natural frequency of the rail beam and its damping ratio, while also noting that intricate dynamical behavior is observed in the system as parameters approach the Bautin bifurcation point [22]. An indicator has been proposed to assess the compatibility between the maglev train and rail beam in terms of stability and bifurcation of the electromagnetic maglev transportation system, defined as the ratio of the size of the stability region to the static deflection of the rail beam, and we have also determined both the equilibrium point stability and the extent of the convergence range under varying parameters [23]. An Amplitude Saturation Controller (ASC) has been proposed to generate saturated unidirectional attractive force control signals for active air gap management, with the stability and convergence of the closed-loop signals demonstrated using the Lyapunov method, and further integration with Radial Basis Function (RBF) neural networks resulted in a Neural Network-Based Supervisor Controller (NNBSC), effectively addressing controller time delays [24]. A model of a two-degree-of-freedom maglev train suspension system was established, incorporating displacement and velocity as feedback control parameters while accounting for controller time delays, and the relationship among feedback control parameters, system parameters, and the critical value of controller time delay was analyzed, revealing that, for fixed system parameters, the critical value of controller time delay depends on vehicle parameters [25]. The electromagnetic suspension system of maglev vehicles utilizes an adaptive neural network controller with input delay compensation and an optimized control parameter scheme to address significant engineering challenges such as external disturbances, input delays, and variations in mass, effectively mitigating air gap vibrations, particularly in the presence of time delay and uncertain dynamics, thereby greatly enhancing suspension control performance [26]. The electromagnetic suspension system employed a Smith predictor to compensate for time delay while introducing a Linear Active Disturbance Rejection Controller (LADRC) to estimate and compensate for the error between predicted and actual time delays, and by treating this error as an internal disturbance, a Smith-LAD-RC suspension control method was proposed, effectively addressing coupling vibration issues between the vehicle and track caused by controller time delay [27]. The objective of this study is to determine the theoretical critical value for time delay in a single electromagnet suspension system and subsequently extend it to a vehicle model and a vehicle-beam coupled dynamic model, enabling the calculation of engineering critical values for time delay in the complex coupled system, and it can be concluded that at higher operating speeds, the impact of time delay on the dynamic response of the maglev system becomes increasingly significant [28]. A coupled dynamic model of maglev vehicles and guideways was established, indicating that enhancing the bending stiffness and damping ratio of the guideway, reducing the gain for gap feedback control while increasing the gain for speed feedback control, and augmenting the damping of the secondary suspension can alleviate the impact of suspension control time delay on the stability of the system's suspension [29].

Through the aforementioned analysis, it can be inferred that contemporary designs for maglev train suspension controllers tend to overlook the impact of time delay on suspension stability. They primarily focus on controller robustness and tracking performance, while neglecting to investigate the relationship between time delay and system stability in a comprehensive vehicle suspension system context. Additionally, there is insufficient elucidation provided regarding the divergence sequence of the coupling system (track and vehicle) during suspension instability.

In this paper, the entire vehicle model, flexible track beam model, and suspension controller model of the maglev train are established in order to address this issue. Subsequently, the impact of a single controller's delay on the dynamic characteristics of each suspension point within the suspension system is examined. Furthermore, the influence of redundant controllers on system stability under identical time delays is investigated. Then, an explanation is provided from the perspective of work performed by the suspension system on the track beam regarding how time delay can lead to suspension instability. Lastly, the solution is explored when the time delay of the system still results in system instability even with the adoption of redundant control mode.

II. VEHICLE-GUIDEWAY COUPLING MODEL

A. TRACK BEAM MODEL

During the analysis of vehicle-guideway coupling vibration, the track beam is commonly simplified as a Bernoulli-Euler beam. Its dynamic characteristics can be described by superimposing the product of mode coordinates and vibration functions for each order. The vertical vibration dynamic equation of the track beam can be established by considering the left endpoint of the track beam as the origin of the coordinate system [30], as shown in equation (1) below:

$$EI\frac{\partial^4 z(x,t)}{\partial x^4} + \delta\frac{\partial^5 z(x,t)}{\partial x^4 \partial t} + \rho\frac{\partial^2 z(x,t)}{\partial t^2} = F_E\sigma(x) \tag{1}$$

where EI represents the bending stiffness of the track beam, δ denotes the damping coefficient of the track beam, ρ signifies the line density of the track beam, and σ represents the position function. These parameters are utilized to transmit positional information regarding the acting force. F_E refers to the external force applied on the bridge, while x denotes the coordinate indicating relative position between electromagnet and track beam. By employing modal superposition method, we can express vertical deflection of an elastic bridge as follows:

$$z(x,t) = \sum_{i=1}^{\infty} \Delta_i(x)q_i(t) \tag{2}$$

where, q_i represents the i-th modal coordinate of the beam, while Δ_i denotes the i-th vibration mode function of the beam, as shown below:

$$\Delta_i(x) = \sqrt{\frac{2}{\rho l}} \sin \frac{i\pi x}{l} \tag{3}$$

where, l is the length of the track beam. Substituting equation (2) into equation (1), multiplying Δ_i both sides of the equation, and integrating the equation from 0 to l, according to the modal orthogonality condition, we can get:

$$\ddot{q}_i(t) + 2\delta w_i \dot{q}_i(t) + w_i^2 q_i(t) = \frac{1}{\rho l} \int_0^l F_E \Delta_i dx \tag{4}$$

The i-th modal frequency of the beam, w_i, can be expressed as follows:

$$w_i = \lambda_i^2 \sqrt{\frac{EI}{\rho}} \tag{5}$$

where, λ_i is the i-th modal wavelength of the beam.

B. VERTICAL DYNAMIC EQUATION OF MAGNETIC LEVITATION

As the vehicle system is symmetric and decoupled in the longitudinal direction, the longitudinal symmetric 1/2 system is taken as the research object in this paper. As shown in Fig. 1. Where, M is the mass of the car body, z_0 is the displacement of the relative balance position of the center of the vehicle body, $\ddot{\theta}_0$ is the rotational angular acceleration of the center of the vehicle body, L_v is the length of the car body 1/2. m_1, m_2 and m_3 are the mass of suspension frame 1, 2 and 3, respectively. z_1, z_2 and z_3 are the displacement of the relative equilibrium position of suspension frame 1, 2 and 3, respectively. $\ddot{\theta}_1$, $\ddot{\theta}_2$ and $\ddot{\theta}_3$ are nods angular acceleration of suspension frame 1, 2 and 3, respectively. J_1, J_2 and J_3 represent the nods inertia of suspension frame 1, 2 and 3, respectively. L_f is the length of single suspension frame 1/2. k and c respectively represent the stiffness and damping of air spring. f_n is the suspended electromagnetic force corresponding to number n. It is assumed that the car body and a single suspension frame have two vertical degrees of freedom, respectively. The ups and downs motion is taken down in a positive direction, and the nodding motion is taken the clockwise direction in a positive direction. The dynamic equation of the maglev train at the rated suspension position is described as follows equation (6)-(13). Levitation force f_n is shown in equation (14). Car body ups and downs:

$$\begin{aligned} M\ddot{z}_0 = &-6kz_0 - 6c\dot{z}_0 + 2k(z_1 + z_2 + z_3) \\ &+ 2c(\dot{z}_1 + \dot{z}_2 + \dot{z}_3) \end{aligned} \tag{6}$$

Car body nods:

$$\begin{aligned} J_0\ddot{\theta}_0 = &(8kL_fL_v - 10kL_f^2 - 4kL_v^2)\theta_0 \\ &+ 2kL_f^2(\theta_1 + \theta_2 + \theta_3) \\ &+ 2k(z_1 - z_3)(L_v - L_f) \\ &+ (8cL_fL_v - 10cL_f^2 - 4cL_v^2)\dot{\theta}_0 \\ &+ 2cL_f^2(\dot{\theta}_1 + \dot{\theta}_2 + \dot{\theta}_3) \\ &+ 2c(\dot{z}_1 - \dot{z}_3)(L_v - L_f) \end{aligned} \tag{7}$$

Suspension frame 1 ups and downs:

$$\begin{aligned} m_1\ddot{z}_1 = &2k(L_v - L_f)\theta_0 + 2k(z_0 - z_1) \\ &+ 2c(L_v - L_f)\dot{\theta}_0 + 2c(\dot{z}_0 - \dot{z}_1) - f_1 - f_2 \end{aligned} \tag{8}$$

Suspension frame 1 nods:

$$J_1\ddot{\theta}_1 = 2kL_f^2(\theta_0 - \theta_1) + 2cL_f^2(\dot{\theta}_0 - \dot{\theta}_1) + (f_2 - f_1)L_f \tag{9}$$

Suspension frame 2 ups and downs:

$$m_2\ddot{z}_2 = 2k(z_0 - z_2) + 2c(\dot{z}_0 - \dot{z}_2) - f_3 - f_4 \tag{10}$$

Suspension frame 2 nods:

$$J_2\ddot{\theta}_2 = 2kL_f^2(\theta_0 - \theta_2) + 2cL_f^2(\dot{\theta}_0 - \dot{\theta}_2) + (f_4 - f_3)L_f \tag{11}$$

Suspension frame 3 ups and downs:

$$\begin{aligned} m_3\ddot{z}_3 = &2k(L_f - L_v)\theta_0 - 2k(z_3 - z_0) \\ &+ 2c(L_f - L_v)\dot{\theta}_0 \\ &- 2c(\dot{z}_3 - \dot{z}_0) - f_5 - f_6 \end{aligned} \tag{12}$$

Suspension frame 3 nods:

$$J_3\ddot{\theta}_3 = 2kL_f^2(\theta_0 - \theta_3) + 2cL_f^2(\dot{\theta}_0 - \dot{\theta}_3) + (f_6 - f_5)L_f \tag{13}$$

Levitation force:

$$f_j = \frac{\mu N^2 A_m}{4} \left(\frac{\Delta i}{\Delta c}\right)^2 (j = 1 - 6, n = 1 - 6) \tag{14}$$

In the equation(14), μ is the vacuum permeability, N is the number of coil turns, Am is the polar area of the electromagnet, Δc is the change in suspension clearance, and Δi is the change in suspension current.

III. DYNAMIC SIMULATION ANALYSIS

Since the maglev vehicle is controlled separately by the levitation controllers, the vehicle static suspension control problem can be decomposed into the control problem of a single suspension electromagnet by the levitation controller. Reference to the literature [31], [32], the double loop PID controller is adopted in this paper, and the schematic diagram with levitation controller delay is shown in Fig. 2.

In the double loop PID controller, the internal loop current control generally is adopted the proportional control method, and it is shown as:

$$u = K_c(i_m - i_c) \tag{15}$$

where, u is the control voltage, i_c is the measured suspension current, i_m is the target suspension current, and K_c is the gain coefficient of the inner loop.

In the outer loop control, the target current i_m is shown as:

$$i_m = K_p\Delta c + K_d\Delta\dot{c} + K_z\ddot{z} \tag{16}$$

where, K_p, K_d, K_z, Δc, $\Delta\dot{c}$ and $\ddot{z}$ are suspension gap feedback coefficient, suspension gap velocity feedback coefficient, suspension electromagnet acceleration feedback coefficient, suspension gap displacement change value, suspension gap

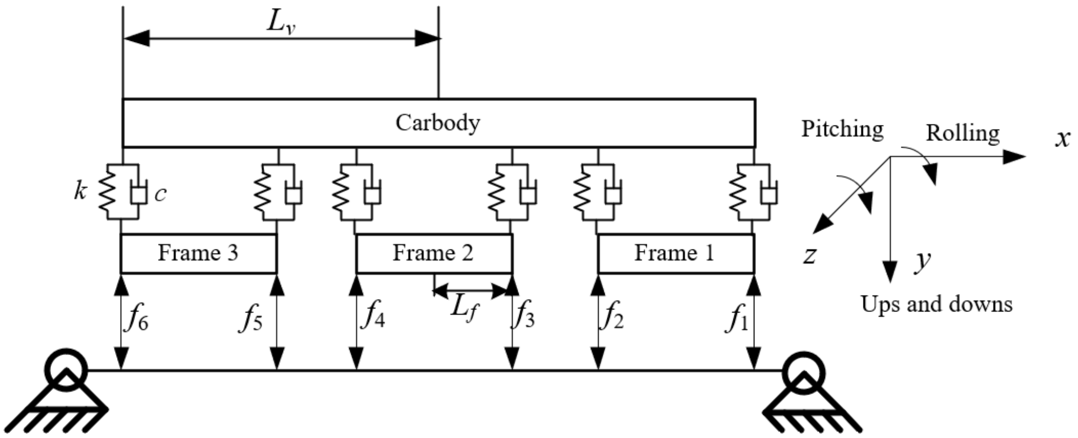

FIGURE 1. Maglev vehicle vehicle-guideway model.

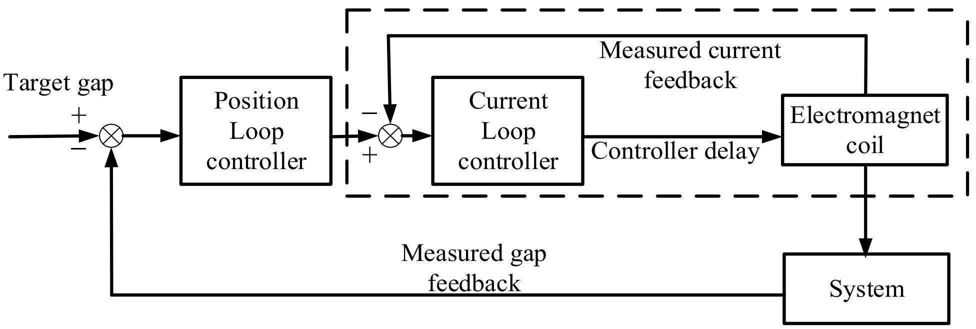

FIGURE 2. Double loop PID controller schematic diagram with controller delay.

velocity change value and suspension electromagnet acceleration change value, respectively(all of the change values are relative nominal suspension position).

Combined with the equation (1)-(16), Matlab/Simulink can be used to establish the corresponding mathematical model of maglev train dynamics. Some parameters of the system are shown in Tab. 1 below.

TABLE 1. System parameters.

Symbol	Quantity	Value
M	Mass	12000kg
m_1-m_3	Mass	2000kg
A_m	Electromagnetic pole area	0.021m^2
l	Track beam length	24m
K_p	Gap feedback coefficient	6000
K_d	Gap velocity feedback coefficient	50
K_z	Electromagnet acceleration feedback coefficient	0.5
K_c	Inner loop gain coefficient	80
k	Air spring stiffness	50kN.m^{-1}
ρ	Track beam line density	2400Kg.m^{-1}

A. INFLUENCE OF CONTROLLER DELAY ON DIVERGENCE SPEED OF THE VEHICLE AND THE TRACK BEAM

The impact of controller delay on the divergence speed of both the vehicle and the track beam will be examined in this section. By focusing on the suspension clearance corresponding to the central positions of suspension frames 1, 2, and 3, and considering the maximum deflection at the mid-span of the track beam, this study aims to provide a more comprehensive analysis of the dynamic characteristics at that position. The primary focus will be on the mid-span position of the track beam for this investigation. Assuming an 8 mm distance between lifting position of the vehicle and track beam and, we establish the following conditions:

Working Condition 1: The controllers of suspension frames 1, 2, and 3 demonstrate control delays of 0.002 s, 0.0003 s, and 0.0006 s, respectively.

Working Condition 2: The controllers of suspension frames 1, 2, and 3 demonstrate control delays of 0.003 s, 0.0003 s, and 0.0006 s, respectively.

The dynamic characteristics of the suspension clearance for suspension frames 1, 2, and 3 are illustrated in Fig. 3 below under Working Condition 1. Conversely, Fig. 4 below depicts the dynamic characteristics of the suspension clearance for suspension frames 1, 2, and 3 under Working Condition 2.

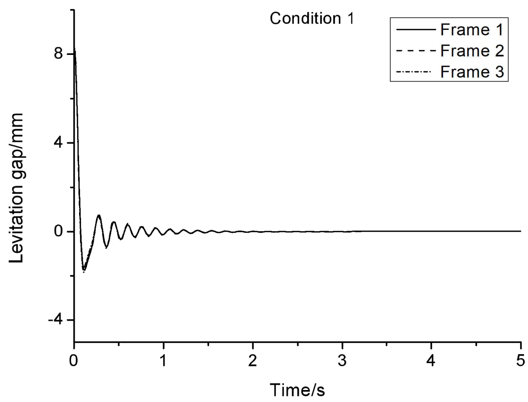

FIGURE 3. **Suspension gap in condition 1.**

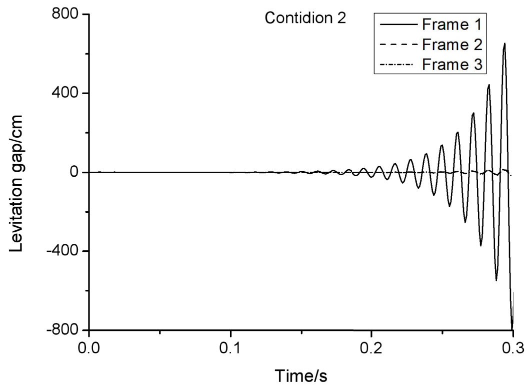

FIGURE 4. **Suspension gap in condition 2.**

According to the analysis presented in Fig. 3, the maglev train demonstrates stable suspension. However, when the controller delay of suspension frame 1 increases from 2 ms to 3 ms, the entire maglev system experiences a loss of stability as shown in Fig. 4. It can be inferred that within the range of 2 ms to 3 ms, certain controller delay points cause vehicle-guideway coupling vibrations in the maglev system. To further investigate this phenomenon, we will examine three specific scenarios: Working Condition 3 with a controller delay of suspension frame 1 at 2.144 ms, Working Condition 4 with a delay at 2.146 ms, and Working Condition 5 with a delay at 2.149 ms while keeping the delays of suspension frames 2 and 3 constant. The mid-span displacement of the track beam and central displacement of suspension frame are chosen as focal points for further analysis. Fig. 5-10 depict the dynamic characteristics under Working Conditions 3-5, respectively.

The mid-span displacement of the track beam, as shown in Fig. 5, 7, and 9, transitions from small amplitude oscillations to a gradual divergence with increasing controller delay. Moreover, this divergence becomes more pronounced as the delays become greater. In Fig. 5-8 analysis reveals that

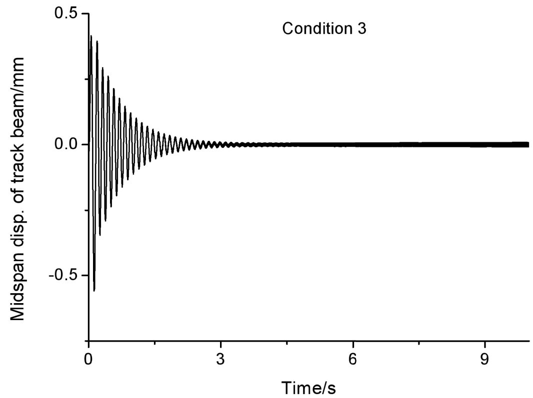

FIGURE 5. **Track beam displacement in condition 3.**

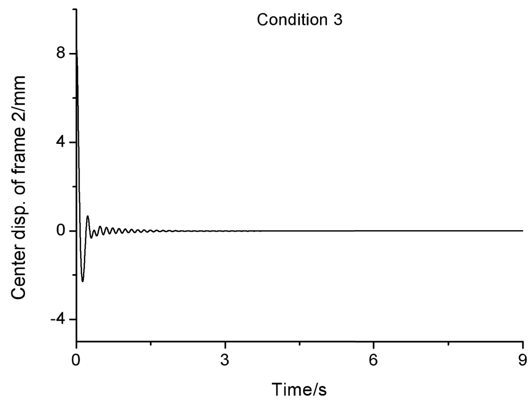

FIGURE 6. **Frame 2 displacement in condition 3.**

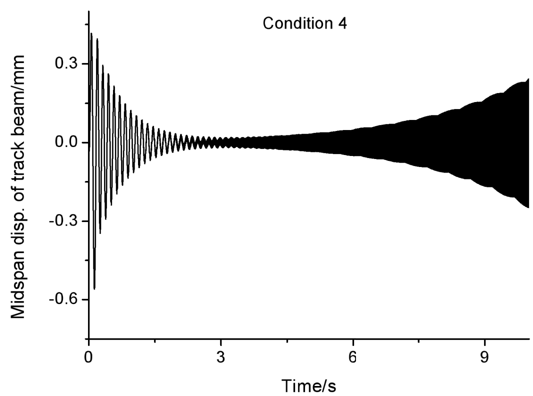

FIGURE 7. **Track beam displacement in condition 4.**

while the mid-span displacement of the track beam exhibits a diverging trend in Working Condition 3 and starts to diverge in Working Condition 4, there are no signs of divergence observed in the central displacement of suspension frame 2 throughout this period. Additionally, Fig. 9-10 depict that the mid-span displacement of the track beam initiates divergence at approximately 3.5 s; however, it is around 6.5 s when the central displacement of suspension frame begins to diverge in Working Condition 5.

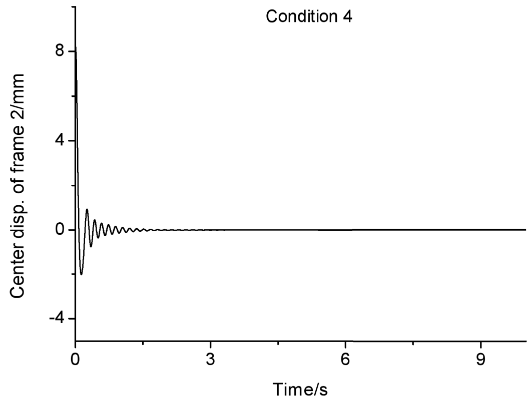

FIGURE 8. Frame 2 displacement in condition 4.

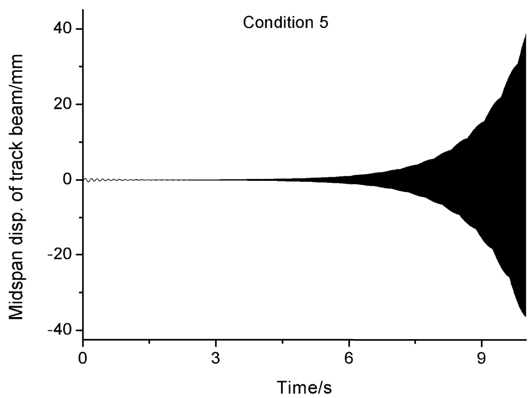

FIGURE 9. Track beam displacement in condition 5.

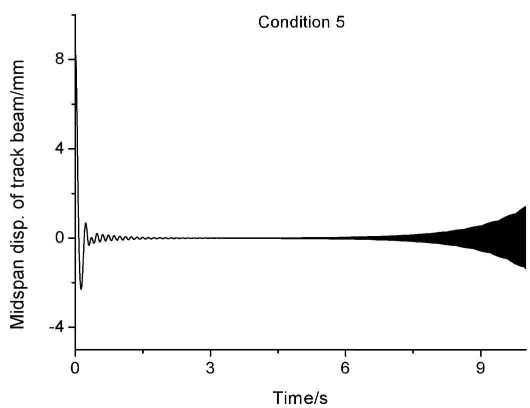

FIGURE 10. Frame 2 displacement in condition 5.

In conclusion, it is evident that the track beam is evidently more susceptible to vibrational disturbances at an earlier stage due to the suspension controller delay-induced vehicle-guideway coupling vibrations, compared to the vehicle system.

B. THE SUSPENSION CONTROL STRATEGY BASED ON REDUNDANCY CONTROL

Through the aforementioned correlation analysis, it becomes evident that the presence of control delay in the suspension controller can result in significant vehicle-guideway coupling vibrations within the maglev system under specific conditions. In the control process of the described maglev system, a centralized approach is adopted by utilizing a single suspension controller instead of employing two controllers to manage a specific suspension point. This control methodology has minimal impact on the dynamic response characteristics of the maglev system when there are no faults or delays in the vehicle's suspension controller. However, if such issues arise, they may severely compromise the dynamic characteristics of the maglev vehicle and lead to pronounced vehicle-guideway coupling vibrations as well as potential suspension instability. Therefore, the objective of this section is to investigate the impact of controller delay on suspension stability in a suspension system where a single suspension point is simultaneously controlled by two independent suspension controllers (the first part of this section focuses on the scenario where only one suspension controller independently controls a single suspension point). For analysis purposes, we will consider a scenario where there is a 2 mm deviation from equilibrium position for the suspension electromagnet under two working conditions:

Working Condition 6: The suspension controller delay for suspension frames 1, 2, and 3 is 0.007 s.

Working Condition 7: At each suspension point of suspension frames 1, 2, and 3, only one suspension controller experiences a time delay of 0.007 s, while the other suspension controller operates without any control delay.

The dynamic response characteristics of the suspension clearance at the central positions of suspension frames 1, 2, and 3, as well as the mid-span displacement of the track beam under Working Condition 6, are illustrated in Fig. 11 – 12, respectively. Meanwhile, Fig. 13-14 depict the dynamic characteristics of the suspension clearance at the central positions of suspension frames 1, 2, and 3 along with the mid-span displacement of the track beam under Working Condition 7.

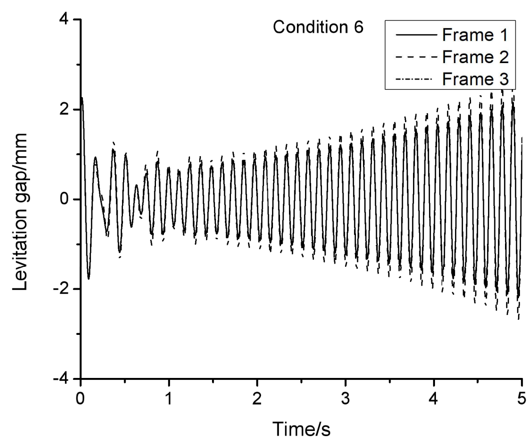

FIGURE 11. Suspension gap in condition 6.

Through the analysis presented in Fig. 11-12, it becomes evident that when the time delay of the controllers exceeds a certain threshold, even with a redundant control mode,

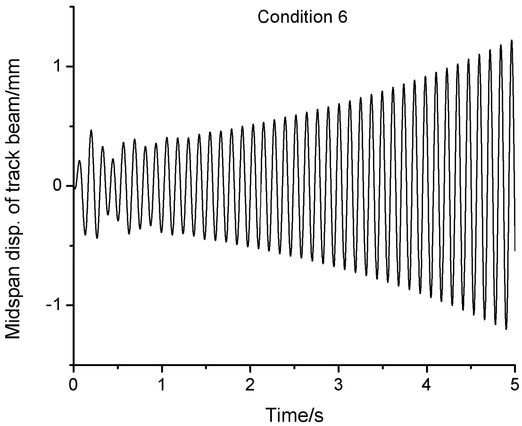

FIGURE 12. Track beam displacement in condition 6.

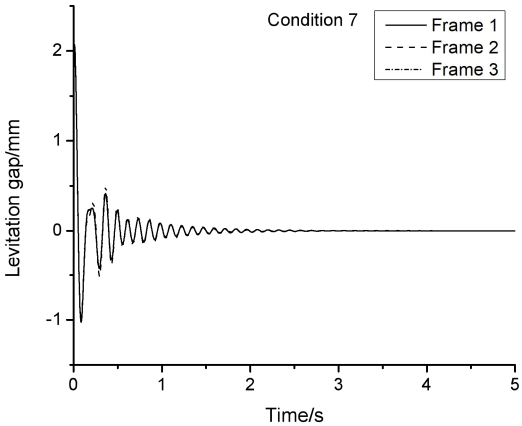

FIGURE 13. Suspension gap in condition 7.

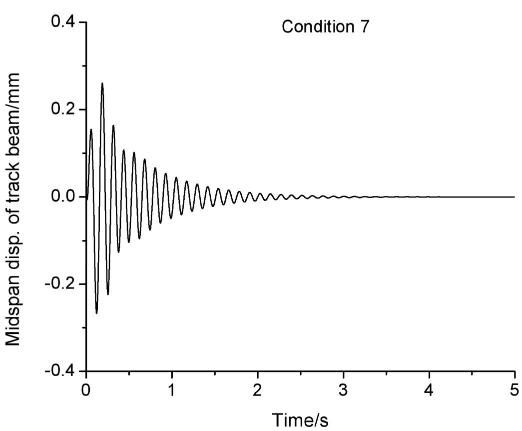

FIGURE 14. Track beam displacement in condition 7.

there is a divergence between the suspension gap and mid-span displacement of the track beam. This ultimately leads to instability in the maglev vehicle. Conversely, as indicated by the analysis in Fig. 13-14, if only one suspension controller experiences a control delay, it does not result in significant vibrations between the vehicle and guideway within the maglev system.

The impact of controller delay on the vibration response characteristics of the vehicle-guideway coupling is further investigated in this section, utilizing the methodologies outlined in references [21], [32], and [33]. Specifically, we focus on analyzing the work performed by the suspension system on the track beam. Since each suspension point is controlled independently, we can equate the static suspension stability of the maglev train to that of a single-iron system (the equivalent parameters of which can be derived from Tab. 1). Equation (17) below [21] presents the linear dynamic equation governing the single-iron suspension system at its nominal position.

$$\begin{cases} m_e\ddot{z} = -2P_i\Delta i + 2P_s(z - b_1) \\ K_c[K_p(z - b_1) + K_d(\dot{z} - \dot{b}_1) + K_z\ddot{z} - \Delta i] \\ = 2L\Delta\dot{i} - 2P_i(\dot{z} - \dot{b}_1) \end{cases} \tag{17}$$

In equation (17), m_e is the mass of equivalent suspended electromagnet. z is the vertical displacement of the suspended electromagnet from the equilibrium position. The pure delay link can be represented by the transfer function $G = e^{-\beta s}$, β is the time delay constant. According to the research results in literature [21], when the control system has a time lag, the dynamic characteristics of vehicle-guideway coupling vibration of the maglev system are mainly affected by the delay of the suspension gap channel. In order to simplify the problem, the effect of suspension clearance delay on the vehicle-guideway coupling vibration response of the maglev system is mainly considered. Therefore, the vibration velocity of the track beam is taken as the input and the suspension force acting on the track beam as the output. The transfer function of equation (17) can be expressed as: $H_{FV}(s) = F(s)/V(s)$.

Here:

$$F(s) = 2m_e s(P_s(K_c + 2Ls) - P_i(2P_i s + K_c(K_p G + K_d s))) \tag{18}$$

$$V(s) = 4P_i^2 s - 2P_s(K_c + 2Ls) + m_e s^2(K_c + 2Ls) + 2P_s K_c(K_p G + s(K_d + K_z s)) \tag{19}$$

Literature [34] pointed out the vibration of track beam can be expressed by sines and cosines of a certain amplitude. Therefore, at a certain moment, the vibration characteristic of the track beam is simulated by the sine function without phase difference, taking the vibration frequency as w and the vibration amplitude as 0.1m/s, i.e. $v(t) = 0.1\sin(wt)$. It can be known that the electromagnetic force acting between the track beam and the suspension system is shown in equation (20) below:

$$F(t) = 0.1\,|H_{FV}(jw)|\sin[wt + \angle H_{FV}(jw)] \tag{20}$$

Further analysis shows that the power of the electromagnetic levitation force acting on the elastic track beam in a time period is shown in equation (21) below [21], [32]:

$$P(w) = \frac{1}{T}\int_0^T F(\psi)V(\psi)d\psi = 0.005Re[H_{FV}(jw)] \tag{21}$$

When the nonlinear characteristics of modal damping are ignored, the absorbed power of track beam damping on the bridge is calculated as $P_b = 0.01\xi w m_1$ [21], ξ is the first-order damping ratio of the track beam, and the value is 0.002. After considering the energy consumed by the track beam damping, the average power of the suspension system acting on the track beam is analyzed as shown in Fig. 15 below, when the controller delay is 6ms and 8ms, respectively.

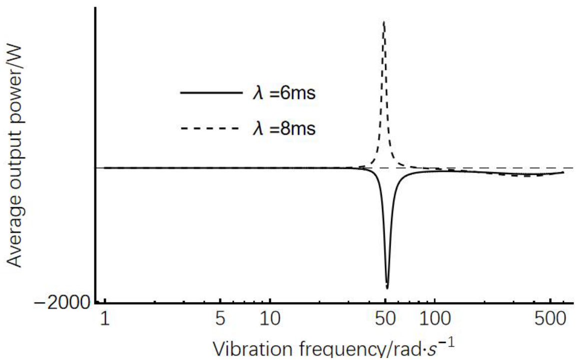

FIGURE 15. Average output power.

According to the analysis presented in Fig. 15, when the controller delay is set at 8 ms and the vibration frequency of the track beam ranges from approximately 35 rad/s to 70 rad/s, it exceeds the energy dissipated by damping of the track beam, resulting in continuous energy output from the suspension system to the track beam. This scenario can lead to significant vehicle-guideway coupling vibrations within the maglev system. Conversely, reducing the controller delay to 6 ms and exceeding a vibration frequency of approximately 35 rad/s allows for negative work performed by the suspension system on the track beam, effectively suppressing vehicle-guideway coupling vibrations in the maglev system. The bending stiffness of the track beam is approximately 19.39×10^9 N/m. Referring to Tab. 1, we can calculate that its first-order modal vibration frequency is around 48.32 rad/s. Therefore, inducing vehicle-guideway coupling vibrations becomes challenging with a controller delay of approximately 6 ms in contrast to readily facilitating severe vibrations with a delay of about 8 ms. Further analysis reveals that as controller delay increases from approximately 6 ms to 8 ms, there is a transition from negative work performed by suspension system on track beam towards positive work indicating an existence of critical delay point likely triggering vehicle-guideway coupling vibrations within range of delays between 6 ms and 8ms.

The further analysis of Fig. 15 reveals that even with a 7ms delay in the suspension controller, the stable suspension of the system can still be maintained if the first-order vibration frequency of the track beam is significantly different from the frequency range of positive energy output to the track beam. In this case, we can set the first-order vibration frequency of the track beam as 20 rad/s downward and 90 rad/s upward. The dynamic response characteristics of the suspension gap at the mid-span position are illustrated in Fig. 16 below. The system can also achieve stable suspension by adjusting the parameters of the suspension controller without altering the system parameters. In this case, a time delay of 7ms is set for the controller, and K_d value is adjusted from 50 to 90. The dynamic response characteristics of the suspension gap are illustrated in Fig. 17 below, while Fig. 18 presents the average power exerted on the track beam by the suspension system.

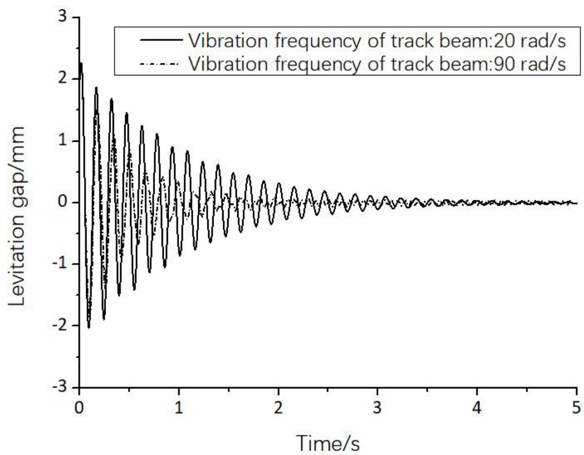

FIGURE 16. Suspension gap in the different vibration frequency.

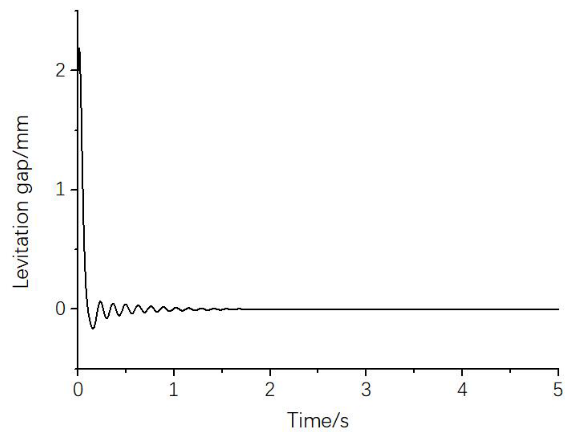

FIGURE 17. Suspension gap when adjusting controller parameters.

It can be seen from the analysis in Fig. 16 that when controller parameters are unchanged, the first-order vibration frequency of the track beam is far away from the frequency range of the suspension system to the normal operation of the track beam by adjusting the relevant parameters of the track beam, and the stable suspension of the system can be achieved when the time delay of the controller is 7ms (the suspension will be unstable in working condition 6).

According to the analysis in Fig. 17, under unchanged system parameters and adjusted controller parameters, a stable suspension of the system can be achieved when the controller

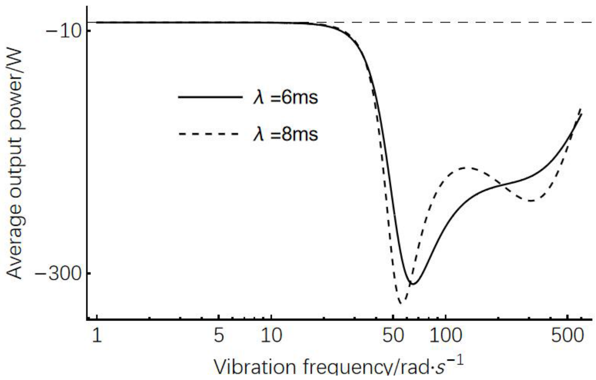

FIGURE 18. Average output power when adjusting controller parameters.

delay is set at precisely 7ms (the suspension will become unstable in working condition 6). The analysis of Fig. 18 reveals that when the relevant parameters of the controller are modified, despite no adjustment being made to the first-order vibration frequency of the track beam, the suspension system effectively dissipates negative energy to the track beam within a circular frequency range of 35 rad/s to 70 rad/s. This capability allows for absorption of vibration energy from the track beam and thereby ensures stability in the suspension system (as evident from Fig. 15, under original controller parameters, system suspension becomes unstable).

IV. CONCLUSION

The preceding analysis leads to the following conclusions to a significant extent:

1) The track beam demonstrates a higher sensitivity to the suspension controller compared to the vehicle system, making it more susceptible to divergent vibrations. Consequently, optimizing the structural parameters of the track beam, rather than focusing on those of the vehicle, can potentially mitigate the coupling vibrations between the vehicle and guideway within the maglev system, assuming that the suspension control system parameters remain constant.
2) When the first-order vibration frequency of the track beam falls within a specific range and the controller's time delay reaches a critical threshold, there is a high likelihood that a significant amount of energy will be transmitted to the track beam by the suspension system. This phenomenon can result in pronounced vehicle-guideway coupling vibrations in maglev trains, potentially leading to suspension instability. Therefore, it is crucial to minimize control delay or increase damping of the track beam while employing dynamic absorbers to effectively absorb track vibration energy.
3) The coupling vibrations between the vehicle and guideway of the maglev train, caused by controller delay, can be effectively mitigated through the implementation of a redundancy control module. When the redundant control mode can not maintain the coupling vibration caused by the delay of the system, the system can be stabilized by adjusting the first-order vibration frequency of the rail beam or adjusting the relevant parameters of the suspension controller.

The dynamic characteristics of the vehicle suspension system and the single-iron suspension system exhibit certain disparities. These include the complexity of the system (resulting from dynamic coupling among its components, thereby forming a more intricate dynamic system), leading to distinct dynamic responses and vibration characteristics compared to single-iron systems. Moreover, the vehicle suspension system demonstrates enhanced load distribution efficiency, which significantly influences overall dynamic performance and stability. Conversely, single-iron suspension systems may result in heightened local stress concentration. Consequently, future studies should delve deeper into exploring these dynamic differences between the two systems in order to gain a comprehensive understanding of their respective performances.

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KEREN WANG received the B.S. and M.S. degrees from Xihua University, Chengdu, China, in 2011 and 2014, respectively, and the Ph.D. degree from Southwest Jiaotong University, Chengdu, in 2019. He is currently working as a Postdoctoral Researcher with Southwest Jiaotong University. His research interests include machine learning, artificial intelligence, vehicle system dynamics, and train-bridge coupled vibration.

WEIHUA MA received the B.S. degree in vehicle engineering from Shandong University, Jinan, China, in 2002, and the Ph.D. degree in vehicle engineering from Southwest Jiaotong University, Chengdu, China, in 2008. Subsequently, he successfully completed a Postdoctoral Fellowship at Southwest Jiaotong University and CRRC Qishuyan Company Ltd., in 2015. He is currently working with the State Key Laboratory of Traction Power, Southwest Jiaotong University. His research interests include locomotive and heavy haul train dynamics, maglev levitation frame design, and vehicle dynamics.

XIAOHAO CHEN received the B.S. degree in mechanical engineering and the Ph.D. degree in vehicle engineering from Southwest Jiaotong University, Chengdu, China, in 2015 and 2022, respectively. He is currently working as a Researcher with Zhejiang Normal University. His research interests include electromagnetic suspension and linear propulsion systems.

RUIMING ZOU received the Ph.D. degree in vehicle engineering from Southwest Jiaotong University, Chengdu, China, in 2019. He is currently working as a Lecturer with the Civil Aviation Flight University of China. His research interests include mechanical system dynamics and risk assessment in transportation systems.