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Actuator optimal placement studies of high-speed power bogie for active hunting stability

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ABSTRACT

We put forward three actuator placements of the high-speed train power bogie to improve the train hunting stability. The active control forces act on the frame, between the frame and the motor, and on the motor by the inertial or retractable actuator, respectively, based on the feedback states of vibration velocity of the front and rear end beams. The feedback gains and the motor suspension parameters in different cases are optimised with the two objectives of system stability margin and control effort. The required actuator outputs of the three cases are compared based on the theoretical analysis with a 8 DOF bogie model. The results show that the three control cases can effectively improve the hunting stability, especially at high speed. The active control of motor lateral movement is helpful to increase the dynamic vibration absorbing function of the motor flexible suspension, and the control output is obviously smaller than the other two control cases. In addition, the influence of system delay on stability was analysed and we could use or avoid the effects of delay on the stability.

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High-speed train bogie; hunting stability; active control; time delay; multi-objective optimisation

1. Introduction

The active controls used in railway vehicles have been receiving much attention. Most of today's researchers are concerned with railway vehicle ride comfort, the tilting train control and the secondary suspension semi-active or active control in order to enhance the vehicle lateral and vertical smoothness, and to reduce high-frequency elastic vibration of the car body [1,2]. With the continuous improvement of the vehicle running speed, more and more scholars pay attention to improve the hunting stability by unconventional suspension or active control. The lateral stability of train employing inerters suspension have been investigated and the inerters are deemed effective in improving the lateral stability [3–5]. Active stability control used in the bogie system can be applied either directly or indirectly [6,7]. The direct control involves the application of an actuator onto a solid wheelset or onto an axle with independently rotating wheels. A practical compromise is to apply a yaw

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torque indirectly at the secondary suspension level, a concept called 'Secondary Yaw Control' (SYC). The actuators may be designed to replace the traditional passive yaw dampers so that active control of the running gear can be introduced without a substantial redesign of the bogie, although this means that radical simplifications of the vehicle design may be limited [8].

With the development of high-speed railway, the active stability is not only used for the newly made railway vehicles for compromising the conflict between the hunting stability and the curving performance [9–11]. With the increase in service life and mileage, the vehicle structure parameters and suspension performances may change due to the tread wear and component failure [12] causing the reduction of hunting stability. Loss of lateral stability results in poor ride comfort, huge wheel/rail action force or even derail accident. Active controls in high-speed trains are great alternatives of passive methods because of their flexibility, which can compensate for the variation of the parameters of the vehicle and guarantee the running performances. There are a lot of literature on using active control onto the solid wheelset or independently rotating wheel to improve the stability of the bogie or wheelset; meanwhile, the control schemes also improve the curving performance [13–15]. However, the application of active control onto the wheelset directly to improve the running stability still has many challenges in practice.

The drive system or motor is suspended flexibly under the bogie frame or under the car body through the swing rods or other elastic. This approach can improve the lateral dynamics performances of locomotives or motorised cars at high running speeds. It has long been understood that the mechanism of flexible suspension of the drive system combined with the elastic connection can decouple the mass and inertia between the drive system and bogie frame. This in turn isolates the lateral movement of the drive system from the bogie frame [16]. This qualitative explanation is not sufficient because the flexible suspension parameters cannot be given out accurately. Alfi et al. [17] analysed the effects of motor connection on the critical speed of high-speed railway vehicles, which shows that the motor mass may act as a dynamic absorber, affecting bogie hunting in a way that improves vehicle stability. The author [18] thought that the mechanism of motor flexible suspension can be considered as mass decoupling to improve the vehicle lateral performances in high-speed locomotive, where the motor suspension damping is relatively large. When the motor suspension damping ratio is about 0.2–0.4 such as in high-speed EMU, the function of motor flexible suspension can be considered as dynamic vibration absorber (DAV). The appropriate motor suspension parameters can be well selected by the theory of DAV.

The active mass damping [19] or tuned vibration absorbers [20] have been applied in civil engineering structure widely, which inspires us to adopt the motor active flexible suspension to improve the vehicle lateral dynamics. The authors [21] found that the active control of the lateral movement of the suspended drive system using SIMPACK can significantly improve the lateral dynamics performance of the locomotive, of which the maximum lateral forces experienced by a wheelset under active control are reduced by up to 25%. We continue some theoretical researches with a simplified bogie model. This method controlling the motor's and the frame's lateral movement is feasible and low risk in practical application.

In the present study, the authors presented an indirect method for improving the bogie hunting stability, by means of controlling the frame lateral vibration [22]. For the high-speed train bogie with the motor flexible suspension, there are three kinds of control configurations to realise the frame lateral vibration control. Previous studies show that the control is effective in improving the hunting stability. Meanwhile, we continue to research the frame lateral vibration control, such as the vehicle dynamics simulation based on commercial software, and even the rolling rig test. In this study, the maximum active control force and energy under the same stability index for the three control configurations are compared. To consider the inherited confliction between the system relative stability and control energy, a multi-objective optimisation is used in the control design. Also, the linear stability of the bogie control system at different running speeds for each control configuration is investigated. Furthermore, the effects of time-delay in the control system on the bogie system performances are discussed.

2. The lateral vibration control of the bogie frame

2.1. The bogie linear dynamics model

The active mass damping [19] or tuned vibration absorbers [20] have been applied in civil engineering structure widely. It is considered as the added mass that provides the inertial reaction force to control the vibration of main structure. The vibration damping of the main structure increases if the vibration velocity of the main structure is fed back. For the power bogie in high-speed train, the lateral flexible suspension motor can be used as the added mass naturally to achieve the active mass damping. In this study, we propose the frame lateral vibration suppression by the active control to further improve the vehicle lateral dynamics performances. A simplified bogie model with hunting freedom is shown in Figure 1. The model describes the lateral movements of a bogie. We assume that the lateral position of the car body is pre-determined. The bogie frame is assumed to be connected with the fixed coordinate system through the secondary suspension, such as stiffness and damping in the longitudinal and lateral directions. The bogie has two solid axle wheelsets which are attached to the bogie frame via suspension springs in the longitudinal and lateral directions. The motor is connected to the frame by the elastic and damping elements. The frame and the wheelset in the model have lateral and yaw degrees of freedom, and the motor has only lateral degree of freedom. In fact, due to the gear transmission box or other devices amounted onto the axle, the non-coincidence between the wheelset's mass centre and geometric centre are ignored in this simplified model for theoretical researches.

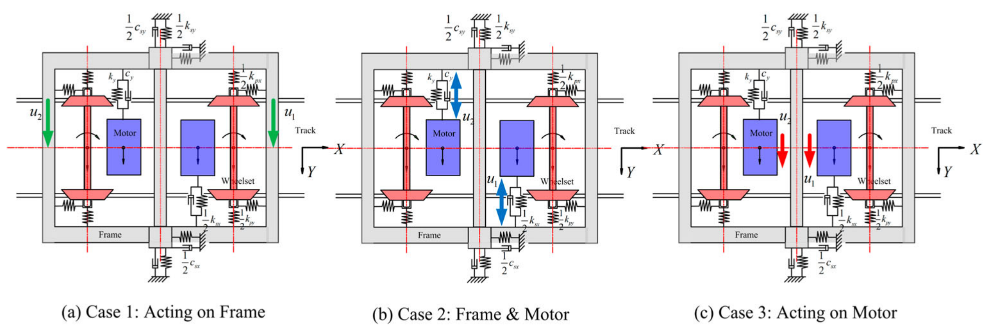

Figure 1. Dynamics model of the bogie with three frame lateral vibration control configurations.

The dynamics model was found to have 5 rigid bodies and 8 degrees of freedom. The model describes the lateral movements of the bogie and reads

$$\mathbf{M}\ddot{\mathbf{x}} + C\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{Q} + \mathbf{E}\mathbf{u}, \tag{1}$$

where $\mathbf{x} = [y_{w1}, \varphi_{w1}, y_{w2}, \varphi_{w2}, y_f, \varphi_f, y_{m1}, y_{m2}]$. The states $y_{w1}, y_{w2}, y_f, y_{m1}$, and y_{m2} are the lateral displacements of wheelset 1, wheelset 2, frame, motor 1 and motor 2, respectively. While $\varphi_{w1}, \varphi_{w2}$, and φ_f represent the yaw angles of wheelset 1, wheelset 2, and frame. The matrices **M**, **C**, and **K**, are the mass, damping and stiffness components of the system. The matrix **E** is the control matrix and **u** is the control effort vector. These matrices and vectors are listed in the appendix. The values and definitions of system parameters are listed in Table A1 [22,23]. The matrix **Q** is the external force vector, such as the wheel-rail forces caused by the track irregularity and the centrifugal force in a curve. At first, we focus on the bogie linear system and its stability, so the **Q** is a zero vector.

In order to better understand the motor flexible suspension mechanism, the expression of lateral stiffness k_{my} and damping c_{my} of the motor suspension are defined as follows:

$$\begin{aligned} k_{my} &= (2\pi f_{my})^2 m_m, \\ c_{my} &= 2\xi_{my} m_m (2\pi f_{my}), \end{aligned} \tag{2}$$

where f_{my} is the motor lateral suspension natural frequency in Hz and it is also called as the motor suspension frequency. ξ_{my} is the motor lateral suspension damping ratio. m_m is the mass of motor in kg.

2.2. The state equation

As shown in Figure 1, we propose three active control cases for the frame lateral vibration suppression. For case 1, two inertial actuators are attached on the front and rear end beams of the frame to exert the active control forces. For case 2, two retractable actuators act between the motor and the side beam of the frame to achieve the frame vibration control. For case 3, two inertial actuators are attached on the motors and the active control forces exert on the front and rear motors transversely. The active damping of the main structure requires the negative feedback of vibration velocity of the main structure [19]. In this study, the vibration velocity of the front and rear end beams of the frame are fed back to calculate the output force of the front and rear actuators. The symbols u_1 and u_2 are the front and rear actuator outputs. The **E** is their vector describing the influence of the control on the system dynamics. For the three cases, the corresponding $\mathbf{E_1}$, $\mathbf{E_2}$, $\mathbf{E_3}$ are different. These matrices and vectors are listed in the Appendix. In order to highlight the effect of the active control on improving the bogie hunting stability, the damping of yaw damper is one fourth of the design value and the analysis of bogie fault state is conducted.

The state equation of the bogie control system can be written as

$$\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}\mathbf{u}(t) + \mathbf{G}\mathbf{d}(t), \tag{3}$$

where $\mathbf{d}(t)$ is the external disturbance such as track irregularity applied on the wheelsets or other external excitation. The state vector is defined as

$$\mathbf{x} = [y_{w_1}, \varphi_{w_1}, y_{w_2}, \varphi_{w_2}, y_f, \varphi_f, y_{m_1}, y_{m_2},$$

$$\dot{y}_{w_1}, \dot{\varphi}_{w_1}, \dot{y}_{w_2}, \dot{\varphi}_{w_2}, \dot{y}_f, \dot{\varphi}_f, \dot{y}_{m_1}, \dot{y}_{m_2}]^{\mathrm{T}}. \tag{4}$$

The system matrices **A**, and **B** and the vector **G** are given by

$$\begin{aligned} \mathbf{A} &= \begin{bmatrix} 0_{8\times 8} & \mathbf{I}_{8\times 8} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C} \end{bmatrix}, \\ \mathbf{B} &= \begin{bmatrix} 0_{8\times 2} \\ \mathbf{M}^{-1}\mathbf{E} \end{bmatrix}, \quad \mathbf{G} = \begin{bmatrix} 0_{8\times 1} \\ \mathbf{M}^{-1}\mathbf{W} \end{bmatrix}, \end{aligned} \tag{5}$$

where the vector **W** is defined as

$$\mathbf{W} = [k_{py}, 0, k_{py}, 0, 0, 0, 0, 0]^{\mathrm{T}}. \tag{6}$$

It can be shown that the whole system is not completely controllable and observable. In order to control the frame vibration completely, we consider the output state feedback control applied onto a subsystem, which consists of a frame and two motors. The bogie primary and secondary suspension loads can be regarded as external interference to the subsystem. The subsystem is completely controllable.

We consider the output state linear negative feedback control $\mathbf{u} = -\mathbf{K}_c \cdot \mathbf{y}(t)$. Where $\mathbf{K}_c$ is the feedback gain matrix and $\mathbf{y}$ is the output state. In order to dampen the lateral and yaw vibration of the frame, the lateral vibration velocity of the front and rear end beam of frames are fed back. For the three cases, $\mathbf{y}$ is the same and it can be expressed as follows:

$$\mathbf{y} = \left[\dot{y}_f + l_e \times \dot{\varphi}_f,\ \dot{y}_f - l_e \times \dot{\varphi}_f\right]^{\mathrm{T}} \tag{7}$$

Using this control law, the closed-loop dynamic matrix of the system reads

$$\mathbf{A}_{cl} = \mathbf{A} - \mathbf{B}\,[\mathbf{K}_c \ \ 0_{2\times 14}] \tag{8}$$

We can try to adjust some feedback gain coefficients to stabilise the closed-loop system $\mathbf{A}_{cl}$. Nevertheless, the stability margin and energy efficiency of the controller should be also taken into account in the design. The reasonable trade-off can be provided by the multi-objective optimal method. These requirements will be set up and defined in the Section 3.

2.3. The mechanism of frame lateral vibration control

The mechanism of the frame lateral control for improving the bogie hunting stability can be analysed by the root locus, which are the eigenvalues corresponding to different running speeds. First, we can analyse the mechanism of the motor flexible suspension. The bogie hunting root locus with different motor suspension parameters is shown in Figure 2. Each root locus is composed of 30 sets of eigenvalues in the speed range of $20 - 600\,\mathrm{km/h}$, and each symbol indicates a corresponding mode. The larger symbol represents a larger running speed. The horizontal axis indicates the mode damping ratio ζ (the negative sign indicates a stable value), which is the ratio of the real part to the modulus of eigenvalues of the matrix $\mathbf{A}_{cl}$ defined in Equation (8). The vertical axis indicates the damped vibration frequency of the mode, which is the imaginary part of eigenvalues. Normally, as the speed increases, the hunting-related mode damping reduces and the hunting frequency increases.

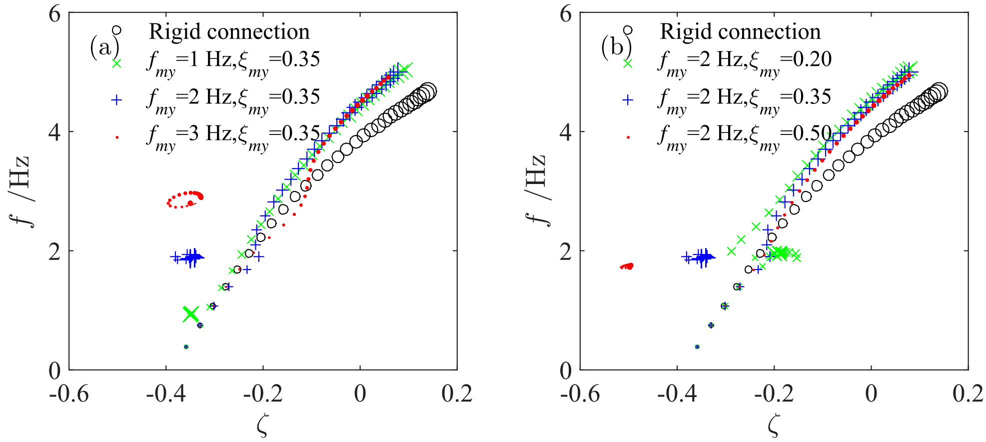

Figure 2. The motor flexible suspension mechanism analysis. (a) The bogie hunting root locus with different motor suspension frequencies and (b) the hunting root locus with different motor suspension damping ratios.

When the ζ is greater than 0, the hunting movement is unstable. Mostly, the low frequency hunting mode determines the bogie hunting critical speed.

The corresponding bogie low frequency hunting mode with different motor suspension frequencies and damping ratios are shown in Figure 2(a,b), respectively. The flexible suspension of the motor makes the sprung mass and yaw inertia decoupling from the motor mass, which leads to the significant improvements of the bogie hunting stability compared to the motor rigid suspension. As shown in Figure 2(b), the minimum ζ is up to -0.3 when the ξ_{my} is set to 0.2. When the motor suspension damping ratio is small, the dynamic vibration absorption appears, which will lead to the suppression of frame lateral vibration. It is the best for the bogie hunting stability at the corresponding antiresonance frequency, while the system stability decreases when the hunting frequency is slightly smaller than the antiresonance frequency [18]. When the train running speed exceeds the speed corresponding to the antiresonance, the stability of the motor flexible suspension system is better than that of the rigid suspension. It is conducive to improving the bogie hunting stability within a wide speed range by adopting the proper motor suspension damping. The corresponding bogie hunting stability decreases when the motor suspension damping ratio is 0.35, as shown in Figure 2(a). The motor suspension frequency ought to be smaller than the bogie hunting frequency corresponding to the usual speed range. In Figure 2(a), the stability of the system with f_{my} of 3 Hz is better than that of the system with f_{my} of 1 and 2 Hz at the speed of 600 km/h. The bogie hunting stability increases as the motor suspension frequency increases at high-speed range. So it is conducive to improving the bogie hunting stability at high speeds by adopting a larger motor suspension stiffness.

The bogie hunting root locus in different control cases is shown in Figure 3. Compared to the uncontrolled system, the bogie hunting stability especially at high speeds can be improved by the three control cases. The bogie hunting root locus for the case 1 is similar to the uncontrolled. The influence of running speed on the motor vibration mode damping ratio is not distinct. In Figure 3(c), the stability of the system is improved obviously when k_d is increased to 1e4. Therefore, if the active control force is large enough, the bogie

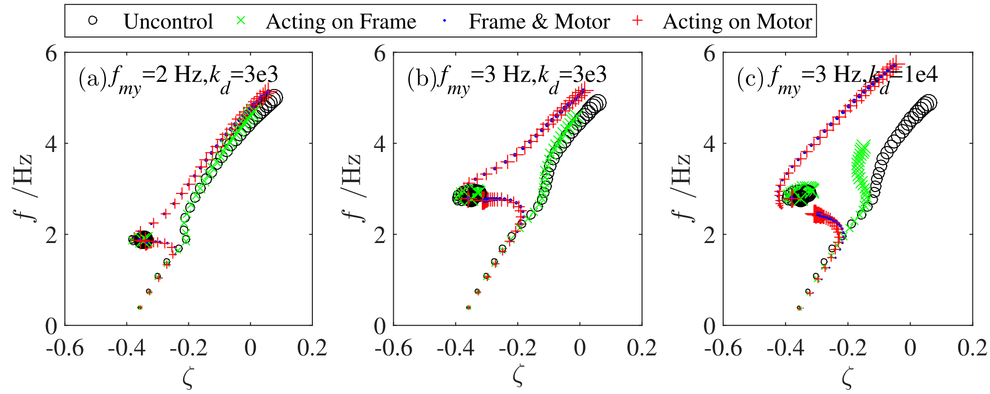

Figure 3. The bogie hunting root locus in different control cases.

hunting stability does not decrease as the running speed increases, and the hunting frequency significantly decreases in the high-speed range. For the cases 2 and 3, the shape of the bogie hunting root locus is consistent. The mode of motor vibration is coupled with the bogie hunting mode, and the root locus are two connected curves. According to the root locus of the two cases, the system stability corresponding to frequencies of about 2 or 3 Hz improves, and it increases with the increase of k_d. The bogie hunting stability corresponding to the frequency close to the motor suspension frequency improves significantly. The active control of the frame lateral vibration significantly increases the effects of dynamic vibration absorption caused by the motor flexible suspension, which leads to an increase of bogie hunting stability.

3. The multi-objective optimal control

3.1. Multi-objective optimal control design

In this study, we take into account the lateral velocity of the front and rear end beams of frame as the feedback states. According to the previous researches, the optimal feedback gains of the front and rear beams are basically the same. In this study, the feedback gains of the two states are assumed to be the same, which are defined as k_d. The suspension frequency f_{my} is used instead of the motor suspension stiffness. The calculation range of f_{my} is from 0 to 10 Hz and the range of the motor suspension damping ratio ξ_{my} (the positive sign indicates a stable value) is 0 to 1. We consider the multi-objective optimisation based on the genetic algorithm to optimise the gain k_d, as well as the motor suspension parameters f_{my}, ξ_{my}. So, k_d, f_{my} and ξ_{my} are the design parameters in each control case for the system discussed in Section 2. In the control system, the system stability margin and energy efficiency of the controller are the objectives of the control design. The multi-objective optimal design problem is stated as

$$\min\{\zeta_{\max}, |k_d|\}, \tag{9}$$

where $\zeta_{\max}$ denotes the maximum ratio of the real part to the modulus of eigenvalues of the matrix $\mathbf{A}_{cl}$ defined in Equation (8), and it describes the stability margin of linear systems. The term $|k_d|$ denotes the absolute value of the control gain k_d. In this study, for fast calculation, the eigenvalues of linear systems are calculated to analyse linear stability, and the influence of irregularities on nonlinear stability cannot be considered. For the stability analysis of linear systems, the control effort cannot be included directly in the objective space, and the feedback gain's absolute value is used instead. By minimising this value, the maximum control force and the maximum active control force are also minimised. To solve this MOOP, the multi-objective optimisation algorithm of NSGA-II is used [24].

3.2. Multi-objective optimal control results

The optimisation of the feedback coefficient k_d and the motor suspension parameters f_{my}, ξ_{my}, are conducted with the method of multi-objective optimisation for the three control cases. The active control force (replaced by $|k_d|$ for simplification) and the system stability $\zeta_{\max}$ are taken as the optimisation objectives. The pareto front for the three control cases at three running speeds is shown in Figure 4. The pareto front illustrates a conflicting relationship between $\zeta_{\max}$ and $|k_d|$. With the increase of control effort, the bogie is more stable. These results highlight the importance of the MOOP design. By having different design options, the decision-maker can choose any point from the solution set that meets the performance requirements. The results show that the system stability can be significantly improved within a certain range by increasing the feedback coefficient for the three control cases. In the same stability margin, the maximum control effort of case 3 is the minimum and case 1 is the maximum. At the running speed of 200 km/h, the uncontrolled system is stable, and its stability can be significantly improved by adopting control cases 1 and 3, while case 2 has little effect. That is to say, the active control forces exerting between the two motors and the frame with a same feedback coefficient have little effect on the system stability when the uncontrolled system is stable. For the running speed of 350 and 500 km/h, the three control cases can improve the system stability effectively. Besides, the system stability achieved for case 1 is the best and case 2 is the worst.

The pareto front also shows that system stability can be improved without active control by optimising the motor suspension parameters. For example, the corresponding $\zeta_{\max}$ for the uncontrolled system is 0 and the system is in the critical stable state at the speed of 350 km/h. However, the $\zeta_{\max}$ of the system without active control decreases to −0.1 after

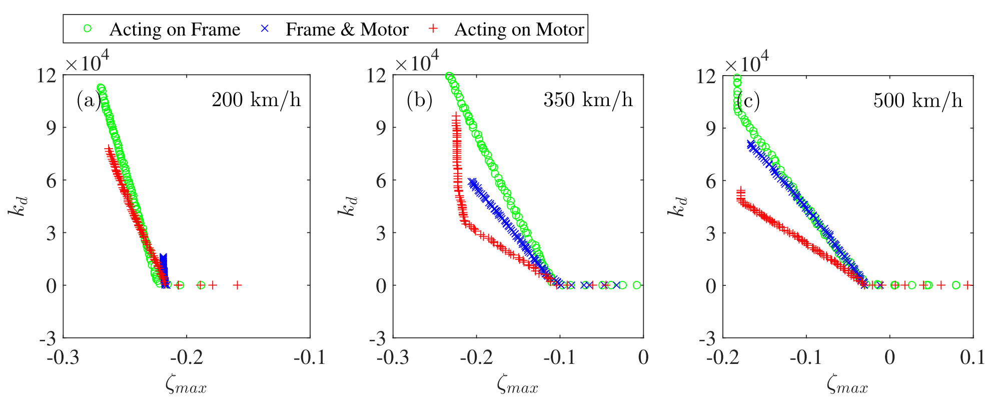

Figure 4. The Pareto front of control parameters optimisation in different control cases.

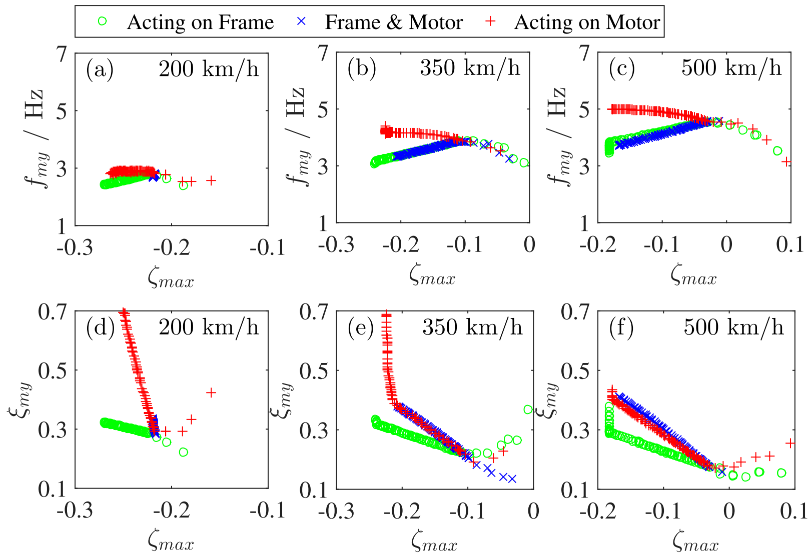

Figure 5. The Pareto set of control parameters optimisation in different control cases. (a)–(c) Motor suspension stiffness optimisation and (d)–(f) motor suspension damping optimisation.

optimising the motor suspension parameters, and it can reach up to −0.22 by adopting any one of the proposed active control cases.

3.3. The motor suspension parameter analysis

The motor flexible suspension parameters have an important influence on the stability of the bogie control system. The motor flexible suspension parameters that can achieve best system stability no matter how much control effort are called optimal motor suspension parameters, which can be calculated by using the genetic algorithm [24]. The optimal motor suspension parameters f_{my}, ξ_{my}, are shown in Figure 5. The bogie hunting frequency, as well as the optimal f_{my} increases as the running speed increases, which verifies the function of dynamic vibration absorption caused by motor flexible suspension. In the condition of no or low control efforts, the corresponding optimal f_{my} at the three speeds are about 3, 4 and 5 Hz, and the optimal ξ_{my} are about 0.3, 0.2 and 0.15, respectively. In the case of active control, the optimal f_{my} for cases 1 and 2 decreases slightly with the increase in stability, while it increases slightly for case 3. The optimal f_{my} for case 3 increases by almost 1,Hz than cases 1 and 2 at high speeds. The optimal ξ_{my} increases as the bogie hunting stability increases for the three control cases, which is more obvious for cases 2 and 3.

4. Comparison of the actuator outputs

The comparison of active control force for the three control cases in the low frequency domain is conducted. The bogie model adopting the three control cases are analysed by time domain integration to further compare the actuator outputs and energy consumption. The time domain curves of active control force are obtained by simulation. Taking into account the actual situation, the motor suspension parameters are set according to

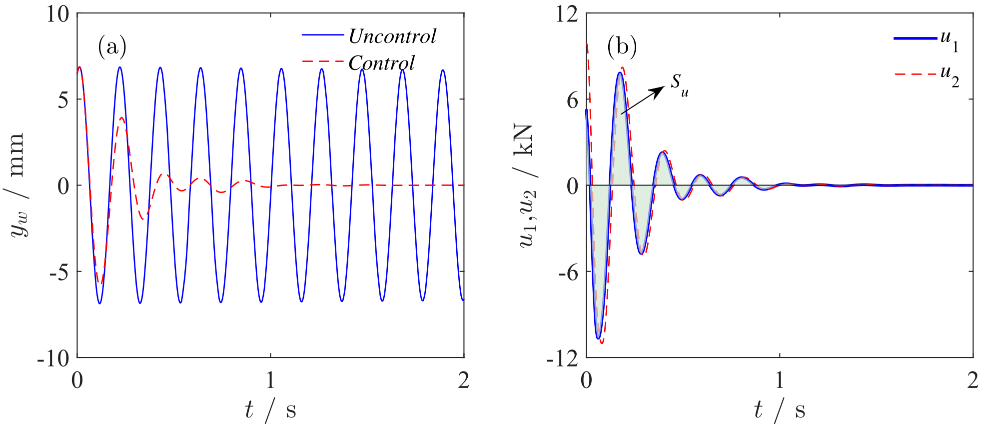

Figure 6. The bogie model simulation results in time domain. (a) The wheelset lateral displacement before and after the frame lateral vibration control and (b) the actuator force $u_{\max}$ and control energy S_u.

Figure 5(b). The f_{my} is set to 3Hz for cases 1 and 2, while it is set to 4 Hz for case 3. The ξ_{my} for the three control cases is set to 0.3. The initial lateral displacement of the guide wheelset is set to 7 mm, and the bogie passes through a smooth straight track. The running speed is 350 km/h, at which the uncontrolled bogie system is in the critical stable state. The wheelset lateral displacements before and after the frame lateral vibration control are shown in Figure 6(a). It can be concluded that the bogie lateral stability is significantly improved by active control. The time domain curves of the front and rear actuators outputs are shown in Figure 6(b). The area enclosed by the time domain curve and the axis is represented by S_u (unit: kN s). Although it is not strict, in a sense, the S_u can be taken as the index of the required energy for the control system.

The required maximum control effort $u_{\max}$ as well as the approximate energy index S_u versus the system stability index $\zeta_{\max}$ corresponding to different feedback coefficient k_d for the three control cases are shown in Figure 7. The variation range of k_d is 0 to 200,000. The bigger the symbol in curves, the larger the corresponding k_d. It can be concluded that the system stability decreases when the k_d increases to a certain value. Increasing k_d has a negative effect on system stability for case 2 at a speed of 200 km/h, while a larger k_d is required to improve the system stability for cases 1 and 3. At a speed of 500 km/h, the system becomes stable as the k_d increases. The corresponding $u_{\max}$ increases, while the control energy decreases. Therefore, the maximum control force should be set as one of the control system optimal objective rather than the total control energy. For the different running speeds, the $u_{\max}$ and S_u of case 1 is the lowest and case 3 is the largest. The condition of 350 km/h is the most typical. The system stability that can be achieved for case 1 is the best and case 2 is the worst. When the $\zeta_{\max}$ is greater than -0.15, the control outputs of case 3 are significantly less than that of cases 1 and 2. When the $\zeta_{\max}$ is -0.13, the $u_{\max}$ and S_u of case 3 are almost half of the values in cases 1 and 2. We can understand that the active control of motor lateral movement is helpful to increase the vibration absorption of the motor flexible suspension, and the control output is obviously smaller than that of the other two control cases. However, if we want to get more stability, a larger active control effort transferred by the motor suspension is needed to control the frame's vibration. It is

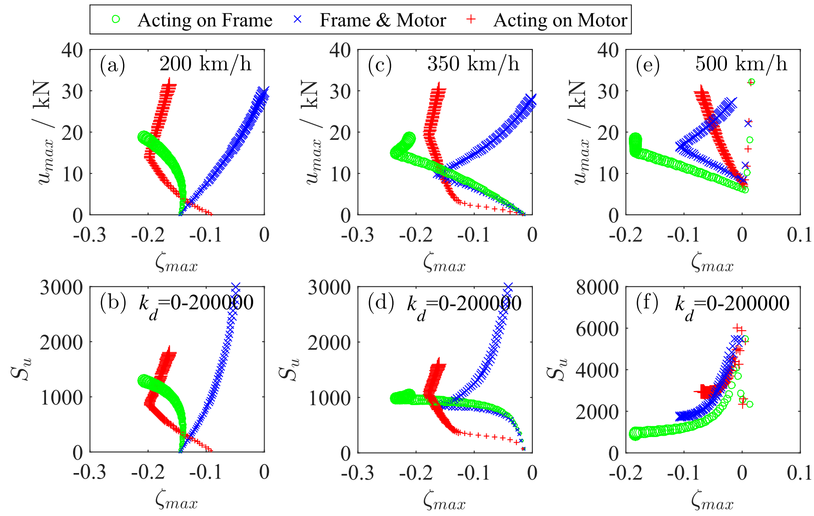

Figure 7. The actuator outputs versus the system stability in different control cases.

well worth noting that the smallest $\zeta_{\max}$ that can be achieved in Figure 7 is larger than that in Figure 4. That is to say, the system stability that can be achieved is worse than the multi-objective optimisation in Section 3.2. That is because the motor suspension parameters in this analysis are specified parameters, while the motor suspension parameters are optimised for Figure 4. So, the optimisation of bogie mechanical structure parameters need to be conducted even if the bogie adopts the active stability control.

5. The time-delay stability analysis

In a feedback control system, the time-delay caused by detecting and processing as well as response delay of the actuator is inevitable. The effect of time-delay on the system dynamic performance is obvious. In this section, the influence of time-delay on the control system on bogie hunting stability is analysed using the CTA method [22,25].

It is quite easy to analyse the stability of a linear time-delay system using the CTA method. In this study, the delayed part of the state vector is extended to 20 points, and

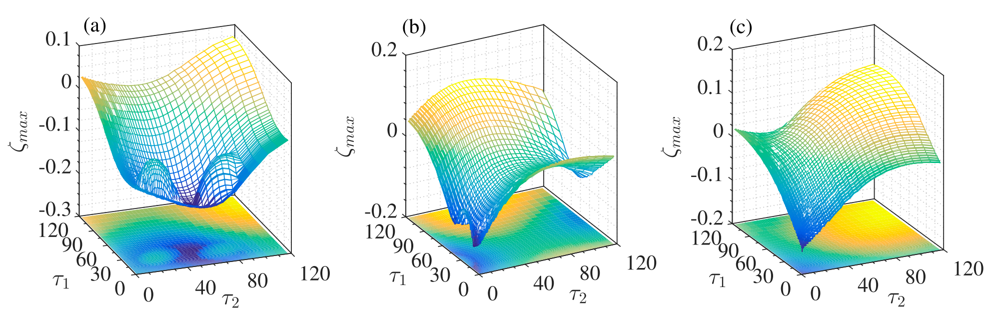

Figure 8. The system stability versus the two actuators output delay (unit: ms) in different control cases. (a) Case1: acting on frame; (b) case 2: acting between frame and motor; and (c) case3: acting on motor.

the feedback gains in three cases are also obtained from the results of multi-objective optimisation in Section 3.2.

The effects of the two actuators output delays, τ_1 and τ_2 on the system stability for the three control cases are shown in Figure 8. The $\zeta_{\max}$ of the system is -0.15 for the three control cases when the time delay is 0 ms. It can be concluded that system stability decreases as the delay increases, especially in cases 2 and 3. The system becomes unstable when the time delay is larger than 40 ms. While an appropriate time delay is conducive to improving the system stability for case 1. The corresponding τ_1 and τ_2 are 20 and 50 ms, respectively, when system stability is the best. However, even if τ_1 and τ_2 are in some low values, the stability of the system may be worse. Therefore, for case 1, the time delay should get more attention, so that we can use or avoid the effect of delay on the stability.

6. Conclusions

(1) For the bogie with motor flexible suspension in a high-speed train, we proposed three active control cases of the frame lateral vibration suppression to improve the bogie hunting stability especially in the failure state of bogie suspension parameters. In the three cases, the actuator output force acts on the frame, between the frame and the motor, and on the motor, respectively. The vibration velocity of the front and rear end beams of the frame are fed back linearly to calculate the output force of the front and rear actuators. The results show that the control cases can effectively improve the stability of the bogie based on the system linear stability analysis.

(2) The feedback gains and the motor suspension parameters in different control cases are optimised with the two objectives of system stability margin and maximum control effort, and the actuator outputs are compared based on the theoretical analysis with a 8 DOF bogie model. The active control of motor lateral movement is helpful to increase the dynamic vibration absorbing function of the motor flexible suspension, and the control output is obviously smaller than that of the other two control cases below the speed of 350 km/h. However, case 1 is more effective when the uncontrolled bogie system is unstable, for instance, at a speed of 500 km/h.

(3) The optimal motor suspension parameters are helpful to improve the active hunting stability. The optimal f_{my} for case 3 increases by almost 1 Hz than cases 1 and 2, and the optimal ξ_{my} increases as the bogie hunting stability increases for the three control cases. The effects of the two actuators output delays on the system stability in three cases are compared. The system stability decreases as the time delay increases. Meanwhile, the time delay in case 1 should receive more attention, so that we can use or avoid the effects of delay on the stability.

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Appendix

The matrices of the bogie system in Equation (1) are listed below.

$$\mathbf{M} = \begin{bmatrix} m_w & & & & & & & \\ & I_w & & & & & & \\ & & m_w & & & & & \\ & & & I_w & & & & \\ & & & & m_f & & & \\ & & & & & I_f & & \\ & & & & & & m_m & \\ & & & & & & & m_m \end{bmatrix}$$

$$\mathbf{K} = \begin{bmatrix} k_{py}+k_{gy} & -2f_\eta & & & -k_{py} & -bk_{py} & & \\ \frac{2\lambda_e l_0 f_\zeta}{r_0} & l_1^2 k_{px}+k_{g\psi} & & & & -l_1^2 k_{px} & & \\ & & k_{py}+k_{gy} & -2f_\eta & -k_{py} & bk_{py} & & \\ & & \frac{2\lambda_e l_0 f_\zeta}{r_0} & l_1^2 k_{px}+k_{g\psi} & & -l_1^2 k_{px} & & \\ -k_{py} & & -k_{py} & & \begin{matrix} 2k_{py}+k_{sy} \\ +2k_{my} \end{matrix} & & -k_{my} & -k_{my} \\ -bk_{py} & -l_1^2 k_{px} & bk_{py} & -l_1^2 k_{px} & & \begin{matrix} l_2^2 k_{sx}+2l_1^2 k_{px} \\ +2b^2 k_{py}+2l_m^2 k_{my} \end{matrix} & -l_m k_{my} & l_m k_{my} \\ & & & & -k_{my} & -l_m k_{my} & k_{my} & \\ & & & & -km_y & l_m k_{my} & & k_{my} \end{bmatrix}$$

$$\mathbf{C} = \begin{bmatrix} \frac{2f\eta}{v} & & & & & & & \\ & \frac{2l_0^2 f_\zeta}{v} & & & & & & \\ & & \frac{2f\eta}{v} & & & & & \\ & & & \frac{2L_0^2 f_\zeta}{v} & & & & \\ & & & & c_{sy}+2c_{my} & & -c_{my} & -c_{my} \\ & & & & & l_2^2 c_{sx}+l_m^2 c_{my} & -l_m c_y & l_m c_{my} \\ & & & & -c_{my} & -l_m c_{my} & c_{my} & \\ & & & & -c_{my} & l_m c_{my} & & c_{my} \end{bmatrix}$$

$$\mathbf{E_1} = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & l_e & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & -l_e & 0 & 0 \end{bmatrix}^{\mathrm{T}}$$

$$\mathbf{E_2} = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & l_m & -1 & 0 \\ 0 & 0 & 0 & 0 & -1 & l_m & 0 & 1 \end{bmatrix}^{\mathrm{T}}$$

$$\mathbf{E_3} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}^{\mathrm{T}}$$

Table A1. The bogie model parameters.

Symbol	Definition	Value
m_w	Mass of the wheelset	1639 kg
I_w	Yaw inertia of the wheelset yaw inertia	822 kg.m^2
m_f	Mass of bogie frame	2328 kg
I_f	Yaw inertia of bogie frame	4080 kg.m^2
m_m	Mass of motor	850 kg
λ_e	Wheel-rail contact conicity	0.2
k_{px}	Primary longitudinal stiffness per axle	60 kN/mm
k_{py}	Primary lateral stiffness per axle	10 kN/mm
k_{sx}	Secondary longitudinal stiffness	0.26 kN/mm
k_{sy}	Secondary lateral stiffness	0.26 kN/mm
c_{sx}	Yaw damper damping (fault/design value)	660/2500 N.s/m
c_{sy}	Secondary lateral damper damping	30 kN.s/m
$2l_1$	Lateral spacing of primary suspension	2000 mm
l_e	Longitudinal distance from end beam	1500 mm
$2b$	Wheel base	2.5 m
$2l_0$	Distance of the Contact spot	1493 mm
$2l_2$	Lateral spacing of the secondary suspension	2200 mm
l_m	Longitudinal distance from motor suspension to bogie center	750 mm
r_0	The wheel rolling radius	0.46 m
f_ξ	The longitudinal creep coefficient	1e7 N
f_η	The lateral creep coefficient	1e7 N
f_{my}	Motor suspension frequency	3 Hz
ξ_{my}	Motor suspension damping ratio (Uncontrol/Control)	0.2/0.5
k_{gy}	The gravitational stiffness	–
$k_{g\psi}$	The gravitational angular stiffness	–
k_r	The contact stiffness of the wheel-rail	500 kN/mm
v	Speed	350 km/h