Research on Parameters of the Curve Negotiation Performance of High-Speed Train Composed of Multiple Vehicles

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Abstract. This paper calculates the wheel/rail nonlinear numerical solution that both changes in trend or in values are relatively consistent conclusion, using different methods of comparative analysis of the cycle dynamics system for studying the influence of the curve negotiating velocity and orbital parameters on the performance of high-speed train. Then based on single-section vehicle system program, a high-speed train coupling system considering vehicle terminal connection device is established. The influence of curve negotiating velocity and orbital parameters on the performance of high-speed train is analyzed. The results show that, with the curve negotiating velocity increasing, the derailment coefficient, rate of wheel load reduction, and the vertical force of wheel rail have a tendency to rise after falling first, lateral force of wheel rail monotone increasing. When the vehicle is in a super-elevation state, the indicators are slightly lower than the equilibrium state. While the indicators with parameters change significantly when the vehicle is in a underelevation state. For instance, when the radius of the curve is 5000m to the radius 6700m, the derailment coefficient increases by 53.6% and rate of wheel load reduction is increased by 44.1%.

Introduction

Many domestic and foreign scholars have carried out much useful research work on the transverse dynamic characteristics of high-speed trains, most of which are based on the analysis and research of dynamic software, such as Simpack, Adams and UM [1]. Due to the complexity of the large vehicle system program, there are few analyses by software programming, most of which are the numerical analysis of half-vehicle or single vehicle. This kind of research neglects the force of the hook force and windshield in the trailer, especially the influence of the force on the lateral dynamic characteristics of the vehicle when the curve passes. Therefore, both at home and abroad about the vehicle coupling dynamics analysis gradually from the half car model to cycling model to consider end connection device of vehicle car coupling model. For example, Knudsen [2] has established a half-vehicle model, analyzing and studying the critical speed and lateral dynamics of a vehicle. X.S.JIN and X.B.XIAO [3-4] establish a half-vehicle model and a full-vehicle model based on the vehicle dynamics theory, respectively. The wheel/rail dynamic problems such as train derailment, rail damage and wear are studied. However, so far, there have been few studies on the effect of coupling curves on the performance of multi-vehicle coupling, and there is no literature on the combination of dynamic software and programming analysis.

Based on the theory of vehicle dynamics, this paper establishes the linear rail vehicle system dynamics equations of each component using the fourth order Runge - Kutta method. Numerical analysis is to test and verify the accuracy and reliability of the program, using international commercial multi-body dynamics software with parameters, with degree of freedom under the bike model and dynamics analysis, comparing and analysising the results of vehicle frame force, lateral wheel set force and lateral displacement, confirming that the single section vehicle system program on the basis of reliable and establishing thinking of end connection device of vehicle car coupled systems, on the dynamics analysis, detailed analysis of the effects of parameters on the curve by the indicators.
Verify and Analyze the Bike Model with the Same Parameters

Cycle Dynamics Model

The established cycle model of emu includes 1 vehicle body, 2 frames, 4 wheel pairs, 1 system suspension and 2 system suspension.

This paper considers the nonlinear factors of wheel / rail contact and the first and second system vibration absorbers in the vehicle system. For the creep force, between wheel and rail, the lateral and longitudinal creep forces and rotational creep torque are calculated by Kalker theory, and then the nonlinear correction is carried out based on Shen’s theory, and the Hertz nonlinear contact theory is used to calculate the normal wheel-rail forces. The flange force is simulated by a piecewise linear function to simulate the contact and separation between the flange and the rail. According to the relative displacement and relative velocity between the components of the system, the expressions of the transverse, longitudinal and vertical forces on the suspension of the first and second systems are calculated. The dynamic equations of the components of the linear rail vehicle system are established. Numerical analysis are got by using the fourth order Runge-Kutta method in the classical explicit numerical integration algorithm.

The Rigid and Flexible Coupling Model Based on Multi-body Dynamics Software Is Established

This paper establishes the system dynamics simulation model with the same numerical analysis parameters and the same number of degrees of freedom, based on the multi-body dynamics software. The center of the left and right axis diameter is selected as the interface node, and the intrinsic mode and the static mode of the elastic wheelset are calculated according to the Craig-Bampton method. The results are read into ANSYS_UM to calculate the generalized stiffness matrix and mass matrix to generate. Fss and Fum files. Based on the UM file, the elastic wheelset is obtained and the rigid flexible coupling dynamic model of the multibody system is established, as shown in Fig1.

Figure 1. Rigid-flexible coupling mode.

Compare and Analyze the Cycle Model with the Parameters

In order to verify the correctness of the established single vehicle simulation program, the results of numerical analysis are compared with the results of large commercial multi-body dynamics software. The vehicle parameters are all true parameters of a certain high-speed EMU, and the circuit is taken as a straight working condition. The track irregularity is only loaded with directional harmonic irregularity, and the function formula is \( Y(t) = 0.5A \sin \left( \frac{2 \pi x(t)}{L} \right) \), where \( A = 10 \text{mm} / \text{L} \), is 20 m, and the calculated velocity is 350 km / h. The results show that the periodic solution is obtained after stabilization. The lateral force of the axle and the lateral displacement of the wheelset are consistent both in the changing trend and in the quantity while the calculated vehicle frame force (lateral force on the axle) is 14.8 kN while the calculated result of UM is 14.5 kN, and the detailed analysis result is shown in figure 2 below. The result of program calculation is a little larger than that of UM, and the main reason is that the two analysis models are different. In program numerical analysis, the rail is considered as elastic rail. While in UM analysis, the rail is considered as rigid rail. The wheel-rail response of the elastic track model is slightly larger than that of the rigid track.
High-Speed Train System Dynamics Modeling

This paper analyzes the force and motion equation of the linked train and establishes the dynamic model of the linked train considering the workshop connection device, based on the program of the single-vehicle system, as shown in figure 3. It is assumed that there is no longitudinal sliding of the vehicle on the rail. Compared with the linear system model of a single vehicle, the nonlinear factors of the wheel/rail contact and the first and second spring vibration damping elements are also taken into account in the analysis of the connecting train curve. The train system consists of two bicycles. The function of the end connection device is considered between the two vehicles, and the bicycle system is proved to be accurate and feasible in the first part.

\[
M_{w} \ddot{y}_{w} = -F_{yfi} - F_{yfr} + F_{yit} + F_{yrt} + N_{hti} + N_{hrt} - F_{ry} + m_u \frac{V^2}{R} \cos \phi_w - g \sin \phi_w)
\]

(1)

\[
I_{w,j} \ddot{\theta}_{wi} = a_\theta (F_{xdi} - F_{xia}) + M_{xi} + M_{ez} + a_{yfr} (F_{xdi} + N_{hrt} - N_{hia}) + d_u (F_{yfi} - F_{yfr}) - I_{w,j} \dot{\phi}_w \Omega
\]

(2)

\[
M_{j} \ddot{y}_{j} = F_{yfi}(2j-1) + F_{yfr}(2j) - F_{yfi}(2j) + F_{yfr}(2j-1) + F_{yfi}(2j) - F_{yfr}(2j-1) + m_i \left( \frac{V^2}{R} \cos \phi_{ie} - g \sin \phi_{ie} \right)
\]

(3)

\[
I_{z,j} \ddot{\theta}_{zj} = I_{z}(F_{yfi}(2j-1) + F_{yfr}(2j-1) - F_{yfi}(2j) - F_{yfr}(2j)) + d_u (F_{yfi}(2j-1) + F_{yfr}(2j) - F_{yfi}(2j) - F_{yfr}(2j-1))
\]

(4)

\[
I_{x,j} \ddot{\theta}_{xj} = -H_{x}(F_{yfi}(2j-1) + F_{yfr}(2j-1) + F_{yfi}(2j) + F_{yfr}(2j)) + d_u (F_{yfi}(2j-1) + F_{yfr}(2j) - F_{yfi}(2j) - F_{yfr}(2j-1))
\]

(5)
The Curve Is Studied by Parameters

This paper establishes a high-speed train coupling system based on the iterative procedure of train/track coupling dynamics, which considers the vehicle-end connection device. The variation rule of the influence of vehicle and line parameters on the performance of train curve passing is analyzed in detail, which provides theoretical basis for the dynamic optimization of high-speed trains. Among them, there are some studies on the influence of the suspension stiffness of 1-series and 2-series of vehicle parameters and the connection device of vehicle end on the curve passing performance. The conclusions obtained are representative[5]. Therefore, relevant parameters are not studied in this paper. At present, however, there is no research on the influence of line parameters and running speed on curve passing performance.

Influence of Curve Radius on Curve Passing Performance

Dynamic simulation calculation takes curve radius of 5000-7600m, running speed of 250km/h, and outer rail super high 110mm. By analysis, figure 4-5 below shows the influence law of curve radius on vehicle curve passing performance.

The analysis results show that the vehicle has the best dynamic performance when the curve radius is 6700mm. When the radius of the curve is less than 6700mm, the dynamic characteristics of the vehicle improve with the increasing of the radius. The derailment coefficient increases by 53.6% and the wheel weight loss rate increases by 44.1% when the radius of the curve is 5000m over 6700m. When the radius continues to increase beyond 6700mm, compared with the curve radius, the section of the curve is in an surplus superelevation state, but its value is already very small, which will not have a significant impact on the dynamic characteristics between the wheel and rail. However, if the curve radius is too large, the maintenance cost and maintenance workload of the line will be increased. At the same time, the dynamic characteristics will deteriorate due to the existence of too high.

Figure 4. Influence on radius of curve radius.                                      Figure 5. Influence on radius of curve radius.

The Effect of Extra Height on Curve Passing Performance

The dynamic simulation calculates that the curve radius is 6700 m, and the running speed is 250 km/h, and the value range of superelevation of outer rail is 60-180 mm. According to the analysis, the influence law of superelevation of outer rail on vehicle curve passing performance is as shown in figure 6-7.
It can be seen from the analysis results that the transverse force of wheel and rail gradually decreases and the derailment coefficient increases, with the increasing of superelevation of outer rail. The vertical force of wheel rail and wheel load reduction rate increase with the superelevation. When the superelevation of outer rail value is 110-120mm, that is, at the point of superelevation equilibrium point, the vibration of the vehicle system is the vibration of the system itself, and the vehicle dynamic performance value is the minimum value. Under the condition of underelevation, the vertical force of wheel rail and wheel load reduction rate decrease with the increasing of superelevation value of outer rail.

The Effect of the Curve on the Velocity

Based on the operating speed and dynamic model of high-speed railway in China, the value of the speed is calculated as 200-300km/h, the curve radius is 6700m, and superelevation of outer rail value is 110mm for dynamic simulation calculation. By analysis, the influence law of curve passing speed on the curve passing performance is shown in figure 8-9.

The analysis result shows that, vehicle derailment coefficient and the rate of wheel weight lightening and wheel/rail vertical forces are minimum values when the curve through the speed of 250 km/h. When the speed is less than this value, the vehicle derailment coefficient and the rate of wheel weight lightening and wheel/rail vertical force decreases monotonously and vehicle dynamics performance improved. When the curve by speed is greater than this value, the vehicles for cant deficiency state, train through the curve method, the dynamic performance simulation of with the improvement of speed increase and monotonous, vehicle dynamics performance gradually deteriorates.

Conclusion

When the vehicle is in equilibrium state, its dynamic performance value is minimum and the state of the train curve passing is optimal. As the speed of the curve increases, the derailment coefficient, wheel weight and load reduction rate and the vertical force of the wheel and rail decrease first and then rise. When the vehicle is in an ultra-high state, comparing with the balance state, the curve declines through various indicators, but changes little. When the vehicle is under ultra-high elevation, the curve changes significantly with the parameters, and the derailment coefficient increases by 53.6% when the radius of the curve is 5000m over 6700m, and the wheel weight loss rate increases by
44.1%. This paper, under ultra-high elevation, verifies that not only the damage of parts such as wheel and rail will be accelerated, but also the accident of wheel derailment may occur.

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References


