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Multi-Body Dynamics Modeling of Heavy Goods Vehicle-Rail Interaction

Lili Liu[1], Jianhua Liu[2], Jihong Zuo[1*]

[1]College of Intelligent Control, Hunan Railway Professional Technology College, Zhuzhou, China
[2]College of Transportation Engineering, Hunan University of Technology, Zhuzhou, China
Email: *zuojihong2006@outlook.com

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Abstract

Based on the principle of vehicle-track coupling dynamics, SIMPACK multi-body dynamics software is used to establish a C80 wagon line-coupled multi-body dynamics model with 73 degrees of freedom. And the reasonableness of the line-coupled dynamics model is verified by using the maximum residual acceleration, the nonlinear critical speed of the wagon. The experimental results show that the established vehicle line coupling dynamics model meets the requirements of vehicle line coupling dynamics modeling.

Keywords

Vehicle-Rail Coupling, Dynamic Modeling, Wheel-Rail Interaction Forces

1. Introduction

In view of the serious challenge of the integrated complexity of the wheel-rail system of heavy-duty trucks, the traditional vehicle-rail dynamics analyses are no longer able to meet the demand for accurate resolution at the technical level. With the deepening of theoretical research and the leap in computer technology, it is now possible to explore this field with the help of advanced multibody dynamics simulation [1]. Especially when dealing with large and complex systems, computer simulation has become a central tool in modern engineering practice. For the study of vehicle-track dynamic interaction, although real vehicle tests provide valuable data, they are costly, difficult to operate, and costly in case of serious accidents such as derailment. In contrast, numerical simulation provides an ideal platform for the design optimisation of new low-energy wheel-rail systems with its fast parametric study and sensitivity analysis capabilities.

To successfully implement the numerical simulation of coupled vehicle-rail

dynamics, it is crucial to construct an accurate mathematical model, which should truly reflect the basic physical properties of the vehicle and rail system. In this paper, based on SIMPACK multi-body dynamics software, we will establish a 73-degree-of-freedom multi-body dynamics model of C80 wagon vehicle-line coupled dynamics, and verify the reasonableness of the vehicle-line coupled dynamics model by using the maximum residual acceleration, and the nonlinear critical speed of the wagon.

2. C80 Locomotive Bogie

The research object is the current common model of heavy-duty locomotive C80, C80 bogie adopts K6 type bogie, K6 type bogie as a key component of the traditional wagon load bearing. Its structure covers a number of key parts: wheel sets, axle box suspension unit (using rubber pads), side beams, pillow spring suspension system, damper unit, centre pin, and constant contact elastic bypass bearings, etc. [2], the 3D effect of the exploded diagram is shown as **Figure 1**.

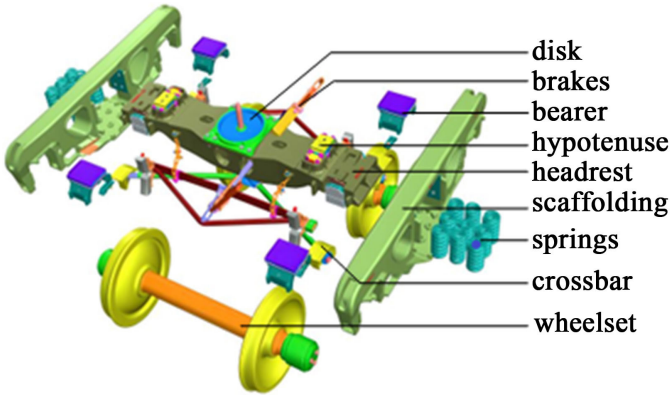

Figure 1. K6 bogie 3D effect explosion diagram.

This complex architecture contains a wealth of nonlinear dynamic properties, such as various gap effects, the role of limiters, the influence of wedge damping, and the friction moment between the heart disc and the side bearings, which all have a significant impact on the overall performance. The heavy-duty vehicle line coupling dynamics model system is mainly composed of vehicle subsystem, track subsystem, and wheel-rail contact system, in which the wheel-rail contact system realises the vibration feedback between vehicle subsystem and track subsystem. In this paper, physical modelling is carried out for C80 wagon, which is a single-section vehicle-rail coupling model for heavy-duty railway, and the interaction between vehicles is not considered.

3. Vehicle-Line Coupling Modeling

3.1. Vehicle Subsystem

(1) Modelling of axlebox positioning

In the modelling process of K6 type bogie, the influence of longitudin-

al/transverse gap between the load carrying saddle and the guide frame needs to be considered. When the relative displacement between the axlebox and the guide frame is less than the amount of clearance, the stop does not produce force; when the relative displacement between the two is greater than the amount of clearance, the stop produces stopping force. At this time, the longitudinal/transverse positioning stiffness is segmented linear stiffness, and the longitudinal/transverse nonlinear elastic suspension force is determined by Eq. (1), and its nonlinear characteristics can be defined in SIMPACK by Input Function

$$F=\begin{cases} K_1x+\left[\max\left(x,\delta\right)-\delta\right]\left(K_2-K_1\right) & x\geq 0 \\ K_1x+\left[\min\left(x,-\delta\right)+\delta\right]\left(K_2-K_1\right) & x\leq 0 \end{cases} \tag{1}$$

Where, x is the longitudinal (transverse) relative displacement, δ is the amount of longitudinal/transverse clearance, K_1 is the longitudinal/transverse positioning stiffness, and K_2 is the contact stiffness.

(2) Modelling of the two-series suspension device

The equivalent friction coefficient model is selected for modelling, and according to the literature [3], the equivalent friction coefficient is defined as follows: the equivalent friction coefficient μ_{de}, μ_{1de} between the friction wedge and the rocker and wear plate when the rocker is moving downward with respect to the sidestand, as shown in equation (2).

$$\begin{cases} \mu_{de}=\dfrac{\mu\left(\sin\alpha-\mu_1\cos\alpha\right)}{\left(1+\mu\mu_1\right)\cos\left(\alpha-\beta\right)+\left(\mu_1-\mu\right)\sin\left(\alpha-\beta\right)} \\ \mu_{1de}=\dfrac{\mu\left(\cos\beta+\mu_1\sin\beta\right)}{\left(1+\mu\mu_1\right)\cos\left(\alpha-\beta\right)+\left(\mu_1-\mu\right)\sin\left(\alpha-\beta\right)} \end{cases} \tag{2}$$

Equivalent friction coefficient μ_{ue}, μ_{1ue} between the friction wedge and the rocker and wear plate when the rocker moves downward relative to the side frame

$$\begin{cases} \mu_{ue}=\dfrac{\mu\left(\sin\alpha+\mu_1\cos\alpha\right)}{\left(1+\mu\mu_1\right)\cos\left(\alpha-\beta\right)-\left(\mu_1-\mu\right)\sin\left(\alpha-\beta\right)} \\ \mu_{1ue}=\dfrac{\mu\left(\cos\beta-\mu_1\sin\beta\right)}{\left(1+\mu\mu_1\right)\cos\left(\alpha-\beta\right)-\left(\mu_1-\mu\right)\sin\left(\alpha-\beta\right)} \end{cases} \tag{3}$$

Where, α, β are the angle between the friction wedge and the contact surfaces of the wear plate and rocking pillow respectively. μ, μ_1 are the friction coefficient of the friction wedge and the contact surfaces of the wear plate and rocking pillow respectively. The equivalent friction coefficient can be calculated only according to the vehicle parameters, and this friction element can be set by SIMPACK software.

(3) Modelling of the heart plate and side bearing device

The C80 locomotive has a unique frame link system that utilises a planar heart disc and dynamically responsive bifunctional elastomeric ball side bearings in a synergistic load-bearing design. With respect to the heart disc assembly, an advanced lubrication-free torque management strategy is used for simulation and

analysis. However, in the case of the normal contact bifunctional elastomeric ball sidebearing, a characteristic of the design is that the ball maintains a small 5 to 6 mm gap with the upper sidebearing surface, which requires that consideration of gap effects must be fully incorporated into the assessment of the sidebearing's performance in terms of longitudinal friction. The calculation can be approximated as shown in equation (4).

$$F_s = \begin{cases} K_z\left(z_0 + d_z\right)\mu_s & d_z \le \Delta z \\ K_z\left(z_0 + \Delta z\right)\mu_s & d_z \le \Delta z \end{cases} \tag{4}$$

Where, z_0 is the by-pass spring pre-compression, d_z is the by-pass vertical deformation, μ_s is the by-pass surface friction coefficient, K_z is the by-pass spring vertical stiffness, Δz is the vertical clearance.

3.2. Wheel Track Contact Mechanics Modeling

In SIMPACK software, the calculation of wheel-rail contact mechanics is divided into three steps: firstly, identifying the location, area size and shape of the contact patch; secondly, calculating the normal contact force at the contact patch; and finally, determining the tangential contact force and tangential torque at the contact patch surface.

The actual contact position between the rail and the wheelset is accomplished by the force element Rail-Wheel Interface, an algorithm that searches for the amount of interpenetration between the rail profile and the wheelset tread in the vertical plane in the reference coordinate system of the rail profile to determine the position and number of contact spots [4]. Two different methods are used in SIMPACK software to deal with the contact patch in the subsequent process, *i.e.*, equivalent elastic wheel-rail contact and discrete elastic wheel-rail contact.

The equivalent elastic wheel track contact determines the actual shape of the contact patch, but instead of using it to directly calculate the normal and tangential forces, it is equated to an elliptical contact patch that produces forces comparable to those produced by the actual contact surface, and an elliptical area comparable to the area of the original contact surface.

The discrete elastic wheel-rail contact uses the actual contact patch to calculate the normal and tangential forces, this algorithm cuts the contact patch into a number of longitudinal slices of constant width, the width of which is the length of the discrete step in defining the rail profile, and the algorithm corrects the problem of the size of the contact surface in order to achieve a better fit to the shape of the contact surface in reality. In this paper, the equivalent elastic wheel-rail contact is used to calculate the wheel-rail contact force.

For the calculation of the normal force, SIMPACK software adopts the Hertz elastic contact theory: it is assumed that the contact patch is an ellipse with long axis a and short axis b. Where a and b are calculated as shown in (5)

$$\begin{cases} a = m\left(\dfrac{3\pi P\left(k_1+k_2\right)}{4\left(A+B\right)}\right)^{\frac{1}{3}} \\ b = n\left(\dfrac{3\pi P\left(k_1+k_2\right)}{4\left(A+B\right)}\right)^{\frac{1}{3}} \end{cases} \tag{5}$$

Where, P is the elliptical contact spot subjected to the normal load, A, B are the constant coefficient of the angle between the plane of the main curvature of the tread ellipsoid of the wheel pair and the plane of the main curvature of the ellipsoid of the head of the rail. m, n are constants related to A, B. k_1, k_2 depend on the Poisson's ratio of the wheel pair with the material of the rail and tensile modulus of elasticity. According to the long and short axes of the ellipsoidal contact surface can determine the normal stress distribution on the ellipsoidal contact surface, as shown in equation (6):

$$p_z\left(x,y\right) = \frac{3P}{2\pi ab}\sqrt{1-\left(\frac{x}{a}\right)^2-\left(\frac{y}{b}\right)^2} \tag{6}$$

In dealing with the evaluation of tangential forces, the Kalker nonlinear rolling contact simplified model is mainly based on the FASTSIM algorithm in the SIMPACK software. The principle is to decompose the elliptical contact interface into separate, transversely equal-width rectangular blocks arranged along the rolling path. These rectangular blocks are further divided into an equal number of small cells, and then the calculation is realised by advancing backwards from the rolling contact front using a recursive chain solution method.

The distribution of the tangential force in the longitudinal, transverse, and spin directions can be obtained from the Kalker's simplified theory of nonlinear rolling contact and the FASTSIM algorithm, as shown in equation (7)

$$\begin{cases} T_x = \iint\limits_F X dxdy = -abGC_{11}v_x \\ T_y = \iint\limits_F Y dxdy = -abG\left(C_{22}v_y + \sqrt{ab}C_{23}\varphi\right) \\ M_z = \iint\limits_F \left(xY - yX\right)dxdy = -\left(ab\right)^{1.5}G\left(C_{22}v_y + \sqrt{ab}C_{33}\varphi\right) \end{cases} \tag{7}$$

where, v_x, v_y, φ are the longitudinal, transverse, spin creep slip rate. C_{11}, C_{22}, C_{23}, C_{33} are the Kalker creep slip coefficient. G is the constant determined by the shear modulus of the wheelset and rail material, which takes the value of $2.1 \times 10^{11}\ N/m^2$ in SIMPACK software, the tensile modulus of elasticity takes the value of 0.28.

3.3. Rail Subsystem

The track subsystem architecture is divided into two key components: the rail unit and the sleeper-roadbed module. From a theoretical point of view, the rail can be analogised to an ideal Timoshenko beam, but in practical numerical simulations, it is usually simplified to a finite Eular model [5]. As for the roadbed

model, its complexity is reflected in the fine treatment of the three-dimensional dynamics of the rails: longitudinal, vertical, and rotational vibrations, while the vertical response of the roadbed itself should not be neglected.

In constructing the analytical framework, the track modelling strategy from the literature is used, in particular, the Euler beam model, which is 3D continuous but with discrete support characteristics, is employed to finely encompass the dynamic behaviour of the rail in vertical, horizontal and torsional deformations. For the rail sleeper, it is positioned as a rigid component in the rail support system, with embedded spring-damping elements to accurately simulate its behaviour in vertical and lateral motion as well as in rotational response, and it serves as a key bridge for the interaction between the rail and the roadbed.

4. Model Validation

The maximum residual acceleration and nonlinear critical speed of trains are key parameters for evaluating the safety and stability of train operation. The correctness of the vehicle-line coupling dynamics model is verified in terms of the maximum residual acceleration when the vehicle is in equilibrium and the nonlinear critical velocity.

(1) Correctness verification based on maximum residual acceleration

The maximum residual acceleration of a train is an important parameter in the analysis of multi-body dynamics systems, used to measure whether the system reaches equilibrium under initial conditions. Usually, when the maximum residual acceleration of the entire system is less than 0.01 m/s^2, it can be considered that the system is initially balanced.

In SIMPACK, a static balance analysis was conducted on the established vehicle line coupled multi-body dynamics model. The vehicle's driving speed was set to 0, resulting in a maximum residual acceleration of 4.2×10^{-5} m/s^2, which is much lower than the threshold of 0.01 m/s^2, indicating that the model is balanced under initial conditions.

(2) Correctness verification based on non-linear critical speed of lorry

The non-critical speed of the wagon is one of the important judgment standards for evaluating the operational stability of the vehicle, which represents the maximum safe speed of the train in the process of high-speed driving. When the train exceeds this speed, serious instability may occur, such as shaking, overturning, etc. In SIMPACK, the convergence of the transverse displacement of the first wheel pair at different speeds is observed by applying sinusoidal excitation on an ideal straight line. When the traverse converges at a certain speed and does not converge at speeds greater than this, then this speed is indicated as a nonlinear critical speed.

In this paper, a sinusoidal excitation with amplitude of 0.007 m and frequency of 5 Hz is taken as the input to the linear line with a length of 30 m. The simulation analysis is carried out in the speed range of 60 km/h - 200 km/h and the results are shown in **Figure 2**.

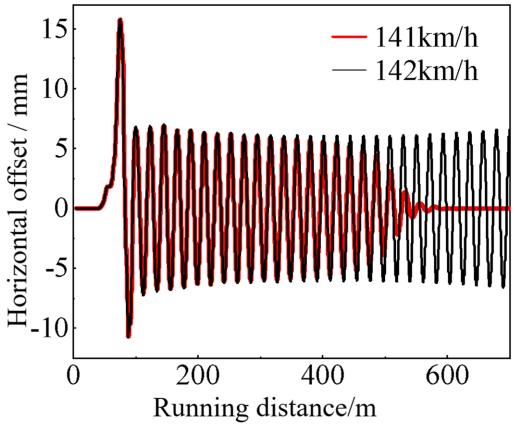

Figure 2. Truck nonlinear critical velocity.

From **Figure 2**, it can be seen that the wheel-rail traverse of the wheelset gradually converges to 0mm at the speed of 141 km/h, and is in the divergence state at the speed of 142 km/h. The nonlinear critical speed of the proposed vehicle-line coupling model is located between 141 km/h and 142 km/h, which is greater than the maximum design speed of 120 km/h for C80 trucks.

5. Conclusion

Based on the multi-body dynamics theory and the vehicle-track coupling multi-body dynamics theory, the basic theoretical support is provided for the construction of the vehicle-line coupling model; secondly, through fully analysing the composition of the vehicle-line coupling model, the vehicle-line coupling multi-body dynamics model of heavy-duty truck C80 is established by using the SIMPACK software; and finally, the correctness of the model is preliminarily verified by the maximal residual acceleration and the nonlinear critical velocity. Finally, the maximum residual acceleration and the nonlinear critical speed are used to verify the correctness of the model. The experimental results show that the model meets the requirements of vehicle-line coupling dynamics modeling.

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Conflicts of Interest

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

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