Analysis of Longitudinal Vibration in Train Sets Caused by Tractive Force Variation

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Variation in driving or braking force can cause longitudinal vibrations of the vehicle, which sometimes causes a fall in adhesion and further slipping and skidding. As such, it is essential to elucidate the relationship between tractive force and longitudinal vibration in consideration of electric and mechanical systems. Understanding this relationship can facilitate improvement of motor control methods. This paper reports on investigations into longitudinal acceleration produced by variation in tractive force, and on results of a comparison between test results using an actual train set and results obtained from numerical simulation.

Keywords: vehicle dynamics of train set, longitudinal vibration analysis, tractive force, coupler force, coupler, buffer

1. Introduction

Railway systems are very energy efficient because of their low rolling friction compared to other means of transportation. Conversely, this characteristic generates railway vehicle slipping between the wheels and rails. Once a wheel slips, it is difficult to generate a large driving or braking force without additional countermeasures. Therefore, electric railcars detect wheel slipping mainly by using data about rotational speed and acceleration of the motor. By effectively using this information, re-adhesion control systems have been developed to improve the tractive effort of trains [1], [2].

Fluctuations in driving force cause longitudinal vibration in a train. In conjunction with this, pitching motion causes axle-weight transfer. These phenomena reduce the adhesion limit, which can cause the wheel to slip. In order to construct an electric motor control method that improves the driving force and reduces the longitudinal vibration of the train, it is necessary to elucidate the relationship between the driving force and the longitudinal vibration of the train while considering the vibration of the electric and mechanical systems.

Previous research led to the development of a numerical analysis method for coupled vertical and longitudinal motions of train set [3]. To study coupler forces, this method uses an elaborate model of the buffer.

We compared the test results and the numerical analysis results for the longitudinal acceleration when fluctuating motor torque is applied. As a result, it is found that the numerical method is applicable to the longitudinal acceleration of the train set.

2. Numerical analysis model

In this chapter, the numerical analysis model for railway vehicles and coupling devices are described. The fourth order Runge-Kutta method is used for the numerical analysis.

2.1 Numerical analysis model for railway vehicle

Figure 1 shows the numerical analysis model of the railway vehicle. This model consists of suspended mass (rigid body $G_b$), mass between primary spring and secondary spring (rigid body $G_t$) and unsuspended mass (rigid body $G_w$). Vehicles are coupled to each other via a coupler and a buffer. Degrees of freedoms shown in Table 1 are given for each rigid body and buffer. The wheels and rails are assumed to roll without slipping. Therefore, only two degrees of freedom for the rotation angle are given for each wheel. Rigid bodies $G_w$ and $G_t$ are coupled by the primary suspension, and rigid bodies $G_t$ and $G_b$ are coupled by the secondary suspension.

![Numerical analysis model of a railway vehicle](image)

**Table 1** Degree of freedom about a railway vehicle in the Fig. 1

| Symbol and meanings | Degree of freedom |
|--------------------|------------------|
| Carbody $G_b$      | $(x_b, z_b, \theta_b, \dot{x}_b, \dot{z}_b, \dot{\theta}_b)$ |
| Truck $G_t(i=1,2)$| $(x_{ti}, z_{ti}, \theta_{ti}, \dot{x}_{ti}, \dot{z}_{ti}, \dot{\theta}_{ti})$ |
| Wheel $G_w(i=1,\ldots,4)$ | $(\theta_{wi}, \dot{\theta}_{wi})$ |
| Buffer displacement $\delta_i(i=1,2)$ | $(\delta_i, \dot{\delta}_i)$ |
2.2 Numerical analysis model for coupling devices

Figure 2 shows the coupling device model. This model consists of two buffers and a concentrated mass \( m_c \) between them. The total length of the coupling device, \( L_{c,n} \), varies with the buffer displacement \( \delta_{c,n} \) and \( \delta_{c,n+1} \), which are positive numbers in the direction of tension. Both ends of the coupler are pin supported to the car body of car \( n \) and \( n+1 \), so that the longitudinal straightness of the coupler is maintained. In the neutral position (near where \( \delta \) equal zero), the gap can be given to the spring characteristics of the buffer in consideration of the gap of the cross joint connecting between coupler and buffer.

The restoring force of the rubber buffer is determined by the nonlinear spring characteristics due to the compressive deformation of the rubber pad, and the amount of energy absorbed depends on the process of compression and tension of the rubber buffer. Therefore, considering the compression or tension process of the rubber buffer, the loading (orange) and unloading (light blue) curves are defined as shown in Fig. 3. In addition, as shown by the red points in the figure, we define a transient curve that asymptotically approaches a load or unloading curve from any starting point within the region enclosed by loading and unloading curves. This enables us to represent the process of any loading and unloading process of the curve as shown by the black arrows in Fig. 3.

3. Track test results and numerical analysis

In this chapter, we confirm the validity of the numerical analysis model for longitudinal vibration by comparing the results between the track test and the numerical analysis.

3.1 Parameters of numerical analysis

A track test was conducted as part of the development of the drive control technology. During this test, the vibration acceleration and coupler force of the railway vehicle under a variable driving force was measured.

The train set used for the track test was composed of four cars as shown in Fig. 4, and the masses of these car bodies used in the numerical analysis were 32.0, 27.5, 26.7, and 31.8 tons, starting with car 1, including the weight of the load. The mass of the bogie was 4.6 tons for the driving bogie and 3.5 tons for the trailing bogie, and the wheel axle weight was 0.94 tons for the drive shaft and 0.95 tons for the trailing shaft. The masses of the main electric motor and gear system related to the drive unit were added to the mass of the bogie frame.

Buffers were double-action type rubber draft gears. The specification of the coupling device were: \( L_{c}=0.8 \) m, while the \( m_c \) at both ends of the train in the form of a pair of tight lock couplers, was 268 kg; the semi-eternal type coupler in the train was 210 kg. In the numerical analysis, it was considered that the buffer characteristics produced different hysteresis depending on the process and that the buffer rubber hardened with aging. As an example, Fig. 5 shows the characteristics of the displacement and load with a buffer displacement of \( \pm 32 \) mm. Here, since the railway vehicles are connected by semi-eternal type couplers, a gap of about 2 mm is considered to be appropriate. This is because it corresponds to the fit tolerance of the cross joint connecting between coupler and buffer. Considering wear in addition, we set a 2.2 mm gap at near the neutral position of the buffer in the numerical analysis, as shown in Fig. 5.

Other values, such as spring stiffness and damping constants, were determined as shown in Table 2 with reference to the design values. The primary suspension’s damping force, which corresponds to the axle damper, acted only in the compression direction. The stiffness of the single link rubber, which is the dominant parameter in the longitudinal direction, was used for the secondary suspension’s stiffness.
3.2 Comparison between track test results and numerical analysis

First, the results of the numerical analysis with and without a gap were compared with the track test results. The drive force for the track tests (Fig. 6 (a)) was applied to the drive shafts of each railway vehicle. In particular, the gray shaded area in the figure, the driving force, which deliberately fluctuates in one-second cycles, is applied to only car 2. Under these conditions, the waveforms of the coupler force measured during the track test are shown in Fig. 6 (b) and (c) as gray dots. In the track test, the coupler force between railway vehicles car 3 and 4 was a constant value of about 5 kN between 0 and 1 seconds. This was because coupling devices were considered to be in tension at the start of the track test.

The driving force variation of the track test is interpolated with a spline curve and used as input for wheel rotation torque analysis. In Fig. 6 (b) and (c), the waveforms of the coupler force obtained from the numerical analysis are shown by the solid red and black lines for the cases with and without gap, respectively. The numerical results show that the coupler force waveforms are in good agreement with the track test from 0 to 4.2 seconds, when the first peak of the coupling force appears. On the other hand, the numerical analysis results for the case with no gaps show that there were discrepancies in the time when the maximum and minimum values of the coupling force occurred in the track tests: for example, after 6.5 seconds in Fig. 6 (b). However, results from the track test and the numerical analysis agreed well with the results of the track test after 6.5 seconds in the case of with a gap.

Figure 7 shows the longitudinal acceleration of the car body for each vehicle from the results of the same track test and numerical analysis as in Fig. 6. Figure 7, which shows the acceleration, also confirms that the results of the track test and the numerical analysis show more consistent waveforms when the gap is set. Even when the driving force was varied at other excitation frequencies, we confirmed that the test results are in good agreement with the track test results when there was a gap.

Next, a driving force varied in small step changes (hereinafter referred to as “step input”) was applied to the drive axle of each vehicle. The results of the track tests using this step input were compared with the numerical analyses of the coupling force and each car’s longitudinal acceleration. Although a “step input” does not actually appear while running on a commercial line, a stepwise torque command was given to the traction motor of Car 1, as one of the test conditions here.

Figure 8 shows the coupler forces and each railway vehicle’s longitudinal acceleration when the driving force is gradually increased from zero to a constant value, and then the driving force was reduced by half (gray shaded area) on
the drive shaft of Car 1. The solid line in the figure shows the results of the numerical analysis (with gap) and the gray dots show the results of the track test.

As shown in Fig. 8, while the driving force of car 1 is halved, the coupler force falls, fluctuating for a period of about 1 second. At this time, since Car 1 is backward in the train moving direction, when the driving force is reduced by half, all coupling devices are in tension, and the coupling force between Cars 1 and 2 is especially large in the tension direction. On the other hand, after the drive force is restored, the coupler force decays with a period of about 1 second around 0 kN, but its response waveform is different during and after the drive force is halved. When the buffer is tensed, the buffer displacement is larger than the width of the gap and the nonlinear characteristics in the buffer rubber compression state dominate the motion, but when the driving force of car 1 is restored, the effect of the gap near the buffer neutral point appears.

As a result, the coupling force and the longitudinal vibration acceleration of the vehicle shows different waveforms during and after the drive force is halved. As shown in Fig. 8, it was found that during the 11-17 second period when the drive power of Car 1 was halved, the cycle of the coupling instrument force variation is in good agreement with the results of the current vehicle test.

From the considerations in this section, we concluded that the effect of the gap is small when dealing with phe-
nomens in which the coupler force is constantly in a state of tension or compression, and that the numerical analysis results are consistent with the actual phenomena with high accuracy if the gap is taken into account when simulating longitudinal vibrations.

From the above, we confirmed that the coupling force and the vehicle’s longitudinal acceleration can be estimated accurately by the numerical analysis, when driving force varies.

4. Frequency analysis of longitudinal railway vehicle’s vibration

This chapter focuses on the frequency components of the vehicle’s longitudinal acceleration caused by a steady fluctuation of driving force.

The driving force and vehicle longitudinal acceleration are shown in Fig. 9. After acceleration comes to a stable value, the driving force of car 1 and 2 are varied with a period of 2 seconds. The gray dots in the figure show the results of the track test and the solid colored lines show the results of the numerical analysis. As in the previous chapters, the positions of the maximum and minimum values in the track test results are identical to the numerical analysis results. By the two second period of vibrated driving force, a two second period of variation in the acceleration of cars 1 and 4 was also observed.

![Fig. 9 Longitudinal acceleration caused by steady state vibration in driving force](image)

![Fig. 10 Power spectral density of vehicle’s longitudinal acceleration](image)
The standard deviations obtained from the acceleration time series data after linear trend removal are shown in Fig. 9. These values are calculated using the data from 7-22 seconds in this figure. The error of the standard deviation of the numerical analysis results against the track test results is generally less than 20%.

In addition, the power spectral density (PSD) was calculated for the same data. This is shown in Fig. 10. In this figure, only the region below 10 Hz is shown, where the predominant frequencies are confirmed by the track test. From this figure, it is confirmed that the track test results and the numerical analysis results are consistent with each other in the vicinity of the two-second cycle variation and its double cycle. The power spectral density of the results obtained by the numerical analysis was found to be consistent with the magnitude of the dominant frequency calculated from track test.

As described above, we confirmed that the numerical analysis enables us to estimate the predominant frequencies of the railway vehicle’s longitudinal vibration. This analysis method enables us to predict the longitudinal vibration status applying the various torque pattern to driving wheels. Thus, this method helps the design of drive control system because longitudinal vibration is taken into consideration.

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