Suppression of Vertical Vibration in Railway Vehicles by Controlling the Damping Force of Primary and Secondary Suspensions

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In order to improve the ride comfort on railway vehicles, it is essential to suppress the vibration of the vertical bending mode of the carbody as well as one of the rigid modes. Therefore, we propose a method to suppress such vibrations by using damping control systems for axle dampers and air springs; the former is to suppress the first mode bending vibration, and the latter is to suppress the rigid mode vibration. Our results indicate that the control method effectively reduces the power spectral density (PSD) of accelerations of the carbody floor.

Keywords: railway vehicle, suspension, damping control, air suspension, bending vibration, variable damper

1. Introduction

To improve the ride comfort on railway vehicles by reducing vertical vibrations, it is common knowledge that this can be achieved by decreasing the carbody elastic vibration centering on the first mode bending vibration. Various measures to reduce the vibration have been studied, as the vibration frequently occurs close to the frequency ranges from 4 to 8 Hz to which human beings are apt to be sensitive, and which affects the ride comfort to a great extent.

In broad terms, two methods have been used in the past to reduce the first mode bending vibration of the carbody. One method is to “design the carbody so that it is free from bending vibration as much as possible” by inserting damping material or installing dynamic dampers; this method aims to disperse the energy of carbody bending vibration [1]. A second method is to “add forces to the carbody to suppress vibration” by installing actuators in parallel with the carbody secondary suspension (normally air-springs); this is known as active suspension [2]. However, these methods have not contributed much to the improvement of ride comfort so far, except in some cases where carbody bending vibration was suppressed.

The authors have therefore developed a new technique to “eliminate forces that would cause carbody bending vibration” based on a new concept without remodelling the carbody [3, 4]. More specifically, the authors have devised a method to reduce carbody vibration by controlling the damping force of the axle dampers which form a damping element of the primary suspension, and by suppressing the vertical vibration of the bogies, which is a major source of carbody excitation.

When the damping force of axle dampers is controlled to suppress the carbody first mode bending vibration, the vibration in the carbody rigid mode (vertical translational and pitching vibrations) becomes relatively more noticeable. Therefore, to improve ride comfort further, it is considered that the carbody vibration in the rigid mode should be reduced in addition to that in the first mode bending vibration. Thus, the authors studied in parallel a method to reduce the vibration in the carbody rigid mode by controlling the damping force of the air-springs, which form the secondary suspension [5].

This paper outlines damping control systems for the primary and secondary suspensions, their effect on reducing the carbody vibration, and the effect when the two damping control systems are used simultaneously; it also covers verification of the effectiveness of the newly developed vibration suppression system.

2. System composition

Figure 1 shows the arrangement of the damping control system. The primary suspension of a railway vehicle is composed of axle springs (coil springs) and variable axle dampers, while the secondary suspension consists of an air-spring with a built-in orifice control valve. The con-
trol system consists of a controller, vertical acceleration sensors installed on the carbody and bogie, and air-spring displacement sensors. The acceleration sensors on the bogie (and the carbody where necessary) and the controller are used to control the damping force of the axle damper. The carbody acceleration sensors, air-spring displacement sensors and the controller are used to control the damping of the air-spring.

2.1 Variable axle damper

Figure 2 shows an axle damper fixed between each axle box and a bogie frame. This damper is a version of the axle damper that is normally used, to which a damping force controlling mechanism is added. The damping force (relief pressure of the hydraulic oil) of the damper and its working direction can be controlled by an external command current.

The author adopted a reverse-type electromagnetic proportional relief valve [6] to control the damping force of the variable axle dampers. Figure 3 shows the characteristics of the damping force of the damper against the relief valve command current when the damper stroke speed is constant. The damping force is highest in rebound mode and lowest in compression mode at the minimum current and vice versa at the maximum current. At a medium current, the damping force takes a minimum value both in rebound and compression modes. As this method can control the damping force in both directions with only one solenoid, it is advantageous from the viewpoint of cost reduction for the dampers, wiring and controller.

The dimensions for installation and the maximum damping force of this variable axle damper are the same as those of the conventional passive axle dampers. Furthermore, compatibility holds with the conventional dampers in regard to installation requirements, including the strength required for the damper attachment bracket. It is possible, therefore, to adopt this system so that it is not limited to newly manufactured vehicles but can also be used on the existing fleet of rolling stock.

2.2 Air-spring with a built-in orifice control valve

2.2.1 Arrangement of the air-spring and its mathematical model

The schematic drawing in Fig. 4 shows the basic arrangement of the air-spring system used for rolling stock, consisting of an air-spring with rubber bellows and a laminated rubber block, an auxiliary air chamber and an orifice installed in the air passageway between them. The orifice is generally installed inside the air-spring to give resistance to the passing air in order to obtain a damping effect. A fixed orifice, 5 mm thick and 13 to 15 mm in opening diameter, is frequently used for this purpose.

On the assumption that the sprung mass \( m_s \) (corresponding to a 1/4 carbody) and the unsprung mass \( m_u \) (corresponding to a 1/2 bogie) only move in the vertical direction and that linearity holds between the airflow passing through the orifice and the differential pressure before and after it, the characteristics of the two-chamber air-spring in Fig. 4 are represented by those of the linear four-element model in Fig. 5 [7] as:

\[
k_{a1} = \frac{\gamma A_2^2 P_0}{V_a}, \quad k_{a2} = (P_a - P_0) \frac{dA_0}{dz}, \quad c_0 = \gamma A_0^2 R_1, \quad N = \frac{V_a}{V_b}
\]

Where the symbols defined are as follows:
- \( z_f \): Vertical displacement of unsprung mass \( m_u \) (bogie)
- \( z_b \): Vertical displacements of sprung mass \( m_s \) (carbody)
- \( z_{bT} \): Relative displacement in the vertical direction between bogie frame and carbody
- \( m_s \): Mass of a 1/4 carbody
- \( A_2 \): Effective area of air-spring
- \( P_0, P_a \): Air-spring internal pressure and atmospheric pressure
- \( V_a, V_b \): Volumes of air-spring and auxiliary air chamber at the neutral position
- \( \gamma, R_1 \): Air density, polytropic index and flow coefficient
- \( dA_0/dz \): Effective area changing rate of air-spring

The symbol \( z_{aT} \) is an internal variable to express the effect of the auxiliary air chamber.
2.2.2 Composition of the air-spring and its mathematical model

We developed an air spring with a built-in orifice control valve, and the internal structure is shown in Fig. 6. The orifice control valve is fixed on the bottom plate in this air-spring, which enables the opening area to be altered by an external command, so controlling its damping force, whereas in the case of the conventional air-spring the fixed orifice is installed inside the air-spring. A current is supplied to drive the valve through a hermetic connector installed at the bottom of the air-spring.

The orifice control valve in Fig. 7 controls the spool position with a solenoid motor to change the diameter of the opening area in a range from \( \phi 3 \text{ mm (minimum)} \) to \( \phi 24 \text{ mm (maximum)} \). To control the damping force, it is desirable to minimize the damping force for the purpose of attaining higher vibration isolation performance. To ensure high damping performance on the lower damping force side (larger orifice diameter side), therefore, the newly developed orifice control valve has two ports for the auxiliary air chamber and an optimized air control port [8].

2.2.3 Basic characteristics of the air-spring with a built-in orifice control valve

To investigate the characteristics of a unit air-spring with a built-in orifice control valve, the authors carried out an excitation test of the air-spring with a load of approximately 7,000 kg (equivalent to a 1/4 carbody) placed on the top. In this test, the authors applied an excitation waveform of a single sinusoidal wave, with an amplitude of 1 to 5 mm and a frequency of 0.5 to 5 Hz, from the unsprung mass at several fixed command values for the orifice control valve. They then measured the response ratio of displacement \( z_T \) to \( z_B \). The results are shown in Fig. 8.

When the voltage applied to the orifice control valve driver becomes higher (in the positive direction), the opening of the orifice becomes wider to reduce the air-spring damping \( c_A \). As a result, the response ratio increases at approximately 1 Hz (natural frequency) and decreases in the frequency band over 1.5 Hz. When the voltage applied to the orifice control valve driver becomes lower (in the negative direction) on the other hand, the opening of the orifice becomes narrower to increase the air-spring damping force and decrease the response ratio at approximately 1 Hz. When the orifice control valve is partly closed, however, the effect of restricting the airflow between the auxiliary air chamber and the air-spring becomes significant. As a result, the characteristics of the air-spring become similar to those of the air-spring without an auxiliary air chamber. When the orifice control valve is fully closed, the state of the air-spring can be approximated to the case where \( c_A \) of the model in Fig. 5 is infinite. This corresponds to a state where the springs \( k_{A1} \) and \( k_{A2} \) alone are connected in parallel with the mass \( m_b \) in Fig. 5. Consequently, therefore, the response ratio has a significantly high peak value at the natural frequency (1.9 Hz under the present test conditions) in the system composed of a mass \( m_b \) and springs \( k_{A1} \) and \( k_{A2} \).

Figure 8 shows that an agreement holds between the
characteristics of an air-spring fixed with a normal $\phi 13\,\text{mm}$ orifice (solid line) and those of an air-spring with a built-in orifice control valve when the power supply is cut (the command voltage 0 V). Therefore, in case the orifice control mechanism of an air-spring with a built-in orifice control valve has failed, it is possible to restore the characteristics equivalent to those of a normal air-spring by cutting the power source.

The damping force can be controlled based on the relationship between the damping coefficient $c_A$ of the air-spring with a built-in orifice control valve and the command voltage applied to the orifice control valve driver. Therefore, the authors identified the value of $c_A$ of the air-spring model in the section 2.2.1. The results are shown in Fig. 9. Based on the value of $c_A$ thus obtained, the authors obtained the equation (1) that expresses an approximate relation between the command voltage applied to the orifice control valve driver and the air-spring damping coefficient.

$$V = \begin{cases} (1/c_{h} + \alpha_{i} - \alpha_{j})/\alpha_{i} & (c_{i} < c_{p}) \\ (c_{h} + \beta_{i}/\beta_{j}) - \beta_{i} & (c_{i} \geq c_{p}) \end{cases}$$

In the equation (1), $\alpha_{i}$ ($i = 1, 2, 3$) and $c_{p}$ are real constants that depend on the excitation amplitude. The solid line in Fig. 9 shows the relation between the command voltage and the damping coefficient calculated by using the approximation equation (1). To control the air-spring damping force, this relationship is used to convert the value of the damping coefficient into a command voltage, which is output for the orifice control valve driver.

![Fig. 9 The identified damping coefficient $c_A$ of an air-spring against the command voltage applied to the orifice control valve driver](image)

### 3. Control law

The authors designed the controllers for axle dampers and air-springs separately on the assumption that the first bending mode and the rigid mode (vertical translation and pitching) of the carbody are independent of each other.

In the following mathematical expressions, the suffixes $i$ and $j$ indicate the positions of bogie and wheel set, respectively, as $i = 1$ (front bogie), $i = 2$ (rear bogie), $j = 1$ (front wheel set) and $j = 2$ (rear wheel set). The symbol (‘*’) in this paper represents the differential with respect to time.

#### 3.1 Axle damper control law

In this study, the authors used the following two axle damper control laws.

#### 3.1.1 Control law not based on a mathematical railway vehicle model [3, 4]

This is a control law to reduce the vibration of the bogie frame and the excitation force that works on the carbody without considering the carbody vibration mode, thereby suppressing the carbody vibration as a result. To control the vibration of the bogie frame, the authors applied the skyhook control law to the vertical translational and pitching components that significantly affect the carbody vibration in a vertical direction. The authors used a filter to integrate the vertical translational acceleration $\dot{z}_{Ti}$ ($i = 1, 2$) and pitching acceleration $\dot{\theta}_{Ti}$ ($i = 1, 2$) of the bogie frame in order to calculate the vertical translational velocity $z_{Ti}$ and pitching velocity $\theta_{Ti}$. We denote the skyhook gain by $c_{s}$ and $c_{p}$. Then, the command voltage $u_{ij}$ for the variable axle damper is obtained as:

$$u_{ij} = \begin{cases} (-c_{s} \dot{z}_{Ti} + c_{p} \dot{\theta}_{Ti})/2 & (i = 1, 2, j = 1) \\ (-c_{s} \dot{z}_{Ti} - c_{p} \dot{\theta}_{Ti})/2 & (i = 1, 2, j = 2) \end{cases}$$

#### 3.1.2 Control law based on a mathematical railway vehicle model [4]

This is a method to suppress the first mode bending vibration of the carbody concentrated by applying a control law based on a mathematical railway vehicle model.

Figure 10 shows the vehicle model used in designing the controller. Apart from the axle box vertical displacement $z_{wij}$, displacements are all described as relative values. They are the axle spring displacement $z_{bij}$, the air-spring displacement $z_{riij}$ and the first bending mode displacement $q_{ij}$. The carbody having a mass $m_{b}$ and pitching inertia moment $I_{z}$ is assumed to be a uniform elastic beam supported with air-springs at two points, with both ends free to move. $\phi(x)$ is the eigenfunction of the carbody first bending mode. Regarding the carbody vibration mode, the authors dealt with only the vertical translation, pitching and first bending modes, which significantly affect the ride comfort.

Under these conditions, the equations of motion for the carbody and the bogie frame are expressed as:

$$M_{z} \ddot{z} + C_{z} \dot{z} + K_{z} z = W_{z} w + D_{z} f$$

$z = [z_{r11}, z_{r12}, z_{r21}, z_{r22}, z_{a11}, z_{a12}, z_{a21}, z_{a22}, q_{1}, \dot{q}_{1}, \dot{q}_{2}]^{T}$

$w = [\dot{z}_{w11}, \dot{z}_{w12}, \dot{z}_{w21}, \dot{z}_{w22}]^{T}$

$f = [f_{s11}, f_{s12}, f_{s21}, f_{s22}]^{T}$

The symbols $M_{z}$, $C_{z}$, $K_{z}$, $W_{z}$ and $D_{z}$ in the equation (3) denote matrices, each having a real constant component. Other symbols $z$, $w$ and $f$ are vectors, each having the components specified below, respectively.

$z$: Relative displacements of different parts of the vehicle

$w$: External disturbance acceleration from axle box positions

$f$: Damping force $f_{s(ij)}$ generated by the axle damper
Kalman Cc

The observation quantity \( y \) used to control the damping force is the vertical vibration acceleration at three points on the carbody floor (immediately above the bogie and at the carbody center) and at the point on the bogie frame above each axle box. The force \( f_{\text{air}} \) generated by the variable damper, which is a component of the value \( f \) in the equation (3), depends on the command value \( u_{\text{cmd}} \) for the damper, the stroke velocity of the damper and its sign. This makes it possible to identify whether the stroke velocity is in the direction of compression or extension.

In designing the controller, however, the damper is assumed to be an actuator having a time lag of the first order with a time constant \( T_{\text{ds}} \) against the command value \( u_{\text{cmd}} \). See the equation (4).

\[
\frac{df_{\text{air}}(t)}{dt} = -f_{\text{air}}(t)/T_{\text{ds}} + u_{\text{cmd}}(t)/T_{\text{ds}} \tag{4}
\]

The equations (3) and (4) give the following equation of state, where \( A, B, C \) and \( G \) are matrices, each having a component of a real number.

\[
x = Ax + Bu + Gw \tag{5}
\]

\[
y = Cx \tag{6}
\]

\[
x = [\dot{z}, \ddot{z}, f]^{T} \]

\[
u = [u_{\text{cmd}1}, u_{\text{cmd}2}, u_{\text{cmd}3}]^{T} + \left[ \frac{1}{2} \sum_{i} (\ddot{z}_{m1} + \ddot{z}_{m2}) + \sum_{i} (\dot{z}_{m1} + \dot{z}_{m2}) + \phi(l_{1}) \right] q_{a} + \left[ \frac{1}{2} \sum_{i} (\dot{z}_{m1} + \dot{z}_{m2}) + \sum_{i} (\ddot{z}_{m1} + \ddot{z}_{m2}) + \phi(l_{2}) \right] q_{b} + \left[ \frac{1}{4} \sum_{i} (\dot{z}_{m1} + \dot{z}_{m2}) + \sum_{i} (\ddot{z}_{m1} + \ddot{z}_{m2}) + \phi(l) \right] q_{c}
\]

\[
y = \frac{1}{4} \sum_{i} (\dot{z}_{m1} + \dot{z}_{m2}) + \sum_{i} (\ddot{z}_{m1} + \ddot{z}_{m2}) + \phi(l)\]

Based on the derived vehicle model, the authors designed a controller. The external disturbance \( w \) for this vehicle model is the vertical vibration acceleration of the axle boxes for axles 1 to 4, which the author regarded as white noise. The author then used an LQG controller to control the vibration of the vehicle. Figure 11 shows a block diagram of the LQG controller. The estimated state of the vehicle \( \hat{x} \) is determined by the Kalman-filter from the observation quantity \( y \) and the command input \( u \). The author designed an optimal controller that enables an evaluation weight for each vibration mode of the carbody and the bogie to be set. As the output \( y \) of the observation equation (6) is the acceleration of the bogie frame and that at the three points on the carbody, it cannot be weighted for each vibration mode as it is. Therefore, in place of \( y \), the author devised a new observation equation (7) having such elements as the carbody vertical translational acceleration \( z_{q1} \), carbody pitching component vertical acceleration immediately above the bogie \( l_{1c} \), carbody first bending mode component vertical acceleration at the carbody center \( \phi(l_{c}) q_{b} \), carbody frame vertical translational acceleration \( z_{q1} \), and carbody frame pitching component vertical acceleration \( l_{w} \). Then, the authors determined the value of \( K \) that gives a state feedback \( Kx \) to minimize the quadratic cost function (8) for a system expressed by the equation (5) and (7).

\[
v_{a} = [\ddot{z}_{q1} l_{1c} \phi(l_{c}) q_{b} \ddot{z}_{q1} l_{w} \ddot{z}_{q1} l_{w} \ddot{z}_{q1}]^{T}
\]

\[
v_{a} = C_{a} x \tag{7}
\]

\[
J = \lim_{r \to \infty} E \left[ \int_{0}^{r} (y^{T} Q y + u^{T} Ru) \; dr \right] \tag{8}
\]

\[
u = -K \dot{x} \tag{9}
\]

The weight matrix \( Q \) in the equation (8) is set to take a large value for the first bending mode that greatly affects ride comfort.

### 3.2 Air-spring control law

As the air-spring control law, the authors applied the skyhook control law in the same way as in Section 3.1.1 to the carbody vertical translational mode and the pitching mode. More specifically, the authors used four acceleration sensors placed at the carbody floor immediately above each air-spring respectively to calculate the carbody vertical translational acceleration \( z_{q1} \) and pitching acceleration \( \dot{\theta}_{b} \). Then, the skyhook forces, \( f_{1} \) and \( f_{2} \), to be generated by the front and rear air-springs are given as:

\[
f_{1} = -c_{a} \ddot{z}_{q1} + c_{\text{af}} \dot{\theta}_{b} \tag{10}
\]

\[
f_{2} = -c_{a} \ddot{z}_{q1} + c_{\text{af}} \dot{\theta}_{b} \tag{11}
\]

To perform skyhook control, it is required to generate the force \( f_{a} \) by the air-spring damping coefficient \( c_{a} \) in Fig. 5. Since \( c_{a} \) is a damping element, however, it...
cannot generate a force having the same sign as that of $f_{si}$ when the signs of skyhook force $f_{si}$ and $\ddot{z}_d$, of air-spring are identical. Therefore, the authors applied the Karnopp approximation [9] as explained below to minimize the damping force if a force cannot be generated in the direction of the skyhook force $f_{si}$.

$$c_A = \begin{cases} \frac{f_{si}}{\ddot{z}_d} & (c_{A_{\text{max}}} \leq \frac{f_{si}}{\ddot{z}_d} \leq c_{A_{\text{max}}}) \\ c_{A_{\text{max}}} & (\frac{f_{si}}{\ddot{z}_d} < c_{A_{\text{max}}}) \\ \frac{f_{si}}{\ddot{z}_d} & (\frac{f_{si}}{\ddot{z}_d} < c_{A_{\text{max}}}) \end{cases} \quad (12)$$

In the equation (12), $i = 1$ and 2. The symbols $c_{A_{\text{max}}}$ and $c_{A_{\text{min}}}$ are the maximum and minimum values of the damping coefficient of the air-spring with a built-in orifice control valve.

4. Excitation test of a vehicle equivalent to a Shinkansen vehicle at a rolling stock test plant

4.1 Test conditions

The authors implemented an excitation test of a vehicle installed with prototype axle dampers and air-springs with a built-in orifice control valve at a rolling stock test plant (Figs. 12 and 13). The test vehicle, equivalent to a Shinkansen vehicle, was 24.5 m long with a carbody mass of $27 \times 10^3$ kg. Natural frequencies in the carbody first bending mode, carbody vertical translational mode and pitching mode are 8.5 Hz, 1.1 Hz and 1.6 Hz respectively.

As the excitation disturbance $w$ (see the equation (5)) to simulate actual running, the authors applied the axle box vertical vibration acceleration measured in Shinkansen vehicle running tests to the test vehicle at a phase difference equivalent to that in 300 km/h operation. As the excitation amplitude was limited at the rolling stock test plant, however, the amplitude used was 0.6 times that during actual running. The data of the excitation waveform was about 110 sec long (equivalent to a running distance of 9.2 km).

Figure 14 shows the overall composition of the controller used for the test, in which the controllers for the axle dampers and the air-springs were designed as a continuous system and installed as a discrete system. The sampling period was 1 ms.

Regarding the PSD of the bogie frame vertical vibration acceleration (Fig. 15 (a)), the acceleration PSD decreased at about 8 Hz by using either control law from that when conventional passive dampers were used (gray) (which is hereinafter referred to as the passive case). A comparison between the skyhook control (light blue) and the LQG control (dark blue), indicates that the skyhook control generally decreased bogie frame vertical vibration acceleration in the frequency band 5 to 10 Hz, while the LQG control selectively decreased vibration at about 8 Hz.

On the other hand, regarding the PSD of vibration acceleration at the carbody center (Fig. 15 (b)), a peak caused by the carbody first mode bending vibration existed around 8.5 Hz in the passive case. However, the skyhook control generally decreased vibration acceleration before and after the peak or in the frequency band 5 to 10 Hz. In contrast, the LQG control selectively decreased vibration around 8.5 Hz. When compared with the passive case, the skyhook control decreased the peak PSD value to 22% and the LQG control to 15%. Thus, the peak value-reducing effect was higher with the LQG con-
When only the damping of the air-spring was controlled (AS control, green), the peak around 1 Hz in the rigid mode (vertical translational mode) decreased to almost half that in the passive case (gray) on the floor at the carbody center (Fig. 16 (a)). When the axle damper and the air-spring were controlled simultaneously (Ax + As control, red), both the peaks at 8.5 Hz and 1 Hz decreased. This proves that a combination of control systems decreases simultaneously both the vibration in the first bending mode and that in the rigid mode.

Figure 16 (b) shows the PSD of vertical vibration acceleration of the carbody floor immediately over a bogie, which presents approximately a similar trend to that at the carbody center. Immediately over a bogie, however, the peak at 1.6 Hz is higher than that at 1.0 Hz. This means that the pitching mode is more influential in vibrating the carbody than the vertical translational mode over a bogie. The air-spring control decreased the PSD peak value both in the carbody vertical translational mode and in the pitching mode. This proves that the air-spring damping control effectively reduces the vibration in the said two rigid modes (carbody vertical translational mode and pitching mode).

4.3 Comparison of the vibration reduction effects through a combination of damping control systems

The authors then compared the vibration reduction effects when different combinations of damping control of both axle damper and air-spring were applied. Figure 16 shows the PSD of the carbody vertical vibration acceleration under the same test conditions as those in the previous section.

![Fig. 15 Results of the excitation test to simulate running on actual tracks at a rolling stock test plant (comparison of the effects of different axle damper control laws)](image)

![Fig. 16 Results of the excitation test to simulate running on actual tracks at a rolling stock test plant (Comparison of the effects through combined damping control)](image)
To study to what extent the $L_T$ value had been improved, the authors then calculated the acceleration power weighted with a frequency dependent function which represents the sensitivity of human beings, and which is used to calculate the $L_T$ value [10]. See Fig. 17 for the calculation results of that power in each octave band. Figure 17 indicates that the vibration in the frequency range with a large component governs the $L_T$ value.

The axle damper control substantially reduced the carbody first bending vibration (the component having a peak around 8.5 Hz in Fig. 16 (a) and (b)) to subsequently reduce the component in the 8 Hz frequency band in Fig. 17 (a) and (b) to 1/5 to 1/4. As a result, the $L_T$ value was improved by 3 to 4 dB. As the component in the 1 to 2 Hz frequency band is extremely small at the carbody center, the $L_T$ value did not improve much. Above the rear bogie, however, the component in the 2 Hz frequency band decreased about 30% to improve the $L_T$ value by about 1 dB. When the two control methods were combined, their effects came out simultaneously to improve the $L_T$ value by about 4 to 4.7 dB.

5. Conclusions

This paper outlines a vibration control system by means of damping control of axle dampers and air-springs and introduces the results of the test of a vehicle installed with prototype damping control elements at a rolling stock test plant. The authors have obtained the following results through the excitation tests of a vehicle equivalent to a Shinkansen vehicle to simulate running on actual tracks:

1. The axle damper damping control reduces the peak value of vibration acceleration PSD in the carbody first mode bending vibration (8.5 Hz) to about 15%.

2. Different axle damper control laws provide different tendency of vibration reduction. The bogie frame skyhook control decreases vibration in the frequency band 5 to 10 Hz, while the LQG control selectively decreases the vibration at 8.5 Hz. The LQG control is more effective than the skyhook control in reducing the peak value of the vibration at 8.5 Hz.

3. The air-spring damping control cuts the peak PSD value of the vibration acceleration in the carbody rigid mode (1 to 2 Hz) to almost a half that when normal air-springs are used.

4. Simultaneous combinations of axle damper control and air-spring control reduce the vibration both in the carbody first bending mode and in the rigid mode at the same time.

5. Damping control improves the vertical ride quality level ($L_T$). In particular, when both the axle damper control and the air-spring control are combined, the ride comfort improves to a degree of 4 to 4.7 dB.

By using a Shinkansen vehicle running test to confirm the vibration reducing effect of this system when running on existing tracks, the authors confirmed that the PSD peak value of vibration acceleration decreased to approximately 20% in the carbody first bending mode and to approximately 60% in the carbody rigid mode. As a result, the ride quality level was improved by up to approximately 3.6 dB, as had been expected in the reference [12].

Irrespective of what axle damper control law was applied, a satisfactory vibration reducing effect was obtained in this study. When the natural frequency of carbody first bending mode exists apart from that of bogie frame vertical translational mode, however, it will be difficult to obtain a vibration reducing effect by the skyhook control. Furthermore, when an influential vibration mode not modelled in the vehicle model exists closely to the natural frequency of the first bending mode, which is the object of control, the skyhook control law was not based on the model may be more effective than the LQG control law. The authors suggest that the appropriate control law should be applied in consideration of the vibration conditions of the object vehicle to improve the vertical ride comfort of the existing vehicles.

The authors will continue to study measures for improving the durability and reducing the cost with this system while aiming at incorporating the results of the study in practical application in the near future.
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