Development of Active Mass Dampers for Reducing Multi-modal Flexural Vibrations of Carbody

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The vertical flexural vibrations of railway vehicle carbodies are needed to be reduced from the viewpoint of ride quality. In particular, plural flexural vibration modes – having natural frequencies around 10Hz – are known to have a major impact on ride quality. To reduce these flexural vibrations, this paper proposes a vibration control method using active mass dampers (AMDs). The feasibility of this method was verified by carrying out excitation tests using a test vehicle with existing actuators at the rolling stock testing plant at RTRI. Following this, a more practical AMD system was designed based on numerical analysis and an actual AMD system with two smaller and lighter actuators was developed. The effectiveness of the vibration control using the newly developed AMD system was confirmed by means of excitation tests using an actual vehicle.

Keywords: vibration control, flexural vibration, carbody, active mass damper

1. Introduction

Recent light-weight and structurally simplified railway vehicle carbodies tend to have complicated three-dimensional flexural vibration modes according to the vibration measurement test results [1, 2, 3]. Such vibration modes often have natural frequencies around 10Hz known as human sensitive frequencies of vertical vibration. In such cases, some clear peaks corresponding to these modes are also observed in the acceleration power spectral density (PSD) measured on the carbody floor, and this means that these vibration modes have an impact on the ride quality of the vehicle. In particular, the modes similar to the first bending mode of a freely supported elastic beam (referred to as "B mode" in this paper) and the diagonal distortion mode (referred to as "D mode" hereafter) are known to be important, since these two flexural vibration modes are commonly observed in most railway vehicle carbodies, from the commuter-type vehicle to the Shinkansen vehicle. Figure 1 shows the mode shapes of these two modes.

This paper proposes a method aimed at reducing these two modes simultaneously by employing two active mass dampers (AMDs). The vibration reduction performance was examined by carrying out excitation tests using a commuter-type test vehicle at the rolling stock testing plant at RTRI. Initially, its feasibility was verified using existing actuators. As the next step, a more practical AMD system was designed based on numerical analysis, and a practical AMD system with smaller and lighter actuators was developed. Finally, the effectiveness of the practical AMD system was verified by means of excitation tests.

2. Feasibility study on multi-modal vibration reduction using AMD

Figure 2 shows an outline and photograph of AMD used for the feasibility study, and Table 1 shows the specifications of the actuators. The AMD is a device which carries out vibration control using inertial reaction force produced by driving the moving body by the actuator. In order to examine the feasibility of the multi-modal vibration control for both B and D mode, this AMD system was constructed using two existing actuators [4]. It is thought that the maximum force and the displacement of the actuators are large enough to realize vibration control of an actual railway vehicle carbody. Figure 3 shows the signal flow of the AMD.

In order to verify the feasibility of the proposed multi-modal vibration control method, excitation tests were carried out using a commuter-type test vehicle at the rolling stock testing plant at RTRI. Figure 4 (a) shows the test vehicle at the plant. A band-limited random signal with a uniform frequency component in the range of 3–30Hz was used as the excitation signal. A condition of the excitation was all-wheel simultaneous excitation in which all wheels are excited at the same time without a time delay between wheelsets. This condition is unrealistic when a vehicle is running on an actual track, but it was considered as suitable for exciting vertical flexural vibration modes, shown in Fig. 1.

The AMDs were mounted on the floor as shown in Fig. 4 (b). It was considered that the AMDs should be placed near the antinode of the target vibration modes to obtain a large vibration reduction effect. Therefore, the two AMDs were mounted on the floor separately, under the seats on either side and at the longitudinal center of the carbody, as shown in Fig. 4 (b) and Fig. 5. Figure 5 also shows the measurement points of the vertical acceleration on the floor.

The control algorithm of the AMDs was velocity feedback control using accelerations $A_{f4R}$ and $A_{f4L}$ which were measured at the measurement points of $f4R$ and $f4L$. Figure 6 shows the block diagram of the controller. The
vertical vibration accelerations on the floor were separated into the signals which accentuate the B and D modes respectively by a mode separator. In this case, the mode separator was defined by the following equations.

$$A_B = A_{f4R} + A_{f4L}$$  \hspace{1cm} (1)

$$A_D = A_{f4R} - A_{f4L}$$  \hspace{1cm} (2)

Control signals ("V1, V2") were generated through an integrator for obtaining the vertical velocity of each vibration mode, band-pass filter ("GB, GD") and control gain ("KB, KD"). These band-pass filters had a role in reducing the influence of the low and high frequencies which were outside of the control frequency range, and compensating phase lags of an actuator.

Figure 7 shows the acceleration PSD of the vertical vibration measured at f4L under passive and active conditions. As shown in the figure, there are two peaks at 7Hz and 10.5Hz which were due to D and B mode respectively under passive conditions. These two peaks were reduced simultaneously under active conditions. This result confirmed the feasibility of the multi-modal vibration reduction effect using AMDs.
3. Downsizing and weight reduction of AMDs

The preceding chapter showed the feasibility of the proposed vibration control method using existing actuators. As the next step, a more practical AMD system with smaller and lighter actuators was examined. The purpose of the next examination was to obtain a sufficient control effect using actuators which could be placed under the seats of the commuter-type vehicle.

The vibration reduction effect is commonly considered to become greater when the maximum force of the actuators is larger. By assuming the time function of response displacements of a moving body of the AMDs as \(x(t) = X e^{j\omega t}\), a relationship between control force \(F_c\) and mass of the moving body \(m\) is shown as (4).

\[
x(t) = X e^{j\omega t}
\]

\[
F_c = m\ddot{x} = -m\omega^2 x
\]

\(X\) denotes the amplitude of displacement; \(\omega\) is the angular frequency; \(t\) is the time; \(j\) is the imaginary unit; \(\ddot{x}\) is the second order differential of \(x\) with respect to time. This equation shows that the control force of the AMDs may be limited by mass and the maximum displacement of the moving body even if the maximum force of the actuator is large enough. Therefore, the relationship between these specifications and the vibration reduction effect was investigated by using a numerical analysis model.

3.1 Numerical analysis model

Figure 8 shows the box-type model used for the numerical analysis in this study. In this model, the carbody is modeled as a box-type structure consisting of elastic plates, elastic beams and solid plates. Each component is connected by means of translational and rotational artificial springs. The model is proposed as a tool for investigating the 3D flexural vibration of railway vehicle carbodies including B and D modes with a small degree of freedom (DOF). A detailed description is available in the literature [5].

The DOF of the original box-type model used in this study was 575. This is enough small as compared with the finite element method (FEM) model which usually becomes so large as to exceed 100,000 in some cases. The calculation costs for the box-type model are low enough as far as frequency response analysis is performed. However, time history response analysis whose calculation load is larger than frequency response analysis is necessary for examining the maximum force of the actuators and the maximum displacement of the moving body. Therefore, a model with a smaller DOF was needed. In order to meet this requirement, we applied a model order reduction technique on the box-type model, and built an analysis model of 14 DOF [6].

Figure 9 shows the results of the acceleration PSD at measurement point f4L. In this figure, the blue line expresses measured data in the above-mentioned excitation test, and the red line shows the calculation result by the 14 DOF model. As shown in the figure, a good level of agreement between the measured results and the calculation results in terms of acceleration PSD was obtained in the frequency range of 5–15Hz.

Figure 10 shows a numerical analysis model of the AMD. This model is a simple system with 1 DOF which consists of a spring, a mass, a damper, and an actuator. The characteristics of the actuator with respect to the relationship between the control signal and the control
force were modeled as a transfer function identified by
the stationary excitation test. The model was added to
the reduced order box-type carbody model, and numerical
calculation was performed under all-wheel simultaneous exci-
tation conditions. The measured and calculated results are
shown in Fig. 11. A good level of agreement between these
results in terms of acceleration PSD was observed under
active conditions in the frequency range 5–15Hz.

3.2 Numerical analysis for downsizing of AMDs

By using the above mentioned numerical analysis
model, the relationship between the specification (maxi-
mum force of the actuator and maximum displacement of
the moving body) of the AMD and the vibration control ef-
fct was studied. The vibration control effect was evaluated
using the amount of change in ride quality level \( L_T \) [7],
which is a power index of vibration acceleration weighted
by human sensibility. The evaluation function is shown in

\[
\Delta L_T = L_{TA} - L_{TP}
\]  

\( \Delta L_T \) denotes the amount of change of the \( L_T \);
\( L_{TA} \) and \( L_{TP} \) are the values of \( L_T \) under active and passive condi-
tion respectively. The ride quality is improved as the value of the
\( \Delta L_T \) gets smaller.

Since the characteristics of the test vehicle’s air springs
were changed from those in the excitation test described
in Chapter 2, the parameters of the numerical calculation
model were similarly changed. In the test vehicle, the nat-
ural frequency of the bouncing mode in which the carbody
vibrates as a rigid body on air springs is around 2Hz. In or-
der to reduce the influence of the bouncing mode, the mode
separator for B mode was changed into a form expressed as
follows.

\[
A_4 A_4 + A_4 - (A_2 + A_6)
\]

The actuators were modeled as force generated by a
moving body in the following calculations.

Figure 12 shows the relationship between the maxi-
mum force of the actuator and \( \Delta L_T \) obtained by numerical
calculation under the condition that the mass of the mov-
ing body is 40kg and the maximum displacement of
which the carbody is 500N. This figure shows that \( \Delta L_T \) becomes smaller
as the maximum force becomes larger, and \( \Delta L_T \) approaches
a specific value.

The relationship between the maximum displacement
of the moving body and \( \Delta L_T \) is shown in Fig. 13. The calcu-
lation conditions were that the mass of the moving body
was 40kg and the maximum force of the actuators was
500N. This figure shows that \( \Delta L_T \) becomes smaller as the
maximum displacement becomes larger.

Figure 14 shows the relationship between the mass of
the moving body and the maximum displacement required
for getting almost the same vibration reduction effect.

Based on the above calculation results, the specifi-
cation of the AMDs can be determined by means of the fol-
lowing steps:

1. Maximum force of the actuators is decided based on
   the vibration control effect expected.
2. Maximum displacement of the moving body is decided
   based on the size of the storage space.
3. Mass of the moving body is decided based on the
   amount of the maximum displacement.
3.3 A performance confirmation test on compact AMDs

Two compact actuators, both of which were commonly used as a motion exciter, were selected under the condition that the expected vibration reduction effect \( \Delta L_T \) was -3dB or less. Table 2 shows the specifications of these compact AMDs. In order to realize the targeted \( \Delta L_T \), maximum force was estimated at more than 200N according to numerical calculation. The actuators were of a size that allowed them to be placed under the seat of the commuter-type test vehicle. Although identical actuators were not available, two different types of actuators with similar specifications with regard to mass, maximum displacement of the moving body and maximum force of the actuators were selected.

An excitation test using the commuter-type test vehicle was carried out at the rolling stock testing plant at RTRI. The compact AMDs were placed under the seats on each side, around the center of the floor, as shown in Fig. 15. The same control algorithm as the above-mentioned numerical analysis was used.

The vibration reduction effect of the multi-modal vibration control using the compact AMDs was examined under all-wheel simultaneous excitation conditions. Figure 16 shows the test results under passive and active conditions. The figure shows that the peaks of acceleration PSD corresponding to B mode (11Hz) and D mode (7.2Hz) were reduced successfully under active conditions. It is shown that multi-modal vibration control was also possible by using the compact AMDs. In addition, the \( \Delta L_T \) value of this test was -3.1dB, which was the same value obtained by the numerical calculation. These results also demonstrated the

| Table 2 Specifications of the compact AMDs |
|------------------------------------------|
| Photograph | |
| Mass of moving body [kg] | 34 | 35 |
| Total mass [kg] | 62 | 50 |
| Maximum displacement [mm] | 25 | 25 |
| Height [mm] | 150 | 220 |
| Maximum force [N] | 490 | 490 |

Fig. 15 Setting condition of compact AMDs under the seat

Fig. 16 Vibration reduction effect on floor (f4L) by using compact AMDs (All wheel simultaneous excitation test)

(a) Simulated running excitation corresponding to 73.5km/h

(b) Simulated running excitation corresponding to 82.8km/h

(c) Simulated running excitation corresponding to 100km/h

Fig. 17 Vibration reduction effect on floor (f4L) by using compact AMDs
validity of the numerical analysis model.

As a next step, excitation tests simulating running conditions were performed. Since the vibration situation of the vehicle changed corresponding to the running velocity, three different velocity conditions (73.5km/h, 82.8km/h, and 100km/h) were adopted. The results of each test are shown in Fig. 17. The vibration reduction effect obtained by applying AMD is presented in the legend as the $\Delta L_v$ value. These figures show that the reduction effect of acceleration PSD was observed around 7Hz and 11Hz under every running velocity condition. These frequencies correspond to the natural vibration frequencies of D and B mode respectively, and the $\Delta L_v$ values were from -3.4dB to -2.6dB. This result shows that multi-modal vibration reduction and ride quality improvement were successfully performed by using two compact AMDs.

4. Conclusion

To reduce flexural vibrations which were known to have a major impact on ride quality, a multi-modal vibration control method using two AMDs was proposed, and the feasibility of the method was examined by carried out an excitation test using a commuter-type test vehicle. As the next step, a more practical AMD system was examined by means of numerical analysis and excitation tests. The results obtained by this study are summarized as follows:

1. The feasibility of the multi-modal vibration control of two typical flexural modes referred to as B mode and D mode was confirmed by means of excitation tests using AMDs each with a total mass of 210kg.

2. A sufficiently accurate numerical analysis model was constructed, and the steps to determine the specifications of the AMD system (maximum force of the actuator and maximum displacement of the moving body) were decided on performing various calculations using the analysis model.

3. Based on the numerical calculations, a more practical AMD system consisting of two actuators each with a mass of 50~60kg was selected, and simulated running excitation tests using the compact AMDs were performed at the rolling stock testing plant. As a result, multi-modal vibration reduction and ride quality improvement were successfully achieved.

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