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Preliminary Performance Test of Active Secondary Vertical Suspension using a Low Power Actuator on a Railway Roller Rig

Y. Sugahara

Vehicle Technology Development Division, Railway Technical Research Institute, Tokyo, Japan

Abstract

This paper investigates a method for effectively applying active secondary suspension to mitigate vibration in the vertical direction to improve ride comfort in railway vehicles. We focus on the fact that, in modern high-speed vehicles, vibrations of higher frequency contribute relatively more to vertical ride comfort than vibrations of approximately 1–2 Hz. To effectively reduce such vibrations, we propose the use of actuators that prioritize high responsiveness over high power.

To verify whether this concept is effective for real vehicles, we developed an actual prototype of a vertical linear actuator based on a linear motor system, applied it to a Shinkansen equivalent vehicle, and carried out vibration excitation tests on a vehicle roller rig. A modal skyhook control of the rigid-body mode and first-order bending mode was applied using four accelerometers mounted on the carbody floor and was particularly effective in reducing the vibration of the rigid-body mode of the carbody in the 0.5 – 4 Hz range and of the bending mode around 9 Hz. The maximum force of each actuator was approximately 1.8 kN (two actuators per bogie). In this case, there was no reduction in vibration reduction performance due to the response delay of the actuators, as observed when hydraulic actuators are used.

Keywords: active suspension, vibration control, vertical vibration, ride comfort, railway roller rig, secondary suspension.

1 Introduction

The lateral vibration control systems that are standard on Japanese Shinkansen trains have significantly improved ride comfort in the lateral direction [1, 2]. As a result, however, vertical vibration now tends to be perceived as relatively large. Consequently, measures to reduce these vibrations are needed.

In addition to lateral vibration control, various studies have also been carried out on vertical vibration control [3]. We have also studied vertical semi-active suspensions in primary and secondary suspensions, which are cost-effective and easy to implement. Vehicle running tests have confirmed that the proposed systems are effective in suppressing vertical vibration and improving ride comfort [4–6]. Secondary vertical semi-active suspension is in service on luxury sleeper trains (also known as ‘cruise trains’) and some sightseeing express trains in Japan [6]. However, active suspension is needed to achieve significant improvement in ride comfort.

Numerous theoretical investigations and excitation tests using scale-models in laboratories have been carried out on vertical active suspension of railway vehicles, and various results have been reported [3]. However, there are very few reports of practical studies using actual vehicles, e.g. vibration excitation tests or vehicle running test using actual vehicles.

Examples of valuable tests on active suspension carried out with full-scale vehicles in the past include those conducted by the Central Japan Railway Company and Kawasaki Heavy Industries (Kambayashi et al.) [7], and by KTH and Bombardier (Qazizadeh et al.) [8]. Both systems in these tests use hydraulic actuators to reduce vertical vibration of the carbody, and both have reported good vibration reduction performance. However, both papers also reported delayed actuator response (time lag) causing a slight degradation in vibration suppression.

In modern high-speed trains, especially in the center of the car body floor, the contribution to ride comfort is often greater for vertical vibrations in the frequency band above 2 Hz than for rigid-body-mode vertical vibrations around 1 Hz [4, 5]. We considered that an improvement in the vibration reduction effect in the frequency band above 2 Hz, even at the expense of a slight reduction in the vibration reduction effect of the rigid body mode around 1 Hz, could lead to superior vertical ride comfort. To this end, considering the results of the previous studies mentioned above, the influence of the small time lag is more important than the magnitude of the actuator force.

Therefore, we propose a change of concept and the use of vertical actuators, which despite a significantly lower force than other actuators employed so far, have a shorter time-delay for the active secondary suspension. To demonstrate the effectiveness of this concept practically, a low-force, short time-delay vertical actuator that can be applied to an actual vehicle is required. This paper thus presents our first prototype vertical linear actuators, developed to verify the validity of the required actuator ratings, and experimentally demonstrates the effectiveness of our proposal through the results of vibration excitation tests on a railway vehicle roller rig testing facility using these actuators.

2 System configuration and control law

2.1 System configuration

The system configuration of the secondary active suspension system is shown in Fig. 1. The system consists of four vertical accelerometers on the carbody floor, four vertical hydraulic dampers, four vertical actuators, and a controller.

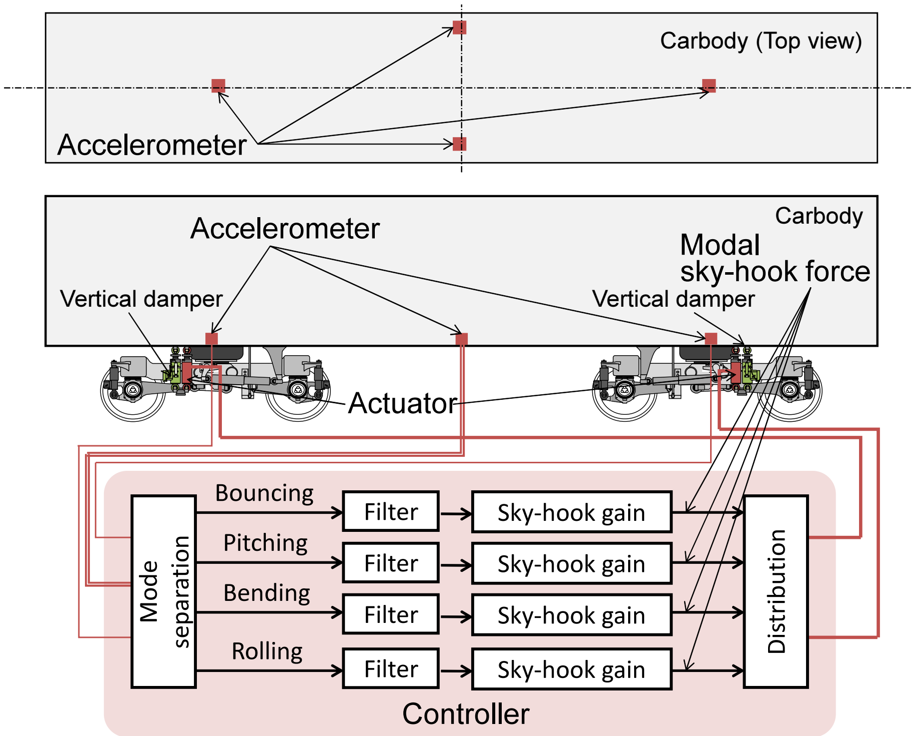


Figure 1: System configuration of secondary active suspension.

2.2 Linear actuator

Previously tested active suspension systems for railway vehicles have used pneumatic, hydraulic, electromechanical, or electrohydraulic actuators in the lateral direction [9–12], and hydraulic actuators in the vertical direction [7, 8]. These actuators were used because they all provide forces of 10-20 kN or more and do not have significant time delays, except for the pneumatic actuators. In contrast, in this case we introduce an actuator using a linear motor. Linear motors generally have a lower power density than rotary motors, making it difficult to obtain a large force. However, linear actuators have an advantage over other actuators in terms of response to small amplitudes and high frequencies, such as minimal mechanical rattle in the drive unit, no compressibility because no working fluid is used, and the same force characteristics can be obtained in expansion and contraction.

The first prototype linear actuator used in this test is shown in Fig. 2. The actuator has a maximum thrust of 2.5 kN and a mass of approximately 31 kg. This actuator has only 10–25 % of the force of actuators that have been applied to actual vehicles to date. The characteristics of the generated force (output) in relation to the desired force (input) are a gain reduction of less than 3 dB and a phase delay of approximately 40 deg at 20 Hz, and the actuator has sufficient responsiveness to control vibrations of approximately 10 Hz.

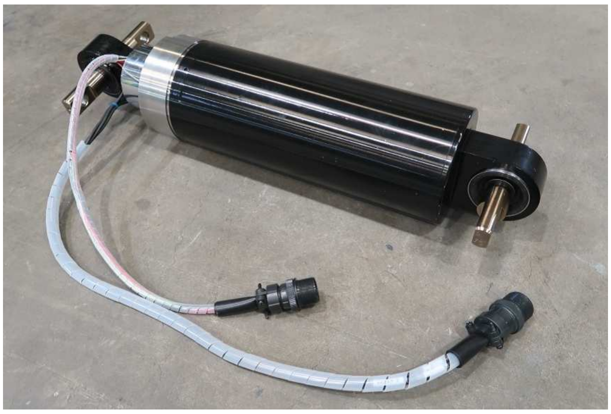


Figure 2: First prototype vertical actuator.

2.3 Secondary vertical damper

Air springs are widely used as secondary springs in modern passenger trains. Particularly in Japan, when air springs are used in passenger cars, an orifice is often provided in the air flow path between the diaphragm of the air spring and the auxiliary air chamber, and the basic damping of the secondary suspension is obtained by providing resistance to the air flow as it passes through the orifice [13]. On the other hand, when active suspension is in operation, the basic damping of the secondary suspension should be kept to a minimum to minimize vibration transmission from the bogie to the carbody.

To achieve this, the orifice of the airspring was removed and a hydraulic variable damper was installed in parallel with the air spring, so that the damping of the hydraulic damper could be switched to minimum during actuator control. When the actuator is uncontrolled, the hydraulic damper provides the basic damping of the secondary suspension instead of the air spring orifice. The first prototype variable vertical damper for a Shinkansen vehicle, as shown in Fig. 3 [14], was used in this study.

2.4 Accelerometer placement and control law

In previous studies, Qazizadeh et al. used four accelerometers to control only the rigid body mode vibration of the carbody [8], whereas Kambayashi et al. used six

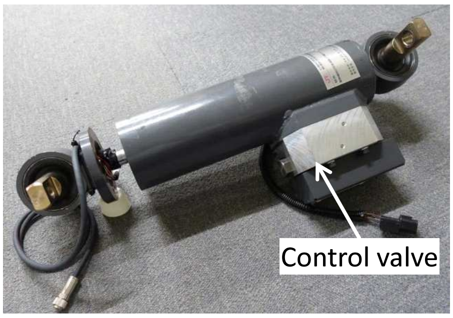


Figure 3: First prototype variable vertical damper for a Shinkansen vehicle.

accelerometers to control the vertical bounce, pitch, roll, and first-bending mode vibrations of the carbody [7]. In this study, a total of four accelerometers were mounted in the positions shown in Fig. 1 to control the vertical translation, pitch, roll, and first-bending modes of the carbody.

A modal skyhook control law was applied to control the actuators, as shown in Fig. 1. The vertical vibration acceleration of the carbody was separated into bouncing, pitching, rolling, and first-bending modes, and each component was integrated through a filter to obtain the vibration velocity for each mode. The vibration velocity is multiplied by the skyhook gain to calculate the skyhook force (desired force of the actuators) for each vibration mode, which is then commanded to the motor driver. This modal skyhook control method has been used in our vertical semi-active secondary suspension systems and is already in service on commercial vehicles [6].

3 Demonstration of system performance at the roller rig testing facility

3.1 Test conditions

Vehicle excitation tests using a test car were carried out at the RTRI railway vehicle roller rig testing facility to evaluate the performance of the active suspension with the prototype actuator. Figure 4 shows a photograph of the test vehicle, which is equivalent to a Shinkansen vehicle. The carbody is of single-skin aluminum alloy construction, with a car body length of 24.5 m, a distance between bogie centers of 17.5 m, and a sprung mass of approximately 32,000 kg, including steel deadweight. However, the car body was unusual in its construction: two half-carbodies made of different structures were welded together to form a single carbody. As a result, this carbody has vibration characteristics which differ from standard Shinkansen carbodies. Bogies with a wheelbase of 2.5 m for the Shinkansen vehicle were used. Adapters for mounting the actuator and vertical hydraulic damper were fitted to the carbody and bogies, and the actuator and vertical hydraulic damper were mounted in parallel with the air springs. As shown in Fig. 5, the actuator and the vertical hydraulic damper extend beyond the rolling stock gauge because no commercial railway line running was carried out in this test. The adaptor is sufficiently strong so no resonance frequencies exist, at least within the excitation frequency range.

Since disturbance is needed to simulate actual running, the vertical and rolling displacements of the roller rig, used as excitation of the vehicle, were estimated on the basis of Shinkansen line longitudinal and cross-level track irregularity data obtained from a track inspection car, and applied to each wheel with a phase difference corresponding to a running speed of 300 km/h. The maximum amplitude of the excitation displacement was limited to 90 % of the actual track irregularity due to the limitations of the rolling stock testing facility. The rotational speed of the roller rig was set to that of the vehicle running at 50 km/h to reduce the load on the axle bearings and the roller rig testing facility.

Figure 4: Test vehicle on railway vehicle roller rig testing facility at RTRI.

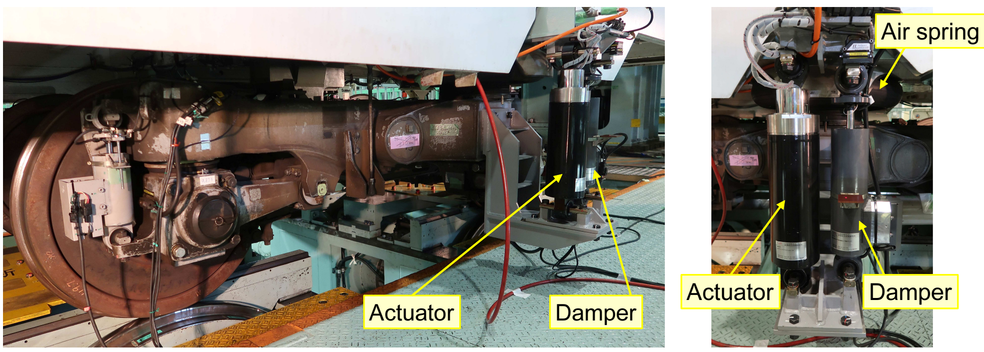


Figure 5: Installation of linear actuator and variable damper on test vehicle.

3.2 Test results

Vibration excitation tests were carried out under the conditions described in section 3.1. Figure 6 shows the power spectral density (PSD) of the vertical vibration acceleration and the roll angular velocity at the floor of the vehicle carbody calculated from the measurement results obtained in these tests.

Without control ('w/o control' in Fig. 6), significant PSD peaks were found at 1.1 and 9.1 Hz, both in the center of the carbody and just above the bogie. Figure 7 shows the carbody and bogie vibration shapes calculated from the vibration accelerations obtained from the accelerometers for measurement (not for control) on the carbody (13 points per carbody) and bogie (8 points per bogie). Figures 6 and 7 show that the PSD peak around 1.1 Hz is due to vibrations mainly in the bounce mode of the carbody (1.1 Hz) in fig. 7(a) and the pitch mode of the carbody (1.2 Hz) in Fig. 7(b). The PSD peak around 9.1 Hz is an elastic vibration with an anti-node at the center of the carbody, as shown in Fig. 7(c), similar to the first bending mode of the carbody but with a slightly more complex vibration shape.

When only the rigid body mode was controlled ('R-control' in Fig. 6), the PSD peak at 1.1 Hz was reduced to approximately 33 % at the center of the carbody and to approximately 17 % just above the bogie. Vibration reduction was generally achieved between 0.6 and 2 Hz (at the center of the carbody) and between 0.5 and 4 Hz (directly above the bogie), and the PSD was almost the same as that without control at higher frequencies.

Next, when the bending mode control was added to the rigid body mode control ('RB-control' in Fig. 6), a vibration reduction effect around 9 Hz was obtained that could not be achieved by the rigid body mode control alone, particularly in the center of the carbody, where the peak value at 9.1 Hz was reduced to approximately 16 %. The effect of reducing roll angular velocity was similar to that of reducing vertical acceleration just above the bogie, with both controls reducing vibration around 0.5–3 Hz, and the PSD values were almost the same as those without control at higher frequencies.

Bending vibration control was very effective in reducing vibrations around 9 Hz; however, vibrations increased around 12 Hz. These vibrations increased as the control gain was further increased or the maximum desired actuator force was increased. Figures 7(d) and 7(e) show vibrations at 12.1 Hz and 13.0 Hz, respectively, both of which have an anti-node in the center of the car body and just above the bogie, and a complex shape similar to that of a third-order bending mode. In the case of recent Shinkansen car bodies, there are few examples of such vibrations occurring significantly at approximately 12–13 Hz, which can be attributed to the unusual structure of the car body used in this study, as indicated in section 3.1.

The effect of the active suspension on ride comfort was evaluated using the 'ride comfort level (L_T)', which is a ride comfort evaluation index mainly used in Japan [15]. The weighted acceleration power was calculated for the vertical vibration acceleration PSD by applying the frequency weight function used to calculate the L_T(this frequency weight has been developed with reference to ISO 2631), and the results of the octave band analysis are shown on the right-hand side of Fig. 6. At the center of the carbody, almost all the power was accounted for by components in the 8 Hz band, with little contribution from other frequency components.

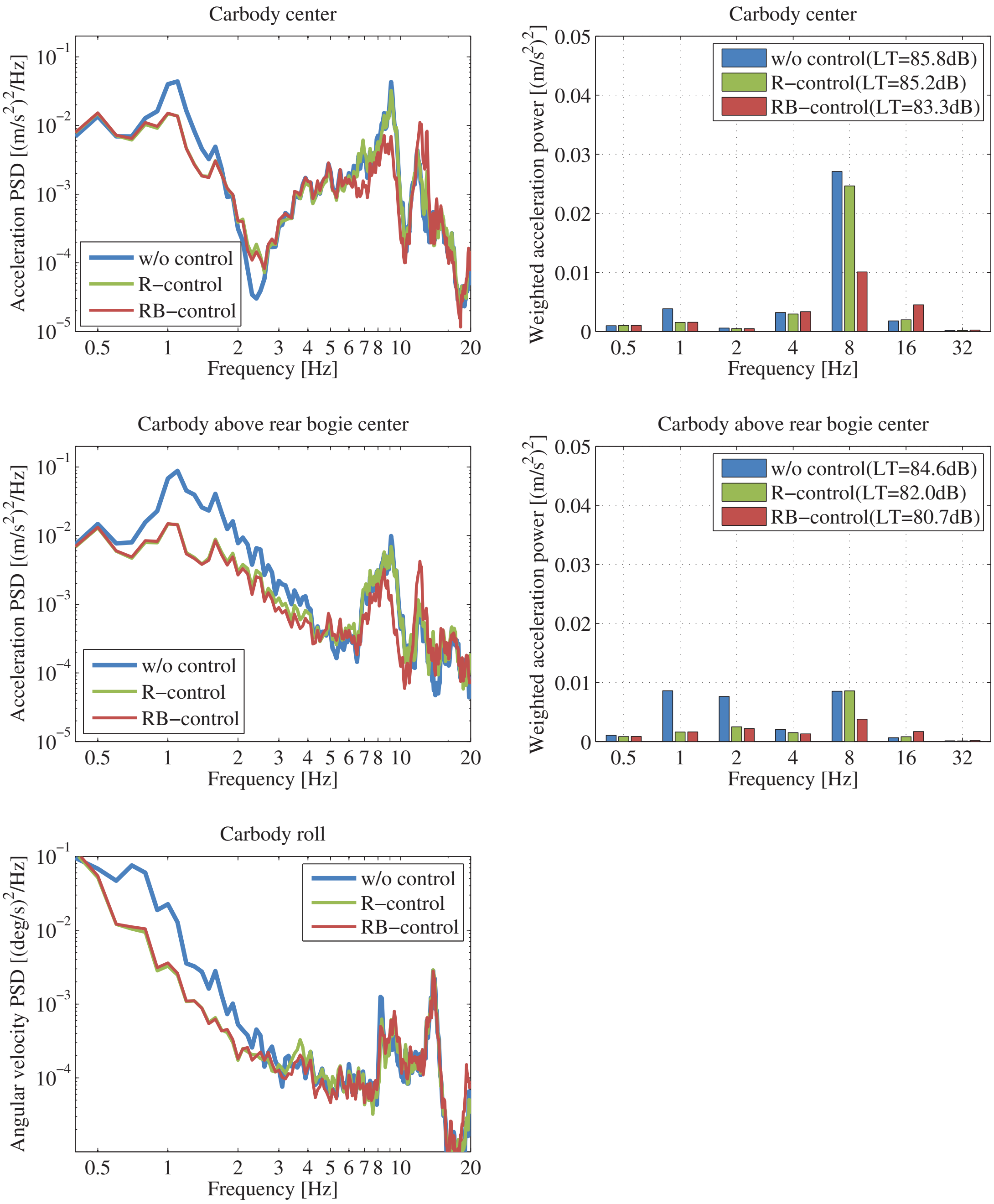


Figure 6: Measured results on carbody floor during excitation tests for vertical acceleration PSD, weighted acceleration power, and roll angular velocity (Simulated running excitation corresponding to 300 km/h).

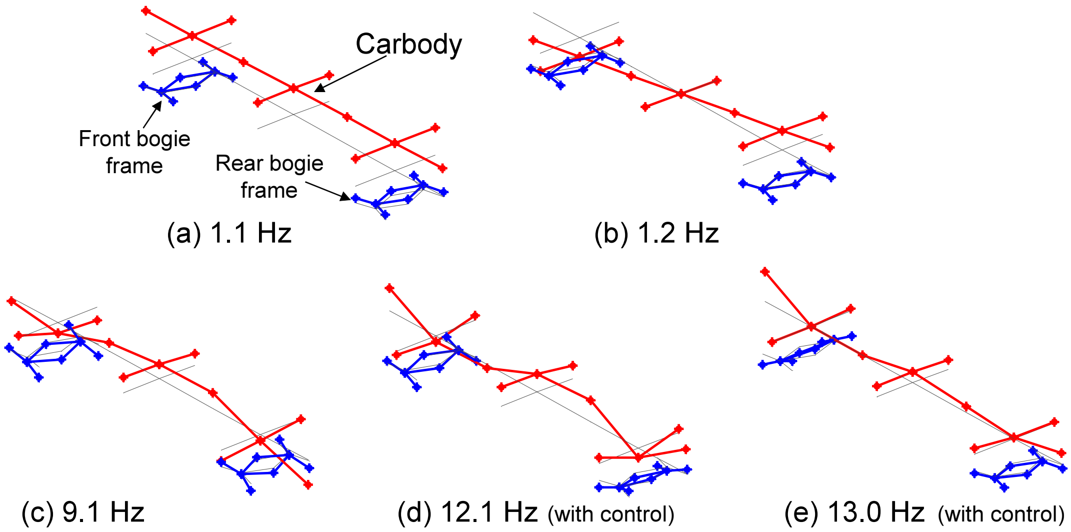


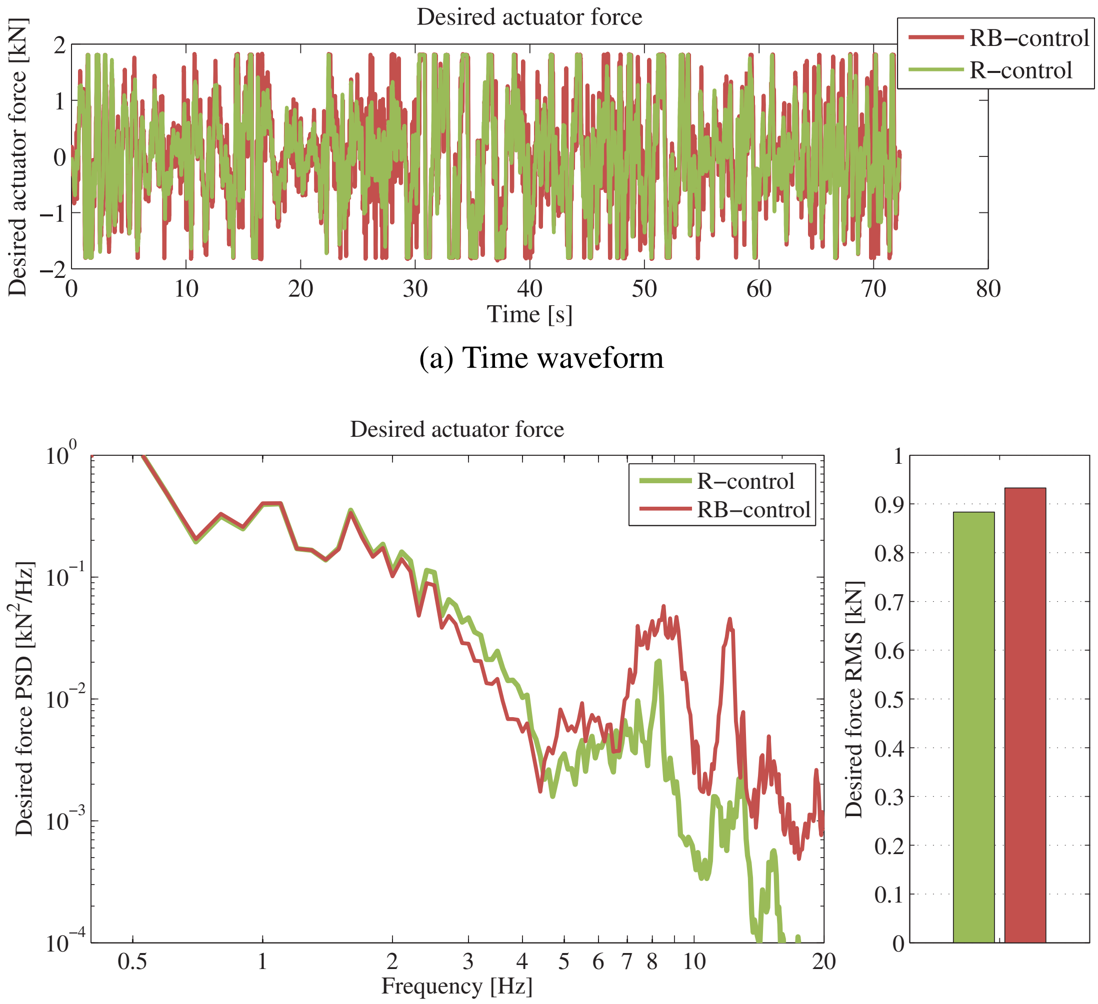
Figure 7: Mode shapes of test vehicle carbody observed during excitation tests.

Therefore, when only the rigid body mode was controlled, the total weighted acceleration power was hardly reduced, although the vertical body vibration around 1 Hz was reduced, and the $L_{\mathrm{T}}$was almost the same as in the case without control. When the bending mode was controlled in addition to the rigid body mode, the acceleration power in the 8 Hz band was reduced to approximately 40 % of that without control, resulting in a 2.5 dB reduction in L_{T}.

On the other hand, in contrast to the center of the car body, the weighted acceleration power was higher in the 1–2 Hz band than in the 8 Hz band directly above the rear bogie. Therefore, even when only the rigid body mode was controlled, the total weighted acceleration power was reduced and the L_{T}decreased by 2.6 dB. Furthermore, when the bending mode control was added, the component in the 8 Hz band was also reduced, resulting in a 3.9 dB reduction in L_{T}.

The actuator force was limited to 1.8 kN by the controller. Figure 8 shows that all the desired forces were limited to 1.8 kN and the ‘RB-control’ was based on the command values during ‘R-control’. From Fig. 8(b), the RMS values for both controls were below 1 kN, and the difference in RMS values with and without bending mode control was approximately 6 %. Comparing the desired force PSD, the ‘RB-control’ desired force is greater than the ‘R-control’ desired force in the frequency band above 6 Hz. This increment corresponds to the desired force in the bending mode control.

The above results show that the prototype actuator can be used to control rigid body mode and bending mode vibrations to improve ride comfort both in the center of the car body and directly above the bogie. The force required to control the bending mode was at most 6 % higher (RMS value) than that of the rigid mode, which considerably improves ride comfort with a small increase in power.



(a) Time waveform

(b) PSD and RMS

Figure 8: Measured desired force of rear actuator during excitation tests (Simulated running excitation corresponding to 300 km/h).

4 Conclusion

In this study, an experimental investigation of vertical active secondary suspension for railway vehicle was carried out. We focused on the fact that, in modern actual high-speed vehicles, the contribution to vertical ride comfort is relatively larger for vibrations of higher frequency than for vibrations of around 1-2 Hz. To effectively reduce such vibrations, we proposed the use of actuators that prioritize high responsiveness over high power.

To verify whether this concept is effective for real vehicles, an actual prototype of a vertical linear actuator based on a linear motor system was developed, applied to a Shinkansen equivalent vehicle, and vibration excitation tests on a vehicle roller rig were carried out. Modal skyhook control of the rigid-body mode and first-order bending mode was applied using four accelerometers mounted on the carbody floor. This was particularly effective in reducing the vibration of the rigid-body mode of the carbody in the 0.5–4 Hz range and of the bending mode around 9 Hz. The maximum force of each actuator was approximately 2.5 kN (two actuators per bogie), with the maximum force limited to 1.8 kN by the controller during actual use in this vibration excitation test. In this case, there was no reduction in vibration reduction performance due to the response delay of the actuators, as observed when hydraulic actuators are used.

In this test, elastic vibrations with complex shapes were generated, which are not common in common high-speed vehicle carbodies, due to the unique structure of the carbody used. The actuator is still in the process of being optimized for this system, particularly in terms of electromagnetics, and there is potential for further improvements in vibration reduction performance. Based on the test results, the actuator will be optimized, and the vibration reduction performance of the system will be evaluated in another vibration excitation test using a common high-speed vehicle carbody and the optimized actuator.

The following points should be noted: When traveling on sections with long-wavelength-large-track irregularities or at relatively low speeds running with large-track irregularities, the lower-frequency vibration component is relatively large. Under these conditions, the advantages of linear motor-based actuators cannot be exploited and power is likely to be insufficient, therefore another type of actuator with more power is suitable.

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