Development of torus-shaped elastic body as a vibration absorber for flexural vibration in railway vehicle carbody and its experimental validation using commuter-type vehicle

Takahiro TOMIOKA*, Satoshi TACHIKAWA** and Yuki AKIYAMA***

*Planning Division, Railway Technical Research Institute
2-8-38 Hikari-cho, Kokubunji-shi, Tokyo 185-8540, Japan
E-mail: tomioka@rtri.or.jp

**Former Graduate School of Engineering, Tokyo University of Agriculture and Technology
2-24-16 Naka-cho, Koganei-shi, Tokyo 184-8588, Japan

***Vehicle Noise and Vibration Lab., Railway Technical Research Institute
2-8-38 Hikari-cho, Kokubunji-shi, Tokyo 185-8540, Japan

Received: 5 August 2016; Revised: 12 November 2016; Accepted: 27 December 2016

Abstract

This paper presents a study to reduce flexural vibrations of rail vehicle carbody, which have large impact on ride quality, by utilizing deformation of an elastic body as a dynamic vibration absorber (DVA). Donut-shape (called "torus" in this study) is proposed as the elastic body considering practical application to rail vehicles. A numerical model to design an elastic torus using finite element (FE) analysis was developed, and it has been demonstrated that the shape and the size of a torus can be determined so as to have desired natural frequency by the FE model. A torus dedicated to a DVA for rail vehicles made of rubber filled with water was designed and manufactured actually. In order to validate the DVA effect of the water-filled elastic torus, a series of stationary vibration tests using a commuter-type test vehicle with stainless steel carbody was conducted. Since the tested vehicle showed two peaks on the frequency response function (FRF) measured on the floor and both of them should be reduced simultaneously to improve ride quality of the vehicle, the authors applied the elastic torus as "double-decker" style (stack a torus up on another torus). Both of the peaks were successfully reduced by applying the double-decker elastic torus. It has been confirmed that the multi-modal vibration reduction effect was achieved because a set of double-decker torus act like a two-degrees-of-freedom (2-DOF) system. Influence on the vibration reduction effect by the difference of the number of applied tori and their distribution on the floor were investigated.

Key words : Railway, Flexural vibration of railway vehicle carbody, Ride comfort, Damping, Dynamic vibration absorber, Vibration of elastic body

1. Introduction

It is widely acknowledged that reduction of flexural vibrations of carbodies in vertical direction is of considerable significance to maintain or improve ride comfort of rail vehicles. This issue is important not only for high-speed rail vehicles travelling over a long distance but also for commuter-type vehicles. According to the authors' previous studies, recent light-weighted and structurally-simplified carbodies tend to have complicated flexural modes, such as deformation of floor and roof in different ways or showing diagonal distortion of carbody cross section (Takigami and Tomioka, 2007, Tomioka, et al., 2012).

There are many difficulties in taking effective vibration reduction measures for those carbodies having a number of complicated flexural modes. The authors consider that it is important to suppress such individual deformation properties and they proposed a method for that by introducing some constraint members to inside of the cabin (Takigami, et al., 2010, 2014). However, the other options without additional members or modification of carbody

structure are also needed. Some vibration reduction measures have been proposed such as utilizing under-floor equipment as a dynamic vibration absorber (DVA; Ishikawa and Sato, 1991) or pasting damping layer consisting of viscoelastic and constraint layers on the carbody shell (Suzuki et al., 1997); however those measures were aiming to reduce specified single flexural mode. This study intends to propose a new multi-modal damping (vibration reduction) device without any additional structural members to the carbody.

The damping device introduced in this study is essentially a DVA utilizing elastic vibration of an elastic body, and our proposal is inspired by the damping effect due to passengers as reported in the authors' former works (Tomioka et al., 2007, Tomioka and Takigami, 2015). The effect of passengers is generally considered as additional mass in designing carbody structure, and natural frequencies of flexural vibrations of a carbody are believed to become lower as the number of passengers increases. Regarding this topic, Kaneda et al. conducted stationary excitation tests using a partial mock-up of a carbody and reported that the acceleration peak level of the floor was decreased with the increasing number of passengers, while the natural frequency of the floor was not changed largely (Kaneda et al., 2000). More detailed vibration measurement tests using several types of actual railway vehicles were carried out by the authors (Tomioka et al., 2007, 2015). They showed that passengers on the carbody behave not as additional mass but as damping. They also revealed that passengers have large and multi-modal reduction abilities of carbody flexural vibrations.

Those results suggest that, if you can appropriately mimic the damping capabilities, it will be an effective countermeasure against flexural vibrations of the carbody and thereby improve ride quality. This idea seems to have connection with so-called the Human-Structure-Interaction (HSI) (Sachse et al., 2003, Busca et al., 2014) which is known especially in the field of civil engineering. The concept of utilizing the HSI to reduce flexural vibrations of large structures such as rail vehicles is considered very unique and interesting subject; and as long as the authors know, no such proposal has been presented.

In this study, a donut-shaped (called "torus" hereafter) elastic body is firstly proposed as the DVA considering practical application; then investigations to develop design procedure of a torus to have desired natural frequency are carried out and a torus dedicated for rail vehicle are actually manufactured; and validation test of vibration reduction performance is conducted by applying the developed elastic torus using an actual rail vehicle.

2. Torus-shaped elastic body and its FE analysis

2.1 DVAs utilizing deformation of some elastic bodies and their suitable shape considering application to rail vehicles

In order to utilize deformation of an elastic or viscoelastic body as a DVA, it is necessary to tune its natural frequency to meet the frequency of target vibration. That is, a design method to determine the natural frequency of the elastic body is needed. There is no established method to calculate natural frequencies of arbitrary shaped elastic bodies. For a simple shape of the elastic body, one of the authors have proposed to use elastic circular cylinder, of which the side wall was made of rubber and filled with water inside, as a DVA against carbody flexural vibrations of rail vehicles (Iwata and Tomioka, 2012), and they showed a very simple formula which describes the relationship between the shape of the elastic cylinder and its natural frequency. They actually designed an elastic cylinder by using the formula and confirmed its DVA effect by using a scale model of a Shinkansen vehicle. According to their research, the elastic cylinder DVA works better as the aspect ratio (height versus bottom radius, in this case) becomes bigger. This means that if the bottom radius of the cylinder is fixed, the DVA effect is getting larger as the height becomes higher. However, such tall elastic cylinders are difficult to be applied to actual rail vehicles, so more realistic shape is needed.

In this study, the authors propose an elastic torus as DVA considering application to actual rail vehicles as shown in Fig. 1. The torus is a shape obtained by rotating an ellipse with two radii of a and b about the z axis. The distance between z axis and the center of the rotated ellipse O' is called rotating radius or major radius R . The authors think that such elastic torus has the following merits (Tomioka and Nihei, 2013).

- Effective: the cross-section deforms totally when the torus is excited vertically in z -direction, and this characteristics yields large effective mass which works as DVA.
- Stable: more stable than cylindrical or spherical elastic bodies when it is placed on the carbody.
- Compact: lower in height than cylindrical or spherical elastic bodies for a given volume.

- Adjustable: easy to adjust its vibration properties such as natural frequency and/or damping ratio by e.g. stacking them one by one, applying some additional mass or damper, etc.

Fig. 1 Donut-shaped elastic torus and its cross-section.

2.2 Development of FE analytical model to design elastic torus with desired natural frequency

Firstly, it was tried to develop a numerical model to design the shape and the size of a torus so that it has a desired natural frequency. Some store-bought torus-shaped air cushions made of rubber were used for a preliminary study. Some excitation tests of the cushions were conducted and the numerical models to express the measurement results were developed by using the commercial finite element (FE) software Nastran. Then a new original torus to have a desired natural frequency was designed by using the developed FE model.

Figure 2 shows the experimental set-up for the preliminary excitation test for the torus-shaped air cushions. We used three tori with different sizes (Okamoto 1541A, 1542A and 1543A, hereafter referred to as "S-size", "M-size", and "L-size", respectively) and their dimensions and mass (in case of being filled with water) are listed in Table 1. The tested torus was placed on the base made of acrylic plate which was attached to the electro-dynamic exciter (IMV VS-50-06). A band-limited random signal with a uniform frequency component in the range of 5-200 Hz was used to excite the torus, and the excitation duration was 120 s for each excitation test. Accelerations of the base plate and the surface of the elastic torus were measured using piezoelectric acceleration pickups (RION PV-94) as shown in Fig. 2. Then the frequency response functions (FRFs) between vertical acceleration of the base plate and the averaged vertical acceleration at the top of the torus (simple arithmetic mean of the vertical accelerations measured at the four points on the top of the torus as shown in Fig. 2(b) with red symbols) were calculated.

Next, the authors developed FE models representing the elastic tori used for the excitation tests. Figure 3 shows one of the FE models. The dimension values shown in Table 1 were used for the three torus models. Tetrahedral elements with ten nodes each and triangular elements with three nodes each were used to express water part and rubber part, respectively. Nodes on the contact area between water and rubber were shared by both of the elements. Each of the three models consisted of 45744 elements and 59002 nodes. Material properties adjusted using the measured data obtained with the L-size torus were applied to all the three models.

Fig. 2 Set-ups for the preliminary vibration test of the three different sizes of store-bought air cushion.

Table 1 Measured size and mass of the tori used for the preliminary study. (Filled with water with 10 kPa.)

	S-size (Okamoto1541A)	M-size (Okamoto1542A)	L-size (Okamoto1543A)
a mm	49.5	50.4	60.0
b mm	45.8	47.2	56.4
R mm	117.0	126.5	142.6
mass kg	5.26	6.04	9.43

Fig. 3 The FE model of the elastic torus.

Frequency response analyses were carried out by applying excitation acceleration in vertical direction to the fixed points with base plate of the torus models and the response accelerations at the torus top were calculated, then the FRF curves were obtained. The FRF curves calculated by the FE models are shown in Fig. 4 together with measured ones. We can see that there is a prominent peak, which is corresponding to the uniform vertical elastic mode of the torus with uniform cross sectional deformation, on each FRF curve. Although a small difference is observed between calculated and measured ones such as on the anti-resonance frequency, the agreement seems very well around the peak of 12.5 Hz of the FRF of L-size torus. For the other sizes, reasonably good agreement is also observed about the peak frequency, and therefore the accuracy of the developed model has been validated. This result indicates that once material properties are determined for specific size of torus, the vertical vibration characteristics at least the natural frequency of the uniform vertical mode of elastic torus can be estimated with sufficient accuracy using the FE model.

Next, the relationship between natural frequency and the size of elastic tori was investigated using the FE model. For simplification, supposing a torus with circular cross section ($a=b=r$ in Fig.1), volume of a torus V can be expressed as $V=2\pi^2r^2R$ which is known as Pappus–Guldinus theorem (Bronstein, et al., 1985). That is, if two of V , r and R are specified, the remaining value can be determined. Considering this relation and the material properties used in Fig. 4, natural frequencies of tori having fixed volume ($V=9.5\times10^{-3}$ m³) were calculated by changing radius of the rotated circle r . The relationship between r and natural frequency f is shown in Fig. 5(a). In addition, natural frequencies for the case that radius of major circle R is varied under fixed value of r were also calculated as shown in Fig. 5 (b). Clear linear relations can be observed in both cases. These results indicate that the shape and size of an elastic torus to have desired natural frequency can be determined by using the FE model.

Fig. 4 Comparison of experimental and numerical FRFs of three sizes of elastic tori filled with water. Material properties of the FE model were adjusted based on the measured FRF of L-size torus, and the same values were used in all FE models. Good agreement is observed about the peak frequencies not only for L-size but also M and S-size torus.

2.3 Development of elastic torus used for DVA

Since the existing tori used for the preliminary study were sold on the market as air-cushions and not assumed to be used with water filled inside, they had a problem to show awkward or non-uniform deformation when water was filled with certain pressure (larger than approx. 15 kPa). It should be noted that the natural frequency of the water-filled elastic torus can be changed according to the inner pressure of the water since the inner pressure is closely related to the

(a) Natural frequency f versus radius of rotated circle r under fixed torus volume of $9.5 \times 10^{-3} \text{ m}^3$.

(b) Natural frequency f versus radius of major circle R under fixed radius of rotated circle $r = 60 \text{ mm}$.

Fig. 5 Relation between natural frequency f and radius of rotated circle r or major circle R of the elastic torus obtained by the numerical calculation using the FE model. These linear relationships indicate that it is possible to design an elastic torus with arbitrary natural frequency under fixed volume by choosing r or under fixed radius of rotated circle r by choosing R .

mass of the water and the tensile force of the rubber skin. Therefore, a torus dedicated to DVA for rail vehicles was designed in this study. The developed torus was made of rubber and usable with water filled inside with inner pressure up to 20 kPa. Its size was determined based on the FE analysis described above.

Figure 6 shows the developed elastic torus. Measured dimension values in case filled with water with 10 kPa are as follows: $a=60.0 \text{ mm}$, $b=59.8 \text{ mm}$, $R=163.0 \text{ mm}$ and the mass is 12.0 kg (water: 10.6 kg, rubber and water injection valves: 1.4 kg). A prominent peak was appeared at 10.5 Hz on its FRF as shown in Fig. 7 when it was excited in the same way shown in Fig.2.

Fig. 6 Developed elastic torus made of rubber filled with water. (Mass of the torus is 12.0 kg including water with inner pressure of 10 kPa.)

Fig. 7 Measured FRF between base acceleration and averaged acceleration at the top of the developed torus.

3. Experimental validation of the DVA effect of the elastic torus by using a commuter-type vehicle

In order to validate the DVA effect of the water-filled elastic torus, a series of stationary vibration tests for a commuter-type test vehicle with stainless steel carbody were conducted. The length, width and height of the carbody were 19.5 m, 2.95 m, 2.67 m, respectively, and the mass of the carbody (not including the bogies) was approximately 12.5 tonnes. The wheelbase of the bogie was 2.1 m and the distance between front and rear bogie centers was 13.8 m.

Figure 8 shows the tested vehicle and the exciter used in the test. The carbody was excited using an electro-dynamic exciter (EMIC $\Sigma 9515$ -AB/AS) with band-limited (5-50 Hz) random signal, and its accelerations were measured by piezoelectric acceleration pickups (RION PV-85 and PV-94) attached to the floor and roof. Note that this stationary excitation testing was aiming at validating the DVA effect of the elastic torus against flexural vibrations of carbody, therefore, the rigid body vibrations (around 1-2 Hz) were out of consideration. The excitation force was applied to a cross beam near the center of a side beam of the carbody floor structure via a driving rod, and the driving force was measured by a load cell (PCB 208B03) inserted between the driving rod and the carbody.

A number of tori up to 20 were mounted on the carbody floor without any fixing. Each torus was filled with water with inner pressure of 10 kPa and the mass was 12 kg/piece. Figure 9 shows the acceleration measurement points on the floor and the distribution conditions of the elastic tori. The measurement results for carbody floor are expressed as FRF curves between excitation force and carbody accelerations measured at the point f4L (longitudinal center of the floor near side wall) and the point f4C (the center of the floor), hereafter in this paper. Here, considering the mode shapes shown later in Fig. 11, which are the target vibration modes to be reduced in this study, to place the elastic tori around the longitudinal center on the floor under the longitudinal passengers' seats of both sides was chosen as normal distribution condition as shown in Fig. 9 (b) and (d). For the case of double-decker torus in which an obvious vibration reduction effect was shown as will become apparent below, distribution condition Fig. 9 (c) was applied in order to check the effect of the number of tori; and conditions Fig. 9 (e) and (f) were added to see the influence of distribution.

Fig. 8 Stationary vibration test using a commuter-type test vehicle.

Fig. 9 Acceleration measurement points and distributions of the elastic tori on the floor.

It should be noted that the elastic tori were placed around the nodal points of the diagonal distortion mode in the case of (e); and in the case of (f), the tori were placed around the anti-node of the bending mode and node of the diagonal distortion mode. In this study, up to 20 tori were applied by considering the weight of four passengers (60 kg per person) as the maximum additional mass.

Figure 10 shows the FRFs for the case without torus. Two peaks are observed at around 7.7 and 14.8 Hz in the FRF curve measured at f4L. These peaks are considered corresponding to the different flexural natural vibration modes of the carbody shown in Fig. 11, and we can understand that those flexural modes have to be reduced simultaneously to improve ride quality of this vehicle.

Fig.10 Frequency response function (FRF) of acceleration on the floor versus excitation force. (Without torus.)

Fig. 11 Natural vibration modes and their natural frequencies of the carbody. (Without torus. Thin black line denotes undeformed shape.)

As already shown in Fig. 7, the elastic torus has a prominent peak around 10.5 Hz on the FRF curve. On the other hand, the target frequencies of the tested vehicle carbody are 7.7 and 14.8 Hz. By considering this situation, the authors applied the elastic tori on the floor as "double-decker" style (stack a torus up on another one) in addition to the normal style (called "single-decker", hereafter). Figure 12 shows the photos of the elastic tori mounted on the floor as the single- and double-decker styles. By introducing the double-decker style, it was expected that a set of two tori to behave a two-degree-of-freedom (2-DOF) system. Since each torus has almost the same natural frequency of 10.5 Hz, the 2-DOF system should have two natural frequencies, one of which is lower and the other is higher than 10.5 Hz. Acceleration pickups were attached on the surface of specified torus as shown in Fig. 12 (b) and (d) to measure vibration of the single- and double-decker torus.

Figure 13 shows the FRFs for the cases 20 tori were applied as single- and double-decker style. The distribution conditions were as shown in Fig. 9 (b) and (d). In the case of single-decker (green line), slight vibration reduction can be observed around 10-13.5 Hz; as for the two target peaks, almost no reductions are seen, but only a little frequency shifts are observed. On the other hand, both target peaks are reduced simultaneously when the elastic tori were applied as double-decker style.

In order to clarify the reason of the different effects between single- and double-decker tori, FRFs between floor acceleration (measured at the center of each specified torus) and the acceleration at the top of the tori (averaged value of the four points on the top of each torus, see Fig. 12 (b)(d)) were calculated as shown in Fig. 14. We can see that the single-decker torus has a single prominent peak around 10.5 Hz and the double-decker torus has two peaks around

(a) 20 tori placed normally (single-decker) around floor center.

(b) Vibration measurement of the single-decker torus.

(c) 20 tori around floor center.
(10sets of double-decker torus.)

(d) Vibration measurement of the double-decker torus.

Fig. 12 Inside of the cabin with 20 elastic tori applied as single- and double-decker style around the center of floor and acceleration pickups attached on the tori and floor.

(a) FRFs at measurement point f4L.

(b) FRFs at measurement point f4C.

Fig. 13 Acceleration FRFs on the floor versus excitation force for the cases with 20 tori. (20 single-decker or 10 sets of double-decker) around floor center.

Fig. 14 FRFs between floor acceleration (as reference) and averaged acceleration at the top of single- and double-decker torus.

6 and 15 Hz. This means that the set of double-decker torus showed the behavior like a 2-DOF system as it had been expected. As shown in Fig. 13, we can see that the FRF values of the carbody floor are reduced around the peak frequencies of each elastic torus, therefore, it can be understood that each elastic torus worked as DVA and the double-decker torus is effective against two different modes of carbody flexural vibration. It should be noted that the two peak frequencies of the double-decker torus are different from those of the 2DOF system consisting of simply combined with the two 1-DOF systems having natural frequency of 10.5 Hz.

We have found that a multi-modal vibration reduction can be realized by applying the elastic torus as double-decker style. The vibration reduction effect by changing the number and distribution of tori are investigated in the following part of this paper.

Figure 15 shows the FRFs when 8 and 20 tori (four and ten sets of the double-decker torus, respectively) were mounted as shown in Fig. 9 (c) and (d). We can see that the two dominant peaks of the FRF are reduced simultaneously as the elastic tori were mounted and the peak values decrease according to the increase of the number of torus. The total mass of the 20 tori was about 240 kg so the mass ratio versus carbody was less than 2% in this case.

Figure 16 shows the results when 20 double-decker torus were mounted with different distributions as shown in Fig. 9 (d) (around the floor center) and Fig. 9 (e) (at both end sections of the carbody floor). In the case of distributed both end sections, only the peak at around 14.8 Hz is slightly reduced because the peak at around 7.7 Hz is corresponding to the flexural mode in which the both end sections become nodal points (see Fig. 11). Note that the measurement shown in this figure was carried out on the day different from those for the other figures. Therefore, very slight differences are observed between the peak values etc. of this figure and those of Figs 10, 13 and 15, even in the same loading condition.

Figure 17 shows comparison of the FRFs for the cases of different distributions of tori in the across-the-width direction of carbody. Although the difference is small, the peak at around 7.7 Hz (corresponding to the diagonal

Fig. 15 Acceleration FRFs on the floor versus excitation force. Comparison of the cases with different number of double-decker torus mounted around floor center as shown in Fig. 9 (c) and (d).

Fig. 16 Acceleration FRFs on the floor versus excitation force. Comparison of the cases with different distribution of the same numbers of double-decker torus as shown in Fig. 9 (d) and (e).

Fig. 17 Acceleration FRFs on the floor versus excitation force. Comparison of the cases with 10 sets of double-decker torus mounted around floor center in the different manners as concentrated (Fig. 9 (f)) and separated to both sides (Fig. 9 (d)).

distortion mode) is reduced largely in case the tori were divided into both sides (red line, Fig. 9 (d)); and the peak reduction at around 14.8 Hz (corresponding to the bending mode) is slightly larger in case of concentrated around floor center (green line, Fig. 9 (f)).

Since the features of flexural vibration such as FRF and acceleration power spectral density (PSD) differs from point to point on the floor, the influences by different loading conditions are needed to be checked widely on the floor. Therefore, average of band limited RMS (root mean square) value of acceleration obtained from all measurement points on the floor is introduced. The average band limited RMS value is calculated as follows for each loading condition;

$$S_{\text{rms}} = \frac{1}{N} \sum_{n=1}^N P_{\text{rms } n} \quad (1)$$

where n and N denote each of the acceleration measurement points and its total number on the floor, respectively ($N=21$ in this paper); $P_{\text{rms } n}$ is the band limited RMS value (called BLRMS, hereafter) of measured acceleration at point n ; and S_{rms} is the averaged value of the BLRMS on the floor. Each BLRMS value $P_{\text{rms } n}$ is determined as the square root of the integral value of the acceleration PSD at point n with respect to frequencies between 5 and 20 Hz. The lower and upper frequencies of the integral calculation are chosen in consideration of the human sensitivity on the vertical acceleration and the frequency band in which the major flexural vibration modes of the tested car are included.

Figure 18 shows the average BLRMS ratios which are normalized as $S_{\text{rms}}/S_{\text{rms}}^{\text{empty}} \times 100[\%]$, where $S_{\text{rms}}^{\text{empty}}$ is the averaged BLRMS of empty case (without torus). From this figure, it can be found that the averaged vibration as a whole on the floor is reduced when elastic tori are applied as double-decker style. The concentrated distribution around

Fig. 18 Average of the BLRMS values for all acceleration measurement points on the floor. (Normalized as empty case = 100%).

floor center was the most effectively reduce the carbody vibration among the conditions applied here. Of course, it is needed further investigation using more realistic excitation condition to check the vibration reduction effect under actual running situation, since the vibration response of carbody is changed according to the excitation condition and the results shown above are different from the actual situation of running on rails. However, those results obtained here are important because it has been confirmed that the elastic torus mounted as double-decker style has multi-modal reduction ability against flexural vibrations of a rail vehicle carbody.

4. Conclusions

This paper has been presented a study to reduce flexural vibrations which have a large impact on the ride quality of a rail vehicle by applying some elastic bodies as dynamic vibration absorbers (DVAs). The results obtained by the study are summarized as follows:

- Considering practical application to rail vehicles, donut-shape (torus) has been proposed as the elastic body.
- It was tried to develop a numerical model to design an elastic torus filled with water by using finite element (FE) analysis, and it has been demonstrated that the shape and the size of a torus can be determined so as to have desired natural frequency by the FE model for Nastran described in this study.
- A torus dedicated to a DVA for rail vehicles was designed based on the FE analysis and manufactured the torus actually. The developed torus was made of rubber and usable with water filled inside with inner pressure up to 20 kPa. The mass of the torus was 12 kg/piece in the case water was filled with 10 kPa, and it had a natural frequency at 10.5 Hz.
- In order to validate the DVA effect of the water-filled elastic torus, a series of stationary vibration tests for a commuter-type test vehicle with stainless steel carbody was conducted. Since the tested vehicle showed two peaks on the FRF measured on the floor and both of them should be reduced, the authors applied the elastic torus as "double-decker" style (stack a torus up on another torus). Both of the peaks were successfully reduced by applying the double-decker elastic torus.
- It was confirmed that the multi-modal vibration reduction effect was achieved because a set of double-decker torus act like a 2-DOF system.
- Influence on the vibration reduction effect by the difference of the number of applied tori and the distribution on the floor were investigated.

The mass ratio versus carbody in this study was less than 2% when 20 tori were mounted. The authors thinks that it is very interesting and important to achieve a multi-modal vibration reduction in an actual rail vehicle equipped with DVAs with such a small mass ratio. Another important result is that the torus shape has flexibility in application to actual rail vehicles such as being able to be mounted in double-decker style as shown in this paper.

In the future work, the authors are going to confirm the effectiveness of the elastic torus under more realistic conditions on the rolling stock testing plant in the Railway Technical Research Institute which can simulate actual running situation. At that opportunity, the authors will also make comparison with the other shape(s) as DVAs. Although water was only used as injected material in this study, the authors are also going to try some more viscous material such as superabsorbent polymer as filling up materials.

References

- Bronstein, I.N., Semenzaef, K.A., Groche, G., Ziegler, V. and Ziegler, D., Taschenbuch der Mathematik (Translated version in Japanese), (1985), p.232, Morikita Publishing Co., Ltd.
- Busca, G., Cappellini, A., Manzoni, S., Tarabini, M. and Vanali, M., Quantification of changes in modal parameters due to the presence of passive people on a slender structure, *Journal of Sound and Vibration*, Vol.333 (2014), pp.5641-5652.
- Ishikawa, R. and Sato, Y., Decrease of vehicle body bending vibration by dynamic damper, *Proceedings of the 68th JSME Spring annual meeting (Vol. C)*(1991), pp.531-533, (in Japanese).
- Iwata, Y. and Tomioka, T., Reduction of flexural vibration of railway vehicle carbody utilizing multi-axis deformation of elastic body, *Transactions of the Japan Society of Mechanical Engineers, Series C*, Vol.78, No.794 (2012), pp.3393-3407 (in Japanese).

- Kaneda, T., Kobayashi, H., Oda, M. and Akiyama, S., Study of human body effect to vibration of railway vehicle body, Proceedings of the 75th annual meetings of JSME Kansai-section, No.004-1, March (2000), 2.29-2.30 (in Japanese).
- Sachse, R., Pavic, A. and Reynolds, P., Human-structure dynamic interaction in civil engineering dynamics: a literature review, Shock and Vibration Digest, Vol. 35, No.1 (2003), pp.3-18.
- Suzuki, Y., Akutsu, K., Maebashi, E., Sasakura, M. and Tomioka, T., Method for flexural vibration damping of rolling stock carbody, Quarterly Report of RTRI, Vol.38, No.3 (1997), pp.123-128.
- Takigami, T. and Tomioka, T., Modal analysis of recent railway vehicles, Proceedings of the 12th Asia-Pacific Vibration Conference (APVC2007) (2007), Paper No. 62.
- Takigami, T., Tomioka, T. and Aida, K., Improving the rigidity of railway vehicle carbodies using inner sub-frames, Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.76, No.765 (2010), pp.1115-1123 (in Japanese).
- Takigami, T., Tomioka, T., Aida, K., Akiyama, Y., Sato, Y. and Imaoka, N., Development of rails for hand straps improving carbodies' rigidity and ride comfort, RTRI Report, Vol.28, No.7 (2014), pp.23-28 (in Japanese).
- Tomioka, T., Takigami, T. and Suzuki, Y., Experimental study on the effect due to passengers on flexural vibration of railway vehicle carbodies, Proceedings of the 12th Asia-Pacific Vibration Conference (APVC2007) (2007), Paper No. 116.
- Tomioka, T., Takigami, T., and Aida, K., Modal vibration characteristics of flexural vibrations in railway vehicle carbodies, Proceedings of the 1st International Conference on Railway Technology: Research, Development and Maintenance (Railway 2012) (2012), Paper No. 33.
- Tomioka, T. and Nihei, T., Reduction of flexural vibration of railway vehicle carbody using deformation of elastic bodies (Design procedure of vibration properties of donut-shaped elastic bodies and their vibration reduction effects on a scale model of a Shinkansen vehicle), Proceedings of the Dynamics and Design Conference 2013 (D&D2013) (2013), Paper No. 244 (in Japanese).
- Tomioka, T. and Takigami, T., Experimental and numerical study on the effect due to passengers on flexural vibrations in railway vehicle carbodies, Journal of Sound and Vibration, Vol. 343 (2015), pp.1-19.