Experimental modal analysis of high-speed railway carriage

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ABSTRACT

Experimental modal analysis of high-speed railway carriage Body in White (BIW) with 7 meters long is presented in this paper. The validity of modal testing is verified by comparing mode shapes of the entire carriage with that of beam structures assembled at the bottom floor. Results show that, the demarcation frequency between low frequency vibration and high frequency vibration is 100 Hz. As below 100 Hz, all 6 global modes are included; while above 100 Hz, the mode count within a 1/3 octave frequency bandwidth is higher than 5, which means it could be regarded as high frequency range in statistic energy analysis. Results of modal testing can be provided to modify the finite element model of the carriage structure.

Keywords: Railway carriage, Modal Testing I-INCE Classification of Subjects Number(s): 13.4.1

1. INTRODUCTION

High speed train carriage body in white (BIW) is such a carriage that all metal components have been welded together, but moving parts (doors, hoods, and deck lids as well as fenders) , the electromotor, chassis sub-assemblies, or trim (glass, seats, upholstery, electronics, etc.) have not been assembled yet. Studying dynamic characteristics of such carriage is significant for noise and vibration reduction design. There are many methods for dynamic study, finite element method (FEM) is an effective one(1), which is also convenient for dynamic and acoustic optimization(2). However, the accuracy of FE analysis depends on the reliability of the model; in order to build such a valid model, it is necessary to obtain structural parameters such as modal frequencies, modal damping and mode shapes by modal testing. In engineering, the first several non-rigid modes are most important for estimating whether the FE model and the real structure are in good agreement, so they need to be identified accurately in experiment.

This paper focuses on experimental modal analysis of high speed train carriage BIW with 7 meters long, and details of modal testing and modal identification are introduced in the following sections.

2. EXPERIMENTAL MODEL

2.1 Model Set-up

The carriage BIW being studied here is cut off from a carriage with 25 meters long, where the width dimension and the height dimension are kept unchanged, so total size is 7m×3.2×m×2.4m. The model is flexibly supported in semi-anechoic room; entire structure and bottom floor are shown in figure 1 and figure2.

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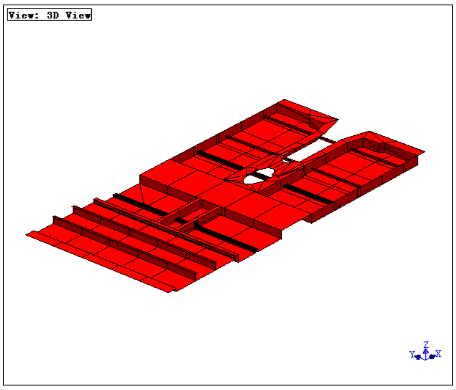
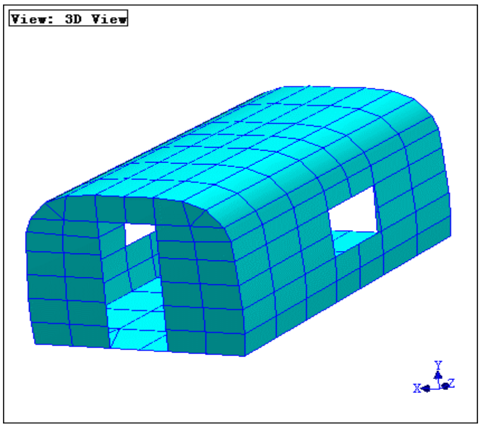
Figure 1 – 7 meters carriage BIW

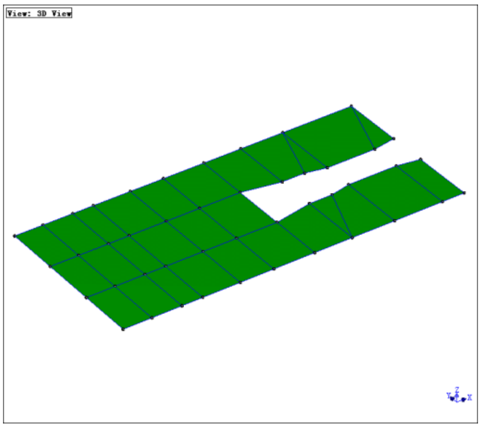


(a) beams under the bottom floor (b) Surface model of the bottom floor

Figure 2 – Beams skeleton and panels under the bottom carriage floor

It can be seen from figure 1 and figure 2 that, most of the model components are aluminium panels with smooth surface; however, the bottom floor is very complicated, with a skeleton containing different kinds of beams. Whether vibrations between beam skeleton and floor panels are in phase or not are unknown; besides, vibration response positions in global modal testing are more likely arranged on the internal surface of the carriage. In this case, local modal vibration of the floor skeleton may not be detected by global modal testing. Therefore, experimental modal analysis on both the entire model and beam skeleton should be studied respectively. Geometry model for modal identification need not to be as the same as the real structure, they can be simplified like figure 3.





(a) Simplified model of the carriage (b) Simplified model of floor skeleton

Figure 3 – Simplified model for modal identification

There are two main ways to excite the structure in modal testing, one way is shaker testing and another is impact testing(3). Shaker testing is used in this paper, and shaker type is HEV-1000 with available frequency bandwidth from 0 Hz to 800 Hz. Excitation position is arranged on the edge of junctions between bogie and bottom floor, as shown in figure 4. White noise is used as excitation signal, accelerometers are used to capture response signal, and transfer functions between response position and excite position are calculated for modal identified.

Figure 4 – Shaker excitation position

2.2 Measurement Point Distribution

The accuracy of mode shapes' detection relies mostly on measurement point's distribution(4). In general, on the condition that total weight of transducers does not affect the structure vibration, the more measurement points are arranged, the better mode shapes will be reconstructed. However, in many cases there are not enough accelerometers to measure the entire structure, thus it is necessary to reduce the number of measurement points appropriately. In order to identify mode shapes, distance between measurement points should not exceed 1/6 of the minimum transverse wavelength. Assumed that transverse wave speed c_s of aluminium panel is 3000 m/s, and the highest modal frequency f_{max} that needs to be identified is 200 Hz, it can be found that the max distance between measurement points is as follow:

$$l_{max} = c_s / 6 f_{max} = 2.5m \tag{1}$$

In this paper, measurement point distance is between 0.8-1.5m on global modal testing, with 36 accelerometers being arranged on the internal surface of the carriage, which will be enough number to detect modes below 200Hz. While on the test of floor skeleton, measurement points are increased to 45 as there are too many beams.

3. Modal Identification Result

Measurement data should firstly be transformed into transfer function of acceleration/force, and then be imported to ME' scope software for modal analysis. All vibration transfer function data are shown in figre 5 and figure 6.

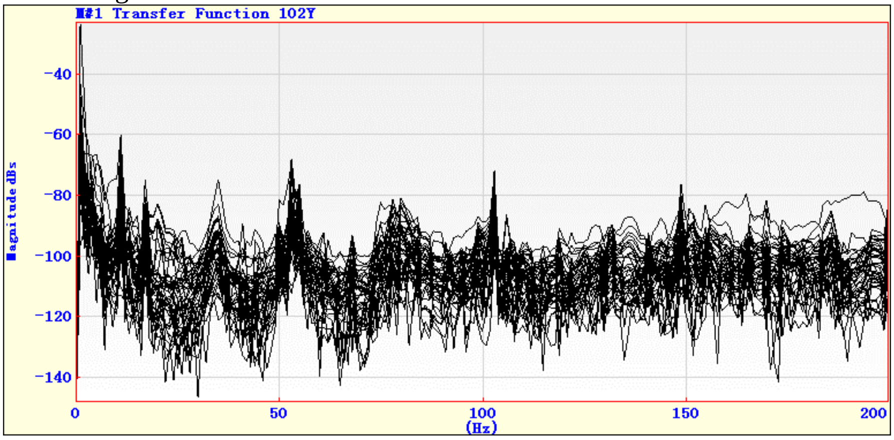


Figure 5 – Transfer function of acceleration/force of the carriage model below 200Hz

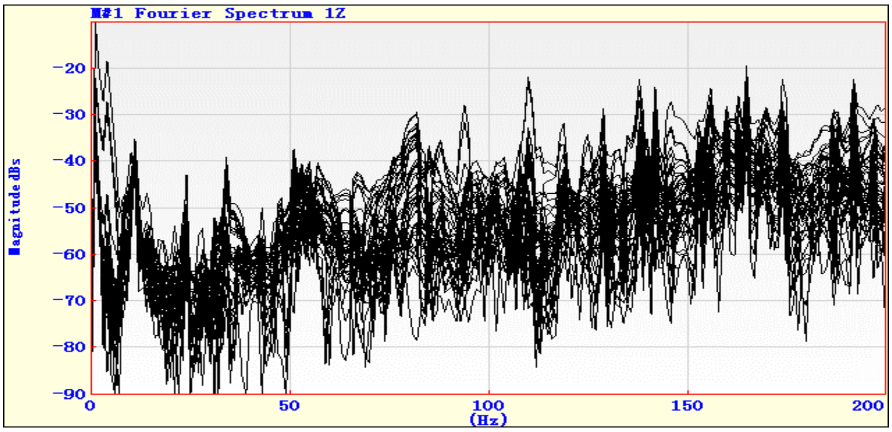
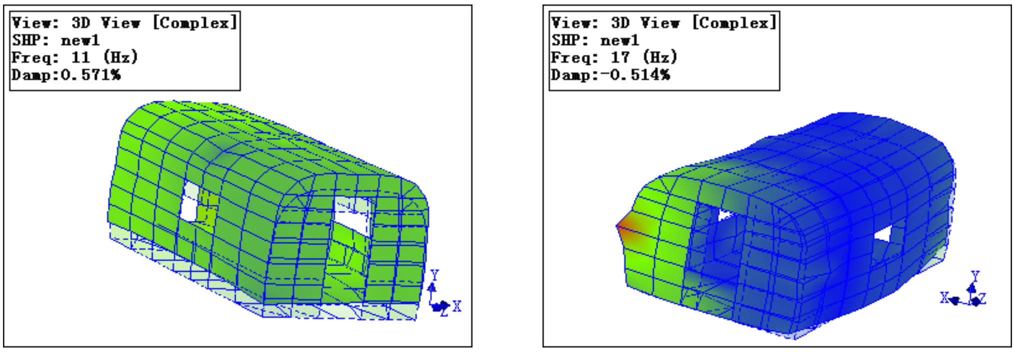


Figure 6 – Transfer function of acceleration/force of floor skeleton below 200Hz

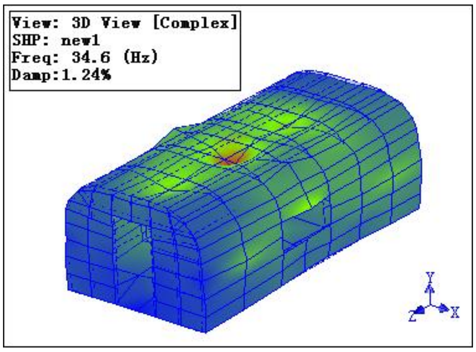
Based on the data of transfer function, modal parameters such as modal frequency, modal damping and mode shapes can be calculated. Results of modal frequencies and damping are shown in table 1 and mode shapes are shown in figure 7 and figure 8 below 100Hz.

Table 1 – Modal frequency and damping of two models below 100Hz

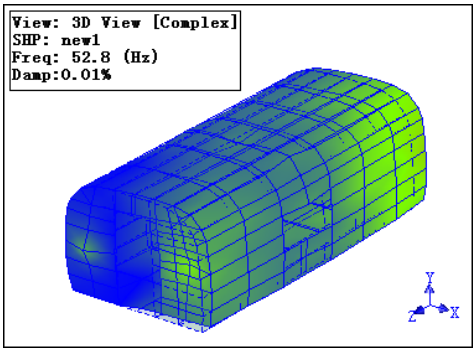
mode	parameters of carriage		parameters of floor skeleton	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
1	11	0.571	10.9	4.32
2	17	0.514		
3	34.6	1.24	32.9	0.997
4	52.8	0.01	51.3	1.16
			55.7	0.77
5	67.2	0.521	67	0.269
6	77.5	0.0262	80.5	0.742
7			86.8	0.154
8	98.6	0.188	95.3	0.0885



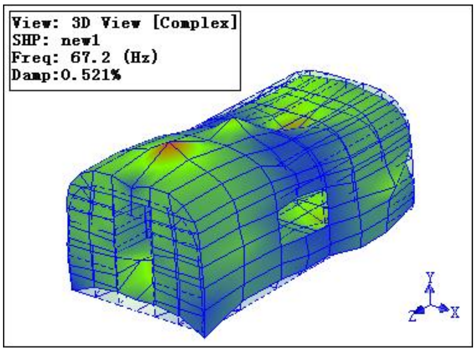
(a) Translational mode on Y axis,11Hz (b) Bending mode shape perpendicular to Y axis



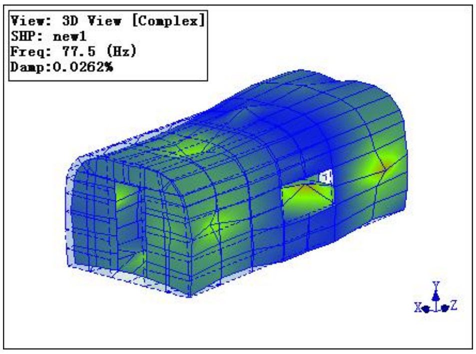
(c) Expansion-contraction mode shape



(d) Shear mode mainly on Z axis

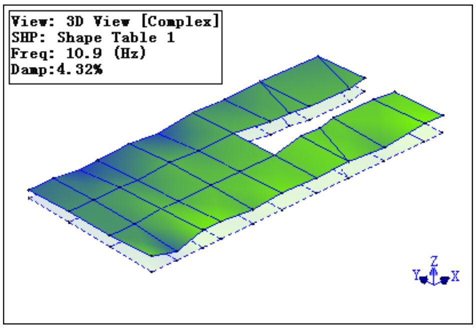


(e) Sinusoidal mode on XYZ

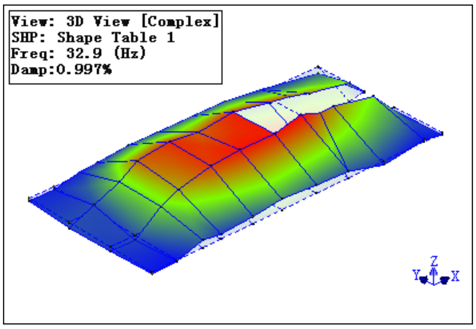


(f) Swallowing mode

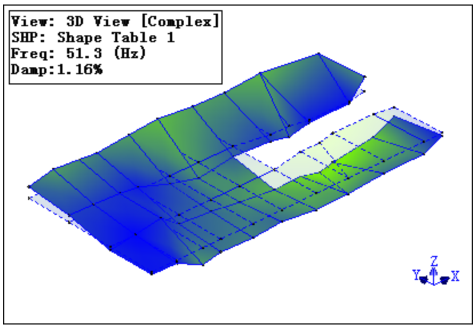
Figure 7 – Global mode shapes of the carriage



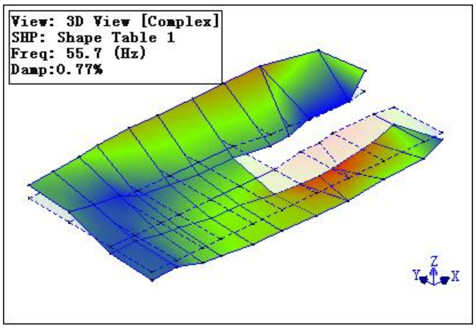
(a) Mode 1: 10.9Hz



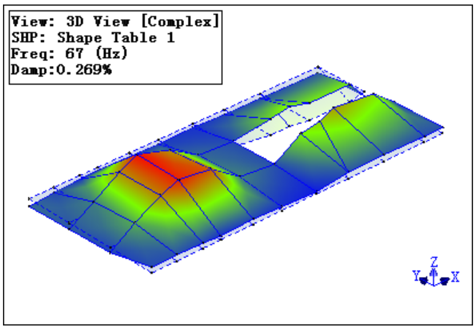
(b) Mode 2: 32.9Hz



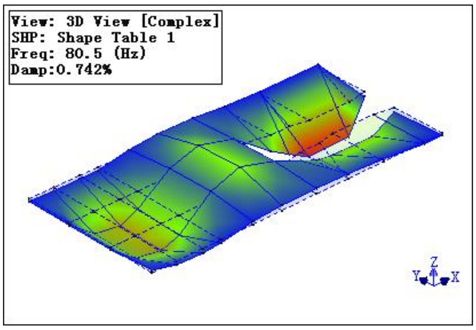
(c) Mode 3: 51.3Hz



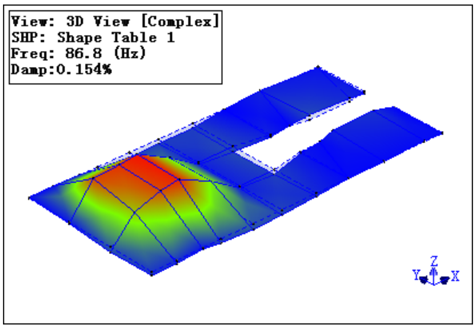
(d) Mode 4: 55.7Hz, the same as mode 3



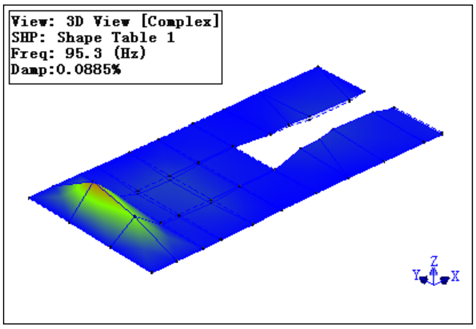
(e) Mode 5: 67Hz



(f) Mode 6: 80.5Hz



(g) Mode 7: 86.8Hz



(h) Mode 8: 95.3Hz

Figure 8 – Mode shapes of floor skeleton

It is observed from table 1, figure 7 and figure 8 that all 6 modes of the carriage below 90Hz are global modes; in these modes, beams under the carriage floor keep the same vibrating performance with floor panels although beam structures are various and complicated. Briefly speaking, mode 1 is a rigid mode and it can be ignored in FE analysis; global mode 2 at 17Hz is bending mode that vibrates parallel to floor plane, which means shear vibration on the floor may be strong but bending vibration is too weak to be detected, therefore, there is no identified result of bending mode on the floor skeleton at 17Hz, as seen in table 1 and figure 8. It can also be seen from figure 8(c), (d) that mode 3 and mode 4 of floor skeleton have the same mode shape, and difference between their modal frequencies is just 4Hz. In other words, modes of beam skeleton at 51.3Hz and 55.7Hz are actually the same mode and they all correspond to the 4^{th} mode of the carriage, as seen in figure 7(d). Results show that all identified modes of the carriage can be matched by modes of floor skeleton except the 2^{nd} mode, and their modal frequencies are also matched. As there are enough measurement points to identify mode shapes of floor skeleton accurately, and by comparing modal results of the carriage with that of floor skeleton, it can be easily found that identified modes of the entire model are dependable.

It should be particularly made clear that, although the mode of beams at 95.3Hz is a local mode and just one beam is vibrating, as seen in figure 8 (h), the relating mode of the carriage is still a global mode. That is to say, beam structures under the carriage floor begin to vibrate separately from the floor above 95Hz, and number of local modes becomes more and more. Particularly, mode number within one-third octave frequency bandwidth above 100Hz is higher than 5, as seen from table 2. That means vibration above 100Hz could be regard as high-frequency vibration in statistic energy analysis(5), and dynamic characteristics of the train carriage can be studied using SEA model.

Table 2 – Modal frequencies between 100-190Hz

mode	Modal frequency of the carriage/Hz	Modal frequency of the beams/Hz
1	106	102
2	111	109
3	119	110
4	136	115

5	141	120
6	146	125
7	151	138
8	155	143
9	162	147
10	166	152
11	174	156
12	178	163
13	186	170
14		174
15		177
16		179
17		185

4. CONCLUSIONS

As described in section 3, all 7 global modes of the carriage are included below 100Hz, while at the first 6 modes, all components make a global vibration, and beams under the carriage floor keep the same vibrating performance with floor panels although beam structures are various and complicated. However, local modes of the entire carriage and beam structures increase unexpectedly above 100Hz, and mode number within one-third octave frequency bandwidth is higher than 5. Results indicate that dynamic characteristics of the train carriage can be studied using FEM below 100Hz, and identified modes as mentioned above can be provided to modify the FE model of the carriage structure; while above 100Hz, SEA may be more efficient for dynamic analysis.

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