An investigation into high-speed train interior noise with operational transfer path analysis method

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Abstract Passengers' demands for riding comfort have been getting higher and higher as the high-speed railway develops. Scientific methods to analyze the interior noise of the high-speed train are needed and the operational transfer path analysis (OTPA) method provides a theoretical basis and guidance for the noise control of the train and overcomes the shortcomings of the traditional method, which has high test efficiency and can be carried out during the working state of the targeted machine. The OTPA model is established from the aspects of “path reference point-target point” and “sound source reference point-target point”. As for the mechanism of the noise transmission path, an assumption is made that the direct sound propagation is ignored, and the symmetric sound source and the symmetric path are merged. Using the operational test data and the OTPA method, combined with the results of spherical array sound source identification, the path contribution and sound source contribution of the interior noise are analyzed, respectively, from aspects of the total value and spectrum. The results show that the OTPA conforms to the calculation results of the spherical array sound source identification. At low speed, the contribution of the floor path and the contribution of the bogie sources are dominant. When the speed is greater than 300 km/h, the contribution of the roof path is dominant. Moreover, for the carriage with a pantograph, the lifted pantograph is an

obvious source. The noise from the exterior sources of the train transfer into the interior mainly through the form of structural excitation, and the contribution of air excitation is non-significant. Certain analyses of train parts provide guides for the interior noise control.

Keywords High-speed train · Interior noise · OTPA method · Acoustical source identification · Transfer path · Source contribution

1 Introduction

With the rapid development of high-speed train, serious noise and vibration pollution have been sensitive and significant issues influencing the passengers all the time. Eade and Hardy [1] studied the interior noise in 1970s and came up with guidelines for noise. Noise and vibration have been affected human's lives for a long time, especially in a relatively enclosed environment, such as in a carriage [2]. Numerous studies had been carried out to show the contribution and analyze the character and mechanism of interior noise, vibration and coupled dynamics [3–7]. Zhang et al. [8] studied influence of the wheel polygonal wear on the interior noise via test and simulation, and provided scientific basis to the wheel re-profiling.

The noise control of high-speed train starts from three primary steps: the source, the transfer path and the receiver. Receiver protection can only ostensibly lessen the impact of noise on the occupants. More attentions should be paid to the complex transfer paths of the sources of the carriages to meet the needs of comfortable riding.

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Actually, the complicated structure and coupling relationships between parts of the high-speed train make it hard to reduce the interior noise and vibration. The contribution amount of each part and transfer path of the noise and vibration needs to be acquired to provide more accurate measures of noise reduction of the high-speed train. Bendat [9] came up with the multiple input/output data process for system identification firstly in 1976. Afterward, the method was developed into engineering application gradually into operational transfer path analysis (OTPA), Gajdatsy and Lohrmann [10] compared classical TPA with OTPA and found out that cross-coupling between input signals led to misestimation of path contributions and numerical problems in the transfer function calculation. Lohrmann [11] compared the transfer functions of OTPA under artificial excitation with that under actual working conditions, and the results obtained showed that there was no significant difference between the composite results of the target response under runup and rundown conditions. Klerk and Ossipov [12] applied the OTPA method to calculate the contribution of vehicle noise source and vibration transfer path, which provided the guidelines to reduce the tire noise, and proposed the singular value decomposition (SVD) method to eliminate crosstalk cancellation (CTC) between input signals. A simulation model was established to verify the validity of the contribution results of the OTPA method by Sitter et al. [13]. The influence of the correlation between the input reference points on the accuracy of the transfer function was explored and the results showed that the more accurate transfer function can be obtained by CTC. Putner et al. [14, 15] used the artificial excitation on the vehicle, analyzed and found out the contribution of the noise sources. Roozen and Leclère [16] used hammer strikes as input signals of OTPA method, which could reduce the crosstalk between the input signals and save the time compared with the actual input condition. The OTPA method was applied in the field of ship noise, and the contribution of ship multiple engines, gearboxes and propellers to ship noise was analyzed by Keizer [17]. Lu et al. [18] studied the structure-borne noise transfer path on the powertrain. A piano was also analyzed with the OTPA method, and the vibration of the soundboard of the piano was found out to be dominant contributor by Tan et al. [19]. Vaitkus et al. [20] proposed an OTPA-D expression to solve some cases of obtained inaccurate results using a simplified test model, however, the accelerations put into use were twice at least than normal method.

Though the transfer path study was developed quite a long time and applied in many areas, while its application in rail vehicles began in recent years. Hardy and Jones [21] studied the British Railways and identified the sources of internal noise which the structure-borne noise from the bogie was a major contributor to the low frequency. Sievi

et al. [22, 23] applied the OTPA method in the field of rail transit to analyze the cause of excess noise in the passenger room above the trailer bogie without traction motor and investigated several train types to find out that the higher the maximum vehicle speed, the higher was the importance of structure-borne noise. Ström [24] studied the OTPA method as a reliable way to identify the noise source contribution via simulation, and applied it to the high-speed train at speed of 120 km/h below the frequency of 500 Hz, which results showed that the airborne noise contributes more in the high frequency bands, while the yaw dampers and traction rods contributed more in the structure-borne noise. Noh [25] applied the OTPA method on KTX high-speed train at the highest speed of 300 km/h, and analyzed the interior noise and vibration transfer paths and the contribution of the components of the train after all. Bouvet and Rissmann [26] studied the interior prediction model and found out that the structure-borne transmission dominates at lower frequencies (typically below 200–300 Hz). Zhang et al. [27] studied the interior noise transfer path and the contribution rate of structure-borne and airborne noise of a high-speed train at 310 km/h. Liu et al. [28] developed an anti-noise operational transfer path analysis to study noise generated in high-speed vehicles. Garg et al. [29] studied the interior noise of high-speed rail with their own app and spectral differences between high-speed and express trains in different lines. Song et al. [30] proposed an input decoupling analysis method into OTPA method and improved the accuracy of results in Chinese high-speed train.

According to the current research status of OTPA application, the method has been able to make quantitative analysis of the contribution of sound source to the receiver, but there are still few researches on the verification of the calculation results. Moreover, the definition of reference point of sound source could be further optimized for the analysis of noise transfer path, especially in a relatively complex structure. This paper would focus on the two points above to carry out research on the transfer path of vibration and noise of a high-speed train.

2 OTPA basic theory and quantification approach

2.1 OTPA theory

Classical transfer path analysis (TPA) is a test-based method to seek the vibration or noise that the sound source transmits through the structure or air to a specified receiver, associated with the source contribution. The total response is as follows:

$$Y(\omega) = \sum_i H_i F_i(\omega) + \sum_j H_j Q_j(\omega), \quad (1)$$

where $F_i(\omega)$ is the structural loads and $Q_j(\omega)$ is the acoustic loads; H_i and H_j are the transfer functions from the reference points to the receivers. The method of TPA focus on the measurement of the frequency response functions (FRF) and the load identification. However, some conditions limit the application of the TPA method. To avoid crosstalk between multiple transfer paths in the system, the frequency response functions of each transfer path between the excitation source point and the response point need to measure separately. Therefore, when there are multiple transmission paths between the source points and the response in the systems, the structure often needs to be disassembled. This also means a huge amount of testing work for a complex system, and since dismantling would change the boundary conditions of the system components. Moreover, the artificial excitation is not the actual excitation normally, and the accuracy of the frequency response function and the analyzed results could not be guaranteed.

The OTPA method makes up the deficiency of the traditional TPA. For this method could be used to measure the system's "source-transfer path-response" relationship and analyze the respective contribution in the actual operating condition of the system in the test, the process of load identification is eliminated. The test and analysis work are greatly simplified.

Based on the assumption of linear time-invariant system, the OTPA could expressed as

$$Y = HX, \quad (2)$$

where Y is the output of the response point; X is the input of the excitation point; H is the matrix of the operational transmissibility. For vibration and noise transfer path analysis, the input and output of excitation source could be in the form of vibration, force or sound pressure signal, etc. By measuring the physical quantities of the excitation reference points and response points under the operational conditions of the system, the input and response sample matrices could be easily constructed as

$$\begin{pmatrix} a_{11} & \cdots & a_{1k} & b_{11} & \cdots & b_{1m} \\ a_{21} & \cdots & a_{2k} & b_{21} & \cdots & b_{2m} \\ a_{31} & \cdots & a_{3k} & b_{31} & \cdots & b_{3m} \\ a_{41} & \cdots & a_{4k} & b_{41} & \cdots & b_{4m} \\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\ a_{n1} & \cdots & a_{nk} & b_{n1} & \cdots & b_{nm} \end{pmatrix} \begin{pmatrix} H_{a1} \\ \vdots \\ H_{ak} \\ H_{b1} \\ \vdots \\ H_{bm} \end{pmatrix} = \begin{pmatrix} Y_1 \\ Y_2 \\ Y_3 \\ Y_4 \\ \vdots \\ Y_n \end{pmatrix}, \quad (3)$$

where n is the total number of working conditions, k is the total number of vibration reference points, and m is the total number of sound reference points; Y_n is the response point under n th working condition; a_{nk} is the response of the k th vibration reference point under the n th working condition; b_{nm} is the response of the m th sound reference point under the n th working condition; H_{ak} is the transmissibility of the k th vibration reference point to the target point; and H_{bm} is the transmissibility of the m th sound reference point to the target point.

Therefore, the transfer function matrix of the system under working condition is

$$\begin{pmatrix} H_{a1} \\ \vdots \\ H_{ak} \\ H_{b1} \\ \vdots \\ H_{bm} \end{pmatrix} = \begin{pmatrix} a_{11} & \cdots & a_{1k} & b_{11} & \cdots & b_{1m} \\ a_{21} & \cdots & a_{2k} & b_{21} & \cdots & b_{2m} \\ a_{31} & \cdots & a_{3k} & b_{31} & \cdots & b_{3m} \\ a_{41} & \cdots & a_{4k} & b_{41} & \cdots & b_{4m} \\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\ a_{n1} & \cdots & a_{nk} & b_{n1} & \cdots & b_{nm} \end{pmatrix}^{-1} \begin{pmatrix} Y_1 \\ Y_2 \\ Y_3 \\ Y_4 \\ \vdots \\ Y_n \end{pmatrix}. \quad (4)$$

This matrix is non-square so the inverse is in a least square sense. The output contribution of each source and transfer path to the response point could be analyzed by matrix H quantitatively. Actually, there are often multiple energy transfer paths between the excitation point and response point in the application, which means the same input will generate responses on multiple paths/response points, namely crosstalk. Especially in the method of OTPA, the inter-coupling exists in the input and response matrixes, since the all the excitation and response are measured at the same time. Such crosstalk would lead to numerical problems in the process of matrix inversion, and the transfer relationship of each transfer channel could not be solved accurately. Therefore, the singular value decomposition (SVD) and principal component analysis (PCA) are applied in OTPA to achieve the CTC.

The response matrix of the reference point could be express as

$$\mathbf{X} = \mathbf{U}\mathbf{S}\mathbf{V}^T, \quad (5)$$

where $\mathbf{X}$ is an $r \times m$ response matrix of the reference point, $\mathbf{U}$ is an $r \times m$ orthogonal unit matrix, $\mathbf{S}$ is an $m \times m$ singular value diagonal matrix, and $\mathbf{V}$ is an $m \times m$ orthogonal unit matrix. Thus, in the OTPA method, it is required that the number of test sample r (number of working conditions) must be more than the number of input points m .

After the SVD is performed on the input sample matrix $\mathbf{X}$ to obtain the principal components (PCs) of the sample matrix, the rank of PCs is shown. In this process, the influence of background noise is reduced and the signal to noise ratio (SNR) is improved. Afterward, the PCs which have nothing to do with the response signal or have little influence can be considered as measurement bias or crosstalk, and it can be discarded and the corresponding singular value can be set to zero to improve the estimation accuracy of the transfer matrix. The PCA is used to reduce the influence of unwished frequency components, in order to reduce the crosstalk. The singular matrix $\mathbf{S}_r$ is expressed as:

$$\mathbf{S}_r = \begin{bmatrix} \sigma_1 & & & & 0 \\ & \ddots & & & \\ & & \sigma_r & & \\ & & & 0 & \\ 0 & & & & \ddots \\ & & & & & 0 \end{bmatrix}. \quad (6)$$

The input pseudo-inverse matrix is

$$\mathbf{X}^+ = \mathbf{V}\mathbf{S}_r^{-1}\mathbf{U}^T. \quad (7)$$

And, the achieved operational transfer relation matrix is

$$\mathbf{H}_r = \mathbf{V}\mathbf{S}_r^{-1}\mathbf{U}^T\mathbf{Y}. \quad (8)$$

Substituting Eq. (8) into Eq. (2), the contribution of each reference point and total synthesized response signal can be derived as

$$\begin{pmatrix} a_{n1} & \cdots & a_{nk} & b_{n1} & \cdots & b_{nm} \end{pmatrix} \begin{pmatrix} H_{a1} \\ \vdots \\ H_{ak} \\ H_{b1} \\ \vdots \\ H_{bm} \end{pmatrix} = \sum_{i=1}^k y_{na} + \sum_{j=1}^m y_{nb} = Y_n, \quad (9)$$

where y_{na} and y_{nb} represent the contribution of vibration and sound reference point under the n th operational condition.

Before the OTPA analysis, the selection of calculation working conditions should be carried out first. When

calculating working conditions, the number of working conditions should be guaranteed not less than the number of paths in order to ensure the invertibility of the transfer coefficient matrix, so as to improve the statically indeterminate degree of linear equations and improve the solution accuracy.

2.2 Quantification approach

Based on the OTPA principle above, the steps of quantification approach are as follows:

- (1) The signals of the target noise and vibration excitation source, the transfer path point and the response position of interest are measured, and the spectrum of the measured point is obtained by Fourier transform.
- (2) Establish the transfer path calculation network according to the requirement that the measuring points are selected as reference points. And, the reference excitation response matrix $\mathbf{X}$ and target response matrix $\mathbf{Y}$ are constructed by selecting the appropriate calculation conditions.
- (3) The SVD and PCA method are performed on the reference matrix $\mathbf{X}$, and remove the secondary components that represent channel crosstalk and measurement bias to achieve the transfer path matrix $\mathbf{H}_r$. Note that if the reference point matrix contains signals of different dimensions such as sound and vibration at the same time, the normalization processing is needed to prevent the error attenuation of the principal component.
- (4) A new set of measurement data is used to construct the input matrix, and the self-power spectrum of the synthesized signal is calculated by solving the transfer relation matrix to verify the correctness of the transfer relation matrix recognition.
- (5) The contribution of each reference point to the target point is calculated by the transfer relation matrix $\mathbf{H}_r$, and the contributions are superposed according to the transfer path.

3 Test and analysis of interior noise in high-speed trains

Noise is transmitted from each of these sources to the interior by both airborne and structure-borne paths [31]. The test of vibration and interior noise is an important means to understand the characteristics of the train and provide working condition data for subsequent train optimization design. Before using OTPA method to analyze the contribution of sound source, it is necessary to have a detailed understanding of train vibration and noise

characteristics and a clear understanding of the noise transfer path, so as to ensure the accurate construction of OTPA analysis model. The “TP03” carriage of interest is a test car of Chinese “Fuxing” high-speed train and the TP03 stands for the trailer vehicle with pantograph (the 3rd carriage from the driving direction), and the operational speed ranges from 160 to 350 km/h.

3.1 Measurement setup

In the manufacturing process of high-speed trains, the air tightness of the vehicle body is strictly guaranteed, so the path of external sound source directly into the passenger room through the gaps can be ignored. In other words, the external sound source of the vehicle is mainly transmitted into the vehicle through the path of sound radiation from the vibration of the panels, and the vibration of the panels in all directions in the passenger room is the main path of the noise from the target points. There are two main excitation sources for plate vibration, one is that the external vehicle noise source excites plate vibration through sound vibration coupling; the other is that the external vehicle vibration source directly causes plate vibration through solid vibration propagation. We will analyze the train body transmission path from the perspectives of in-room route contribution and external sound source contribution.

The noise or vibration response in the carriage can be regarded as the superposition of the contributions transmitted to the vehicle through different paths by various excitation sources outside the vehicle. Combined with the noise mechanism of high-speed train and the distribution characteristics of sound source, the noise transfer of the train body is simplified as showed in Fig. 1.

The distribution of sound measurement points in the carriage is shown in Fig. 2, according to the ISO 3381[32]. In order to understand the noise distribution along the length direction in the carriage, the measurement points are located at the front end (mic1), the middle part (mic2) and the back end (mic3) of the carriage, 1.2 m above the floor. And another measurement point is set in the inter-coach gap, 1.6 m above the floor (mic4), noting that the mic3 is above the bogie and beneath the pantograph.

A spherical array with 50 channels (B&K WA1565W004) was applied to achieve the purpose of interior noise identification on each measurement point in Fig. 2, as shown in Fig. 3. The spherical array has the same height as each measurement microphone point. Moreover, the access door keeps closed during the test, for the leak noise would greatly affect the measurement results.

It could be seen that the space inside the train can be divided into six areas, and the whole space can be regarded as a cube. The six outwardly extended cubes will divide the

Fig. 1 Schematic diagram of interior noise transfer path

sphere into six areas, and the grid points within the six azimuth areas will be determined. The 50 mics on spherical array surface could achieve the sound pressure via the beamforming method which could calculate the sound pressure of the space around, and the camera captured all aspects of the surrounding space. Unified spherical system coordinates implemented the correspondence between sound source and space position to achieve the goal of sound source localization and quantitative. The operational speed ranges from 160 to 350 km/h. The frequency band of interest is 20 to 1600 Hz.

3.2 Noise and vibration characters and transfer paths of high-speed trains

In this section, typical working conditions are selected to make a preliminary analysis of the reference point noise in the carriage, as well as the sound source noise such as

Fig. 2 Measurement point layout in the carriage

wheel-rail area, pantograph area and the side wall surface. The main influencing factors of the noise in the carriage are identified, and the distribution and variation characteristics of the noise are determined which provide help and guidance for the construction of OTPA analysis and transfer model.

3.2.1 Identification and characters of interior noise

The sound power at each grid point was calculated, and the sound power at each grid point in the corresponding region was superimposed according to the above regional division rules to obtain the energy contribution rate of the six directions of the speed from 160 to 350 km/h. Since the vehicle structure can be regarded as symmetric in the left and right directions, the contribution rate of the left and right side was combined into the contribution rate of the side in the following analysis. The interior noise identification in the back-end points with the condition of pantograph lifting was shown in Fig. 4.

As can be seen from the Fig. 4, the energy contribution rate from the sidewall is the largest, reaching about 25%. However, considering that the radiation area of the sidewall was twice than that of the other directions, the energy

contribution rate of the sidewall was about 12% and did not account for the main component. Without considering the sidewall area, the contribution rate in the interested frequency band of the top and the bottom ranked the first two positions, among which the top contribution rate was the largest. The contribution rate of the front was significantly lower than that of the rear; i.e., the noise from the middle of the carriage was significantly lower than that from the inter-coach gap. In addition, as the speed increases, the energy contribution rate of the bottom, sidewalls and the rear shows a trend of gradual decrease, while that of the top and front have a trend of gradual increase. Specific sound source location of the reference point mic3 in the carriage of 160 km/h and 350 km/h is presented in Fig. 5, where the first three peaks in the frequency domain are also presented with the overall sound intensity cloud map. The range of the sound intensity contour is 3 dB.

It can be learnt that the main noise sources are the sidewalls, rear and the bottom parts below the speed of 250 km/h. However, the main noise sources are the top, bottom and rear above 300 km/h. The highlight in 630 Hz shows that the wheel/rail noise plays a dominant role in all speeds. And, the aerodynamic noise plays an important part only in high speeds. In practice, the pantograph noise

Fig. 3 Spherical array of interior noise identification

Fig. 4 Energy contribution rate in each direction

would affect the interior noise in the center frequency of 200 and 250 Hz due to the vortex shedding at 300 and 350 km/h, respectively. The influence of sidewall becomes

significant at 350 km/h which indicate that the noise of the panels from each direction is obvious. That is because the air excitation caused vibration on the panels (airborne noise) turns into more and more serious with the increasing speeds. Measures of air excitation should be taken especially for high speeds.

3.2.2 OTPA approach

The results of the array could only present the distribution and the sound intensity inside the carriage, and further knowledge could be gotten from the OTPA method. The OTPA results show the quantitative numerical results of every defined sources which present the detailed contribution of each parts of the train. Based on the sound source identification results of the interior noise before, the contribution analysis by the OTPA method is further carried out in detail in this section. The spread of noise to the carriage would cause the vibration of the interior panels, and then the vibration radiation is transmitted to the interior noise (structure-borne noise) in the well-sealed

Fig. 5 Sound intensity maps of interior noise at 160 and 350 km/h

carriage. Therefore, the vibration of interior panel is the main source of noise. Setting the mic3 at the back end as the standard sound target response point, the OTPA analysis was carried out with the vibration of each directional plate on the cross section of the standard point set as the reference point response; the contribution of sound sources from all directions in the compartment could also be obtained. The calculation model of noise contribution

transfer path for the interior panels at back-end measurement points in TP03 is shown as Fig. 6.

The vibration measurement points of the panels and the noise measurement points in the inter-coach gap area are the reference points. The calculation process of OTPA is first to calculate the transmissibility and contribution of the reference point to the target point and then add the contribution of all the vibration reference points of the panels in accordance with the direction to get the total

Fig. 6 Calculation model of transfer paths in all directions

contribution of the vibration of the panel in a certain direction. In the process of establishing the transfer path model, some considerations are taken into account. Firstly, the contribution of the front part is not taken into account. The reason is that the sound source identification results show that the front part in the interested frequency band has the lowest energy contribution rate. The front corresponds to the direction of the middle of the carriage which has no main sound source distribution. Secondly, only the vibration of the left sidewall panel is applied to calculate the vibration contribution of the sidewall, for the train body is symmetrical in the structure, so the left and right sidewalls of sound source distribution could also be regarded as roughly the same. Moreover, the transmissibility to the target point obtained by using the vibration of the left panel has included the sum of the left and right contributions, and there is no need to distinguish the vibration on both sides. Finally, the noise in the inter-coach gap is greater than that in the carriage due to its weak sound insulation performance. Noise sources outside the train may also be transmitted to the end of the carriage through the inter-coach gap. Although the closing of the end door could reduce the noise in the inter-coach gap during the whole testing process, the contribution of the rear part is still large in the sound source identification results. The noise in the inter-coach gap still needs to be taken into account, since the noise in the inter-coach gap would be transmitted to the carriage through the vibration and sound radiation of the end door and the gap of the end door.

The comparison spectrum between the synthesized response of the target point and the measured response shows that the overall trend of the two curves is very close,

Fig. 7 Spectrum of the synthesized and measured results

and many significant peaks are in good fitting in Fig. 7. The red line represents the numerical superposition of the contribution from the sources to the target point, while the black line is the numerical value of the practical test. All these are non-weighted. The frequency bandwidth of this spectrum is 20–1600 Hz. Therefore, the following contribution analysis could be carried on.

3.2.3 Contribution results analyses of train panels

Figure 8 shows the mic1–mic4 results at all speed levels with the lifted pantograph.

The results indicate that mic4 has the largest sound pressure level in all speed levels, since the inter-coach gap is a relatively small reverberation field and has weak sound insulation measures. Of them, mic3 is the largest target point inside the carriage, for it gets more information components such as bogie and pantograph. Further, sound pressure level of mic1 is larger than that of mic2 and smaller than that of mic3, for it is above the bogie without the pantograph. It shows that mic2 has the smallest sound pressure level, for it is in the middle of the carriage and only shows the interior reverberation of the carriage. In order to refine the analysis object, mic3 is chosen to be further studied and it contains the signal of the bogie and the lifted pantograph (the following results are from mic3).

The distribution of interior sound sources is various at different speeds. Figure 9 shows the total contribution of sound sources from all directions to target point noise calculated by OTPA at different speeds.

In different operational conditions, the contributions of sound sources in all directions are different, and the variation trend of the noise with speed is similar to that of target point. At each speed, the contribution of the

Fig. 8 Measured results of target points

sidewalls is the largest, while the noise contribution of the inter-coach gap is the least. However, the rear part of the inter-coach gap is not the least contribution. The reasons for this are manifold, for example, the reflection of the end door on the sound wave may increase the contribution of sound power in the rear part. The vibration contributions of the floor and sidewalls are close, and it is obviously greater than that of the roof and inter-coach gap below 250 km/h. The increasing rate of the roof vibration contribution is obviously faster than that of the floor and sidewalls, and the gap is gradually reducing. The first two sound sources become sidewalls and the roof when the speed is above 300 km/h, and the contribution of the roof vibration is greater than that of the floor. The difference below 250 km/h from the array results in Fig. 4 might mainly be due to the calculation. The calculated OTPA data contain

Fig. 9 Source contribution in all directions at various speeds

all the sensors (acceleration and sound) in the test section. The sensors are set on the panels which are close to the roof, floor and sidewalls. However, the array microphones are totally in the interior sound field and the calculation is an average result of the area. It might fade the difference.

For instance, at 350 km/h, the law of overall noise and spectrum of vibration contribution of panel parts in each direction are carefully analyzed below. Another significant point to be considered is that the result of reference point response superposition is different from that of sound pressure level superposition. Since the responses of each reference point are all partially correlated, the total value of the response does not follow the principle of independent sound sources superposition and can only be superimposed with the amplitude with phase. Figure 10 shows the spectrum of the contribution of panels in each direction.

In the spectrum, the contribution of the sidewalls is the largest, the contribution of the roof is the second, and the contribution of the floor is the least, at the center frequency of 160, 200, and 315 Hz. The contribution of roof is significantly greater than that of floor and sidewalls, and similar conclusions are obtained for sound source identification before, at the center frequency of 250 Hz, which is due to the vortex shedding of the pantograph. At the center frequency of 630 Hz, the contributions are nearly the same. Actually, according to the analysis before, the vibration and noise in bogie part are obvious at 630 Hz center frequency band. Further study is still need to be done.

Fig. 10 Spectrum of the source contribution at 350 km/h in 1/3 octave band

3.2.4 Contribution result analysis of structure-borne and airborne noise in train

The contribution of noise transfer in the carriage is discussed in the previous section. The main source of noise in the carriage is mainly caused by the vibration of the interior panels. According to the classification of excitation types, the excitation sources of the train are mainly composed of airborne and structure-borne.

Only two significant sound sources, pantograph and bogie, are considered in the airborne sound source. These two sound sources are located in the upper part and the lower part of the car body, respectively. The sound source area is relatively independent, and the crosstalk between them can be ignored. Considering the symmetry of bogie structure, the vibration measurement points are only arranged in the center and left side under the car body. For the middle area, the lateral damper and the connection point between the traction rod and the train body are located near the center pivot, so these two paths are combined and only the vibration measured point of the traction rod is arranged. For the left area connection points, the anti-roll torsion bar, vertical damper, air spring and anti-yaw damper are all located in the narrow area at the end of the cross beam, so these four paths are combined and only the anti-roll torsion bar vibration measurement points is arranged. A newly comparison spectrum between the synthesized response of the target point and the measured response shows that the overall trend of the two curves is very close and the significant peaks are well matched. Substituting the airborne and structure-borne excitation into the transmissibility matrix, the result shows as Fig. 11.

It can be seen that the overall trend of the two spectrum curves is very close, and many significant peaks can be well matched, so the following contribution analysis can be carried out.

The curve of the contribution of airborne and structure-borne noise to the target point overall noise with speed is presented in Fig. 12.

In different operational speed conditions, the contribution variation trend of airborne and structure-borne noise of total noise at the target point is the same. The contribution of airborne noise is always less than that of structure-borne noise at all speeds. The contribution of structure-borne noise is close to the total value of target point noise, which is dominant below the speed of 250 km/h. With the increasing speed, the difference between the airborne and the structure-borne is decreasing and is reduced from 10.5 dBA at 160 km/h to 1.5 dBA at 350 km/h.

The following is also taking the operational speed of 350 km/h as an example due to the space limitation. For each correlated path, it should be analyzed in detail

Fig. 11 Spectrum of the synthesized and measured results

Fig. 12 Structure-borne and airborne noise changing at various speeds

combined with the spectrum of contribution. The airborne and structure-borne noise are shown in Fig. 13.

It can be observed that there are significant peaks of the structure-borne noise in 50 and 630 Hz, while the significant peaks of airborne noise appear in 250 and 630 Hz. The contribution of structural-borne noise is larger than that of airborne noise in almost all frequency bands, which also indicates that the interior noise in the carriage is dominated by the structure excitation in high-speed. Moreover, the almost identical spectrum shape for airborne and structure-borne is observed across the interested frequencies range and structure-borne noise is higher than airborne noise in the interested frequency band, which are unlike to the previous results in Ref. [31]. This might be due to the different train types and the fact that the airborne

Fig. 13 Spectrum of airborne and structure-borne noise in 1/3 octave band

transmission loss is improved for trains with a higher maximum speed, but the structure-borne transmission loss is not to the same extent [23].

The airborne noise is divided into pantograph and bogie part. The airborne noise is shown in Fig. 14.

From Fig. 14, the peaks of pantograph part appear in 250 and 630 Hz, while the peaks of bogie part appear in 160–250 Hz and 630 Hz. The total contribution and spectrum indicate that pantograph part is the main source of the airborne noise.

The spectrums of the contribution of each structure excitation source in vertical and lateral direction are given in Fig. 15.

From Fig. 15, the characteristics of vertical and lateral vibration contribution of the traction rod are similar, and significant center frequency bands are located in 160, 315 and 630 Hz. The contribution in 630 Hz is nearly the same as the structure-borne noise. The characteristics of vertical and lateral vibration contribution of the anti-roll torsion bar are slightly different. The peak in vertical direction is very significant at 630 Hz, but the peaks of other frequency bands are not obvious. The significant center frequency band of lateral direction vibration is 160 and 630 Hz, and the values of the two peaks are basically the same. The significant center frequency band of the pantograph base is about 250 Hz, and it is close to the value of structure-borne noise.

In order to clearly compare the contribution of each sound source in the significant frequency band of target point noise, Fig. 16 presents the sound sources contributions in the center frequency bands of 160, 200, 250, 315 and 630 Hz.

The sound sources in each band can be clearly compared in Fig. 16. The contribution of each sound source in

Fig. 14 Spectrum of airborne sources in 1/3 octave band

160 Hz ranking the first two is the traction rod in vertical and lateral vibration, and that the case of 315 Hz seems similar as that of 160 Hz. The contribution of lateral vibration of anti-roll torsion bar is almost the same as that of lateral and vertical vibration of the traction rod in 200 Hz. The pantograph base is the largest in 250 Hz, which indicates that the excitation on pantograph is obvious and it is largely due to the vortex shedding. The vibration of anti-roll torsion bar in vertical direction becomes the largest in 630 Hz.

4 Conclusions

This paper presents the interior noise character and the contributions of a high-speed train by applying the spherical array and the method of OTPA. The following conclusions can be drawn:

- (1) For consideration of symmetrical structure, the contribution of the left sidewall and the right sidewall was combined to obtain the largest contribution of the acoustic source in the direction of the sidewalls to the target point, followed by the contribution of the roof and floor. The vibration contribution of floor is obviously greater than that of roof in low speeds, while the roof vibration increases rapidly with the speed. And, the contribution of roof is greater than floor when the speed is above 300 km/h. Reasonable shock pad or damping measures would be taken within the economic cost.
- (2) In different speeds, the contribution of structure-borne noise is greater than that of airborne noise. However, with the increasing speed, the difference between the contribution of the airborne noise and the

Fig. 15 Spectrum of structure-borne sources in 1/3 octave band

Fig. 16 Contribution of different parts in remarkable frequency band

structure-borne noise decreases gradually. The noise of pantograph and bogie part is mainly transmitted to the train body panels through structure excitation and then into the carriage target point.

- (3) The main affected frequency band of each source is different. The traction rod dominates in 160 to 315 Hz. The pantograph base is only obvious in 250 Hz, noting that pantograph noise is only a typical character in TP03 carriage, especially with lifted pantograph. And, the vibration of anti-roll torsion bar dominates in 630 Hz.

The OTPA method is efficient in complicated experiments. However, the potential weakness is the calculated accuracy, especially when the sources are highly

correlated. More precise data achievement and processing would be the main object in further work.

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