Wheel/rail Noise above 10 kHz Generated on a Curved Sections

Tsugutoshi KAWAGUCHI

Takeshi SUEKI

Toshiki KITAGAWA

Noise Analysis Laboratory, Environmental Engineering Division

It is well-known that wheel/rail noise from 250 Hz to 4 kHz has a greater influence on wayside noise along railway lines. However, when a train runs on a gently curved section, noise due to wheel and rail vibrations above 10 kHz could have a greater contribution to the total wayside noise. This paper therefore investigated the contribution of the high-frequency noise to wayside noise on a high-speed railway line through field tests and static experiments. It was found that wheel noise above 10 kHz is generated mainly by the leading wheel on the outside rail of each bogie, and that the noise level depends on train speed.

Keywords: railway noise, wheel/rail noise, curved track, field test, static experiment

1. Introduction

Measurements and theoretical studies on wheel/rail noise have been carried out widely and thoroughly. The wheel/rail noise from 250 Hz to 4 kHz has a greater influence on wayside noise along railway lines. However, when a train runs on gently curved track, the noise due to wheel and rail vibrations above 10 kHz (referred to as high-frequency noise) could have a greater contribution to the total wayside noise. Kawaguchi et al. [1] reported that, during a train passage through a curve, the wheels on the outside rail are the dominant noise sources and, after a train has passed, the outside rail is predominant on the curved section of a low-speed railway line. Also, on a curved section of a high-speed railway line, high-frequency noise could be a greater contributor to total wayside noise [2]. However, the characteristics of this high-frequency noise remain unclear. In this paper, the authors investigated the contribution of high-frequency noise to railway noise in field tests and in static experiments through shaker excitation of a curved section on a high-speed railway line. Furthermore, the source of the high-frequency noise is localized in the field tests with a directional microphone.

2. Contribution analysis of high-frequency noise

2.1 Outline of measurements

(1) Field tests

Field tests were performed on a curved section of a high-speed railway line where high-frequency noise had been detected. The track was of ballasted construction with a curve radius of 2,500 m and continuous welded rail (60 kg/m). The measured cross-section was set in the curved section. Measurements of the wayside noise and rail vibration were performed while trains with 16 cars passed at approximately 220-270 km/h. Figure 1 shows the cross-section of the test site. The sound pressure was measured with 1/4" microphone (20 Hz-100 kHz), installed 3.5 m from the outside rail in the horizontal direction and at corresponding height of the center of the wheel axle. Accelerometers (1 Hz-20 kHz) were installed at the head, web and foot of both the inside and outside rails. The accelerometers on the head measured both vertical and horizontal vibra-

tions. The accelerometers on each web and foot measured horizontal and vertical vibrations, respectively. The measured data were converted into A-weighted sound pressure/vibration velocity levels for each 1/3 octave band frequency. Figure 2 shows the typical time histories of the sound pressure level with time weighting, F (FAST). It was found that the high-frequency noise was generated during the passage of the train. The time-averaged A-weighted sound pressure/vibration velocity levels were converted to the equivalent continuous sound pressure level and the equivalent continuous vibration level, respectively. The average time is shown in Fig. 2. Furthermore, the sound power per unit rail length was estimated by using the equivalent continuous vibration velocity. The surface area of the vibrating rail was represented by the area per unit rail length, provided that the rail vibrates uniformly in the direction normal to its surface. The acoustic radiation efficiency is assumed to be unity.

(2) Static experiments

Static measurements of the rail vibration and wayside noise were made during excitation with a shaker (referred to herein as "shaker tests"). The arrangements of the accelerometers and microphone were the same as those in the field tests in Subsection 2.1 (1). Measurements were performed with the track in unloaded condition, and the shaker made the rail vibrate properly up to 20 kHz. The wheel/rail interaction force is considered to have both vertical and horizontal components when a train runs on the curved track. The shaker was installed on the railhead of each rail alternately, and the rail was vibrated constantly in either the vertical or horizontal direction. To reduce the noise produced by the shaker itself, the shaker was

Fig. 1 Cross-section of test-site set up. The shaker on the railhead is installed only when a static experiment is performed

Fig. 2 Time histories of wayside noise 3.5 m from outside rail

placed 15 m from the measured cross-section. In addition, the shaker was enclosed in a box with sound absorbing materials on its inner surface. Under these conditions, the induced waves propagated along the rail, and the rail noise radiated from the propagating wave could be detected. The measured data were converted into A-weighted sound pressure/vibration velocity levels normalized to the unit force generated by the shaker for each 1/3 octave band frequency.

2.2 Method for determining the wheel/rail noise contribution

Figure 3 schematically shows the experimental method for determining noise contribution. The contribution of wheel/rail noise was determined as follows:

- (1) To determine the excitation direction, the results obtained in the field tests were compared with those in the shaker tests on the vibration distribution of each rail. The determined direction was expected to be the same as the predominant direction of the wheel/rail interaction force in the field tests.
- (2) By using the results obtained in the shaker tests, the transfer function between rail vibrations and rail noise was estimated from the results of the excitation direction determined in step (1). This function corresponds to the level difference between the sound power per unit rail length and rail noise.
- (3) By combining the results of the transfer function with the sound power of the rail obtained in the field tests, the noise radiated from the outside and inside rails in the field tests was estimated.

Fig. 3 Experimental method for determining noise contribution

- (4) The contribution of the wheel noise was estimated by subtracting the rail noise from the wayside noise measured in the field test. Furthermore, the noise contributions of the outside and inside wheels were determined by combining the results of the total wheel noise and the energy ratio between outside and inside rail vibrations obtained in the field tests.

2.3 Results

- (1) Sound power of rails during train passage

Figure 4 shows the frequency characteristics of the sound power of the rails during the passage of trains. These results show the arithmetic average of the sound power level for trains running at 230-239 km/h. The sound power of the outside rail is clearly greater than that of the inside rail. This suggests that the wheel/rail interaction force acting on the outside rail is greater than that on the inside rail. Similar trends were seen at other train speeds but they are not discussed here for the sake of brevity. It is suggested therefore, that the outside wheel is the dominant source of wheel noise in the band above 10 kHz. Furthermore, the predominant locations of the rail noise were the head and web of the outside rail.

- (2) Predominant direction of wheel/rail interaction force

Figure 5 shows the equivalent continuous vibration levels of the outside rail obtained in the field tests and the shaker tests. As an example, the direction of excitation on the outside rail in the 16 kHz band was derived as follows. When the outside rail is excited vertically [see Fig. 5 (a)], the predominant component in the shaker tests is vertical vibration in the rail head. However, the rail vibration induced by vertical excitation is not similar to the behavior in the field tests. This is because the level difference between the vertical vibration in the rail head and the horizontal vibration in the web in the shaker tests was by approximately 15 dB greater than that in the field tests. When the rail was excited horizontally [see Fig. 5 (b)], the level difference between these vibrations was similar to difference observed in the field tests. Moreover, the results obtained in the two types of test were similar since the vertical vibration in the foot is 10–20 dB lower than the vibrations in the head and web. Therefore, for the results in the 16 kHz band in Fig. 5, it was confirmed that the predominant direction of the wheel/rail interaction force is horizontal.

Fig. 4 Sound power level of rails at 230 to 239 km/h. The black and blue plots denote the results of outside and inside rails. ○, Horizontal vibration in the rail head; △, vertical vibration in the rail head; □, horizontal vibration in the web; ×, vertical vibration in the foot, respectively

(a) Vertical excitation

(b) Horizontal excitation

Fig. 5 Vibration distribution of the outside rail at 230 to 239 km/h. The black and blue plots denote the results in running and shaker tests. The legends are the same as those in Fig. 4

Table 1 gives the estimated predominant directions of the wheel/rail interaction force. A horizontal force is clearly generated on the outside rail above the 16 kHz band. The wheel/rail interaction force acting on the outside rail is greater than that acting on the inside rail. This suggests that the high-frequency noise is associated with the horizontal components of the wheel/rail interaction force.

(3) Estimated contributions to wheel/rail noise

Figure 6 shows the contributions to wheel/rail noise in the band above 10 kHz. This was estimated from the rail

Table 1 Predominant directions of the wheel/rail interaction force acting on the rail during train passage

		○: Vertical direction, ●: horizontal direction			
	Train speed [km/h]	1/3 octave band center frequency [Hz]			
		10 k	12.5 k	16 k	20 k
Outside rail	230-239	○	●	●	●
	250-259	○	○	●	●
	270-274	○	○	●	●
Inside rail	230-239	○	○	○	○
	250-259	○	○	○	○
	270-274	○	○	○	○

vibrations in the field tests and from the associated transfer functions. The dominant noise source is clearly the outside wheel. This trend was similar to a previous study on a low-speed railway line. It was found, in addition, that the dominant frequency of the wheel noise depended on train speed. Furthermore, the total energy was greater as the speed of the train is lower. This means that the wheel/rail interaction force depends on train speed. These also indicate that the wheels are the most important source of wheel/rail noise above 10 kHz in a wider speed range.

3. Source localization of wheel noise

The results of the previous tests demonstrate that the wheels on the outside rail are the most important noise source during the passage of a train. To develop an understanding of the high-frequency noise, an attempt was made to identify the properties of each wheel on the outside rail as it ran through a curve, which makes a significant contribution to total wayside noise.

3.1 Characteristics of paraboloidal apparatus

The type of high-speed train considered here has two-axle bogies, in which the distance between the two axles presents a 2.5 m wheelbase. In this train, the wheelbase is the shortest of all the distances between two neighboring wheel axles. It is therefore necessary to distinguish the noise generated from two wheels embedded in a bogie on the outside rail. In previous field tests with a narrow-gauge railway line, an omni-directional microphone was installed close to the track [1]. However, the measurements with an omni-directional microphone were not suitable for the present purpose. Therefore, a directional microphone, i.e. the paraboloidal apparatus, was adopted.

Figure 7 shows a photograph of the paraboloidal apparatus, which comprises an acoustic mirror with a paraboloidal shape and a 1/4" omni-directional microphone (20 Hz-100 kHz) set at its focal position. The diameter of the mirror was 0.4 m. The difference in sound pressure level as measured by the present apparatus and by an omni-directional microphone at the same measuring point in a free sound field is referred to as the gain factor. In Fig. 7, the gain factor measured when the point source is set parallel to a straight line passing through the front of the apparatus (x -axis) in an anechoic room [3] is also shown. The

Fig. 6 Contributions of each component to the sound pressure due to wheel and rail vibrations above 10 kHz band. The size of each pie chart denotes the magnitude of total noise in the band above 10 kHz. The sections in the pie charts denote the energy ratio of each noise source in relation to total noise

Fig. 7 Parabolic apparatus and directional pattern in 10 kHz band

focal position of the apparatus was set 4 m away from a point source, the same distance as in the field tests. The spatial resolution of the apparatus was defined as half the distance between two source positions at which the gain factor drops by 10 dB (referred to as the resolution width). Figure 8 shows the frequency characteristics of the gain factor and the resolution width. The theoretical values of the parameters were calculated in reference to Ref. [3]. The gain factor obtained in the experiment exceeded 20 dB. The resolution width agreed well with the calculation, and the resolution width above the 10 kHz band was smaller than the wheel radius ($= 0.43$ m). This confirms that the apparatus is suitable for noise source localization.

3.2 Field tests

(1) Outline of sound-localization field tests

The field tests were performed on the curved section of track described in Section 2.1. The paraboloidal apparatus was installed 4 m from the center of the outside rail and at the height of the center of the wheel axle. The measured results were then converted into A-weighted sound pressure levels for each 1/3 octave band frequency band. The equivalent continuous sound pressure level L_{eq} was obtained during the passage of one wheel, and the averaging time was the passage time when the wheel ran in the range of ± 1.25 m (half of the wheel base) of the measured cross-section along the track.

Fig. 8 Directional characteristics of parabolic apparatus

(2) Results and discussion

Figure 9 shows the time histories of the sound pressure levels measured using the paraboloidal apparatus with the time constant of 5 ms. Peaks can be seen at the leading wheel axes of each bogie in the time histories of the overall levels including the frequency components above 10 kHz band. These peaks are divided into two parts. This is due to the fact that the wheel noise was screened by the axle box and the axle box suspension, and that the sound pressure was canceled out on the axis of wheel axes by the interference of sound wave radiated from wheel vibrations [4]. The scale of the axle box was twice the order of the resolution width (see Fig. 8 (b), 0.25 m-0.375 m) of the paraboloidal apparatus above 10 kHz band. At these peaks, the difference between the overall level and the partial overall level below 8 kHz was more than 10 dB. This also indicates that high-frequency noise has a significant influence on overall noise level. In Subsection 2.3 (3), the predominant noise source above 10 kHz band was the outside wheels during the passage of a train. Fig. 9 shows that the predominant source in the band above 10 kHz is the leading wheel on the outside rail of each bogie. The noise contribution of the following wheel to the overall noise is much lower than that of the leading wheel.

Figure 10 shows the values of L_{eq} for each 1/3 octave band frequency band during the passage of one wheel. These results show the arithmetic average values of L_{eq} from trains running at 230-239 km/h. At the leading wheel of each bogie, L_{eq} was greater than for the following wheel on the same bogie in the band above 12.5 kHz. This means

Fig. 9 Time histories of the wayside noise by the paraboloidal apparatus with the time constant of 5 ms at 232 km/h. —, Partial overall level below 8 kHz; - - -, overall level. Bottom vertical lines represent the center positions of the wheel axes

Fig. 10 Equivalent continuous sound pressure level at 230-239 km/h during the passage of one wheel

that the predominant noise source in the band above 10 kHz is the outside leading wheel. Figure 11 shows Leq in the 16 kHz band during the passage of one wheel. Leq tends to be greater as the train speed is lower. This trend is more apparent for the intermediate cars (cars No.4-14). This suggests that the high-frequency noise depends on the axle position of the outside wheel and also on the train speed. This also indicates that (i) the wheel/rail contact behavior depends on the axle position and (ii) the contact properties between the outside leading wheel and the rail depend on the train speed.

Authors

Tsugutoshi KAWAGUCHI
Assistant Senior Researcher, Noise Analysis Laboratory, Environmental Engineering Division
Research Areas: Railway Noise

Takeshi SUEKI, Dr. Eng.
Senior Researcher, Noise Analysis Laboratory, Environmental Engineering Division
Research Areas: Railway Noise

Fig. 11 Equivalent continuous sound pressure level at different train speeds in the 16 kHz band during the passage of one wheel

4. Conclusions

The authors investigated the contribution of the wheel/rail noise above 10 kHz to wayside noise by carrying out field tests and static experiments with shaker excitation of a curved section on a high-speed railway line.

The findings were as follows:

- The wheels on the outside are the dominant noise sources during the passage of a train. This trend was also observed in a previous study on a low-speed railway line.
- The high-frequency noise is associated with the horizontal component of the wheel/rail interaction force, and the predominant sources of rail noise were the head and web of the outside rail.
- Source localization with a directional microphone showed that wheel noise above 10 kHz is generated mainly from the leading wheel on the outside rail of each bogie. The noise was found to depend on the axle position of the outside wheel and also on the train speed.

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Toshiki KITAGAWA, Ph.D.
Senior Chief Researcher, Head of Noise Analysis Laboratory, Environmental Engineering Division
Research Areas: Railway Noise