Influence of rail fastener stiffness on railway vehicle interior noise

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Abstract

More attention has been paid in recent years to the interior noise of railway vehicles. It has been observed that the interior noise can increase in some locations where vibration-isolation measures are used in the track structures. In order to assess the influence of vibration isolation measures on the noise levels inside railway vehicles, a field measurement campaign has been carried out. The vehicle interior noise has been measured when a train is running at different speeds over the same non-ballasted track section fitted with two types of rail fastener of different stiffnesses. Additional measurements of axlebox vibration, train floor vibration, exterior noise and rail vibration are used to investigate the influence of the fasteners further. The experimental results are compared with simulations performed using the TWINS model, considering the wheel/rail interaction, by focusing only on the relative differences between the two fastener systems. The axlebox vibration and rail vibration are predicted for a unit roughness input and the differences in rolling noise are also obtained. The predicted differences in axlebox vibration, rail vibration and rolling noise are in broad agreement with the measurement results. The results show that the fasteners with a lower stiffness cause a noisier interior environment. Around 125 Hz and in the frequency range 315 to 1000 Hz, the noise levels are higher for the more elastic fastener, with an average level difference of 3 dB in the latter frequency range. It appears from the shape of the level difference spectra that airborne noise has most influence between 100 and 400 Hz and structure-borne noise has more influence between 500 and 1000 Hz.

Key words: Elastic rail fastener; interior noise; stiffness; railway track; acoustic response

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1. Introduction

Modern railway trains are required to provide their passengers with a comfortable ride in order to maintain commercial competitiveness: an important aspect of such comfort is the internal noise and vibration. However, less attention is often paid to the vibration and noise within railway vehicles than to the external environmental noise [1].

The most important source of noise in railways is the rolling noise caused by wheel and rail vibration. For trains in tunnels, however, the main environmental impact of running trains is ground-borne noise, caused by vibration propagating through the ground to nearby properties, where it radiates low frequency noise. Decreasing the stiffness of the track is one of several countermeasures deployed against ground-borne noise [2, 3]. In [4] it is shown from measurements that, when the track support stiffness is reduced, lower levels of vibration on the floor of the tunnel are clearly seen. Various types of vibration-isolating track-forms are used in practice, including resilient rail fasteners, under-sleeper pads, under-ballast mats, booted sleepers, or floating slab tracks [5, 6]. Each mitigation measure on track has a particular insertion loss and effective frequency range [6]. In these studies of the vibration reduction, the interior noise has not been taken into account.

However, many people complain that the noise inside the trains becomes louder in particular areas where vibration reduction measures are installed. Wang et al. [7] presented measurements of in-car noise and floor vibration from a metro line with different track systems. It was found that the in-car noise was much higher on floating slab sections and there was a difference of about 4 dB between a type of soft fastener and the standard baseplate track. Higher noise levels were found on curved sections and in at least one case this was associated with rail corrugation.

In recent decades, researchers have begun to pay attention to the interior noise problems in vehicles, including investigations of source mechanisms [8]-[12], transfer path analysis [13] and interior noise evaluation. Eade and Hardy [8] discussed the mechanisms by which noise reaches the interior of a rail vehicle, including airborne and structure-borne paths. They pointed out that the noise spectrum inside modern trains is dominated by low frequency components due to the increased isolation against airborne sound transmission and the increased acoustic absorption at high frequencies. The most important sources of interior noise include wheel/rail rolling noise, traction noise and noise from fans including the heating, ventilation and air-conditioning system; at high speeds aerodynamic noise is also important. Noise is transmitted from the various sources to the interior by means of both airborne and structure-borne paths. Structure-borne noise tends to dominate the lower frequencies and airborne noise the higher frequencies [5]. The airborne paths involve transmission through the floor, walls, windows, doors and the gangway connection as well as through gaps in the door seals.

Kim et al. [14] evaluated the interior noise of an urban railway vehicle when it passed straight, curved, turnout and rail lubricator sections. From the sound pressure spectra in one-third octave bands, when the speed was 60 km/h on straight track, the noise between 160 Hz and 1250 Hz contributed most to the A-weighted interior noise. Noh et al. [15] studied the interior noise characteristics of high-speed trains. They found that, at the speed of 150 km/h, the spectra of interior noise were dominated by low and mid-frequency components. The highest A-weighted interior noise levels occurred between 500 Hz and 1250 Hz. At the speed of 300 km/h the spectra of interior noise were dominated by low frequencies. Zhang et

al. [16] investigated the interior noise characteristics of a Chinese high-speed train running on both a slab track and a ballasted track and either at the ground surface or in a tunnel. The results of field tests showed that, when the train ran at 200 km/h on slab track, at the different measuring positions the interior noise levels were higher by 0 to 2.5 dB(A) than on ballast track. At higher train speeds, the differences in interior noise between the slab track and the ballast track became smaller.

Shi et al [17] predicted the interior noise below 300 Hz in the cab of a subway train running at 60 km/h caused by vibration of the train panels by applying the vehicle-track coupling dynamics as the excitation. Shi et al. [18] also established a vehicle-track coupled dynamic model, a finite element model and an acoustic boundary element model to calculate the noise up to 5000 Hz in the interior passenger spaces of a high-speed train running at 200 km/h caused by track irregularities and ascertained the distribution of the acoustic features. Liu et al [19] calculated the noise below 250 Hz inside the passenger compartment of a high-speed train at 300 km/h. In these three papers, the FRA Class 5 irregularity spectrum was adopted as the excitation. Moreover, no comparisons were made with experiments. However, generally the American standard track irregularity spectra are suitable for wavelengths in the range 3-300 m [20]. For a train speed of 40 km/h, a wavelength of 3 m corresponds to an excitation frequency of 3.7 Hz and even for a speed of 300 km/h it is only 28 Hz. Clearly shorter wavelengths are required for noise predictions.

Zhang et al [21] analysed the contributions of interior noise of a high-speed train between 100 Hz and 3150 Hz through measurements and simulations based on statistical energy analysis (SEA). They used the model to identify the contributions from different panels and concluded that the noise from the bogie region is an important source. Zheng et al [22] combined various methods including multi-body dynamics, finite element analysis of the carbody and fast multipole boundary elements into a framework based on SEA they called statistical acoustic energy flow (SAEF). They used this to simulate the full-spectrum interior noise of a high-speed train which gave good agreement with measurements but no insight was given into the contributions of different paths or components.

Recent research has also focused on the noise radiation and transmission behaviour of the extruded aluminium panels from which modern rolling stock is often constructed. The sound transmission loss (STL) of such extruded aluminium panels is less satisfactory than flat panels with the same surface density. Xie et al [23] presented an SEA model to predict the vibroacoustic behaviour of aluminium extrusions used in railway vehicles. Kim et al. [24] proposed a prediction method of the STL of the aluminium extruded panel using finite element analysis. Zhang et al. [25] modelled aluminium extrusions using wavenumber finite element and boundary element methods and studied the dependence of the STL on the cross-section geometry. Sui et al. [26] modelled the vibrational responses of the extrusion in the low frequency range and measured the transfer mobility and vibration energy of the panel.

The parameter used in ISO 3381 [27] and GB 14892 [28] to evaluate the interior noise is the A-weighted equivalent continuous sound pressure level. However, the A-weighted sound pressure level has not been found to correlate well with perceived acoustic comfort in rail vehicles [1]. In particular, the influence of low frequency noise on people is underestimated. Eade and Hardy [8] suggested that acceptable levels of interior noise should be specified in terms of Preferred Speech Interference Level or Loudness Level. Furthermore, there are no standardised criteria to evaluate the low frequency noise inside railway vehicles. In contrast,

in ISO 14837 [29] there are some regulations to measure and predict ground-borne noise caused by rail systems in the frequency range 16 Hz to 200 Hz.

The aim of this paper is to compare the interior noise in a metro vehicle when running over the same track when fitted with rail fasteners of different stiffness. The approach taken is mainly experimental. In addition to measurements of interior noise, axlebox vibration, train floor vibration, exterior noise and rail vibration have also been measured and are used to investigate the differences further. All these quantities were measured at the same time to avoid the influence of any other changes in the conditions. The measurements are described in Section 2 and the results are discussed in Section 3. Following this, in Section 4, numerical models are used to investigate the reasons for the differences.

2 Description of the measurements

2.1 Test track

A series of measurements were undertaken at the Comprehensive Rail Transportation Test Line at Jiading campus, Tongji University, China. The test line currently consists of a length of 678 m of electrified standard gauge track. For the purpose of the current tests a section of slab track of length 28 m was installed in the test line and fitted with two types of rail fastener. The test section is a straight line.

The two types of fasteners are fitted with a rubber pad and a plastic insulator. The first type (Fastener A) has a static stiffness of 30–35 MN/m (TB/T 3396.3 [30]). The second (Fastener B) has a static stiffness of 10–15 MN/m (EN 13146-9+A1 [31]). These stiffness values correspond to the gradual application of a preload of 70 kN (100 kN for Fastener B) over a period of about a minute. The two types of fasteners were both installed on the same section of track in such a way that each one could be removed during the testing of the other (Fig.1 and Fig. 2). The spacing between fasteners of each type is 0.6 m. The rail roughness was not measured but as the same rails are used for both types of fastener it can be considered to be invariant.

Fig. 1. Test site during construction showing Fastener A attached to the sleepers and Fastener B in between sleepers

Fig. 2. Test site after construction

2.2 Train

The test train consisted of two metro vehicles, a power car and a trailer car, with an axle load of 145 kN. The length of each car is 22.4 m. The nominal wheel rolling diameter is 0.84 m. There was no internal decoration or seating. The air conditioning inside the car was switched off for the duration of the tests. The running speeds were 20, 30 and 40 km/h. For each vehicle speed at least four different train passages were measured; the results presented are the average of these multiple train passages.

2.3 Vibration and noise measurement points layout

According to the Noise Limit and Measurement for Train of Urban Rail Transit standard (GB 14892) [28] and referring to ISO 3381 [27], six points were used to measure the interior noise, as shown in Fig. 3 and Fig. 4. The microphones were located at heights of 1.2 m and 1.6 m above the vehicle floor along the centreline of the trailer car. M1 and M2 were close to the driver's cab and located in the central axis of the side doors. M3 and M4 were in the centre of the car, also opposite side doors. M5 and M6 were close to the connection between the two cars and were located directly above the bogie. Three accelerometers (V1, V2 and V3) were installed on the floor of the same vehicle directly below the noise measurement points (see Fig. 3 and Fig. 4).

Beside the track, two microphones were used to measure the exterior noise (Fig. 5 and Fig. 6). These were located at 2.0 m from the track centreline at the same height as the top of the rail (P1), and at 7.5 m from the track centreline and 3.5 m above the rail head (P2). Due to various objects, including a wall, the environment did not correspond to a free field.

Two accelerometers were used to measure the vertical vibration of the axlebox (Fig. 7 shows one of them). Six accelerometers were used to measure the vertical vibration of the track, as shown in Fig. 8. Accelerometers A1 and A2 are located on the rail foot; the others are located on the slab or the ground surface but are not used here.

Fig. 3. Layout of the interior vibration and noise measurement points

Fig. 4. Photograph of instrumentation in the vehicle

Fig. 5. Exterior noise measurement points

Fig. 6. Photograph of exterior noise measurement

Fig.7. Measurement point for vertical vibration of axlebox

Fig. 8. Location of accelerometers on the track

3 Measurement Results

3.1 Interior noise spectra

During the field measurements, at least four train passages were measured for each vehicle speed. It was ensured that the maximum difference between the results for a given condition was less than 3 dB, otherwise the measurement was repeated. The microphones calibrated using a sound level calibrator before and after testing; the deviation between the calibration before and after the tests was less than 0.5 dB. To illustrate typical signal-to-noise ratios, Fig. 9 shows the average interior noise for Fastener B at Point M5 at 20 km/h and 40 km/h, together with the background noise. The differences between the signal and the background noise are greater than 10 dB over most of the frequency range. Only at 50 Hz and 63 Hz are the differences smaller than this (4.1 dB at 50 Hz, 6.6 dB at 63 Hz and also 8.9 dB at 5000 Hz for 20 km/h; 6.2 dB at 50 Hz and 8.8 dB at 63 Hz for 40 km/h).

Fig.10 shows the linear (i.e. unweighted) sound pressure level in 1/3 octave bands at different interior measurement points for the two fastener systems. It can be seen that the spectra are dominated by the noise in the low frequency range. There is a broad peak between 125 and 200 Hz, which is larger at Points M1 and M5 than in the centre of the car at Point M3.

Fig.11 and Fig.12 show the dependence of the sound pressure levels on train speed. At low frequencies these results are nearly independent of speed, but above 250 Hz the sound level increases with increasing speed by up to 9 dB for a doubling of speed, which is typical of rolling noise [5].

Fig.13 shows the level difference between the results for Fastener B and Fastener A at two measurement points. Lines are shown for all three speeds. In the frequency range 315 to 1000 Hz, the noise levels are higher for Fastener B, with an average level difference of 3 dB. Also at 100-125 Hz Fastener B has a higher noise level than Fastener A. At other frequencies the level difference is smaller and in some cases. Fastener A shows higher noise levels than Fastener B, in particular at 63 Hz and 125-200 Hz, the latter corresponding to the peak in the spectra in Figs 9-11.

Fig. 9. Average interior noise spectra of Fastener B with background. (a) 20 km/h; (b) 40 km/h

Fig. 10. Interior noise spectra at M1, M3, M5 at 40 km/h. (a) Fastener A; (b) Fastener B

Fig. 11. Interior noise spectra at M3 at 20, 30, 40 km/h. (a) Fastener A; (b) Fastener B

Fig. 12. Interior noise spectra at M5 at 20, 30, 40 km/h. (a) Fastener A; (b) Fastener B

Fig. 13. Difference in interior noise spectra: Fastener B minus A; (a) at M3 and (b) at M5

3.2 A weighted total linear sound pressure level

The total sound pressure level is a simple and direct parameter to quantify the sound level. Although the A-weighted level has some limitations, it is still the quantity specified in the standards [27, 28] so it is used here to summarise the differences between fastener systems.

Table 1 Total sound pressure level within the train (dB (A) re 20 μ Pa)

Speed	Classification	M1	M2	M3	M4	M5	M6
20km/h	Fastener A	74.3	71.6	65.8	65.8	73.1	69.1
	Fastener B	76.9	74.3	66.8	66.8	75.9	71.7
	Difference	2.6	2.7	1.0	1.0	2.8	2.6
30km/h	Fastener A	76.5	74.0	67.6	67.5	75.6	71.7
	Fastener B	78.4	75.9	68.7	68.7	77.3	73.7
	Difference	1.9	1.9	1.1	1.2	1.7	2.0
40km/h	Fastener A	79.0	76.5	70.2	70.2	77.9	74.2
	Fastener B	80.1	77.5	71.8	71.6	79.3	75.9
	Difference	1.2	1.0	1.6	1.4	1.4	1.7

The test results under different running speeds are shown in Table 1. In general, when using the softer Fastener B, the noise levels are greater than those with Fastener A. As the running speed increases, the difference between the two fasteners mostly becomes smaller. Because positions M1, M2, M5 and M6 are above a bogie, they are likely to be more influenced by structure-borne noise transmitted from the bogie. Conversely positions M3 and M4 are near the centre of the vehicle and will be more influenced by airborne sound. However, from Fig. 12 these positions all show similar trends.

3.3 Vibration spectra of the vehicle floor

The vibration measured on the car floor at 40 km/h is shown in Fig.14 and Fig.15. Fig.14 shows the vertical vibration at the point V2 in middle of the car, while Fig.15 shows the vertical vibration at V3 above the bogie. The level difference between the results for the two fasteners is shown in Fig. 16. From this it can be seen that, in the frequency range 250 to 1250 Hz, the vibration level at V2 is larger for Fastener B, with an average level difference of 1.5 dB. For position V3 above the bogie Fastener B gives a higher vibration level with an average difference of 2.2 dB than Fastener A for all frequencies above 80 Hz.

Fig. 14. Floor vibration at V2.

Fig. 15. Floor vibration at V3.

(a)

(b)

Fig. 16. Vibration of Fastener B minus A; (a) at V2; (b) at V3

3.4 Vibration spectra of axlebox

Figure 17 shows the acceleration of the axlebox at 40 km/h. In the low frequency range up to 250 Hz the vibration for Fastener A is generally greater than for Fastener B, whereas at higher frequencies this is reversed. This should be a direct indicator of differences in structure-borne transmission.

(a)

(b)

Fig. 17. Vertical acceleration of axlebox. (a) 1/3 octave band spectrum; (b) Fastener B minus A

3.5 Exterior noise and vibration spectra

Besides the structure-borne noise transmission, the interior noise is also the result of airborne transmission from outside, in particular from rolling noise. Fig.18 shows the linear sound pressure level measured at the exterior microphone positions P1 and P2 during the passage of the train at 40 km/h. These microphone positions were shown in Fig. 5. The level difference between the two fasteners is shown in Fig. 19. Similar level differences are seen for the two microphone positions. As can be seen in Fig. 18 and 19, the noise with Fastener B is greater than that for Fastener A between 160 Hz and 500 Hz by up to 2.6 dB, but the noise from Fastener A is greater for frequencies above and below this range.

Fig. 18 Exterior noise: (a) at P1 and (b) at P2.

Fig. 19. Exterior noise of Fastener B minus Fastener A at P1 and P2

Fig. 20. Rail vibration. (a) 1/3 octave band spectrum; (b) Fastener B minus A

Figure 20 shows the vertical rail vibration during train passages at 40 km/h. The vibration with Fastener B is larger than that for Fastener A below 500 Hz and also at high frequency, but between 500 and 1600 Hz the vibration is greater for Fastener A.

4 Comparison with theoretical model

To give insight into the effect of the fastener stiffness on interior noise and to provide some interpretation of the experimental results, it is useful to make comparisons with theoretical models. However, a comprehensive model of the airborne and structure-borne noise transmission into the vehicle interior would require validated sub-models of all components [21, 22]. These include the wheelset, bogie frame, carbody structure and interior acoustics. Moreover, the suspension system elements, including springs, dampers and other connections in both primary and secondary suspension, are critical to the structure-borne transmission and would require detailed validated models.

By focusing only on the relative differences between the two fastener systems, it is possible to use a simplified approach. The differences in the structure-borne noise component will be directly related to the corresponding differences in the axlebox vibration, whereas the differences in airborne noise component will be related to the differences in exterior rolling noise.

In this section comparisons are made with models of the wheel/track interaction contained within the TWINS model [32]. Although some of the input parameters are unknown, by using plausible values it can be explored whether the measured differences are consistent with the model.

4.1 Track dynamics

The most important track dynamic properties are the vertical point mobility (velocity for a unit force) and the track decay rate. To estimate these, the parameters listed in Table 2 are used for the track with the two fastener systems. The dynamic stiffness values are assumed to be a factor of 2 larger than the static stiffness values quoted in Section 2.1 above. In addition, a second layer of stiffness is included in Fastener B to represent the upper pad above the metal baseplate. For this fastener an intermediate mass is also included between

the two elastic layers. The loss factors are chosen based on previous experience and to some extent to improve the fit with the measurements as seen below.

Table 2. Assumed track parameters

Rail	Fastener A	Fastener B
Vertical bending stiffness (MNm ²)	6.42	6.42
Rail mass per length (kg/m)	60	60
Rail shear coefficient	0.4	0.4
Rail loss factor	0.02	0.02
Upper pad		
Pad vertical stiffness (MN/m)	–	120
Pad vertical loss factor	–	0.35
Baseplate parameters		
Baseplate mass (kg)	–	6.0
Fastener spacing (m)	0.6	0.6
Lower pad		
Pad vertical stiffness (MN/m)	70	30
Pad vertical loss factor	0.25	0.15

Based on the parameters in Table 2, the point mobility is calculated using a model of a Timoshenko beam continuously supported on one or two elastic layers [5]. The results are shown in Figure 21(a). At low frequencies Fastener B has a higher mobility than Fastener A, reflecting its lower stiffness. For both tracks the mobility has a resonance at which the rail mass bounces on the support stiffness. This occurs at around 220 Hz for Fastener A and 120 Hz for Fastener B. Above this frequency free waves start to propagate in the rail and the mobility tends to that of an infinite beam. A second peak occurs for Fastener B at around 900 Hz associated with the second layer of resilience. Above 1 kHz the response is dominated by the rail itself so no differences are found between the two fasteners.

The same model can be used to predict the track decay rates, which are important for rolling noise: a low track decay rate corresponds to a high noise level [5]. These are shown in Figure 21(b). The decay rate can be seen to fall from an initial value around 10 dB/m above the resonance frequencies identified above. For Fastener B a second peak in the decay rate occurs around 900 Hz due to the internal resonance of the baseplate mass between the two layers of stiffness.

Fig. 21. (a) Estimates of track mobility; the wheel mobility and contact spring mobility are also shown. (b) Track decay rates for the two rail fasteners.

4.2 Axlebox vibration

The wheel/track interaction force F at circular frequency ω can be estimated using the following equation [5]:

$$F = \frac{i\omega r}{Y_r + Y_w + Y_c} \quad (1)$$

where Y_r is the rail mobility, Y_w is the wheel mobility, Y_c is the mobility of the Hertzian contact spring and r is the roughness amplitude. The rail mobility has been obtained in the previous section. For the vertical wheel mobility, a simple model is used representing the wheel by a mass, m , in series with a stiffness, K_w [5]:

$$Y_w = \frac{1}{i\omega m} + \frac{i\omega}{K_w} \quad (2)$$

Although this does not capture the high frequency modes of the wheel, it gives a reasonable representation of the point mobility for frequencies up to 1k Hz [2]. For the value of m , half the wheelset mass is used, i.e. 827 kg. The stiffness is set to 5 GN/m with a loss factor of 0.1. The mobility is also shown in Figure 21(a).

The Hertzian contact stiffness k_H is taken as 1.28 GN/m and gives the following mobility, also shown in Fig. 21(a),

$$Y_c = \frac{i\omega}{k_H} \quad (3)$$

The axlebox vibration can be estimated by multiplying the contact force (Eq. (1)) by the wheelset mobility (Eq. (2)). The predicted axlebox vibration for a unit roughness is shown in Fig. 22(a). A peak occurs at low frequency which is the resonance of the wheelset mass bouncing on the track stiffness. This resonance can also be identified as the frequency at which the wheelset and track mobilities have equal magnitude, see Fig. 21. It shifts to a lower frequency when a softer rail fastener is used. The difference in axlebox vibration between the two fastener systems is also shown in Fig. 22(b) and compared with the measurements. These show similar trends, with Fastener B giving a lower vibration than Fastener A between about 50 Hz and 200 Hz in the predicted results; in the measurements the difference is not quite as large but extends to 400 Hz. Considering the simple model of the wheelset used here this level of agreement is acceptable and can be used to explain the trends seen in the measurements.

(a)

Fig. 22. (a) Estimates of axlebox vibration for a unit roughness input. (b) Level difference between the two rail fasteners (B minus A). Measured level differences are also shown.

4.3 Rail vibration

The rail vibration at the wheel/rail contact can similarly be obtained from the product of the wheel/rail force and the rail mobility. Results for a unit roughness are shown in Fig. 23(a). A peak is seen at the same resonance frequency as for the axlebox vibration. Above this frequency the rail vibration is approximately equal to the roughness for frequencies up to about 1 kHz; at higher frequencies the contact spring mobility is higher than that of the rail and the rail response drops with increasing frequency. However, the differences between the two fastener systems are limited to low frequencies.

The noise radiated by the track is determined by the average vibration over the length of the train. This is shown in Fig. 23(b). Where the track decay rate is lower (see Fig. 21(b)) the average vibration is greater. Thus Fastener B has a larger average response than Fastener A between 100 and 500 Hz and a smaller response between 500 and 2000 Hz, in addition to the differences already seen at low frequencies. In Fig. 23(c) the level differences between the average rail vibrations for the two fastener systems are shown and compared with the measured results. Again these show similar trends, at least below 2 kHz.

Fig. 23. Estimates of rail vibration for a unit roughness input. (a) Vibration at the wheel/rail contact point; (b) Average vibration over the length of one vehicle; (c) Level difference between the two rail fasteners (B minus A). Measured level differences are also shown.

4.4 Rolling noise

Finally the TWINS model [32] is used to predict the rolling noise. This includes the same track model used in the previous sections but the wheel is represented by a finite element model. As details of the wheel geometry are not available, a model of a similar wheel with the same diameter has been used. The lateral interaction force is also included in the model together with lateral dynamic properties of the wheel and track; however, the noise from the rail is dominated by its vertical vibration. The differences in average sound pressure level at positions P1 and P2 between the two fasteners are shown in Fig. 24 and compared with the corresponding measured differences.

These predicted noise spectra show similar trends to the rail vibration, with the noise from fastener B exceeding that from fastener A in the frequency region 125 to 500 Hz and the opposite trend from 630 to 2000 Hz. Above 2 kHz the wheel noise dominates and there are negligible differences between the two fastener systems. The measured signals also show similar trends although the level differences are smaller than from the predicted results.

Fig. 24. External noise level differences (fastener B minus A) from TWINS and measurements

4.5 Discussion

As seen from the results in Figs 22-24, the structure-borne and airborne components of sound transmitted to the vehicle interior are expected to be affected differently by the change in fastener system. Changing from fastener A to B, the axlebox vibration is reduced between 63 and 250 Hz and increased at higher frequencies. Conversely the rail vibration is increased for frequencies below 60 Hz and 100-600 Hz and reduced above 600 Hz. The exterior rolling noise is increased between 100 and 600 Hz and reduce at other frequencies. As the interior noise above the bogie is increased between 100 and 2500 Hz on the whole, but there is a dip between two peaks, this suggests that both airborne and structure-borne noise components contribute to the interior sound. Perhaps surprisingly it appears from the shape of the difference spectra that airborne noise has most influence between 100 and 400 Hz and structure-borne noise has more influence between 500 and 1000 Hz.

In the measurement situation, in which the train is located between two walls, the airborne sound transmitted through the doors and sidewalls will be greater than in free field but may be less than in a tunnel. In addition the situation in an operational line may involve different roughness on different tracks; corrugation is known to grow more rapidly on some track types and this may be a factor in the differences in rolling noise observed on some vibration-isolating tracks

5 Conclusions

The influence of rail fastener stiffness on railway vehicle interior noise has been investigated in a controlled study by a series of field measurements and numerical simulations.

Comparative measurements were made on a test track in which only the rail fastener was changed. Besides the interior noise, the measurements also included the vibration of the vehicle floor, axlebox vibration, rail vibration and exterior noise, which help to give insight into the sources of interior noise.

From the interior noise measurement, it is found that when the softer rail fastener was used, the noise levels around 125 Hz and in the frequency range 315 to 1000 Hz were greater than those with the stiffer fastener. The average level difference in the range 315 to 1000 Hz was 3 dB. Similarly, from the measured vibration of the vehicle floor the vibration above the bogie with the more elastic fastener had a higher level than that with the stiffer one, with an average difference of 2.2 dB for all frequencies above 80 Hz. In the middle of the vehicle, the vibration between 315 Hz and 1000 Hz was higher for the more elastic fastener.

Differences in axlebox vibration are indicative of differences in the structure-borne noise. The test results showed that, apart from a peak at 50 Hz, the vibration for the stiffer fastener was generally greater below 250 Hz whereas that for the softer fastener was greater above 315 Hz.

Exterior noise propagating into the vehicle through airborne transmission paths is another important source of interior noise. Rolling noise, due to vibration of the rails and wheels, is the most important component of the exterior noise. From the vertical rail vibration during train passages, the vibration with the softer fastener was larger below 500 Hz, but smaller between 500 and 1600 Hz. The exterior noise level difference showed a similar trend; the noise with the more elastic fastener was greater between 160 Hz and 500 Hz by up to 2.6 dB, but was lower for frequencies above and below this range.

It is very difficult to build a sufficiently detailed and validated vehicle model as there are many parameters that influence the noise transmission. To give some insight into the measured results, models are used from the TWINS model for rolling noise considering the wheel/rail interaction. The aim is only to determine the differences in the predicted noise spectra. After adopting reasonable parameters the vibration of the axlebox and rail are predicted for a unit roughness input. Calculations of the noise indicate that the component due to the vertical vibration of the rail is dominant in much of the frequency range. The noise from the softer fastener exceeds that from the stiffer one in the frequency region 125 to 500 Hz. The opposite trend occurs from 630 to 2000 Hz due to the presence of an internal resonance in the two-stage fastener, which gives an increase in the track decay rate. The measured signals also show similar trends although the level differences are smaller than those from the predicted results.

From the field measurements and the numerical simulations, it can be concluded that the more elastic fastener can cause increased noise in the vehicle interior, although at some frequencies the opposite occurs. The spectrum is influenced by both structure-borne and airborne paths, which are dominant in different frequency ranges.

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