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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 315–1000 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.

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

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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 0–2.5 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–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 (Figs. 1 and 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.