A STUDY OF THE USE OF VEHICLES WITH SMALL WHEELS FOR DETERMINING THE COMPONENT OF NOISE FROM THE TRACK

DJ Thompson ISVR, University of Southampton, Highfield, Southampton SO17 1BJ.
CJC Jones ISVR, University of Southampton, Highfield, Southampton SO17 1BJ.

1. INTRODUCTION

Railway rolling noise is generated by wheel and track vibrations, that are induced by the combined surface roughness of the wheel and rail. In order to understand these phenomena, a theoretical model, TWINS (Track-Wheel Interaction Noise Software) has been developed and validated for the European Rail Research Institute [1-3]. It has been used successfully to design quiet wheels and tracks [4-6].

The fact that the noise originates from both wheel and track vibrations means, in practice, that responsibility for noise control lies with both the vehicle operators and the track authority. Recent changes across Europe, driven by national government and EU policy, have led to separation of these two functions into different companies. Therefore it has become important to be able to identify experimentally the contributions of wheel and track to the overall noise level [7]. In order to achieve this, various measurement methods have been proposed [8,9]. One of these is the so-called quiet reference vehicle method. This is intended to measure the roughness-to-noise transfer function of the track by running an intrinsically quiet vehicle over it. An equivalent method would then be required to determine the roughness-to-noise transfer function of the vehicle, along with measurements of the roughness.

In this paper, the potential of four types of vehicle for use in the quiet reference vehicle method is investigated theoretically using the TWINS model. It is shown that two of these are inadequate, while the other two could be used with some precautions. In particular, the track component of noise is shown not to be wholly independent of the dynamic properties of the wheel. Other factors that must be taken into account are the combined roughness of the wheel and track and differences in the contact filtering due to the finite contact patch length. The latter effect can be minimised by an appropriate choice of vehicle load.

2. THE TWINS MODEL

The TWINS model [1] is intended to represent in detail the origins of the sound emitted by a wheel rolling on the track. An outline of the model is given in Figure 1. It performs a linear frequency-domain prediction of the noise from wheel, rail and sleeper vibrations induced by the combined surface roughness of the wheel tread and rail running surface. TWINS consists of a number of modules to calculate the wheel and track dynamic response, their interaction and the resulting noise radiation. The wheel vibration is modelled using the finite element method and its properties are passed to the TWINS module through a list of natural frequencies, mode shapes and modal damping values. The track is modelled using an infinite beam on a continuous spring-mass-spring foundation, representing the rail pad, sleeper and ballast. The coupling is performed in terms of a matrix of point receptances (frequency response functions) of the wheel, the track and the contact spring between them.

Extensive field tests have been carried out to validate this model and have shown that it is capable of predicting the overall A-weighted sound level to within about 2 dB [2]. Results in individual one-third octave bands are subject to a greater uncertainty, although this is in part due to uncertainties in the roughness input to the system. Recently, the model has been used to design a series of prototype 'low noise' wheels and tracks which were then tested in the field. Results from these measurements, which covered a range of less conventional designs as well as conventional reference vehicles and track, were compared with TWINS predictions and found to give similar levels of agreement to the earlier tests [3].

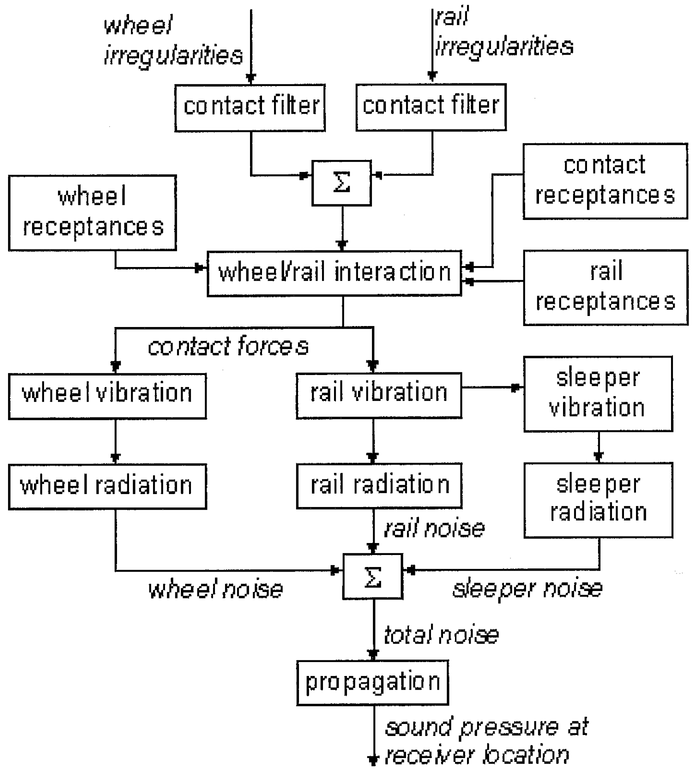


Figure 1. Flow chart of the TWINS prediction model for rolling noise.

3. QUIET REFERENCE VEHICLE METHOD

In the EU project STAIRRS (Strategies and Tools to Assess and Implement Noise Reducing Measures for Railway Systems) several methods for measuring the separate track- and vehicle-radiated components of rolling noise are being developed [7,8]. One such technique relies on having a 'quiet reference vehicle' so that only the dominant track noise is measured.

It is intended to separate, where possible, the total noise at the track side, $L_{p,tot}$ into wheel and track components:

$$L_{p,tot}(f_{to}) = 10\log_{10}\left(10^{L_{p,veh}(f_{to})/10} + 10^{L_{p,tr}(f_{to})/10}\right) \quad (1)$$

where $L_{p,tot}$ is the total noise at a standard microphone position (e.g. at 7.5 m from the track), $L_{p,veh}$ is the component of this noise radiated by the vehicle (wheel vibration) and $L_{p,tr}$ is the component radiated by the track vibration. Each of these quantities is expressed in one-third octave frequency bands, with f_{to} the band centre frequency. Furthermore, the vehicle and track components are expressed in terms of transfer functions between roughness and noise:

$$L_{p,veh}(f_{to}) = L_{H,veh}(f_{to}) + L_{r,eff}(f_{to}) \quad (2)$$

$$L_{p,tr}(f_{to}) = L_{H,tr}(f_{to}) + L_{r,eff}(f_{to}) \quad (3)$$

where $L_{r,eff}(f_{to})$ is the *effective combined roughness* one-third octave spectrum, which is the incoherent sum of wheel and rail surface roughnesses, allowing for the contact filter effect. $L_{H,veh}(f_{to})$ is the transfer function in one-third octave bands from unit roughness excitation to the vehicle-radiated component of sound and $L_{H,tr}(f_{to})$ is the transfer function for the track-radiated component of sound. It is the objective of the quiet reference vehicle technique to enable the measurement of $L_{H,tr}$ while other measurements would be used to obtain $L_{H,veh}$ and $r(f_{to})$.

In the quiet reference vehicle method, a vehicle with a low transfer function $L_{H,veh}$ is to be used to characterise the track. The noise measured during the passage of the reference vehicle is then assigned to the track,

$$L_{p,tot(ref.vehicle)}(f_{to}) \approx L_{p,tr}(f_{to}) \quad (4)$$

and hence, if the total effective roughness is known, the track transfer function $L_{H,tr}$ can be obtained from equation (3). The vehicle transfer function $L_{H,veh}$ is required to be at least 10 dB less than $L_{H,tr}$ in all frequency bands of interest. If this condition is satisfied, the approximation in equation (4) is correct to within 0.5 dB. If an error of 1 dB could be tolerated, then $L_{H,veh}$ would be required to be at least 6 dB less than $L_{H,tr}$ in all frequency bands

4. WHEELS CONSIDERED

One way of achieving a reference vehicle with a sufficiently low transfer function $L_{H,veh}$ might be to use a vehicle with small wheels, preferably with thick, straight webs. Such wheels are known to produce very little noise [10-12]. Four such existing wheel designs have been considered as candidates for use in the quiet vehicle method. These are:

- 730 mm diameter curved web wheels,
- 680 mm diameter straight web wheels,
- 640 mm diameter solid web wheels,
- 360 mm diameter solid web wheels.

The diameter in each case is less than the conventional size of 920 mm or thereabouts. A standard freight wheel of diameter 920 mm is considered here for comparison.

5. CONTACT FILTER EFFECTS

The separation of vehicle and track transfer functions indicated by equations (1-3) relies on the assumption that the effective roughness $r_{eff}(f_{to})$ will be the same for different wheels. If the wheels are smooth compared to the track, the total roughness will be dominated by that of the track which will be the same in each case. However, the effective roughness, r_{eff} is also influenced by the contact patch – wavelengths that are short compared to the contact patch length are attenuated (filtered) in their effect.

The size of the contact patch is a function of the diameter of the wheel and the static wheel load, as well as the rail head profile. The effect of the contact patch size on the noise for small wheels must therefore be examined against that for a conventional size wheel. A numerical model for the contact filter [13], that is implemented in TWINS, is used to calculate the effect on the contact filter of changes in wheel diameter and load. Figure 2 shows the contact filter effect for three diameters (left) and three loads (right). The frequency scale applies for a vehicle speed of 100 km/h, whereas the wavelength scale applies for all speeds. A rail head radius of curvature of 0.3 m is assumed throughout. It can be seen that the contact filter effect has a similar shape in each case, but that this shape is moved towards higher frequencies as the diameter is reduced or as the wheel load is reduced, in each case due to a shortening of the contact patch.

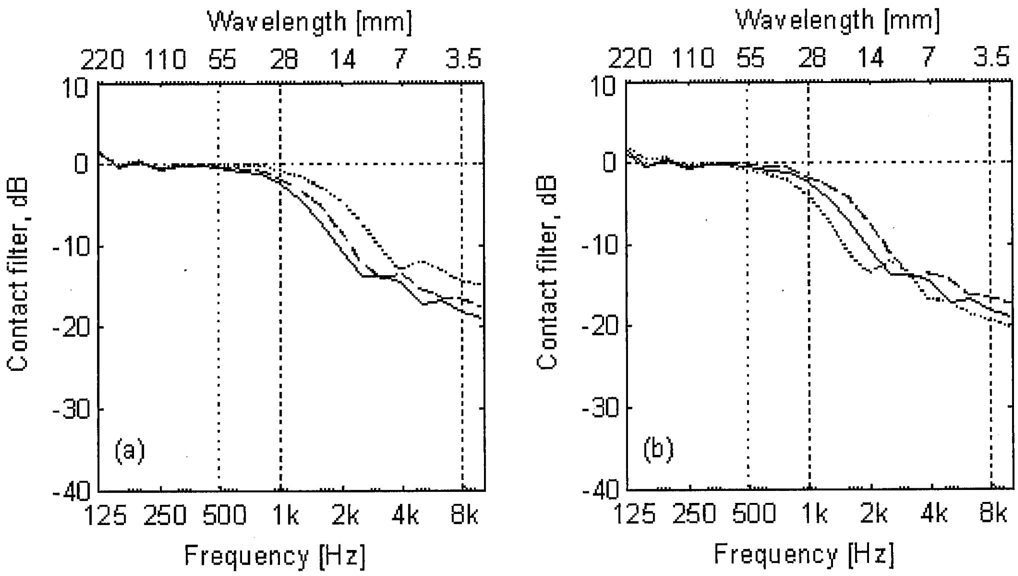


Figure 2. *Average contact filter effect calculated using numerical model for 100 km/h. (a) For 50 kN wheel load and various diameters: —— 920 mm, – – – 680 mm, ······· 360 mm. (b) For 920 mm diameter and various wheel loads: —— 50 kN, – – – 25 kN, ······· 100 kN.*

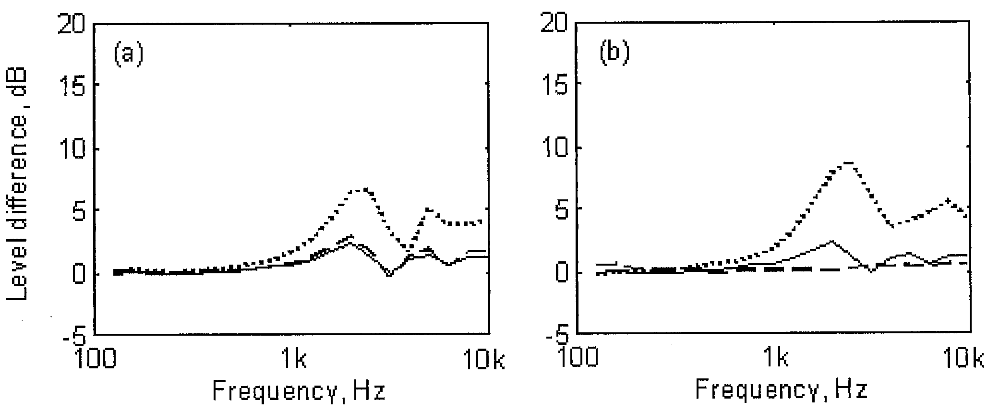


Figure 3. *Difference in contact filter effect for 100 km/h due to various wheel diameters relative to 920 mm wheel. (a) In each case for 50 kN load, —— 730 mm, – – – 680 mm, ······· 360 mm. (b) For expected operational load, —— 730 mm, 50 kN, – – – 680 mm, 83 kN, ······· 360 mm, 25 kN.*

In Figure 3 differences are shown between the contact filter effect applying in various situations, relative to the reference case of a 920 mm wheel with a load of 50 kN. In the left-hand graph, results for three wheel diameters are shown each with a load of 50 kN. These therefore represent the differences between two curves as in Figure 2(a). At low frequencies the differences are small. At higher frequencies, the differences are limited to 2-3 dB for 730 or 680 mm wheels whereas for a wheel of 360 mm diameter differences of up to 7 dB are found. When account is taken of the expected wheel load of the vehicles considered, Figure 3(b), it can be seen that where the wheel load is reduced the differences increase. However, from the result for the 680 mm wheel, it is seen that it is possible to compensate for the reduction in diameter by an increase in the wheel load. For the 640 mm wheel, although not shown here, it has been found that the contact filter effect could also be fully compensated by using a wheel load of 86.5 kN.

6. ROLLING NOISE PREDICTIONS

Using the TWINS model, predictions of rolling noise have been carried out for a standard 920 mm wheel and the four candidate 'quiet' wheels running on a standard track of bibloc sleepers in ballast with a typical moderate stiffness rail pad. The predictions were made in terms of the sound power radiated by a single wheel and the associated track vibration. Differences in sound power spectra can be expected to replicate differences in the sound pressure spectra at the standard measurement distance of 7.5 m. A train speed of 100 km/h has been used throughout.

Figure 4 shows the wheel, rail and sleeper components of sound power for the standard 920 mm wheel, along with the total noise spectrum. The spectra are unweighted, although the overall A-weighted level is also quoted in the legend. From this it is seen that the sleeper dominates at low frequencies, the rail in the middle of the frequency range and the wheel at high frequencies, in this case from 1.6 kHz. This high frequency range is characterised by natural modes of the wheel with an axial component on the web and a radial component on the tread.

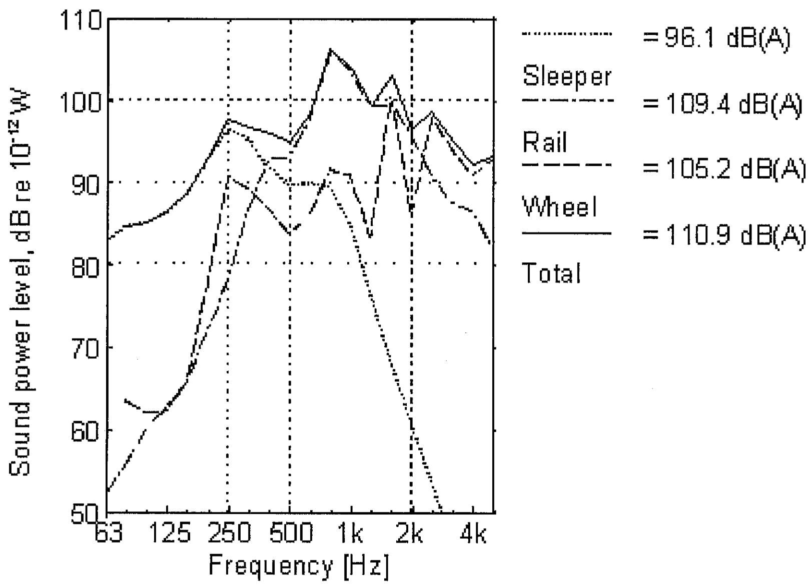
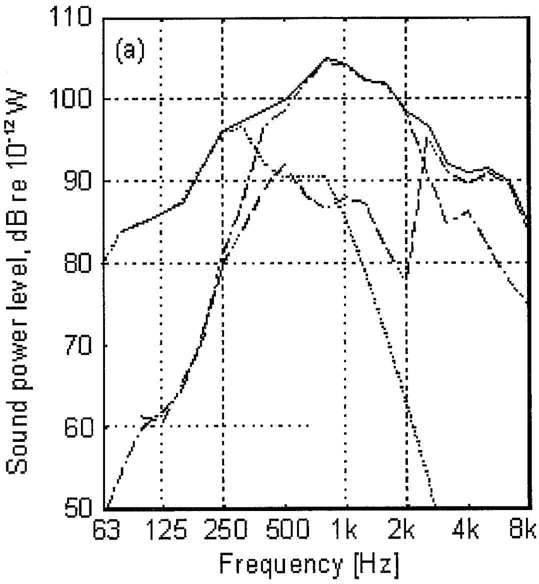
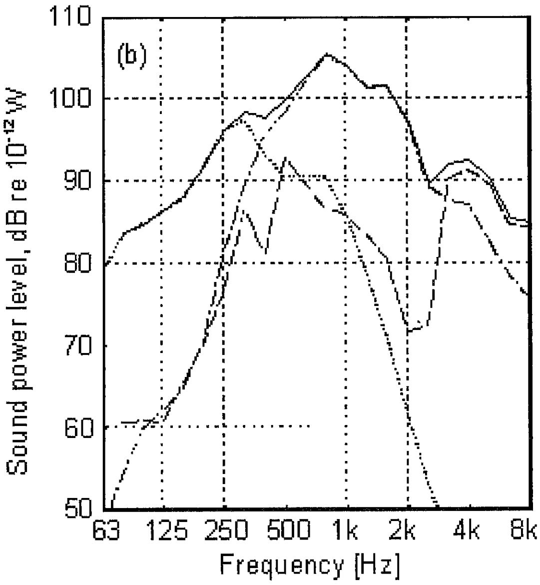


Figure 4. TWINS sound power prediction for the 920 mm diameter standard freight wheel at 100 km/h with a 50 kN wheel load.

To fulfil the requirements for the quiet vehicle method, the wheel component should be reduced to well below the track component (rail plus sleeper) in the whole frequency range of interest, shown here as 63 to 5000 Hz. This requires a small reduction in the region of 250 Hz as well as a much greater reduction at high frequency. The latter can be achieved by a reduction in level of the high frequency modes or an increase in their natural frequencies. A reduction in the level of up to 20 dB would be required, which is unlikely to be achievable using damping treatments [4]. It is, therefore, more profitable to attempt an increase in the natural frequencies. This is the reason for studying smaller wheels, as a reduction in the diameter can significantly raise these natural frequencies [9].





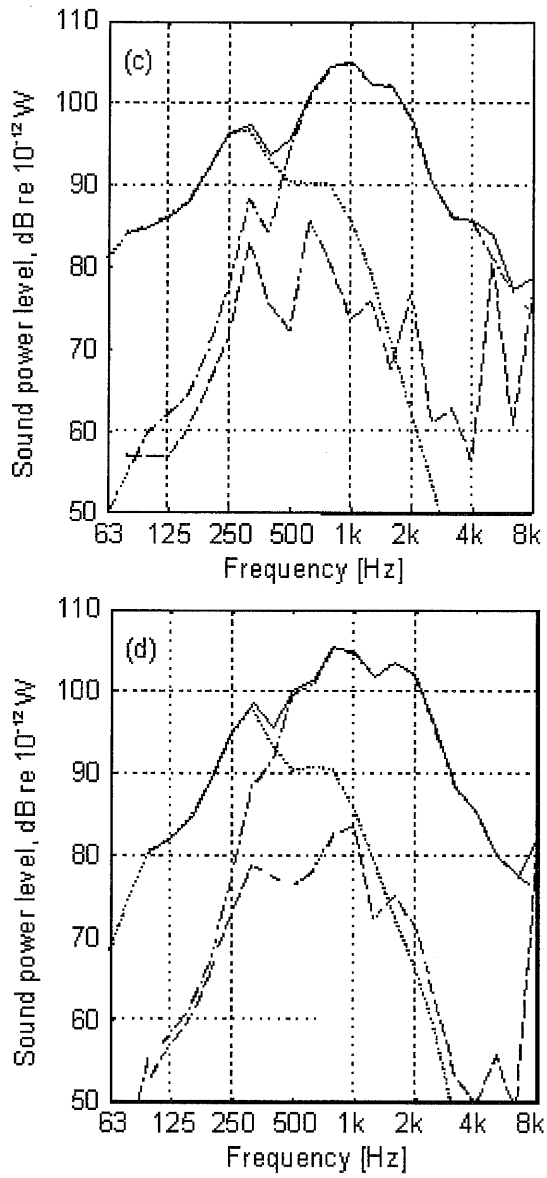


Figure 5. TWINS sound power predictions for the candidate wheels at 100 km/h (a) 730 mm diameter wheel with 50 kN wheel load, (b) 680 mm diameter wheel with 83 kN wheel load, (c) 640 mm diameter wheel with 86.5 kN wheel load, (d) 360 mm diameter wheel with 25 kN wheel load. —— total, – – – wheel, ······· rail, – · – · sleeper.

In Figure 5, equivalent results are given for the four candidate wheels. As the wheel diameter is reduced, and as the web thickness is increased, the frequency at which the wheel noise component rises sharply is increased. Thus for 730 mm this rise occurs at 2.5 kHz, for 680 mm at 3.15 kHz, for the 640 mm solid web wheel at 5 kHz, and for the 360 mm wheel in the 8 kHz band.

From these results, it is already clear that the two solid web wheels would be satisfactory as quiet reference vehicles up to the 4 or 6.3 kHz bands. The other two wheels, however, would only be satisfactory for frequencies up to 2 or 2.5 kHz. Above this limit, the wheel component is still larger than the track component. Even the addition of wheel damping, which could reduce the high frequency component from the wheel by up to about 6 dB, would not be sufficient to extend the frequency range over which the wheels could be used.

This is shown further in Figure 6, in which is plotted the ratio between the vehicle and track transfer functions, $L_{H,veh}$ and $L_{H,tr}$, as calculated using TWINS. The left-hand figure shows the results from the above analysis for a standard track while the right-hand figure shows the results that would be obtained for a damped track [6]. This is clearly a more difficult test for the method as $L_{H,tr}$ is reduced. These figures confirm that only the 640 mm and 360 mm wheels are satisfactory for use up to 4 kHz. Moreover, it can be seen that the vehicle transfer function $L_{H,veh}$ is at least 10 dB smaller than the track transfer function $L_{H,tr}$ in this frequency range for these two wheels; on the other hand for the larger wheels this is not the case, although the difference is always greater than 6 dB. These conclusions hold for both the standard track and the damped track.

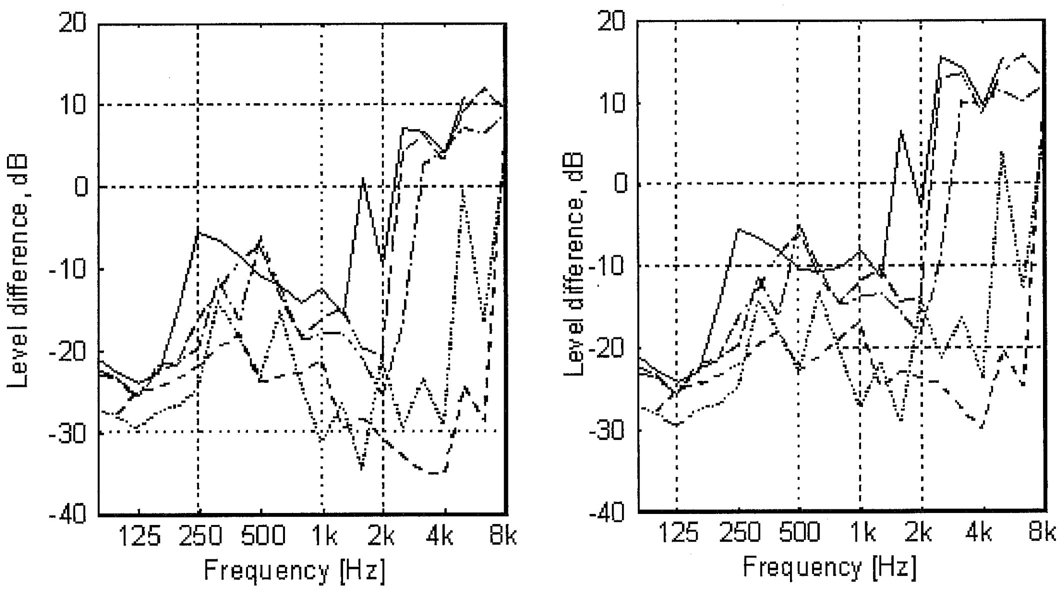


Figure 6. Difference between $L_{H,veh}$ and $L_{H,tr}$ for each wheel type for (a) standard track, (b) damped track. —— 920 mm, – – – 730 mm, – · – · 680 mm, ······· 640 mm, - - - 360 mm.

Finally it is necessary to check whether the track transfer function, as measured by the quiet reference vehicle method, is the same as that which applies to another wheel. Figure 7 shows the level difference between $L_{H,tr}$ for the various candidate wheels and that for the standard 920 mm wheel for the standard track. Clearly there are some differences.

At around 400 Hz, the result for all four wheels is up to 4 dB greater than that for the 920 mm diameter wheel; this is due to higher lateral rail vibration for these wheels. Conversely, at high frequencies the difference is around -2 dB (-5 dB for the 360 mm wheel) due to differences in contact stiffness and in the frequency range where the wheel modes have an influence. Thus, the noise from the track in a given one-third octave band can vary by up to ±4 dB. Allowance would need to be made for this if the quiet reference vehicle method were to be used in practice, either by using TWINS predictions or by using rail vibration measurements.

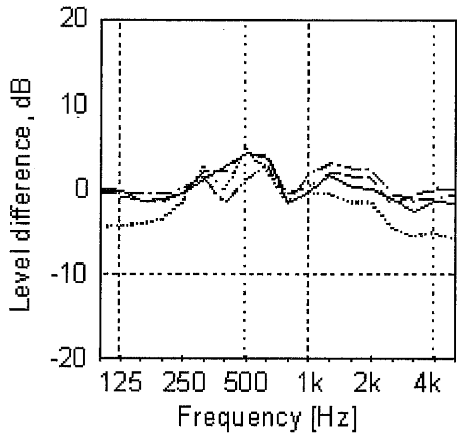


Figure 7. *Difference in $L_{H,tr}$ compared to the 920 mm wheel prediction, —— 730 mm, – – – 680 mm, – · – · 640 mm, ······· 360 mm.*

In addition, the overall noise from the track increases slightly for smaller wheels due to the changes in the contact filter. Thus the overall A-weighted sound power level from the rail, which is 109.4 dB(A) for the reference track, increases by about 1 dB for the 730, 680 and 640 mm wheels and by 2 dB for the 360 mm wheel.

7. CONCLUSIONS

Using the TWINS theoretical model for rolling noise it has been possible to assess the viability of using several designs of wheel in the proposed quiet reference vehicle method for measuring the track component of noise. It has been shown that two of the designs, with diameters 730 mm and 680 mm, would not be acceptable for measuring the whole frequency range intended. The other two, which had solid webs and smaller diameters of 640 mm and 360 mm, could be used up to 4 kHz and 6.3 kHz respectively. Both satisfy the requirement in this frequency range that the wheel noise is at least 10 dB below the track noise, even for a damped track. Of the two, the 640 mm wheel is preferred as it has least effect on the track transfer function and the contact filtering of the roughness. Even so, some correction for the effects of the wheel dynamics on the track response may be necessary. The contact filter effect can be virtually eliminated by an appropriate choice of wheel load.

8. ACKNOWLEDGEMENTS

The work described here has been performed within the EU fifth framework project STAIRRS (Strategies and Tools to Assess and Implement Noise Reducing Measures for Railway Systems), contract 1999-RD-10442, project number: B99/99/S12.107978-B66131122, coordinated by ERRI.

9. REFERENCES

1. D.J. Thompson, B. Hemsworth and N. Vincent 1996 Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method. *Journal of Sound and Vibration*, 193, 123-135.

2. D.J. Thompson, P. Fodiman and H. Mahé 1996 Experimental validation of the TWINS prediction program for rolling noise, part 2: results. *Journal of Sound and Vibration,* 193, 137-147.

3. C.J.C. Jones and D.J. Thompson 2001 Extended validation of a theoretical model for railway rolling noise using novel wheel and track designs. Presented at 7th International Workshop on Railway Noise, Portland, Maine, USA, October 2001.

4. C.J.C. Jones and D.J. Thompson 2000 Rolling noise generated by wheels with visco-elastic layers. *Journal of Sound and Vibration*, 231, 779-790.

5. N. Vincent, P. Bouvet, D.J. Thompson and P.E. Gautier 1996 Theoretical optimization of track components to reduce rolling noise. *Journal of Sound and Vibration*, 193, 161-171.

6. D.J. Thompson, C.J.C. Jones and D. Farrington 2000 The development of a rail damping device for reducing noise from railway track. *Internoise 2000,* Nice, France, 28-30 August 2000, vol. 2, 685-690.

7. P. Dings 2001 Characterisation and classification of railway noise sources. Presented at 7th International Workshop on Railway Noise, Portland, Maine, USA, October 2001.

8. M.G. Dittrich and M.H.A. Janssens 2000 Improved measurement methods for railway rolling noise. *Journal of Sound and Vibration*, 231, 595-609.

9. International Standards Organization 2001 Railway applications - acoustics - measurement of noise emitted by railbound vehicles, draft standard ISO/DIS 3095:2001.

10. D.J. Thompson 1993 Wheel-rail noise generation, Part II: wheel vibration. *Journal of Sound and Vibration* **161**, 401-419.

11. P.E. Gautier, N. Vincent, D.J. Thompson and G. Hölzl 1993 Railway wheel optimization. *Proceedings of Internoise 93, Leuven*, 1455-1458.

12. C.J.C. Jones and J.W. Edwards 1996 Development of wheels and track components for reduced rolling noise from freight trains. *Proceedings of Internoise 96, Liverpool*, 403-408.

13. P.J. Remington and J. Webb 1996 Estimation of wheel/rail interaction forces in the contact area due to roughness. *Journal of Sound and Vibration* **193**, 83-102.