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ROLLING NOISE GENERATED BY RAILWAY WHEELS WITH VISCO-ELASTIC LAYERS

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The noise-generating characteristics of two types of railway wheel design have been studied theoretically. These are "resilient wheels" in which a viscoelastic layer is located between the type and the web, and wheels with constrained layer damping treatments applied to the web. A method of predicting the rolling noise of these wheel types using the TWINS rolling noise model has been developed. For this, a modal description of the wheel must be constructed. A finite element model is used to calculate the mode shapes and modal masses. The modal damping is predicted by a complex modal analysis of the finite element model in which a material-specific damping parameter is used. Analyses have been carried out for a number of resilient wheels with different stiffnesses of their resilient layer, including the case where the wheel becomes a conventional one by specifying the resilient element as steel. The sound power radiated by both the wheel and the rail are shown to be dependent on this stiffness. A number of configurations of wheels with constrained layer damping treatments have been analysed taking into account the frequency variation of the properties of real damping materials. Significant reductions in the wheel sound power are shown to be possible.

1. INTRODUCTION

In the present climate of environmental improvement, railway wheelset manufacturers are required to produce new designs of wheel that generate a lower contribution to rolling noise. For this purpose, the procedure embodied in the TWINS software for predicting the sound power generated by wheels on ballsted track has become well established for conventional wheels and has been validated experimentally [1, 2]. An objective of the work presented here has been to extend the application of TWINS to two types of wheel with viscoelastic components, namely, (1) wheels with a layer of resilient material located between the tyre and the web (produced in order to reduce the impact forces on the track), and (2) wheels damped with constrained layers of viscoelastic material applied to the surface of the web (wheels with such damping treatments are produced primarily to reduce curve squeal). As well as presenting a method by which the rolling noise of these designs of wheel may be predicted, this paper investigates the influence of the designs on the wheel rolling noise.

In the rolling noise model, the vibrational behaviour of the wheel is characterized by its "modal basis" [3], i.e., the data comprising the mode shape, frequency, mass and damping for all modes of the wheel in the frequency range of interest (up to about 6000 Hz). A finite element model is used to generate the natural frequencies, mode shapes, and modal masses. For conventional, solid steel wheels, the modal description is completed by selecting modal damping ratios from measurement experience. A modal summation is then used to calculate the point receptance of the wheel at the wheel/rail contact which is entered, along with rail receptance, etc., into a model of the wheel/rail rolling contact mechanics. This produces a prediction of the dynamic contact forces acting on the wheel and the rail. A second modal summation for the wheel predicts the response of the wheel to these forces at a number of locations of the tyre and web. These vibrational responses are used to estimate the sound power from the wheel by means of multiplication by radiation efficiencies for various cylindrical and annular surfaces to which the wheel surface geometry is approximated [4].

The method can be followed for the two types of wheel with viscoelastic layers, i.e., by producing a modal basis for the wheel. However, a number of aspects of the model must be considered carefully and these are addressed in this paper.

2. PREDICTION OF THE VIBRATION RESPONSE OF WHEELS

2.1. VIBRATIONAL MODES OF A RAILWAY WHEEL

For use with TWINS, the modal basis of the wheel, comprising the resonance frequencies and the corresponding mode shapes, is usually predicted using an axisymmetric analysis within a proprietary finite element (FE) package. The mode shape data required in the calculation, consist of the normalized modal response of the wheel at the nominal wheel/rail contact position and at up to six points distributed on a single radial cross-section of the wheel. These allow the vibrational responses at these positions to be calculated for use in predicting the sound power. The radial cross-section is taken to be at the azimuthal angle $\theta = 0$ such that the (out-of-plane) mode shape displacement of the wheel at the position (r, z, θ) is given by

$$u(r, z, \theta) = \bar{u}(r, z)\cos n\theta$$

where $\bar{u}$ is the normalized displacement on the radial cross-section, and n is the number of nodal diameters in the mode. The normalization of the mode shapes is defined by a modal mass which is included in the modal parameters data.

In the case of conventional railway wheels, it has been found that an FE model comprising only the wheel itself, constrained at the cylindrical interface with the axle, is sufficient to represent the mode shapes of all significant modes with $n \geqslant 2$. In the case of wheel modes in which $n = 0$, the mass of the axle and its tensile stiffness play a part in determining the natural frequency and the modal mass. For wheel modes with $n = 1$, the mass and the bending stiffness of the axle take part in determining the natural frequency and modal mass. Other modes of the wheelset with $n = 1$, which are predominantly bending modes of the axle rather than modes of the wheel, are not significant in the radiation of sound from the wheel.

For conventional wheels, it has been found that the $n = 0$ and 1 modes of the wheel, on which the axle has a significant effect, have generally higher damping than the modes with $n \geqslant 2$. Moreover, most of such modes occur at relatively low frequencies where the wheel component of noise is much less than that of the track. For these reasons, it has been found to be sufficient to base modal data for the rolling noise prediction entirely on an FE model of the wheel without the axle.

The damping of the wheel is defined by the inclusion, in the modal parameters file, of a modal damping ratio for each mode. This can be taken from specific measurements, from experience of measurements on similar railway wheels, or by calculation.

2.2. PREDICTION OF MODAL DAMPING

Based on measurements of the damping of all-steel railway wheels, modal damping ratios can be assumed on the following basis: $n = 0$ modes, $\zeta_i = 0{\cdot}001$; $n = 1$ modes, $\zeta_i = 0{\cdot}01$; $n = 2$ modes, $\zeta_i = 0{\cdot}0001$. The material loss factor, η for steel is between 2×10^{-5} and 3×10^{-4}, depending on the type of steel [5]. Wheel modes with $n \geqslant 2$ have a modal damping ratio ($\zeta \approx \eta/2$) which is hardly greater than this, any slight increase probably being due to friction at joints in the wheelset structure. At this level of damping, the exact value used for the damping ratio is not critical for the rolling noise prediction as the effective damping introduced by the interaction at the wheel/rail contact is considerably greater [6]. The higher damping values that have been found to be associated with modes with $n < 2$ probably arise because of the influence of damping in the bearings of the wheelset.

For the case of a wheel in which one or more of the materials has a high loss factor compared to steel, the modal damping can be calculated by other means. For the present work a complex modal analysis has been used.

In this case the FE formulation for the unforced structure is of the form

$$(-[M]\omega^2 + \mathrm{j}\omega[C] + [K])\{u\}\mathrm{e}^{\mathrm{j}\omega t} = 0,$$

where $[M]$ and $[K]$ are the global mass and stiffness matrices and $\{u\}$ is the vector of displacements. The global damping matrix $[C]$ is constructed from element damping matrices using a damping parameter, d, which is related to the loss factor of the element material by $d = \eta/\omega$. The above equation leads to the quadratic eigenvalue problem

$$[K]\{\phi_i\} + \bar{\lambda}_i[C]\{\phi_i\} = -\bar{\lambda}_i^2[M]\{\phi_i\},$$

where the complex eigenvalue can be expressed as $\bar{\lambda}_i = \sigma_i \pm \mathrm{j}\omega_i$. Using these terms, ω_i represents the circular natural frequency of the mode and $\sigma_i = \omega_i\zeta_i$ represents the damping, where ζ_i is the modal damping ratio. To construct the FE formulation, the material damping parameter, d, has been calculated at a specific circular frequency, ω_d, as $d = \eta(\omega_d)/\omega_d$. The modal damping ratios may then be approximately calculated from the complex eigenvalues as

$$\zeta_i = \frac{\sigma_i}{\omega_i^2}\,\omega_d\,\frac{\eta_v(\omega_i)}{\eta_v(\omega_d)},$$

where η_v is the frequency dependent loss factor of the viscoelastic material. This method is only precise when $\omega_d = \omega_i$. However, this estimate has been tested for a range of values of ω_d and found to give consistent results for all modes in the frequency range of interest when a value for ω_d is chosen in the middle of this range, e.g., at 3000 Hz.

3. ANALYSIS OF A SERIES OF RESILIENT WHEELS

3.1. WHEELS MODELLED

To illustrate the analysis for a resilient wheel, a typical design for a light rail vehicle (LRV) has been chosen. The geometry of this 740 mm diameter wheel is shown in Figure 1. The modal parameters have been generated for five widely differing materials forming the resilient component between the tyre and the web using an FE model of the wheel only. Five cases have been studied. In the first the resilient element is given the properties of steel. The four others cases, called “rubber 1”, “rubber 2”, “rubber 3” and “rubber 4”, are given Young’s modulus values of $2{\cdot}3 \times 10^9$, $5{\cdot}75 \times 10^8$, $1{\cdot}44 \times 10^8$ and $3{\cdot}6 \times 10^7$ N/m^2 respectively. For each of the resilient materials, a Poisson’s ratio of 0·4 and a loss factor of 0·1 have been used. Apart from the all-steel version, the four levels of resilience represent a factor of 4 in Young’s modulus between each successive step. Rubber 1 represents a rather stiff rubber, with rubbers 2, 3 and 4 progressively softer. The track parameters, which have not been changed throughout the study, represent a typical ballasted track. The rail pads have been attributed a vertical stiffness of $3{\cdot}8 \times 10^8$ N/m. All the analyses have been carried out for a train speed of 100 km/h and a typical roughness spectrum for a tread-braked wheel has been used.

In the case of a resilient wheel (with a sufficiently soft resilient layer), the first $n = 1$ radial mode represents the frequency at which the tyre becomes decoupled in the radial direction from the web part of the wheel. In this mode, neither the tyre

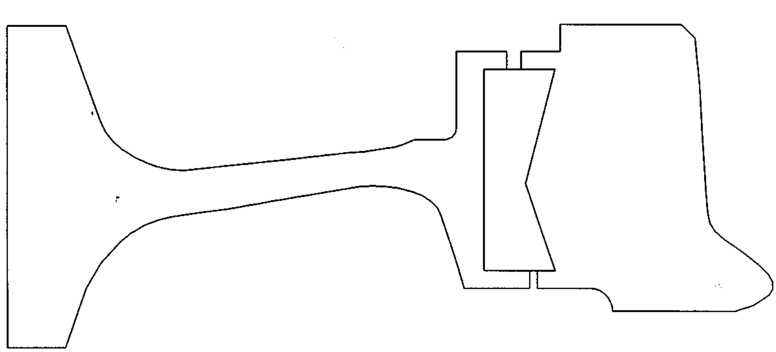

Figure 1. Geometry of the LRV resilient wheel studied.

nor web deform in cross-section but the tyre moves radially in opposite phase to the web and the axle. The mass of the axle has a significant influence in determining this natural frequency so, in this case, an FE model which includes the axle is used. Other modes have not been exchanged as their resonance frequencies change much less.

To calculate the acoustic radiation with the rolling noise model, four axial response positions have been used on the web of the wheel. A single axial response position has been used on the decoupled part of the tyre.

On the conventional wheel, the radiation from the inner and outer surfaces of the tyre partly cancel each other out at low frequencies. This leads to a lowering of radiation efficiency. In reference [4], predictions are given of the radiation efficiency of a conventional wheel for radial vibration of the whole tyre and of only the running surface of the tyre. These results are virtually identical above about 250 Hz, showing that this cancellation between inner and outer surfaces is only effective at low frequencies. The radiation efficiencies used in the rolling noise model are based on a curve fitted to the results above 250 Hz. It is therefore justifiable to use these results for resilient wheels as well as conventional wheels.

3.2. COMPARISON OF WHEEL SOUND POWERS

The wheel sound powers for the five cases are shown in Figure 2. In Figure 3, the sound power components due to the axial and radial vibration of the wheel are

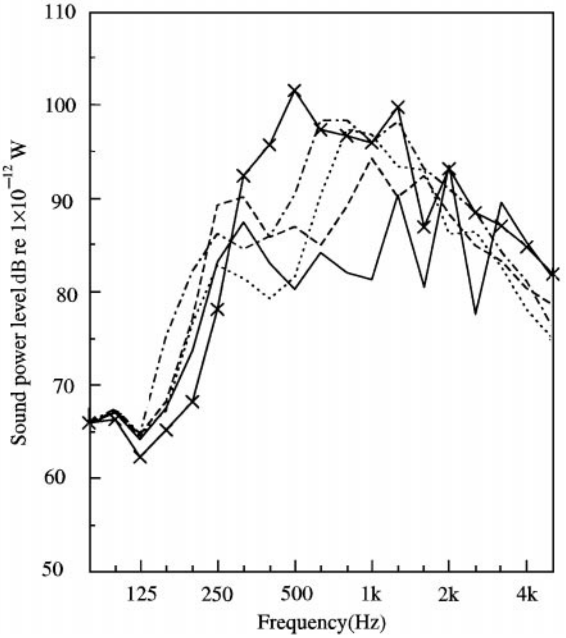

Figure 2. Comparison of wheel sound power levels for resilient wheels: —— steel, 98·5 dB(A); ---- rubber 1, 99·9 dB(A); .... rubber 2, 102·3 dB(A); -·-·-·- rubber 3, 104·7 dB(A); ×—× rubber 4, 105·3 dB(A) (spectra unweighted).

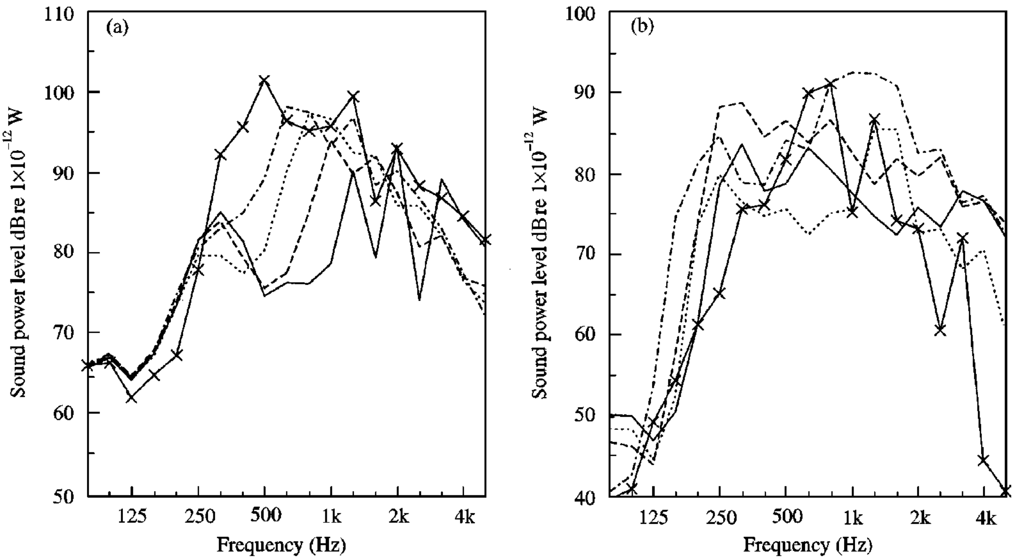

Figure 3. Sound power level due to (a) radial response on wheel tread and (b) axial response: —— steel, ---- rubber 1; .... rubber 2; -·-·-·- rubber 3; ×—× rubber 4.

plotted separately. The total wheel sound power is mostly dominated by the radial component in which the effect of reducing the stiffness of the resilient layer is apparent as an increase in level, particularly in the frequency region 500–1000 Hz. The frequency at which this increase occurs, although reducing with reduced stiffness, is not directly related to the decoupling frequency of the wheel tyre on the web, either radially or axially. This effect is, in fact, related to the relative amplitudes of the wheel and rail receptances and will be discussed further below.

The radiation from the axial component of vibration on the wheel, Figure 3(b), also increases with reduced stiffness. The axial component is dominated by the web and it might be expected that decoupling of the web from the tread would reduce the axial component for reduced stiffness. However, the increase occurs because the decoupling of the web at a progressively lower frequency allows modes to exist in the web at a lower frequency since this part of the wheel is no longer constrained by the stiffness of the tyre.

3.3. EFFECT OF INCREASED DAMPING

Noting that inclusion of the resilient layer in the example wheel design leads to an increase in its radiated sound power, it is of interest to evaluate the effect of the increased damping in the wheel. This has been calculated by using the modal bases for the resilient wheels with the modal damping ratios that would be assumed for conventional steel wheels (Section 2.2). The inclusion of damping is found to be responsible for a reduction in the sound powers by 4·4, 4·7, 3·9 and 2·5 dB(A) for the rubber 1, rubber 2, rubber 3 and rubber 4 wheels, respectively, indicating that the role of the resilient layer in damping is important.

3.4. EFFECT ON TRACK RADIATION

The overall A-weighted sound power levels for the rail and sleeper are shown alongside those of the wheels in Figure 4. From this it can be seen that the overall noise level mainly follows that of the rail, since this is the dominant source. This is because the wheel studied, being designed for LRV application, is quite small so that, in its all-steel form, it produces 12 dB(A) less noise than the track. The progressive increase in noise from the wheel with increasing resilience can also be seen in this figure. The effect on the track sound power of the resilient wheel is greater than would be expected for different designs of conventional wheel although it is still smaller than the effect on the wheel component. Figure 4 shows that the total sound power may go up or down with a change in resilience. It is lowest, in these cases, for the stiffest rubber.

3.5. COMPARISON OF WHEEL AND TRACK RECEPTANCES

Figure 5 shows the wheel and track vertical point receptances at the wheel/rail contact for the resilient wheels. The track receptance has two well-damped peaks at approximately 120 and 630 Hz. These are due to resonance of the track on the ballast stiffness, and mass of the rail on the rail pad stiffness respectively.

There is a rise in the wheel receptance at the frequency of the first significant wheel mode which occurs at 1000 Hz in the case of the rubber 1 wheel. This frequency corresponds to the 1-nodal-diameter radial mode. At frequencies above this, there is a high density of wheel modes which leads to a rise in the receptance of the wheel above that of the rail. This results in an increase in the response of the wheel to the forcing at the contact and a corresponding rise in the sound power from the wheel in these one-third octave bands. The rise in the wheel sound power

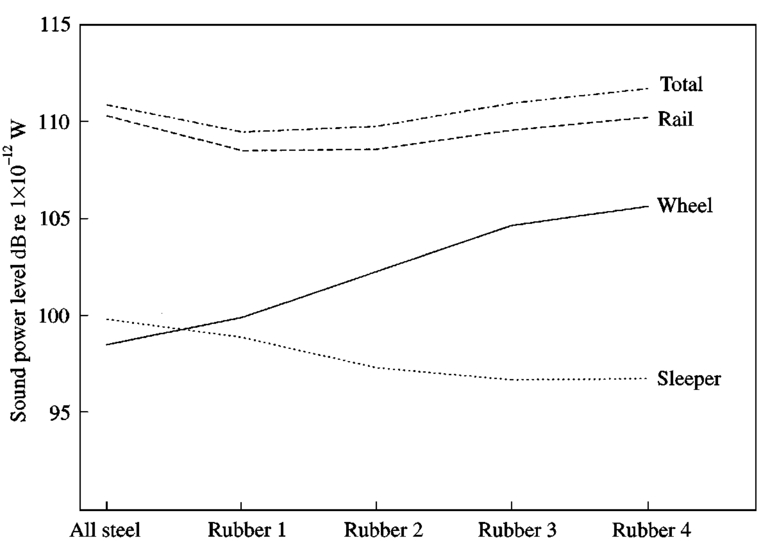

Figure 4. Overall A-weighted sound power levels for resilient wheels.

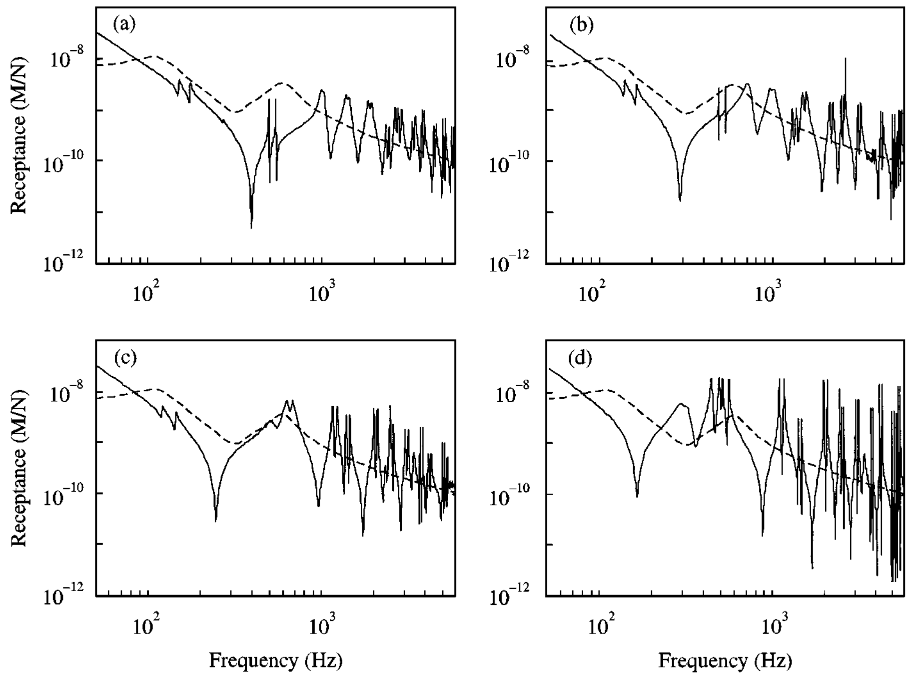

Figure 5. Wheel and rail vertical receptance for the four resilient wheel cases; (a) rubber 1; (b) rubber 2; (c) rubber 3; (d) rubber 4 (——, wheel; ----, rail).

shown in Figures 2 and 3(a) for all four wheels can be seen clearly to correspond to the frequency at which the wheel receptance begins to exceed that of the rail. This occurs at lower frequencies for rubber 2, 3 and 4 wheels due to the greater resilience. Since the rail receptance is still greater than that of the wheel between the wheel modes, the noise from the rail summed over a one-third octave band is not greatly affected.

4. WHEELS WITH CONSTRAINED LAYER DAMPING TREATMENT

4.1. ANALYSES

A study of wheel damping has been carried out on a standard 920 mm diameter wheel. This has a curved web profile designed to allow for the thermal/stress conditions of tread braking. It is shown in Figure 6 which also shows the extent of the damping treatment. In addition, an 860 mm wheel has been studied. This has a cross-section designed to generate less noise but to allow the wheel to be used in place of the 920 mm wheel with tread braking. The track model and train speed used are the same as for the study of the resilient wheel. As a basis of comparison, the sound power components were first predicted for each wheel without added damping treatment. The contact position on the wheel was set at an offset of 15 mm from the nominal running position at the centreline of the tread. This offset gives rise to a typical lateral component of excitation at the contact.

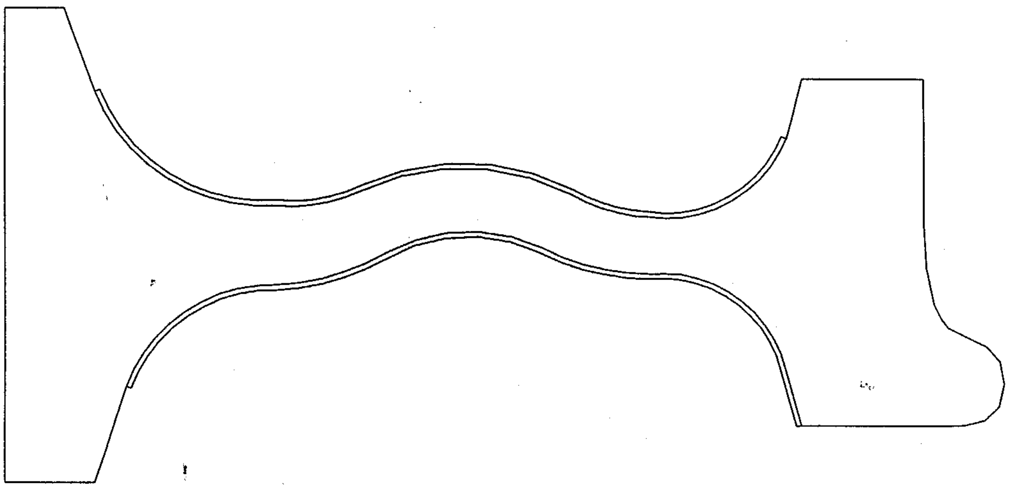

Figure 6. Profile of the 920 mm wheel showing the coverage of the damping treatment.

A number of predictions of the wheel noise reduction associated with different configurations of damping treatment have been carried out using the parameters for three readily available, suitable damping materials. The parameters of the damping materials have been taken at 0 and 20°C from the manufacturers' nomogramme data.

The natural frequency, mode shape, and modal masses have been assumed to be little affected by the addition of the damping layers and so the finite element model for the undamped structure has been used to derive these parameters. The complex modal analysis method described in Section 2.2 has therefore been used only to predict the modal damping of the complete structure. In these noise predictions, the full frequency dependence of the damping material loss factor is taken into account. However, it is not possible to interpolate the effect of Young's modulus in a similar way. It has therefore been checked that the effect of the frequency dependence of the damping material Young's modulus is, unlike that of the loss factor, insignificant in terms of the sound power predicted.

The effectiveness of the damping treatment is dependent on the stiffness of the constraining layer. For practical reasons a limit of 1 mm thickness of steel has been considered as it would be necessary to form this layer to the shape of the wheel.

4.2. RESULTS

Figure 7 compares the sound power spectra for the 920 mm wheel, the undamped 860 mm wheel and for the 860 mm wheel with the best damping material at 20°C. This case is for a 1 mm layer of the viscoelastic material and a 1 mm thick steel constraining plate. It was found generally that a 1 or 0·5 mm thickness of

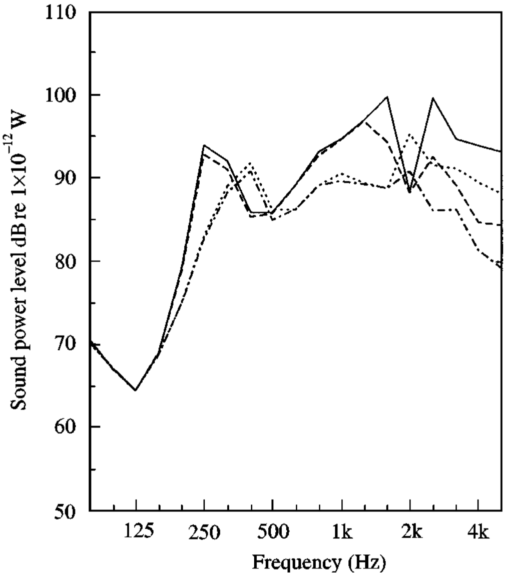

Figure 7. Comparison of wheel sound powers for the damped and undamped 920 mm wheel, and the damped and undamped 860 mm wheel when both are tread-braked: —— 920 mm undamped, 106·7 dB(A); ---- 920 mm damped, 101·8 dB(A); .... 860 mm undamped, 101·8 dB(A); -·-·-·- 860 mm damped, 98·8 dB(A).

damping material performed similarly well but that if the layer were 2 mm thick no significant reduction in rolling noise was predicted due to the damping treatment.

It can be seen that the damping treatment has reduced the sound power in the 2000 Hz and higher frequency bands by around 5 dB but not very much at lower frequencies. In the Figure, the spectra are shown unweighted, but the overall A-weighted sound power levels are given in the caption. From these it can be seen that the damper is predicted to reduce the noise from the 920 mm wheel by 3·7 dB(A) and from the 860 mm wheel by 3·0 dB(A). The smaller change in level for the 860 mm wheel is due to the different shape of the spectrum for the 860 mm wheel that is a result of its already improved cross-sectional shape with respect to noise generation. Since the spectrum of the damped 860 mm wheel is fairly flat between 300 and 2000 Hz, the overall wheel sound power would not be reduced significantly by further increases in damping.

At higher train speeds, the 2000 Hz and higher frequency bands would be more important in the overall level and the reduction in the overall level with the same damping treatment would be higher. The spectrum shape would also be more favourable in this respect for disc-braked wheels because of the different shape of the combined wheel and rail roughness spectrum. Figure 8 shows the predicted sound power spectrum for the two wheels, damped and undamped, when used with disc braking (i.e. applying a typical disc-braked wheel roughness spectrum). The reduction associated with the application of the damper is 5·1 dB(A) for the 920 mm wheel and 3·9 dB(A) for the 860 mm wheel.

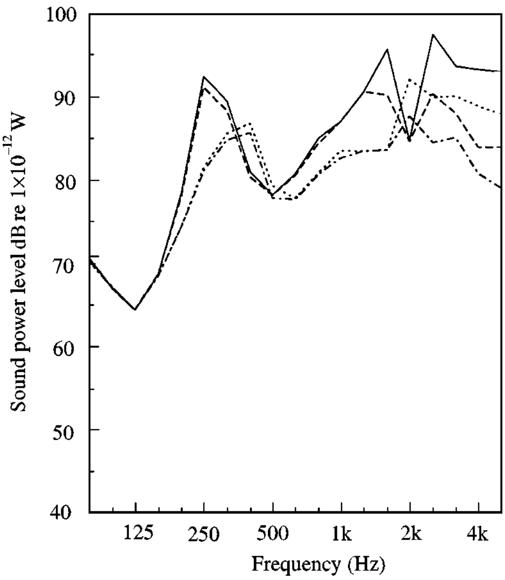

Figure 8. Comparison of wheel sound powers for the damped and undamped 920 mm wheel, and the damped and undamped 860 mm wheel when both are disc-braked: —— 920 mm undamped, 103·8 dB(A); ---- 920 mm damped, 98·7 dB(A); .... 860 mm undamped, 99·0 dB(A); -·-·-·- 860 mm damped, 94·8 dB(A).

4.3. EFFECT OF CONSTRAINING LAYER THICKNESS

In order to assess whether the 1 mm steel constraining plate is sufficiently stiff to obtain the maximum damping effect from the damping material, three analyses were carried out for a 1 mm viscoelastic layer at 0°C for the 860 mm wheel with tread-bracked roughness. The reduction in wheel sound power for constraining layer thicknesses of 1, 2 and 4 mm were predicted to be 2·3, 3·1 and 3·8 dB(A), respectively, indicating that significantly greater damping could be achieved if the constraining plate could be made stiffer.

5. SUMMARY

A technique of modelling the rolling noise from resilient wheels and wheels with constrained layer dampers has been developed and some example analyses have been carried out. The method uses a combination of a finite element complex modal analysis and the TWINS software.

The use of a resilient wheel design has been shown to have a significant effect on the track noise as well as a strong effect on the wheel noise. The noise from the wheel, in the case of the small LRV wheel design that has been studied, increases with decreasing stiffness of the resilient element. The total noise level, in this case, is dominated by the rail which has been shown to vary by a smaller amount than the wheel noise but can increase or decrease depending on the stiffness of the resilient element.

The reduction in the wheel noise associated with applying constrained layer damping treatments to a standard 920 mm wheel and an 860 mm wheel has been predicted. It has been shown that the noise reduction due to the damper is near the limit achievable for the tread-braked 860 mm wheel design at 100 km/h. This is due to the spectrum of sound power containing significant components in frequency bands below 2000 Hz where wheel damping has little effect. The damping treatment has been shown to be more productive on a disc-braked wheel because of the shape of the spectrum of excitation. If a tread-braked 920 mm standard wheel were replaced with the 860 mm acoustically improved design with damping treatment, the wheel sound level is predicted to be reduced by 7·9 dB(A). In the case of disc braking, the corresponding reduction in wheel noise is 9 dB(A).

ACKNOWLEDGMENT

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