inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 1.3

DESIGN OF A RAILWAY WHEEL WITH ACOUSTICALLY IMPROVED CROSS-SECTION AND CONSTRAINED LAYER DAMPING

C. Jones*, D. Thompson, A. Frid***, M. Wallentin******

* ISVR, Southampton University, Highfield, SO17 1BJ, Southampton, United Kingdom

** ISVR, Highfield, SO17 1BJ, Southampton, United Kingdom

*** ABB, Corporate Research, S 72178, Vasteros, Sweden

**** CHARMEC, Chalmers University, S 41296, Gothenburg, Sweden

Tel.: +44 2380 593224 / Fax: +44 2380 593190 / Email: cjcj@isvr.soton.ac.uk

Keywords:
RAILWAY NOISE, WHEELS, DAMPING

ABSTRACT
The rolling-noise generating characteristics of a railway wheel design have been studied theoretically. Two aspects of design for reducing noise have been investigated, optimisation of the cross-sectional shape of the wheel and addition of a constrained layer damping treatment. In this the thermo-mechanical behaviour of tread-braked wheels must be taken into account. To produce significant noise reduction, the damping achieved by the constrained layer treatment must exceed the effective 'rolling damping'. A finite element model is used to produce the modal basis of the wheel and to predict the modal damping of a wheel with visco-elastic layers a method of complex eigenvalue analysis has been used. The analysis indicates that the wheel component of rolling noise from the UIC standard tread-braked freight wheel can be reduced by more than 5 dB(A) by a combination of the two measures.

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 ballasted track has become well established and has been validated experimentally [1, 2]. The standard UIC 920 mm diameter freight wheel has a curved web profile designed to allow for the thermal/stress conditions of tread braking. Generally a wheel cross-sectional shape that is improved from the point of view of rolling noise generation can be achieved by straightening the curve in the web to reduce out of plane motion and stiffening in order to raise some resonance frequencies out of the frequency range of rolling excitation. However this leads to increased stresses during tread braking. Here a new cross-section has been produced as a compromise to generate less noise but still to allow the wheel to be used in place of the 920 mm wheel with cast-iron block, tread braking. A study of wheel damping is also performed.

2 - PREDICTION OF ROLLING NOISE FROM RAILWAY WHEELS

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, natural frequency, mass and damping for all modes of the wheel in the frequency range of interest (up to about 6000 Hz). A finite element (FE) model is used to generate the natural frequencies, mode shapes, and modal masses [1]. For wheels with no added damping, the modal description is completed by selecting modal damping ratios from measurement experience. In order to predict for wheels incorporating layers of visco-elastic material, a method of predicting the modal damping ratios is required to complete the modal basis. In this work these have been predicted using a method based on a complex modal analysis of the FE model of the wheel with different material damping for the steel and visco-elastic material [4]. 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 on the tire and web. These vibration 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 [5].

3 - WHEEL SHAPE

Figure 1 shows the cross-sectional shape of the UIC standard 920 mm freight wheel which formed the basis of the study and the design for an 860 mm diameter wheel which has been derived. In designing this wheel, the acceptance criteria in terms of maximum axial deflection, residual stress and fatigue stress [6] were checked using a thermo-mechanical finite element analysis simulating a prescribed drag-braking cycle. The method of proving a wheel's acceptability by FE prediction is allowed in the standard [6].

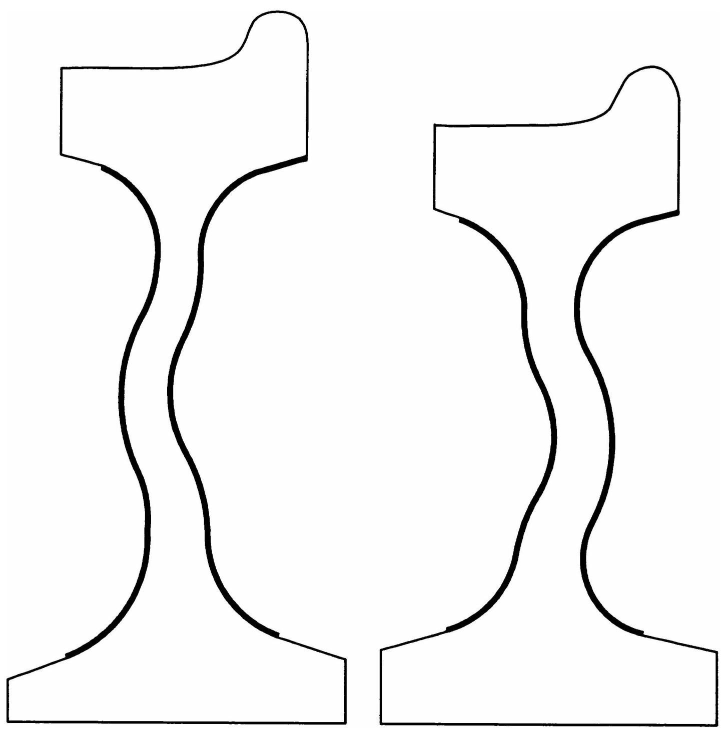

Figure 1: Profile of the 920 mm wheel (left) and 860 mm wheel (right) showing the coverage of the damping treatment.

4 - DAMPING

In addition to the improvement in the wheel shape, a number of predictions of the wheel noise reduction associated with different configurations of damping treatment have been carried out. Figure 1 also shows the extent of the damping treatment studied.

For conventional railway wheels without added damping treatment, the modal damping ratios can be based on experience of measured values and comparisons of measured and calculated accelerations. For the modes with $n \geq 2$, modal damping ratios can be assumed to be ζ_i=0.0001. The material loss factor, η for steel is between 2 $\times 10^{-5}$ and 3 $\times 10^{-4}$, depending on the type of steel and the modal damping ratio ($\zeta \approx \eta/2$) 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 ('rolling damping') is considerably greater [7]. Higher damping values have been found to be associated with modes with n=0 ($\zeta_i \approx$0.001) and n=1 ($\zeta_i \approx$0.01). These probably arise because of the influence of damping in the bearings of the wheelset especially where the modes involve a strong component of axle bending.

An important criterion for the choice of the visco-elastic damping materials is their ability to survive high temperatures during severe braking events. Predictions of these temperatures, which can reach in excess of 200°C on the back of the tire, were produced as part of the thermo-mechanical modelling of the wheel shapes. The temperature and frequency dependent parameters of materials were taken from manufacturers' nomogramme data. 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.
Predictions of the wheel sound power for a train speed of 100 km/h were made for the 920 mm and 860 mm wheel designs with a large number of configurations of damping treatment in which the material and the thicknesses of the constraining plate and visco-elastic layer as well as the extent of the damping treatment were varied. In these predictions, the full frequency dependence of the damping material loss factor was taken into account. It was found that the effect of the frequency dependence of the Young's modulus is, unlike that of the loss factor, not significant in terms of the sound power predicted. The performance of the damping treatment was predicted for damping material properties at 0°C and 20°C. For brevity only the results for 20°C are presented here.
Figure 2 presents the modal damping ratios achieved by a damper comprising a 1 mm layer of visco-elastic material and a 1 mm thick steel constraining plate. All modes indicated have frequencies below 6000 Hz. It was found generally that the performance of a 1 mm or 0.5 mm thickness of damping material performed similarly well but that if the layer were 2 mm thick no significant reduction in rolling noise due to the damping treatment was predicted. It can be seen from Figure 2 that the damping ratios obtained with the damper are high compared to the untreated wheel (damping ratios around 10^4). The figure additionally presents some values of equivalent 'rolling damping' taken from [7]. It can be seen that the damping treatment is successful in exceeding these values.

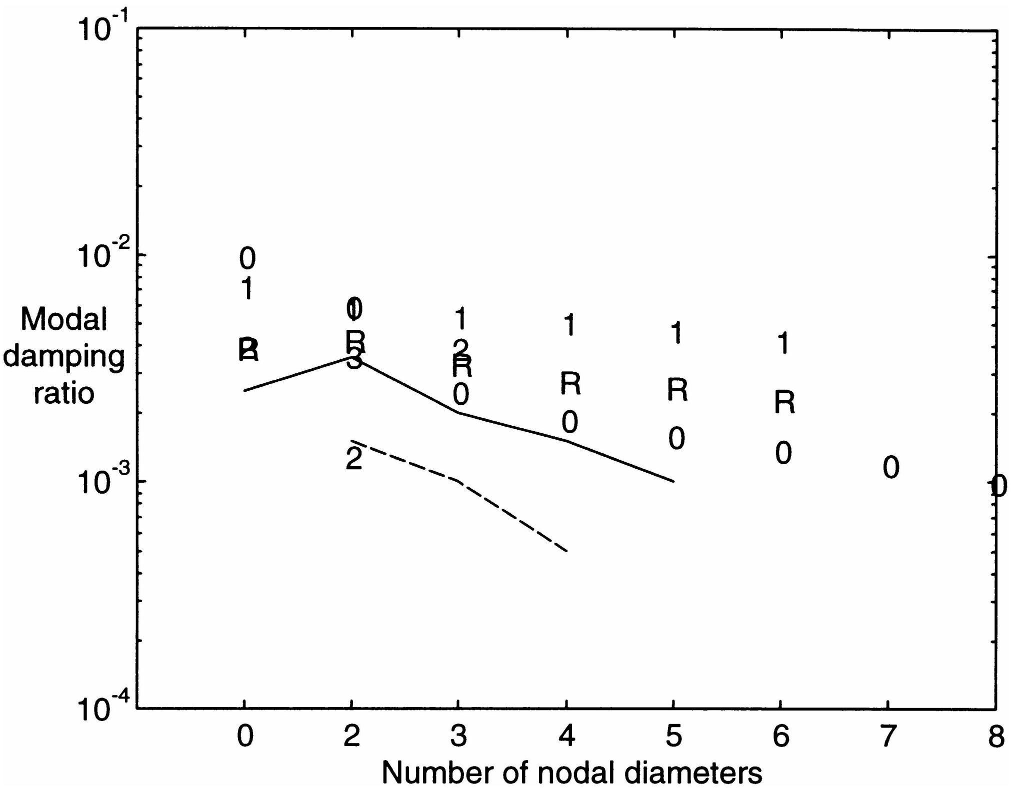

Figure 2: Modal damping ratios for the 860 mm wheel with damper; modes identified by 0: 0-nodal circle, 1: 1-nodal circle, 2: 2-nodal circle and R: radial modes; the lines indicate the 'rolling damping' for 1-nodal circle modes (–) and radial modes (- -) given in [7].

5 - ROLLING NOISE RESULTS

Figure 3 compares the sound power spectra at 100 km/h for the undamped 920 mm and 860 mm wheels, and for the two wheel designs with the optimum damping treatment at 20°C. These spectra are for the

sound produced by the wheel only. In interpreting the results, it must be understood that the total of rolling noise is made up also by components radiated by the rail and sleepers.
The A-weighted results indicate a reduction of 3.4 dB for the change of wheel cross-section. This prediction is in line with the results for this wheel obtained in running tests carried out as part of the Silent Freight and Silent Track projects [8].
It can be seen that the damping treatment has reduced the sound power in the 2500 Hz and higher frequency bands by more than 5 dB for both wheel designs, but very little reduction is achieved at lower frequencies. This corresponds to the range of frequency in which the wheels exhibit resonances that are significant in the generation of rolling noise. In the figure, the spectra are shown unweighted, but the overall A-weighted sound power levels are given in the caption.
From Figure 3 it can also be seen that the damper is predicted to reduce the noise from the 920 mm wheel by 2.7 dB(A) and from the 860 mm wheel by 2.0 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. The studies have shown that greater damping effect could be achieved for stiffer constraining plates. However, since the spectrum of the damped 860 mm wheel is fairly flat between 300 Hz and 2000 Hz at this train speed, the overall wheel sound power would not be reduced very much more by further increases in damping.
It is well known that replacing cast-iron block tread brakes with disc brakes reduces noise because the wheel roughness is reduced at wavelengths correspond to frequencies around 500 to 1000 Hz at 100 km/h. Starting from a disc-braked wheel roughness, the overall noise reduction achieved both by shape optimisation and damping would be greater because of the more favourable spectrum shape. The predicted reductions in wheel sound power for the two wheels due to the damper when used with disc braking (*i.e.* applying a typical disc-braked wheel roughness spectrum) are 3.9 dB(A) for the 920 mm wheel and 3.1 dB(A) for the 860 mm wheel. The reduction achieved using a damped 860 mm wheel compared to the undamped standard 920 mm wheel is therefore 6.6 dB(A) for the disc-braked case. Of course, for disc-braked wheels greater noise reduction could be achieved by further improvement to the wheel shape since the constraints imposed by the need to limit thermal stresses during braking are removed.

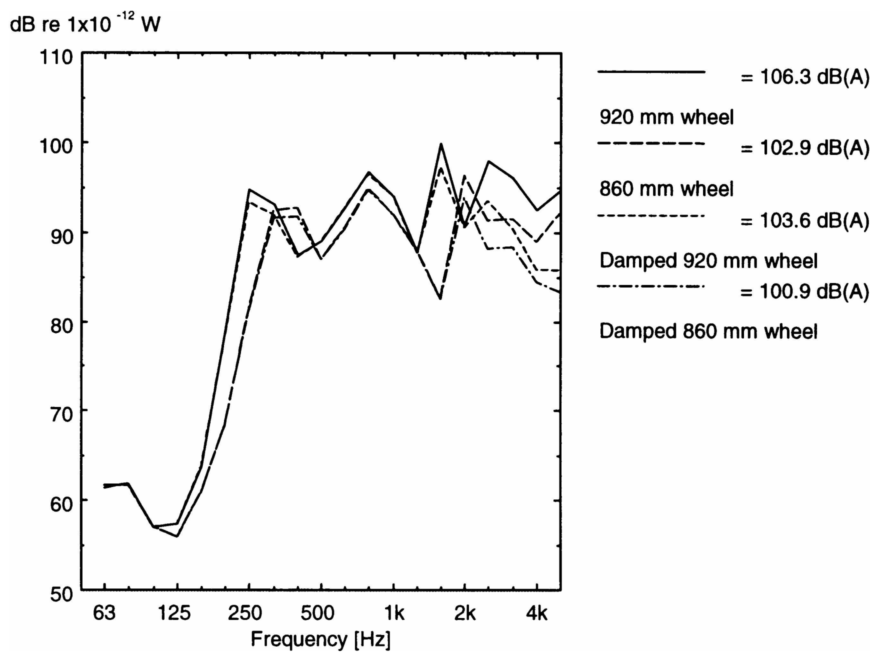

Figure 3: Comparison of wheel sound powers for damped and undamped wheels.

In a recent initiative by the International Union of Railways (UIC), cast-iron brake blocks on freight vehicles are being replaced by those of composite materials. This leads to a combined roughness spectrum close to that with disc-braked wheels and so will reduce noise. However, composite brake blocks lead to higher temperatures at the tire during braking than cast iron blocks do. This will lead to a shift in the balance between design for thermal stresses and noise generation. In these circumstances the principle of designing cross-sections for low noise as well as low stress remains important if some of the acoustic benefit of adopting composite brake blocks is not to be lost by the need to use low-stress wheels.

6 - SUMMARY

A technique of modelling the rolling noise from wheels with constrained layer dampers has been developed. The method uses a combination of a finite element complex modal analysis and the TWINS software. Using this, a design of wheel that reduces noise generation while still meeting the criteria for limiting thermo-mechanical stresses due to braking has been derived. The reduction in wheel noise associated with replacing a standard 920 mm wheel with a compatible 860 mm wheel has been predicted to be 3.4 dB(A). This increases to 5.4 dB(A) with the application of a constrained layer damping device. The damping treatment has been shown to be more effective on a disc-braked wheel because of the shape of the spectrum of roughness excitation. The limits of allowable stresses for tread braked wheels will, in the near future, be made more stringent. This is partly as a result of the replacement of cast-iron with composite brake blocks − itself a move to reduce noise. Under these circumstances the principle of designing the cross-section with noise generation in mind remains important in order to achieve the greatest acoustic gain of the lower wheel roughness.

ACKNOWLEDGEMENTS

The work was carried out under the Brite Euram project No. BE 95-1238 'Silent Freight', Contract No. BRPR-CT95-0047, funded by the European Commission and co-ordinated by the European Rail Research Institute.

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