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

I-INCE Classification: 4.7

THE DEVELOPMENT OF A RAIL DAMPING DEVICE FOR REDUCING NOISE FROM RAILWAY TRACK

D. Thompson*, C. Jones*, D. Farrington**

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

** Corus Rail, Moss Bay, Derwent Howe, CA14 5AE, Workington, Cumbria, United Kingdom

Tel.: +44 23 80592510 / Fax: +44 23 80593190 / Email: djt@isvr.soton.ac.uk

Keywords:
RAILWAY NOISE, WHEEL/RAIL NOISE, TRACK VIBRATION, DAMPING

ABSTRACT
The component of railway rolling noise radiated by the track dominates in the frequency range up to about 1.5 kHz and often can be more important than the wheel component for the overall noise level. A promising means to reduce the track noise is to increase the damping of the rail and so reduce the length over which vibrations are transmitted along the rail. In order to increase the damping of the rail significantly, particularly at frequencies below 1 kHz, a tuned mass-spring absorber system has been designed. To cover a wide range of frequencies, multiple tuning frequencies are used along with a visco-elastic material with a high loss factor. Suitable materials have been found from tests on samples and prototypes of the damper have been built and tested both in the laboratory and in the field. Results are very promising with reductions of the track component of noise of about 6 dB being found.

1 - INTRODUCTION

Reduction of rolling noise at source can be more attractive than high noise barriers, but this requires a thorough understanding of the mechanisms. Rolling noise is generated by the excitation of the wheels and track by their combined roughness. The relative importance of the noise radiated by the wheels, rails and sleepers depends on the design of wheels and track, the roughness spectrum and the speed of the train. A theoretical model, 'TWINS', for the generation of this noise has been available for some time [1, 2] and has led to the investigation of a number of noise-reducing technologies. This paper describes the development and testing of a rail damping device for reducing the vibration of the rail.

Figure 1 shows a typical one-third octave band spectrum of the sound power predicted for a single wheel and the associated track vibration. In this case, the sleeper is the dominant source of noise below 400 Hz, the wheel is the major source in the 1600 Hz band and above but it is the rail which is the greatest source, being responsible for the broad peak in the spectrum between 500 and 2000 Hz. The sound radiation from the vertical vibration of the rail is greater than that from the lateral component in this situation.

The sound power produced by the rail in a particular frequency band is inversely proportional to the decay rate of vibration along the rail, Δ in dB/m, since a high decay rate implies a short radiating length. The stiffness of the rail pads, located between the rail and sleepers, has a strong influence on the levels of noise from the track as it affects the isolation of the sleeper and the rail decay rate [3]. An optimum stiffness of rail pad exists at which the rail and sleeper components of sound are equal. However, this is very stiff compared to the pads now commonly adopted, low stiffness pads being used to reduce track component damage and track maintenance costs. The starting point for reducing the rail noise has been taken as a typical ballasted track with UIC 60 rails, monobloc concrete sleepers and a dynamic rail pad stiffness of 200 MN/m per pad. The spectra presented in Figure 1 represent this track. A damped mass-spring absorber system is considered in order to obtain high damping of the rail particularly in the range 500 to 2000 Hz.

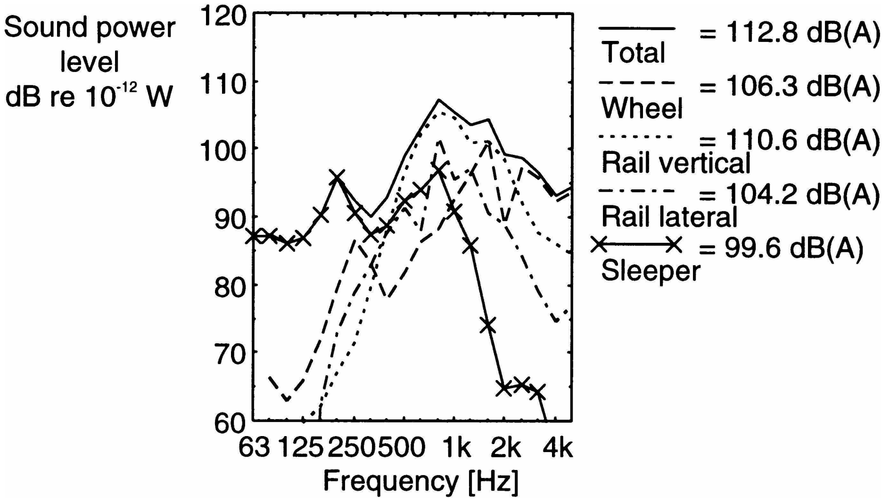

Figure 1: Predicted rolling sound power generated per wheel for a standard 920 mm freight wheel running on a standard track at 100 km/h.

2 - INITIAL STUDIES OF TUNED ABSORBERS

Initially the concept of a tuned absorber system is studied using a simple track model, based on a Timoshenko beam, to identify appropriate parameters. Firstly, the effect of the absorber on the rail vibration is calculated for an unsupported rail. This decay rate is then added to that for the untreated rail in track, as shown in Figure 2, which corresponds to a single continuous absorber, tuned to 800 Hz, with a mass of 17.5 kg/m and a loss factor of 0.35. The sound power is then calculated by modifying the component due to each wave according to the difference in combined decay rate.

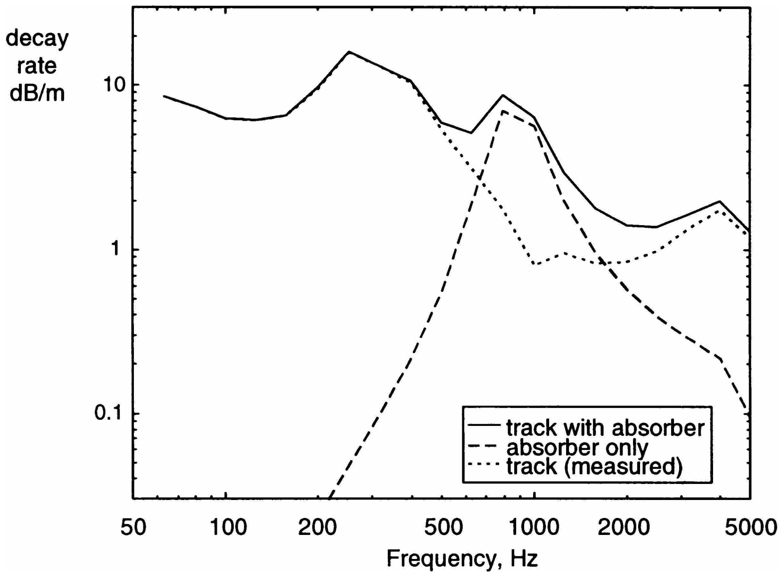

Figure 2: Track decay rates for vertical vibration of the reference track with a continuous tuned absorber system added.

By carrying out an extensive parameter study, it was found that the reduction in noise from the rail can be maximised by using a tuning frequency around 800 Hz. The optimum frequency depends slightly on the train speed. The performance increases with increasing mass; the value of 17.5 kg/m being used for practical reasons. The damping loss factor affects the breadth and height of the peak in the decay rate; it is found that increasing the loss factor above 0.5 produces no further reduction in overall noise level. Another novel method of broadening the peak is to use multiple tuning frequencies. The parameter study has therefore been extended to study the noise reduction for a system with two tuning frequencies. The optimum configuration gave a noise reduction of 5.9 dB(A), which is a gain of 0.6 dB over a single-

frequency absorber. Moreover, a two-frequency system is more robust for variations in train speed, train type or temperature. A three-frequency system, also studied in the same way, is found to give only very small improvements over the two-frequency system.
Although the two-frequency system could be produced using two separate single-degree-of-freedom systems, an alternative is explored in which a two-degree-of-freedom system is added to the rail. Not all combinations of frequency and mass ratios are then possible, but the optimum system that is found corresponds closely to the optimum of the two SDOF systems, and the noise reduction is identical.

3 - IMPLEMENTATION

The highest priorities in designing railway track are safety, low cost of maintenance and standardisation. For new methods of reducing noise to be accepted, they must be capable of being fitted to the established infrastructure design with minimal change to railway practice. Therefore, the present development has concentrated on a design of rail damper that may be fitted to the standard UIC60 rail section. It is confined to a region at the base of the web and top inner part of the rail foot, that does not interfere with track handling machinery, nor with the rail clips.
The rail absorber is designed with two masses on each side of the rail, separated from each other and from the rail by an elastomeric material. The system has two resonances in the vertical direction in the desired ratio. Finite element models of the rail and absorber cross-section are used to study the design in detail, allowing the Young's modulus and loss factor of the elastomer to be specified. The specification has been sent to a number of suppliers and for many material samples the dynamic Young's modulus and loss factor over the frequency range from 300 Hz to 3000 Hz were measured, thereby guiding the suppliers toward the design of the most suitable material. In addition, a number of test samples of 15 cm lengths of rail were assembled with the absorber using the different materials and tested to ensure that the expected tuning frequencies are obtained in practice.
Figure 3 shows the material parameters for one of the materials which is used in subsequent prototypes. In order to meet the requirement for a high loss factor, the transition between the glassy and rubbery phases of the elastomer occurs in the working temperature range. There will therefore be a dependence of both parameters on frequency and temperature for a given elastomer.

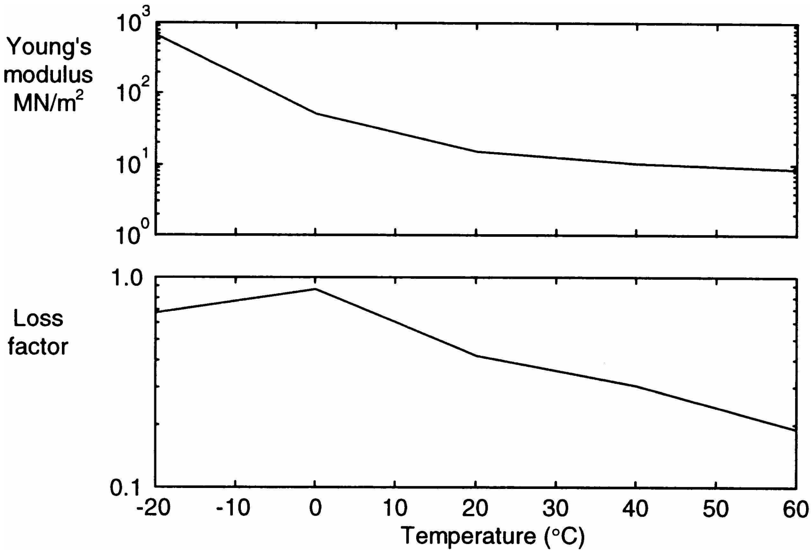

Figure 3: The temperature dependent Young's modulus and loss factor of the elastomer.

4 - DETAILED MODELLING

For the detailed evaluation of the specific design, a periodic structure model of the track is used [4]. In this, a short slice of the rail cross-section and damper are modelled using finite elements. The slice is then assumed to be repeated to form an infinite structure. The decay rates are calculated allowing for the frequency and temperature dependence of the material properties of the elastomer. The resulting predicted decay rates of the free rail with absorbers are presented in Figure 4. Measured decay rates, taken on a 4 m length of rail fitted with the absorber, are also given.
The noise reduction can be predicted using these decay rates, as in Figure 5 which is based on the measured decay rates. Thus at 20°C a reduction of the rail noise of 5.4 dB(A) is expected. From

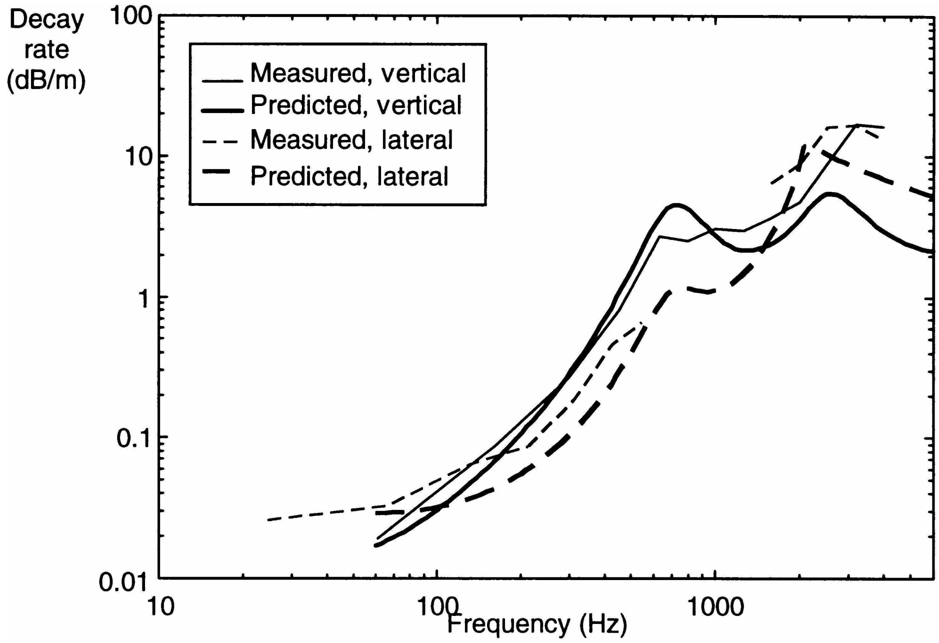

Figure 4: Predicted decay rates compared to measurements on a 4 m long freely suspended rail.

the predicted decay rates the reduction is 5.6 dB(A). A prediction of the decay rates for the elastomer properties at 0°C shows that, due to the stiffening of the material and consequent raising of the operating frequencies of the absorber, its benefit would be reduced to 3.4 dB(A). This effect must be taken into account in further refinement of the absorber. However, it must also be offset against the changes in decay rate of the untreated track with temperature.

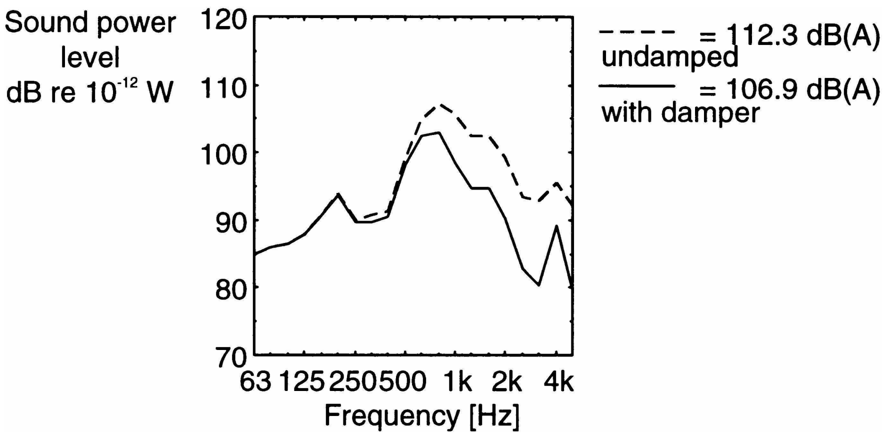

Figure 5: Predicted rail sound power with and without the absorbers.

5 - FIELD MEASUREMENTS

Field measurements on a track fitted with a prototype rail absorber were conducted in May 1999 as part of the 'Silent Track' project [5]. The decay rates in the track were measured using impact excitation and rolling noise measurements were made using a test train. The rail temperature was in excess of 30°C, which meant that the rail pads were softer than was the case in the previous decay rate measurements (Figure 2), and the rail noise component would initially be higher. On the other hand, the rail damper elastomer would also have a lower stiffness and therefore a lower tuning frequency, found to be around 500 Hz.

An average of the sound pressure at three microphones, 3 m from the near rail at different heights, is used. The levels measured from a test train passing at 100 km/h with and without the absorbers are shown in Figure 6. The measurement shows an overall reduction of 5.6 dB(A). In order to show the reduction in track noise most clearly, these results are for a low-noise wheel [6,7]. This had no effect on

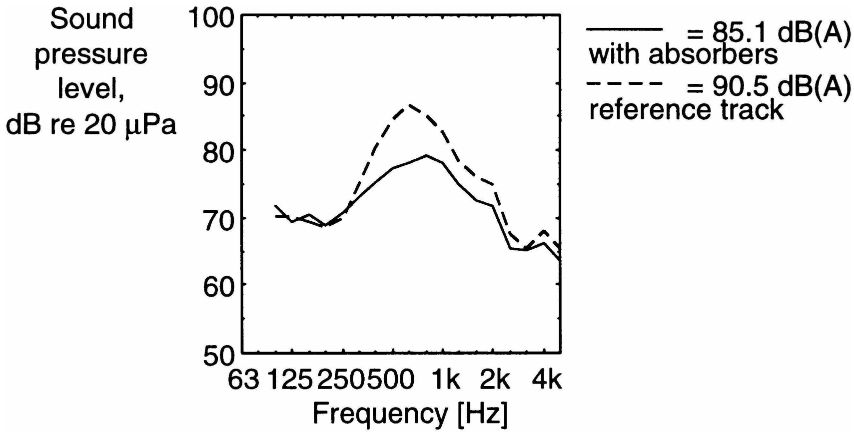

Figure 6: Results of measurements of rolling noise at 100 km/h using a vehicle fitted with a noise reducing wheel. Average from three microphones at 3 m from near rail.

the track component of noise. However, the overall measured noise is influenced significantly by the noise from the wheel in the 1600 Hz band and above. The full reduction of the track noise is therefore not seen in these bands. It is estimated that the track noise is reduced by 6.0 to 6.2 dB(A) by the absorber. These results are consistent with the predictions.

6 - SUMMARY

A damping tuned absorber to reduce the rail component of rolling noise has been developed which gives a broad band reduction in noise between 500 and 2000 Hz. An initial parameter study was used to obtain the main parameters which were realised in a design using steel masses and an elastomer formulated to a specification. Laboratory testing confirmed the results of detailed modelling. Field tests of a prototype absorber demonstrate that it reduces rail noise by about 6 dB(A) for a modern track design with soft rail pads. This falls at low temperatures when the elastomer properties become stiffer. The benefit is reduced for a track with a stiffer pad, but the design has been optimised for soft pads as they are being more widely used for reasons of reduced track damage.

ACKNOWLEDGEMENTS

This work was undertaken as part of the Silent Track project funded by the European Union (Brite Euram project BE 96-3017, contract BRPR-CT96-0258).

REFERENCES

1. **D.J. Thompson, B. Hemsworth and N. Vincent**, Experimental validation of the TWINS program for rolling noise, part 1: description of the model and method, *J. Sound. Vib.*, Vol. 193(1), pp. 123 - 136, 1996

2. **D.J. Thompson, P. Fodiman and H. Mahé** , Experimental validation of the TWINS program for rolling noise, part 2: results, *J. Sound. Vib.*, Vol. 193(1), pp. 137 - 148, 1996

3. **N. Vincent, P. Bouvet, D.J. Thompson and P.E. Gautier**, Theoretical optimisation of track components to reduce rolling noise, *J. Sound. Vib.*, Vol. 193(1), pp. 161 - 172, 1996

4. **D.J. Thompson**, Wheel-rail noise generation part III: rail vibration, *J. Sound. Vib.*, Vol. 161, pp. 421 - 446, 1993

5. **B. Hemsworth**, An overview of the Silent Freight and Silent Track projects, In *Internoise 2000, Nice, France*, 2000

6. **C.J.C. Jones, D.J. Thompson, A. Frid and M.O. Wallentin**, Design of a railway wheel with acoustically improved cross-section and constrained layer damping, In *Internoise 2000, Nice, France*, 2000

7. **P. Bouvet, N. Vincent, A. Coblentz and F. Demilly**, Rolling noise from freight railway traffic: reduction of wheel radiation by means of tuned absorbers, In *Internoise 2000, Nice, France*, 2000