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ROLLING NOISE FROM FREIGHT RAILWAY TRAFFIC: REDUCTION OF WHEEL RADIATION BY MEANS OF TUNED ABSORBERS

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ABSTRACT
Rails, sleepers and wheels are the predominant acoustic sources for freight rolling noise. This article deals with the optimisation of dynamic absorbers fitted on a wheel with an optimised shape. The theoretical methods used for the design of tuned absorbers are presented. A finite element model is used to calculate the wheel modal parameters. The wheel vibration response and radiation are calculated with the TWINS (**T**rack **W**heel **I**nteraction **N**oise **S**oftware, developed and validated with the support of the European Rail Research Institute, ERRI) software. It is shown that in combination with a wheel shape optimisation, tuned absorbers lead to a wheel noise reduction of about 10 dB(A). Dynamic absorbers are developed and set on prototype wheels. These wheels are mounted on a test train in order to assess the actual noise reductions during field measurements: a good agreement is found between measurements and calculations.

1 - INTRODUCTION
Rolling noise is the dominant source of external noise from freight trains. Hence railway wheelset manufacturers are required to produce new design of wheel that generate a lower contribution to rolling noise. In the frame of the "Silent Freight" European project, several concepts of optimised wheels have been developed.
These optimised wheels were designed from calculations, mainly based on the TWINS software. To validate these "low-noise" components, prototype wheels were manufactured, and a measurement campaign on a test train running on a test track was carried out.
This paper presents the calculation method used to design dynamic absorbers fitted on a wheel with an optimised shape. The physical phenomena accounting for wheel radiation are discussed in section 2. The results of calculations are presented section 3.1, and are compared with the validation measurements in section 3.2.

2 - WHEEL MODELLING

2.1 - Rolling noise mechanism
Wheel and rail roughness introduces a vertical relative displacement between wheel and rail. The extent to which wheel or rail moves in response to this relative displacement depends on their respective mechanical receptances, in the vertical direction. These receptances (calculated with TWINS) are compared in figure 1, for a freight configuration.
Below 1600 Hz, the rail vertical receptance is much higher than for the wheel: the relative displacement is mainly converted into rail vibration, the wheel vibration remaining very low. Consequently, the track turns out to be the major radiator over the $100 - 1600$ Hz range, with a high contribution in the 800 $-$ 1000 Hz one-third octave bands. The wheel response is essentially related to excitation of its natural frequencies with a significant radial amplitude at the contact point (radial modes called Rn and 1-axial

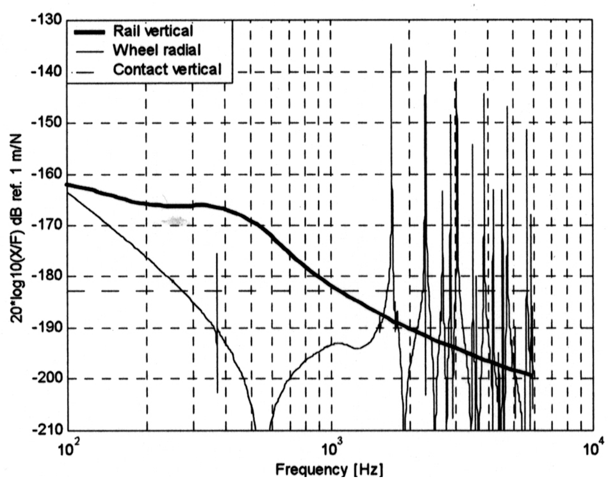


Figure 1: Track, wheel and contact receptances at wheel/rail interface; vertical direction; freight configuration (UIC60 rail with twin bloc sleepers and ballast, freight wheel).

modes called $1Ln$, n being the number of nodal diameter). For conventional freight wheels, these modes generally occur at frequencies higher than 1600 Hz.
Increasing the modal damping of these modes is a potential way to reduce the wheel noise. This can be achieved with tuned absorbers, fitted on the wheel.

2.2 - Procedure to design dynamic absorbers on wheels

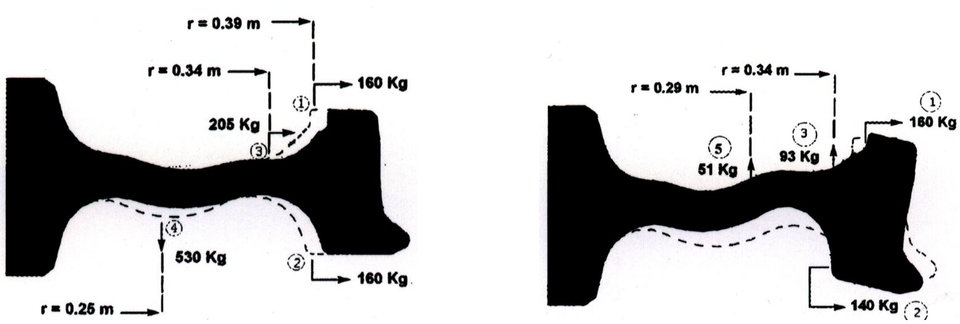
In order to damp one of these modes by means of dynamic absorbers, the general procedure is the following.

Selection of the active mass m_a of the absorber:

To damp a given wheel mode N, the total active mass of absorbers implemented around a wheel circumference is related to the apparent modal mass M $_N$ at the connecting point of the absorber on the wheel by:

$$\mathrm{m_a} = 2\mu \mathrm{M_N}$$

μ being the "mass ratio". In practice, μ is chosen between 0.05 and 0.2. In order to minimise the mass of the absorbers, their connecting points must be chosen close to a vibration anti-node, where the modal mass is lower.
The concept of apparent modal mass is illustrated figure 2, for the main radiating modes of the wheel the $R2$ (2238 Hz), $1L2$ (2623 Hz) modes. The points 1 and 2, on the inner side of the tread, represent the best locations: dynamic absorbers, acting in radial direction, can be designed to damp radial and 1-axial modes simultaneously.



(a): Radial mode $R2$ (2240 Hz). **(b):** Axial mode $1L2$ (2620 Hz).
Figure 2: Apparent modal mass at specific point on wheel.

Selection of the tuning frequency f_a of the optimal loss factor:

The determination of the other parameters is straightforward. The optimum tuning frequency f_a to damp the wheel mode f_N and the optimum loss factor η of the rubber are given by:

$$f_a = \frac{f_N}{1+\mu} \text{ and } \eta = 2\left(\frac{3}{8}\frac{\mu}{1+\mu}\right)^{1/2}$$

For example, for $\mu = 0.1$, one obtains $\eta = 0.36$. However, in practise, values of about $\eta = 0.2$ are easier to achieve with rubber.
Although a damper is tuned to a reference frequency f_a related to the treatment of one specific mode, the higher modes of the wheel are also affected by the damper. This effect, called here "residual" damping, allows the tuned absorbers to be efficient on a broad frequency range, although they are tuned at discrete frequencies.

2.3 - Wheel modelling

In TWINS, a modal approach is used to describe the wheel. For conventional wheels (steel monobloc), the modal parameters (frequency, mode shape and mass) are issued from a Finite Element Analysis, preferably using a axisymetric model. The wheel receptance at the contact point is calculated with a modal summation, and used – with the rail and contact receptances – to estimate the forces at the wheel rail interface. The wheel vibrational response is then calculated by modal summation, and the use of modal radiation efficiencies allows to estimate the acoustic power from the wheel.
For wheels equipped with dynamic absorbers, the main difficulty is to estimate the modal damping with a sufficient reliability. The prediction of modal damping can be achieved using a complex modal analysis [4], or by analysing the ratio between the modal elastic energy stored in the rubber and the modal elastic energy of the whole wheel [5]. An axisymmetric mesh of the wheel with the absorbers is used, the active mass of the absorber being uniformaly spread over the wheel circumference (Figure 3). Several models were built, to study the influence of absorbers parameters (active mass, number of tuning frequencies, rubber damping ...).

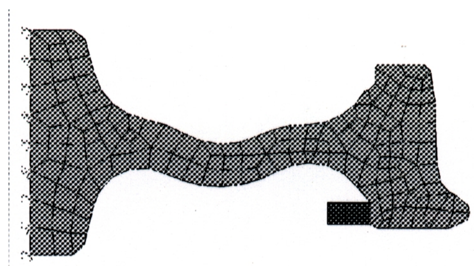


Figure 3: Axisymmetric mesh of the wheel with one set of dynamic absorber, tuned at a single frequency (not to scale).

3 - RESULTS

3.1 - Parameters and results of calculations

Two configurations of dynamic absorbers have been considered (Table 1). Calculations were carried with the first set of absorbers, and with the two sets. The figure 4 gives the acoustic gain provided by these 2 configurations.

	Set 1	**Set 2**
Modes mainly treated	R2, 1L2	R3, 1L3
Absorber location	Point 2	Point 1
Active mass	25 Kg	15 Kg
Tuning frequency	2200 Hz	3000 Hz
Damping loss factor η	0.2	0.2

Table 1.

The implementation of the two sets of tuned absorbers on the 860 mm optimised wheel shape provides a reduction of 5.3 dB(A) of the wheel sound power. Compared to a reference 920 mm wheel, the wheel acoustic power is reduced by 10 dB(A) (5 dB(A) due to the shape optimisation and 5 dB(A) due to the

tuned absorbers. Most of the acoustic gain is due to the first set of dynamic absorbers (25 kg of active masses tuned at 2200 Hz).
Although the wheel sound power is spread over a broad frequency range (2000 − 5000 Hz) a single tuning frequency is still acceptable, thanks to the residual effect of the absorbers on the higher wheel modes. Furthermore, it was shown that the wheel noise reduction was not very sensitive to frequency mis-tuning of the absorbers.

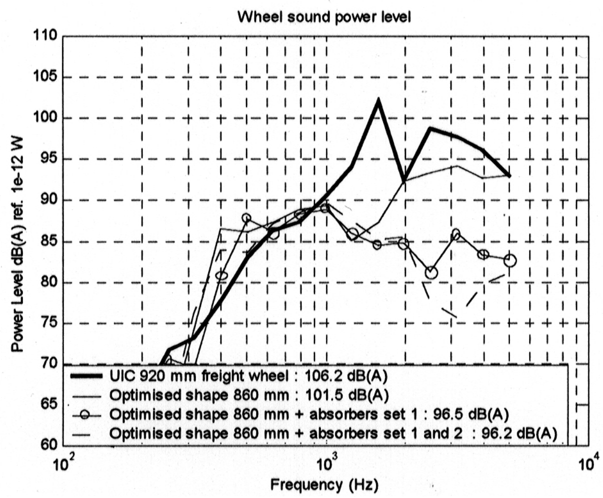


Figure 4: Sound power radiated by one wheel, at 100 km/h (dB(A), calculation).

3.2 - Experimental results

Specifications were defined, based on F.E.M. and TWINS calculations (total active mass of 25 kg, tuned at about 2200 Hz). The prototype wheel is presented figure 5. Nine identical dampers are implemented on the inner side of the wheel tread. Each damper is made of steel piece, connected to the wheel tread with a rubber material.
Due to the track contribution to the overall noise, the wheel noise reduction has been assessed indirectly, by two ways:

- firstly, by tuning the calculated modal basis to the measured modal basis, and calculating the wheel radiation with TWINS. A wheel noise reduction of 9 dBA was estimated.
- secondly, by measuring the wheel vibration velocity on a rolling train (see figure 6) and by deriving the wheel sound power from the vibration velocity. A wheel power reduction of about 7 dBA has then been found.

Furthermore, the radiation of the absorbers was found negligible (using velocity measurements on the absorbers, and assuming radiation efficiency of baffled pistons).

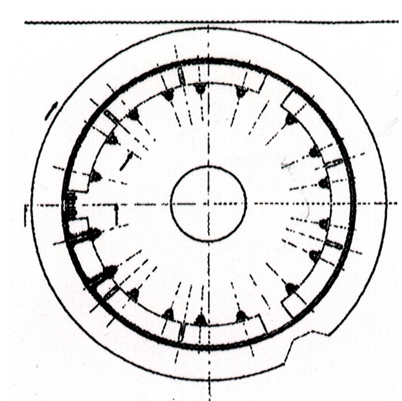
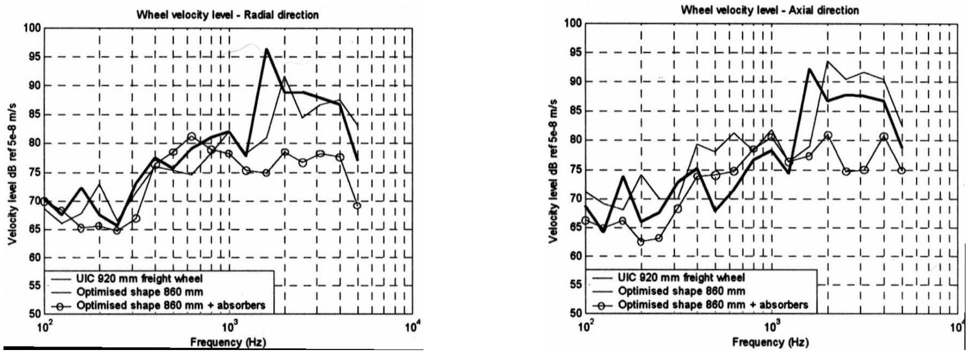


Figure 5: Prototype wheel equipped with tuned absorbers.



(a): Wheel tread, radial direction. **(b):** Wheel web, axial direction.

Figure 6: Measured wheel vibrations, when rolling at 100 km/h.

4 - CONCLUSION

Compared a standard freight wheel with a 920 mm diameter, the combination of wheel shape optimisation and dynamic absorbers can lead to a 10 dBA reduction of the wheel acoustic power. Only one tuning frequency is sufficient to provide a significant decrease of the wheel sound power.

Prototype wheels were manufactured, based on these calculations. The experimental noise reductions were found in good agreement with the computational results.

ACKNOWLEDGEMENTS

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