DEVELOPMENT OF A VIBRO-ACOUSTICAL METHODOLOGY FOR THE DESIGN AND VALIDATION OF LOW NOISE RAILWAY WHEELS

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Summary

The development of a feasible low noise emission wheel is an extremely actual goal as the sensibility to environmental problem rapidly grew up in the past few years.

As a fundamental part in this task it is necessary to define the experimental methodologies to test and qualify low noise wheels. This work presents an analysis procedure to investigate the dynamic and acoustical behaviour of a railway wheel. A modal analysis of a standard Lucchini wheel under free and loaded conditions is carried out with standard procedures, while custom acoustical tests (sound power emission and directivity) are described. The procedure led to a correlation between the acoustical behaviour and the vibration modes of the wheel and to understand how each eigenmode participates at the noise emission for defined excitation conditions.

In the second part of this work results relative to some damped wheel prototypes are shown. Constrained-layer damping techniques were used to increase the structural loss factor in the frequency ranges typical of rolling, impact and squealing noise.

Key Words: Railway wheel design, noise reduction, damped wheels, vibro-acoustical tests.

Introduction

Railway noise is an extremely actual topic that certainly needs high resources and advanced researches to be tackled. Several authors published very interesting results, and a good collection of recent papers can be found in the proceedings of the Fifth International Workshop on Tracked Transit System Noise (1996).

This work deals with the procedure to design and to verify the vibro-acoustical properties of a railway wheel. Even if the wheel is only one of the responsibles of railway noise, the other being clearly the track, numerous researches are actually in progress to reduce as more as possible its emission. As a step towards the design of new low noise wheels, Lucchini CRS developed a methodology for the qualification of a generic wheel. The 890 mm monobloc wheel that equips the Pendolino and ETR500 trains of the Italian State Railways FS was chosen to validate these procedures.

Experimental modal analysis of the free wheel

One of the fundamental steps in the vibration analysis of a wheel is the investigation of its modal behaviour. Several authors performed modal analysis on undamped (Thompson, 1991) or ring-damped (Lopez, 1996) wheels and wheelsets (Bracciali, 1994), and a complete analysis of the standard wheel is presented here.

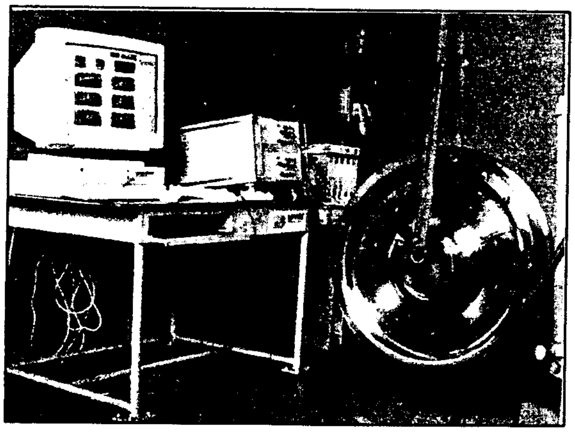

Fig. 1. *Standard 890 mm wheel elastically suspended for free-free experimental modal analysis.*

The wheel was suspended and excited with an instrumented hammer, reproducing the "free-free" conditions (Fig. 1). The measurement chain consists of a workstation and an 8-channel front-end, while the excitation and the response were measured with a piezoelectric load cell and two triaxial accelerometers respectively. Measurements in the three orthogonal directions were performed in 110 points; excitations were supplied in two different points, one on the tread and one on the web, at 45° degrees w.r.t. the local cylindrical reference triad through a prism glued to the surface. These excitations allowed the analysis of coupled modes and the analysis of circumferential modes, that are probably negligible in the acoustic emission of the wheel but that must be considered anyway in the complete dynamic model of the wheel. Their extraction will certainly prove to be fundamental for future analysis on traction/braking conditions and for FEM models validation. Accelerometers fixed by bees wax enabled the measurement of 660 FRFs up to 8 kHz with a very high resolution (0.5 Hz).

Eigenmodes description

The wheel exhibited a very high modal density (about 50 modes up to 7 kHz with a slight concentration in the 3÷4 kHz band) with very low damping ratios, typically lower that 0.1%. Eigenmodes are as more complicated as the frequency increases, even if a relatively precise classification can be made. It is worth to underline that the web is curved and that a clear distinction between radial and axial modes can not be made. The modes are grouped here on the basis of the wheel elements partecipation to eigenmodes.

In flexural modes the deformations mainly affect the tread while the web and the hub remain practically undisturbed. In these modes, that are 8 up to 7 kHz, the tread moves in the axial direction and rotates around its virtual constraint on the web, generating in the latter only little vibrations. This is valid for all these modes but for the first (325 Hz), where a global flexural deformation of the tread and the web is observed. For the flexural mode of order *n*, *n+1* nodal diameters are observed.

Radial modes are more complex and more numerous (they are 31 up to 7 kHz) and show a high coupling between the tread and the web; also the hub is sometimes involved in the movement. In these modes the tread moves radially (in plane movement) while the web moves axially. Radial modes can be classified in two families:

- a first family, to which belong most of these modes, where the lateral sides of the tread move radially and in counterphase, leading to a corresponding axial movement of the adjacent parts of the web. Also here the number of nodal diameters is *n+1* where *n* is the mode order; axial web displacements are present between two nodal diameters;
- a second family, with less modes, where both lateral sides of the tread moves radially in phase and the web is stressed radially leading to some flexure due to its curvature.

Other eigenmodes were identified in the investigated frequency range: two axial modes with respectively one and two nodal circles, three modes where the hub rotates around a wheel diameter and a circumferential mode where the tread and the hub rotate in counterphase around the wheel axis. As an example, the point FRF on the tread and two eigenmodes are shown in Figs. 2 and 3.

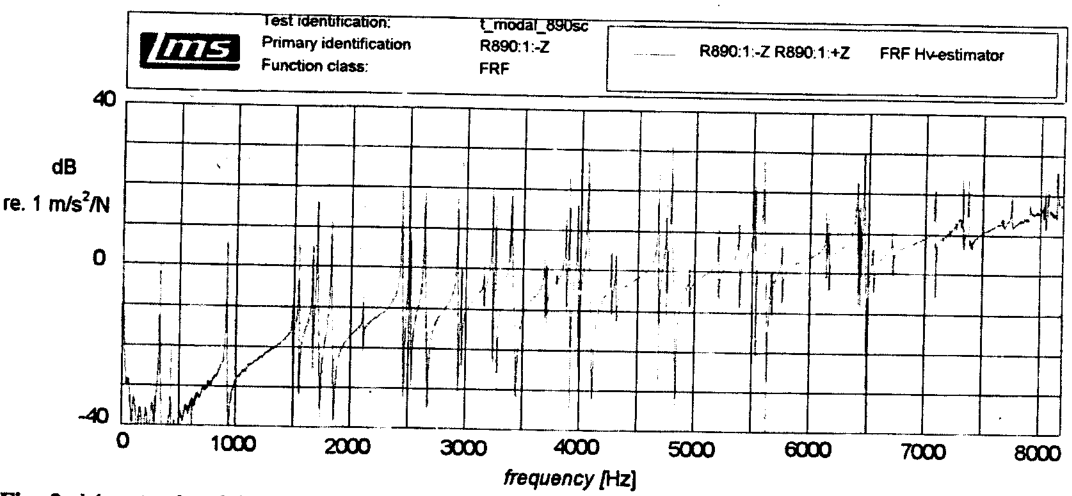

Fig. 2. *Magnitude of the point FRF on the wheel tread. Measurement direction is at 45° with respect to the local reference system.*

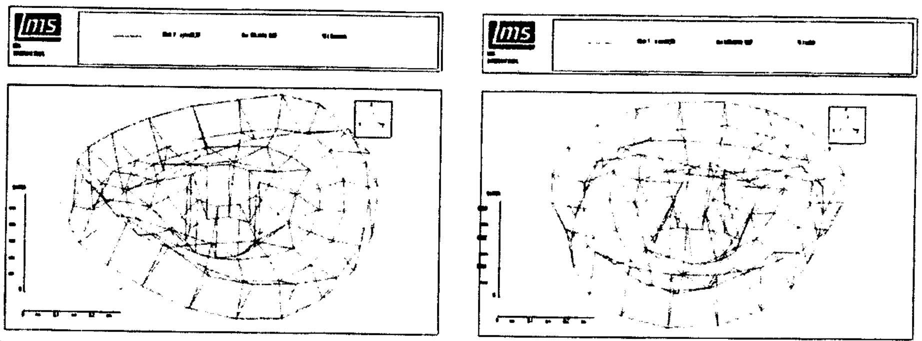

Fig. 3. *1st flexural mode (f=325 Hz, ζ=0.05%) and 1st radial mode (f=1485 Hz, ζ=0.03%).*

Experimental modal analysis of a dummy wheelset vertically statically loaded

The idealized condition of a free suspended wheel is particularly convenient for comparison tests but it is too far from the real operating conditions of a wheel. Lucchini CRS is planning the construction of a roller test bench dedicated to acoustical measurements; as a first step to approach real conditions, results obtained by statically loading the wheel are presented here.

The wheel was fitted on a dummy axle of the QPR1 test bench used in the Lucchini plant mainly for strain gauge tests on wheel prototypes. The bench can oleodinamically supply controlled vertical loads (Q) on the axle spindle and lateral loads (Y) on a dummy rail on which the wheel is supported (Fig. 4). Modal investigations were performed under several vertical loads (5, 25, 50, 75 and 100 kN); the wheel was vertically supported on a 30 cm long UIC 60 rail. The wheelset was excited with an electromagnetic shaker and point FRF is measured with an impedance head. Two excitation points were chosen, one close to the wheel-rail contact inclined of 45° w.r.t. the wheelset plane, and one in a higher position to excite the system in a more general way. Measurement points were placed on the wheel, on the axle, on the rail, on the test bench structure and on the floor, to investigate the dynamic behaviour of all the system, including the surrounding.

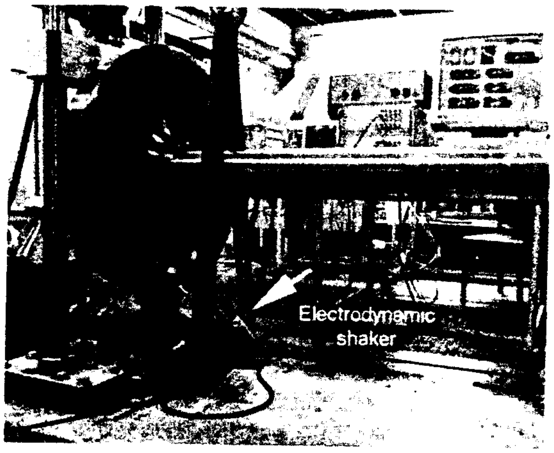

Fig. 4. *The wheel mounted on the dummy axle and loaded under the QPR1 test bench.*

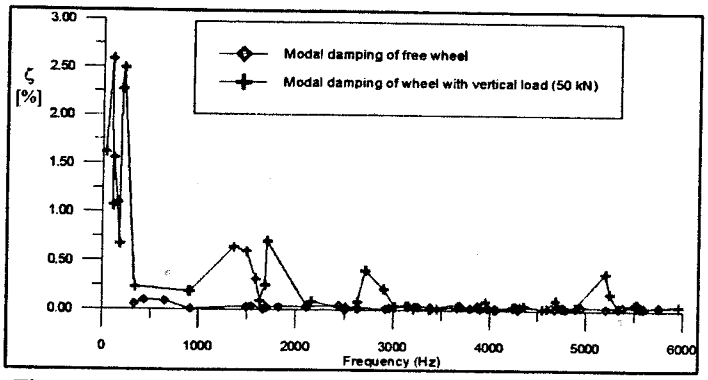

Fig. 5. *Damping ratio for free wheel and 50 kN vertical loaded wheelset.*

Eigenmodes description

Compared to the suspended wheel, the modal density is practically unchanged above 300 Hz; below there are eight modes were the axle bends and the wheel moves rigidly. Resonance frequencies and damping ratios remain constant varying the load except for a limited number of modes where damping increases probably because the input energy flows through the rail to the surrounding leading to an equivalent higher damping (Fig. 5).

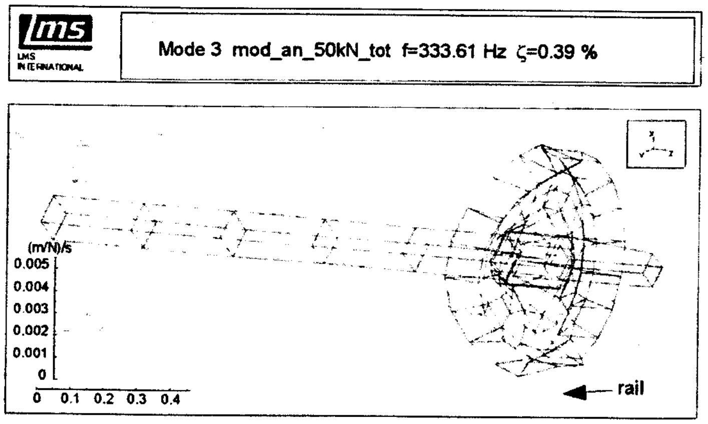

Fig. 6. *Eigenmode of 50 kN vertically loaded dummy wheelset*

Eigenmodes can be classified with the aforementioned rules as radial and flexural modes. The rail partecipates to almost all modes and it can not be considered rigid at all, and this is particularly evident for the wheel axial modes (Fig. 6). The tread tends to roll on the rail because of the punctiform contact. Hub radial movements disappeared, while tread and web movements are very similar to the suspended wheel case.

Experimental acoustical analysis

The complete description of the vibro-acoustical behaviour of a wheel requires the measurement of its sound emission properties.

The parameters chosen to qualify and to quantify noise emission are the sound power and the directivity. The sound power emitted by a source can be measured through sound pressure measurements on a grid (ISO 3744, 1994), sound intensity measurements at discrete points (ISO 9614-1, 1993) and sound intensity measurements by scanning (ISO 9614-2, 1996). In this work the ISO 3744 approach was used, as an essentially free field over a reflecting plane was available at the Lucchini plants. The wheel stands horizontally, is elastically supported on a pedestal and it is excited at 45° by an acoustically isolated electrodynamic shaker (Fig. 7, left); measurement surface, according to the norm, is a 90° 2.2 m radius arc that constitutes the hemispherical measurement surface on which microphones are placed. The side of the wheel that faces the floor is acoustically isolated by a sealed lead sheet, such that the measured emission is relative only to the upper side of the wheel.

The ratio of the pressure over the force (FRF) was measured each 10° on the measuring arc to obtain information on the directivity of the sound emission (Fig. 7, right). This analysis is important as the directivity lobes can identify the wheel as a monopolar or dipolar source (Remington, 1976, Scarano, 1991).

The sound power was calculated according to ISO 3744 again normalizing the measured pressure to the input force. The normalized 1/24 octave band sound power emission shown in Fig. 8 allows to assign a sound power value to eigenmodes in each band.

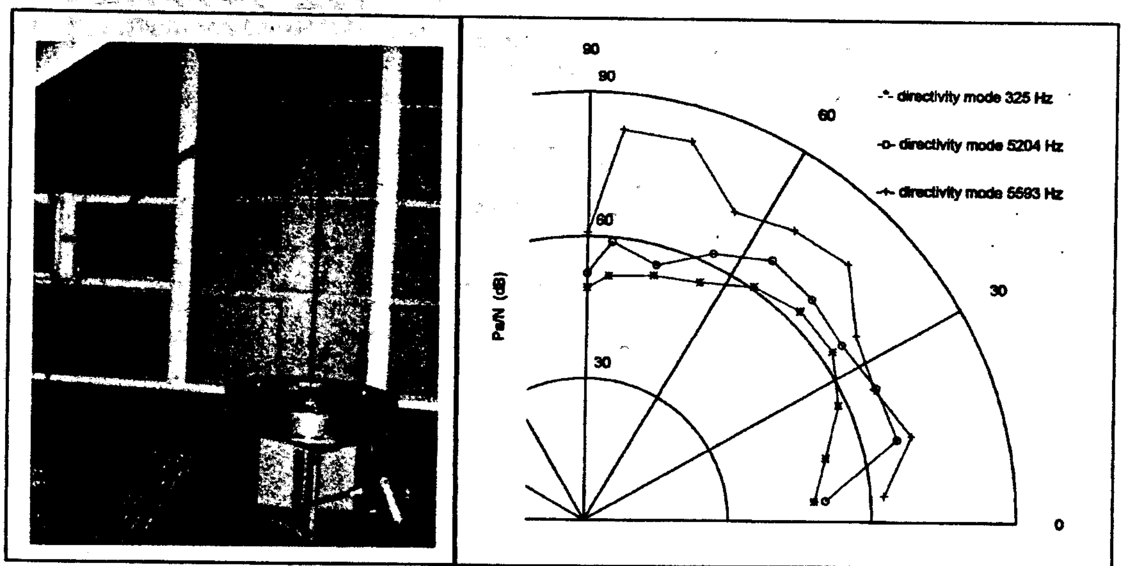

Fig. 7. *Wheel during mounting for acoustical tests with standing arc for microphone support. Shaker soundproof enclosure and lower surface lead cover are not shown for clarity (left). Directivity of the sound emission for 1st flexural mode (f=325 Hz, ζ=0.05%), 7th flexural mode (f=5204 Hz, ζ=0.02%) and radial mode with highest emission (f=5593 Hz, ζ=0.03%) (right).*

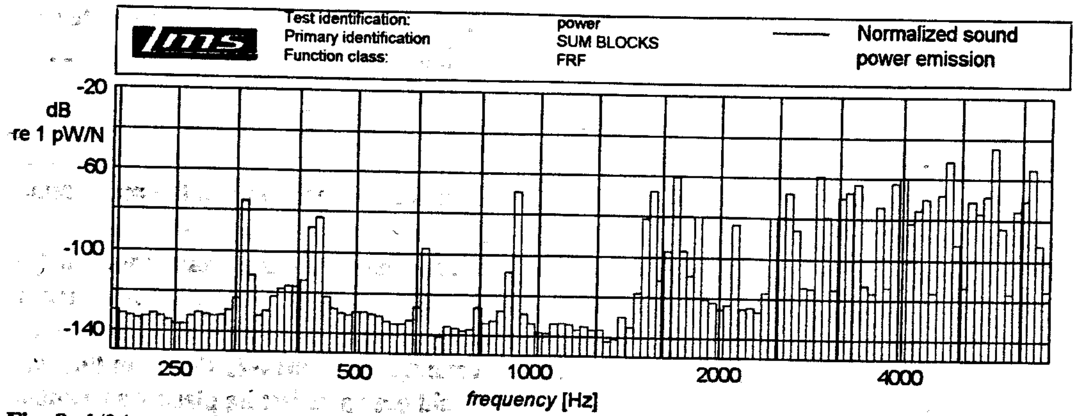

Fig. 8. *1/24 octave band normalized sound power emission.*

Analysis of some damped wheels

The procedures described in this work were applied to several 890 mm standard wheels treated with viscoelastic materials and steel rings in the well-known constrained-layer configuration (Nashif, 1985). The tested solutions are sketched in Fig. 9. The damping obtained with the different solutions is shown in Fig. 10 where distinction is made between radial and flexural modes.

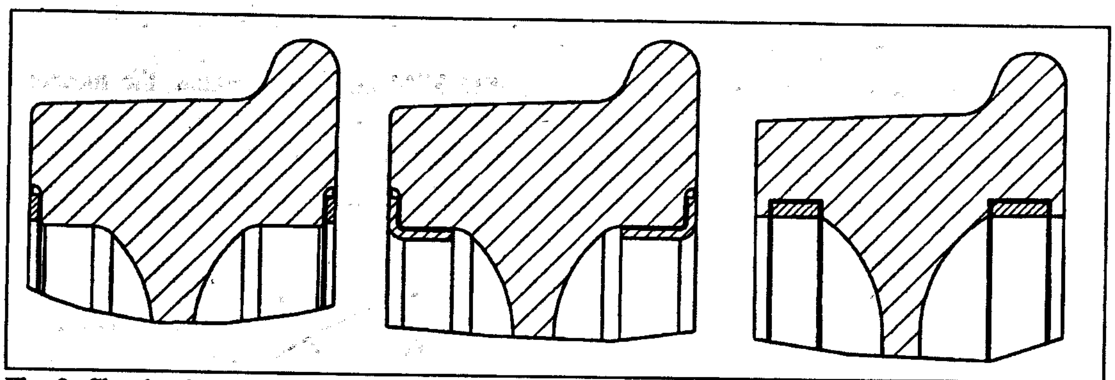

Fig. 9. *Sketch of tested solutions for constrained-layer damped wheels: lateral rings (left), L-shaped rings (center), internal rings (right).*

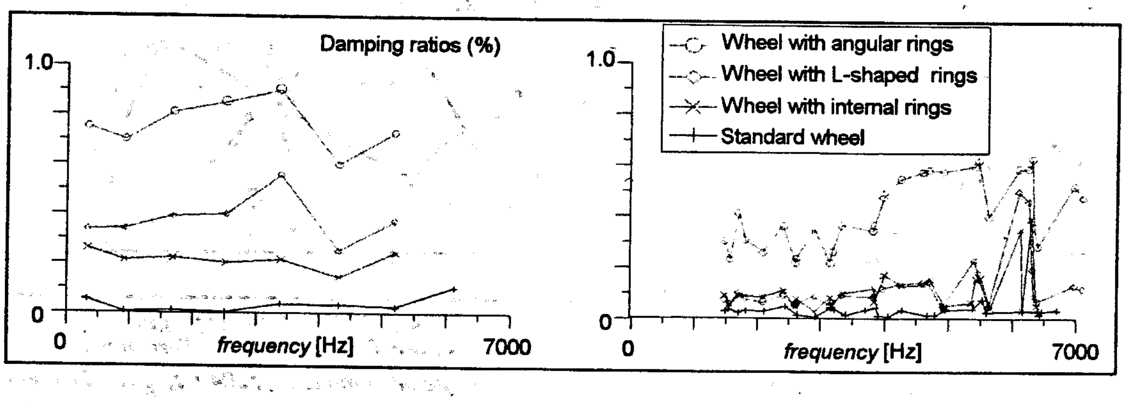

Fig. 10. *Damping ratios for flexural (left) and radial (right) modes of the constrained-layer damped wheels.*

Conclusions

In this work a methodology of vibro-acoustical analysis developed by Lucchini CRS was described. The methodology can be usefully applied to characaterize both standard and damped wheels and to evaluate at a laboratory stage the efficiency of noise reduction solutions.

This research will be continued by investigating the relationship between laboratory measurements and field measurements, made for example by using array of microphones (Barsikow, 1987) or the device to measure the noise under the axlebox developed by Bracciali (1994a).

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