STRUCTURAL DESIGN AND ACOUSTIC OPTIMIZATION OF A RAILWAY BOGIE NOISE SCREEN

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SUMMARY

Railway noise has become a problem since the introduction of restrictive noise regulations that limit daily exposure allowed for people living close to railway lines; the problem is even more important for high speed lines as the noise increases with train speed. One of the most promising devices to reduce the noise spread around by a train is a screen applied in front of the wheels. This work deals with the optimized integrated structural/acoustic design of a screen to be applied to the bogies of a passenger coach, but the design procedure can be applied to any kind of railway vehicle. The acoustical design is based on sound absorption properties of Helmholtz resonator, for which a custom model was developed, and on simulation with FEM acoustical models. FEM techniques were also used to verify structural stiffness and resistance under the loads arising from railway service.

The proposed solution was tested with a mockup; results prove the potential advantage of high absorption panels over conventional multilayered panels. Simulations highlight the convenience to design panels tuned on the sound emission of each vehicle.

Key Words: Railway noise, noise abatement, Helmholtz resonator, FEM acoustical models.

INTRODUCTION

The use of noise barriers or enclosures is a common engineering practice when the source noise reduction proves to be too expensive and when it is nonetheless necessary to protect workers or people from high noise exposures. The problem is harder to solve when the source consists of vehicles (city traffic, highways, railways) of different types running at different speeds.

Railway line characteristics are such that, except for particular situations, an efficient noise barrier is very expensive and it has a heavy visual impact. Inversely a noise screen fitted as close as possible to the wheel, that is one of the main noise sources, can be tuned on the emission of the single wheel type and will then be effective wherever the vehicle runs at whatever speed.
In this work the design procedure for a noise reduction screen with structural, acoustical and functional properties optimized for a specific coach is detailed. In the following paragraphs:
- geometrical and stiffness constraints for a noise screen to be applied to the bogies;
- noise emission properties of a specific coach;
- structural FEM design of the frame of a screen to be applied to the selected coach;
- acoustical design and optimization of the screen, with experimental and FEM techniques;
- laboratory tests on a prototype of the optimized screen

will be illustrated.

SELECTION OF THE COACH, SPECIFICATIONS AND STRUCTURAL DESIGN OF THE SCREEN

The almost unanimous opinion of scientific community is that the main source of train noise is the wheel-rail contact, at least in the 50÷250 km/h range. Mutual forces exchanged in this region induce wheels and rails vibrations that generate noise. A comprehensive review and bibliography of railway noise generation and properties lies outside the scope of this paper, and it can be found in the proceedings of the Fifth International Workshop on Tracked Transit System Noise (1996) to which the interested reader is referred.

Tests conducted with a custom on-board device (Bracciali, 1994, 1994a, 1997) allowed the authors to measure sound pressure levels and 1/12 octave-band noise composition up to 300 km/h under the axlebox of several vehicles without wind disturbances and with high reliability and repeatability. Noise measured for the FS F85 bogie of the Electrical Measurements Coach is shown in Fig. 1, where a pronounced peak at around 500 Hz with almost constant amplitude at any speed is observed while other frequency components grow up with the speed. At low speeds F85 noise is hence concentrated at lower frequencies, which are clearly harder to be attenuated.

A noise screen to be applied to the bogie frame was developed (Fig. 2), as the application of a screen directly fixed to the axleboxes would require too high strength and rigidity as axlebox accelerations are extremely high. The following constraints were considered for the screen design:
- flexibility of suspensions and normal kinematics of the bogie (translations and rotations) must be ensured without any damage to the screen and/or to the bogie;
- the screen must not violate geometrical limits for vehicles even in the case of primary suspension failures;
- structural resistance and stiffness must be guaranteed for any load combination (inclu-

ding aerodynamic loads), with the lowest possible static and dynamic stresses and strains;
- materials used for screen manufacturing must be cheap, fireproof, washable, non toxic, recyclable and they must neither absorb nor release particles;
- for retrofit applications the screen assembling should be easy with minimum modification of the bogie frame.

For the selected F85 bogie a survey indicated 30 mm as the maximum thickness of the screen that respects geometrical constraints.

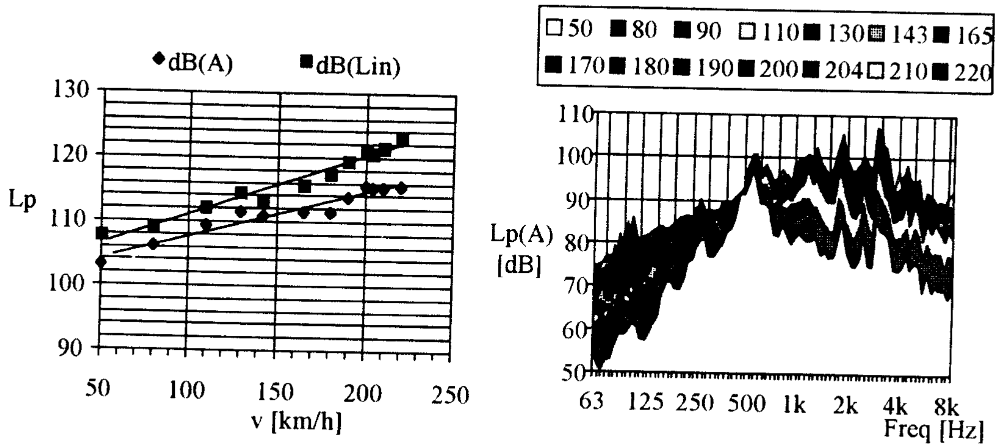
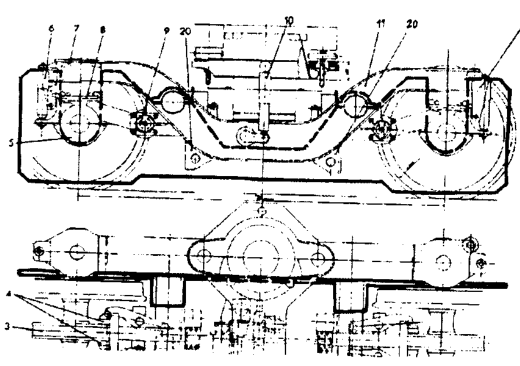


Fig. 1. Sound pressure levels Lp(A) measured under the axlebox of a F85 FS bogie (left) and 1/12 octave band frequency spectra for indicated speeds (right).

The very reduced thickness and the limited number of available fixing points on the bogie frame clearly constrast with the required screen stiffness. The final structural design consists of a 1 mm silent steel sheet stiffened with a rectangular 70*75*1 mm hollow profile with several ribs. The total mass is about 33 kg, and the first eigenmode (flexural) is at 37 Hz.

To compute deformations in operating conditions, not exactly defined, a unit acceleration in horizontal and vertical directions has been imposed in the 0÷50 Hz frequency range. Resulting maximum displacements (Fig. 2) are in the order of 30 mm and Von Mises equivalent stresses reach a maximum of about 1.2 MPa. Even with maximum bogie frame acceleration peaks (the bogie ride is considered unstable when the bogie frame acceleration exceeds 8 m/s2 for 6 consecutive cycles), the rigidity and the strength of the screen seem to be ensured anyway. No aerodynamic loads were considered in the design as they strongly depend on the actual air flow around the bogie; nonetheless the very high strength and rigidity of the screen seem to be largely sufficient to prevent aerodynamic problems.



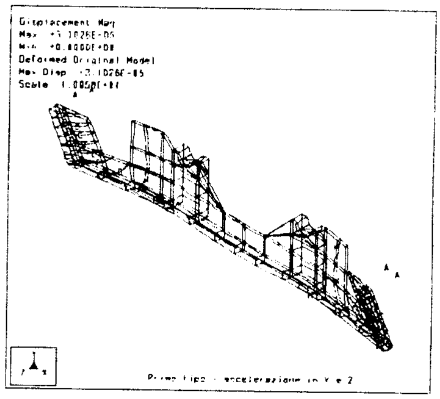


Fig. 2. Sketch of the noise screen applied to a F85 bogie (left). The screen is mounted beetwen the wheels and the bogie frame and is fixed with two collars on the connecting bars of the frame. Displacements under a 1 m/s2 vertical and horizontal acceleration in the range 0÷50 Hz (right).

ACOUSTICAL DESIGN AND OPTIMIZATION OF THE SCREEN

The noise reduction efficiency of several multilayered screens was simulated through FEM acoustical models using the ANSYS, code; the screen stratification was optimized through original design and verification techniques.

FEM simulation of the screen noise reduction efficiency.

To estimate the efficiency of the chosen screen type, a FEM model with noise point sources simulating wheel and rail emission, structural shell elements representing the screen and fluid elements was developed. The modeled area (Fig. 3) is 8 m wide by 3 m high and is delimited below by the ground, on the left side by a line simulating the wheel, above and on the right side by open air. As the element size must be smaller than 1/10 of the shortest wavelength considered, to keep the number of elements within reasonable limits a 2D model was used with an upper frequency of 1 kHz and nevertheless the number of elements was about 27000 (refining the mesh around the screen). It is worth noting that a 2D model considers the train as a cylindrical source and thus underestimates the efficiency of the screen.
The distribution of pressure level at the source was assumed linearly variable from a maximum close to the rail (to take into account the noise emitted by the rail) to a minimum at the top of the wheel (115 dB and 103 dB respectively at 250 km/h, for example). The type of the structure of the screen as a sound barrier has a limited importance, provided it has a reasonable transmission loss. More important is the height of the screen over rail level; two different heights, i.e. 180 mm, respecting the normal geometrical limits for railway vehicles, and 90 mm, still acceptable for the experimental coach, were simulated. In addition, the effect of the absorption coefficient of the wheel surface and, more important, of the surface of the screen facing the wheel were explored.

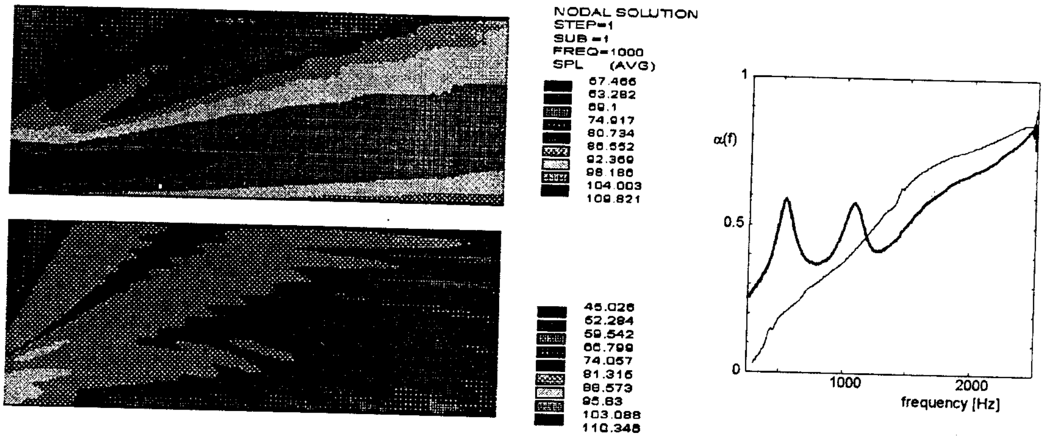


Fig. 3. (Left) Free-field noise map without screen at 1 kHz (top). Noise map after the application of an absorbing screen at 90 mm above rail level (bottom). Absorption coefficients: α_{wheel}=0.1; α_{screen}=0.9. (Right) Measured absorption coefficient of a specimen of a stratification with Helmholtz resonators (thick line) and of a "standard" multilayered solution with the same thicknesses (thin).

On the upper and right sides of the FEM model a unity absorption coefficient (α=1) was used, simulating the free field. The ground had α=0.1 and the left side, simulating the wheel, had α=0.1 or α=1 to simulate the opposite cases of reflecting or absorbing wheel surface. The side of the screen that faces the wheel was simulated with α=0.1 or α=0.9. The screen efficiency is expressed by the average sound pressure level on the vertical line at 7.5 m from the track axis, i.e. at one of the standard distances for train noise survey tests. The results of some simulations at 1 kHz are shown in Table 1 and in Fig. 3. These values, clearly only indicative, show the following trends:

- the application of a screen seems to be very effective for noise reduction;
- it is very important to reduce to a minimum the distance between the rail level and the bottom of the screen. This may violate standard geometrical limits, but it should be properly taken into account the fact that the screen is very close to the wheel, preventing possible hits of external objects;
- in this first approximation simulations the rail level above ground level has not been considered, probably leading to an overestimation of the screen efficiency;
- it is very important that the internal side of the screen is highly absorbing, otherwise noise reflections on the wheel can increase sound pressure level at the measuring site.

Screen height from rail level	180 mm	180 mm	90 mm	90 mm
α screen	0.9	0.1	0.9	0.1
Noise reduction at 7.5 m	-10.8	-9.7	-16.8	-15.2
Screen height from rail level	180 mm	180 mm	90 mm	90 mm
α screen	0.9	0.1	0.9	0.1
Noise reduction at 7.5 m	-14.1	-11.6	-20.8	-17.3

Tab. 1. FEM estimated 1 kHz noise level reduction for absorbing (α=0.9, top) and reflecting (α=0.1, bottom) wheel. Average level at 6.8 m from the wheel without screen are respectively 93.6 and 97.7 dB.

Definition of the screen tipology and acoustical simulation

As remarkable gain on noise abatement can be obtained through the increase of the absorption coefficient of the internal side of the screen, a non-standard solution was investigated as the screen overall thickness, including the structural frame, must be less than 30 mm and a "classical" solution (multilayered absorbing materials) is inefficient at the lower frequencies.
The only possibility to obtain a high a, at least for some frequencies, with this thickness is the use of Helmholtz resonators (Beranek, 1992). The Helmholtz resonator frequency can be easily modified acting on the geometrical parameters (neck diameter, volume) of the resonators. Two problems were preliminarly solved in the design process:

- absorbing material manufacturers normally do not provide some physical properties like flow resistivity and structure factor. These values were determined with the techniques described in Braccesi (1997) for many commercial materials potentially applicable to the screen;
- reliable models to estimate the damping given by absorbing material inserted into the cavity are not available in literature, and moreover available models of Helmholtz resonators, typical of the analysis of exhaust gas systems for road vehicles, proved to be unefficient to describe the behaviour of multilayered panels. A custom simplified model that estimates a for a Helmholtz resonator from experimental tests was developed. This model uses both the geometrical configuration and the acoustical properties of absorbing materials inserted into the cavity.

Optimal results were obtained by using innovative acoustical materials and a specifically studied geometry. The final composition of the screen, starting from the internal side, consists of a sheet of porous aluminium, a layer of absorbing material, a steel sheet on which Helmholtz resonator's necks were bored (with diameters corresponding to 500 Hz and 1.25 kHz resonance frequency), an air layer (i.e. the cavity of the Helmholtz resonator) and a "silent steel" constrained layer damped sheet to reduce external noise radiation due to structural vibrations.
Compared to a classical multilayered panel, the prototype screen has a better behaviour up to around 1.25 kHz, i.e. the range of action of the resonators, while, at higher frequencies, its performances are still sufficient (Fig. 3, right).

ACOUSTICAL LABORATORY TESTS ON A PROTOTYPE OF THE SCREEN

Some acoustical tests were performed on a square screen in a free-field environment using a large loudspeaker driven by a white noise signal (Fig. 4, left). The screen was elastically suspended very close to the loudspeaker, and the sound levels were measured at 6.8 m (corresponding to the standard 7.5 m from the track axis distance) and 13.6 m (for free field assumption verification). The height of the panel above rail level was rescaled with respect to FEM simulations by using geometrical similitude; as the loudspeaker diameter is about 1/3 of the wheel diameter, heights have been proportionally reduced to 30 and 60 mm.

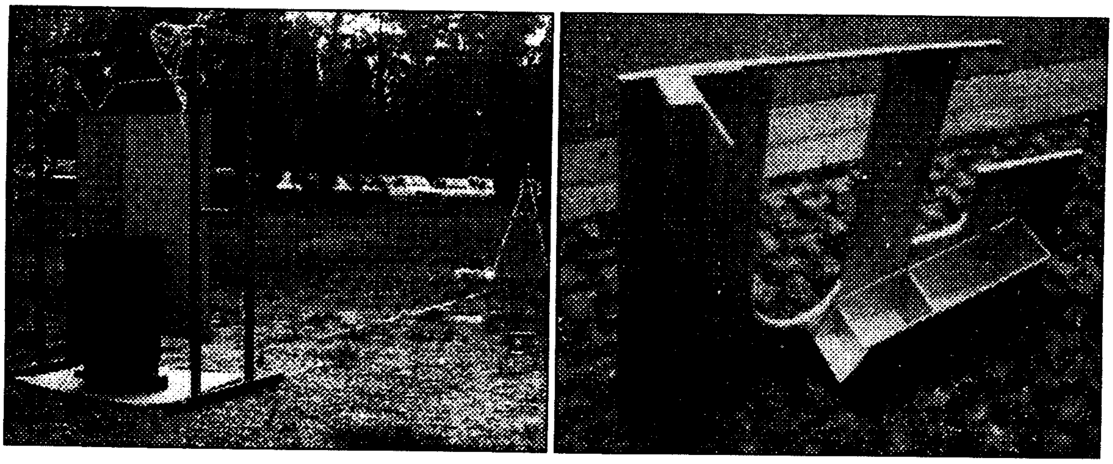


Fig. 4. Measuring location view (left). Absorbing panel prototype for a single wheel of the F85 bogie (right).

The proposed solution was compared with a standard screen by computing the difference in the insertion loss (IL) between the two solutions (Fig. 5, left). The greatest differences in IL are present only at Helmholtz resonators natural frequencies that were tuned to the highest emission ranges of the F85 wheel. As the frequency composition of noise changes with speed, the global emission level is reduced by the IL difference accordingly.
Noise emitted by the loudspeaker was rescaled to on-board measured noise taking into account the distance between the source and the measuring location (6.8 m), resulting in different efficiencies for screens with or without Helmholtz resonators at different heights from the rail level. The Lp(A) results at different velocities are shown in Fig. 5 (right), where the superiority of the absorbing screen in evident; in particular, the screen with Helmholtz resonators at 60 mm from the rail level has a better efficiency than a traditional panel at 30 mm. The average reduction obtained is about 3.5 dB(A) at all velocities for the 30 mm solution; if this were considered inapplicable, the 60 mm solution, that respects the geometrical limits imposed by international railway norms, is only 0.5 dB(A) worse, and therefore is still very advantageous.

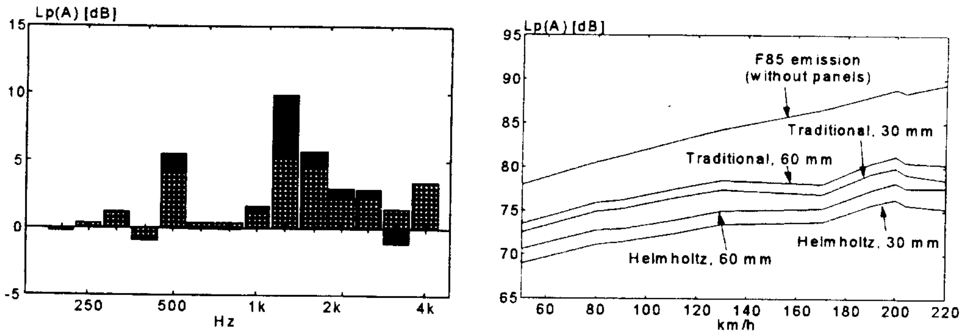


Fig. 5. (Left) Measured Insertion Loss difference between traditional and absorbing panels at 60 mm (grey) and 30 mm (black) above rail level. (Right) SPL(A) estimated at 7.5 m from the trach axis for the F85 bogie at different heights above rail level.

CONCLUSIONS

The fact that bogie screens can be extremely effective for railway noise reduction is widely known. In this research the procedure for the correct design of a screen with optimized sound absorption capabilities was developed. Performances exhibited by this screen during laboratory tests seem very promising, such that their use could be very effective, reaching a noise abatement of about 10 dB(A) at 7.5 m from the track axis vs. the less than 7 dB(A) reduction obtained with traditional panels. It is important to underline that the advantage is still more important at the lower velocities (typical of city crossings) where emission is predominant at lower frequencies that are harder to abate with light panels.

The integrated numerical and experimental procedure the for design and the optimization developed in this research led to a light, stiff and sound reduction effective bogie screen. The aim of the authors is to continue this work testing full scale screens (a prototype of which is shown in Fig. 4, right) during normal railway operation, to investigate those effects that have not been considered at a laboratory stage (aerodynamical effects, noise from other wheels, etc.) and to develop even more advanced types of panels.

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