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ACOUSTIC RESONATOR FOR MITIGATION ACTIONS ON UNDERGROUND RAILWAY LINES

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Abstract

Noise pollution caused by underground railway lines is particularly strong in open sections, where levels of noise equal to or superior to noise produced by traffic reach the surrounding receivers. In order to minimise the acoustic impact, it is not always possible, for aesthetic or structural reasons, to build ad-hoc noise barriers. It is therefore necessary to consider less invasive actions that can reduce noise close to its source, taking into account the fact that noise produced by train transit is mainly due to the wheel-track contact. In this paper an acoustic resonator is presented, designed to reduce noise produced by the wheel-track contact in the Rome underground railway lines. Considering that the train emission spectrum, measured close to the permanent way, has a peak around 350 Hz, the acoustic resonator was tuned on this frequency interval and designed using the space between the safety gangway and the support plane of the railway line. The value of the resonator's absorption coefficient was verified experimentally on a prototype built for the purpose.

1. ACOUSTIC CHARACTERIZATION OF UNDERGROUND TRAINS

Trains are acoustic sources that can be characterised by an average emission spectrum. Considering usual underground train speeds (50 – 80 km/h), noise is almost entirely due to the wheel-track contact, being generated by the rolling of one of the two contact surfaces. The following parameters influence the wheel-track contact and the associated noise:

- The condition of the tracks, for example presence of rail corrugation;
- Presence of joints on the tracks;
- The state of the rolling strip of the wheel rims.

Usually rail corrugation is the predominant source of noise. The emitted acoustic power is centred around a band of frequencies that depends on the corrugation wavelength and on the train speed:

$$f = \frac{100u}{3{,}6d} \qquad \text{[Hz]} \qquad (1)$$

Where: f is the frequency, u is the train speed in km/h and d is the distance between two corrugation peaks in cm.

In the case under study the measured corrugation wavelength was between 4 and 6 cm; considering a train speed of 50 km/h, the main emission spectral components were estimated to be between 230 and 350 Hz.

This estimation was verified with measurements close to the tracks.

The average spectrum was found for each train transit (always for a 50 km/h speed). The spectrum maintains its form with a good approximation and has a peak around 315 Hz (fig.1), thus confirming the previous estimation based on the corrugation properties.

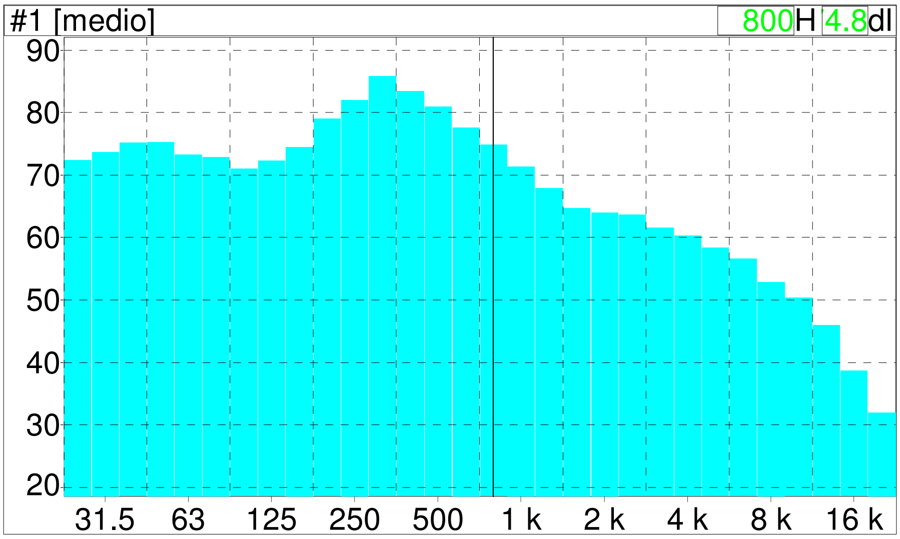

Figure 1 – Emission spectrum produced by a train in transit.

2. REDUCTION OF THE NOISE PRODUCED BY THE WHEEL-TRACK CONTACT

In order to reduce the noise produced by the wheel-track contact, a resonant absorbent acoustic system has been designed using the space existing underneath the safety gangway. Since the sound absorbing system is placed close to the noise source, it is possible to intercept and attenuate the direct noise produced by the wheel-track contact (fig.2).

The system is made up of resonant and bi-absorbent acoustic septa (fig.3), absorbent on two sides, placed vertically underneath the gangway, between the track floor and the technical airshaft that is used as a resonant cavity. The inner surfaces of the cavity are treated with sound absorbing plaster. The incident noise is reduced in part by the exterior vertical panel and is further reduced by the resonant cavity.

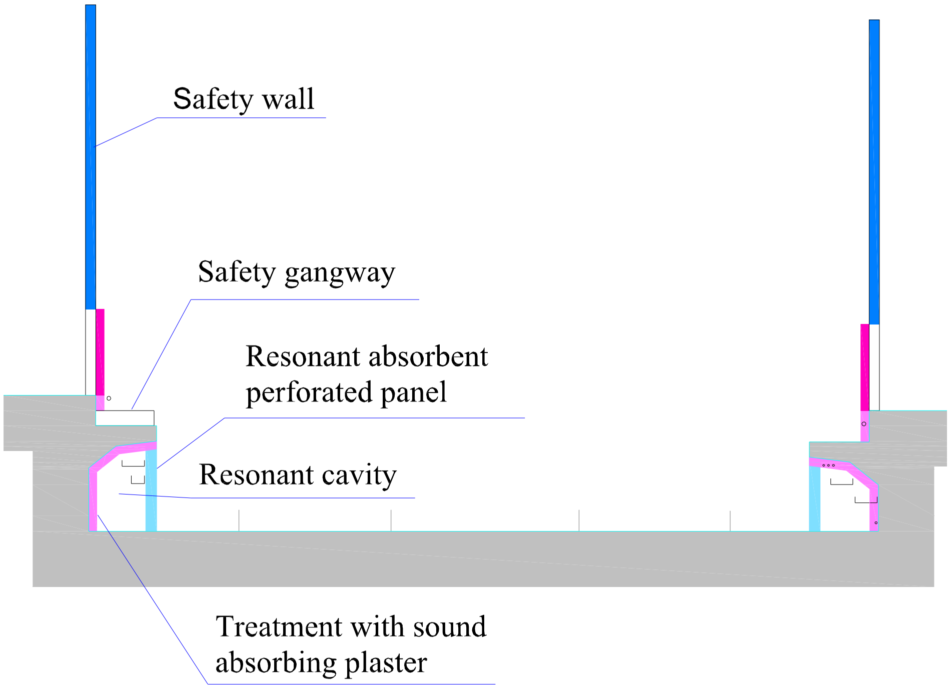

Figure 2 – Resonant cavity.

The resonant absorbent perforated panel behaves as a set of Helmoltz resonators, each with a neck, corresponding to the hole in the panel and a cavity, made up of part of the volume of the airshaft between the panel and the wall. Figure 3 shows a draft of the system.

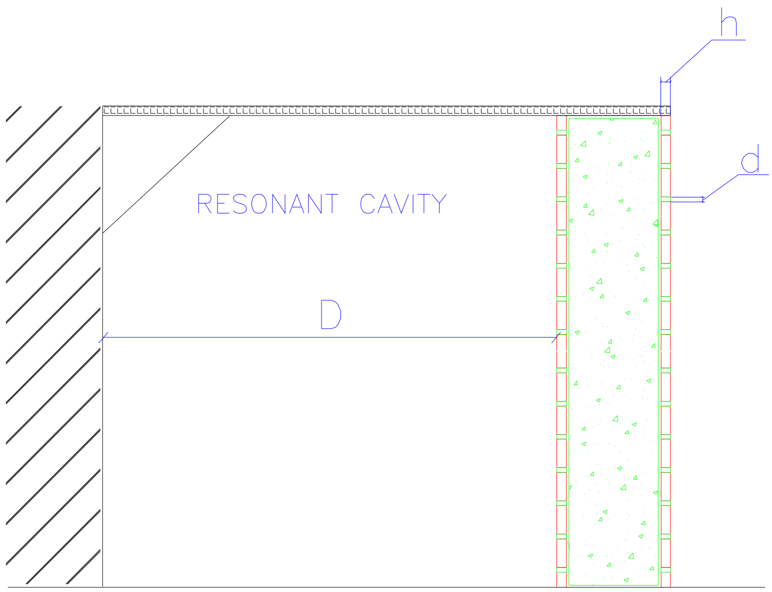

Figure 3 – Draft of the resonant cavity.

The design of the panel was made by choosing the correct thickness of the panel, the size of the holes and the percentage of perforation, in order to maximize the absorption around the frequency which corresponds to the maximum noise emission.

The maximum resonant frequency of the cavity can be estimated with the following relationship [2]:

$$f_r = \frac{c}{2\pi}\sqrt{\frac{\sum S_f / S}{D(h+0{,}8d)}} \quad \text{[Hz]} \qquad (2)$$

where c is the speed of sound in air, S_f is the total area of the holes, S is the panel surface, D is the distance between the panel and the wall, h is the thickness of the panel and d is the diameter of the holes.

In this specific case, (figs. 3 and 4) the perforated panel has the following characteristics:

- Panel-wall distance (corresponding to the depth of the airshaft): D = 50 or 60 cm;
- Perforation percentage = 30%;
- Panel thickness: h = 10 mm;
- Hole diameter: d = 5 mm.

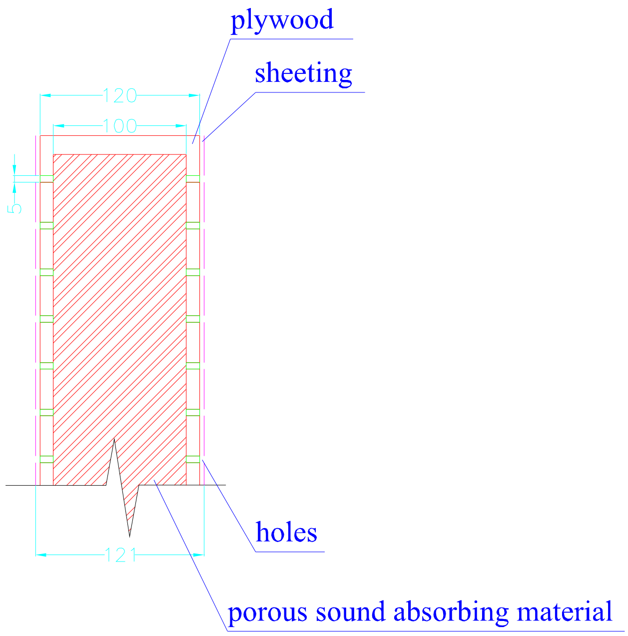

Figure 4 – Cross-section of the panel.

3. EXPERIMENTAL DETERMINATION OF THE ABSORPTION COEFFICIENT

From the draft shown in fig.3, a trial section that replicates the real situation was built in the lab; the absorption coefficient of the acoustic resonator was then determined from this prototype.

The intensimetric method was used by applying an impulsive, directional, pink noise source. The incident angle was set equal to the real one, between the plane of the track closest to the resonator and the centre of gravity of the vertical panel (fig.5).

The incident and reflected impulsive sound was measured with a bi-channel acoustic intensity gauge. This operation was done for each third of octave band, and for various reflection angles. The resulting reflection diagram was substantially symmetrical so the reflection acoustic intensity was measured, at various angles, on a plane orthogonal to the panel, passing through its centre of gravity. The measurement points were taken on a semicircle lying on this plane.

For each frequency band, the absorption coefficient was determined with:

$$a_j = \frac{\sum_{i=1}^{n}\left[\left(1 - r_{\alpha i}\right)\right]}{n} \tag{3}$$

where:

- aj is the absorption coefficient relative to the Jth band;
- n is the number of measured reflection angles;
- $r_{\alpha i}$ is the reflection coefficient relative to angle α_{ri}.

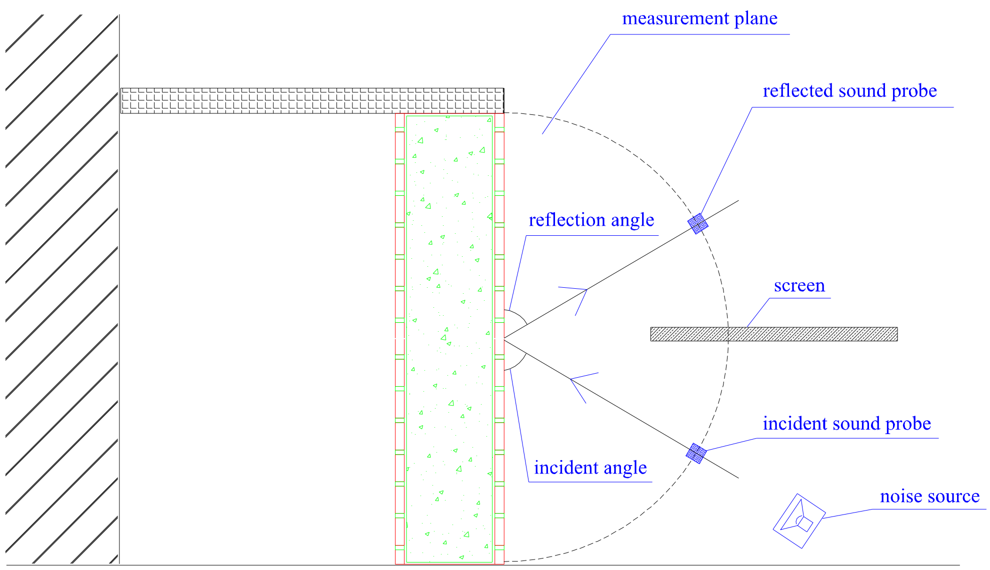

Figure 5 – Measurement setup.

The value of the reflection coefficient $r_{\alpha i}$ was determined with the following expression:

$$r_{\alpha i} = \frac{J_{\alpha i}}{J_I} \tag{4}$$

where:

- $J_{\alpha i}$ is the reflection acoustic intensity measured along the direction determined by angle α_{ri};
- J_I is the incident acoustic intensity.

The experimental measurements carried out on the prototype are shown in fig.6. The maximum absorption occurs around 315 Hz, that is around the maximum noise emission frequency for underground trains.

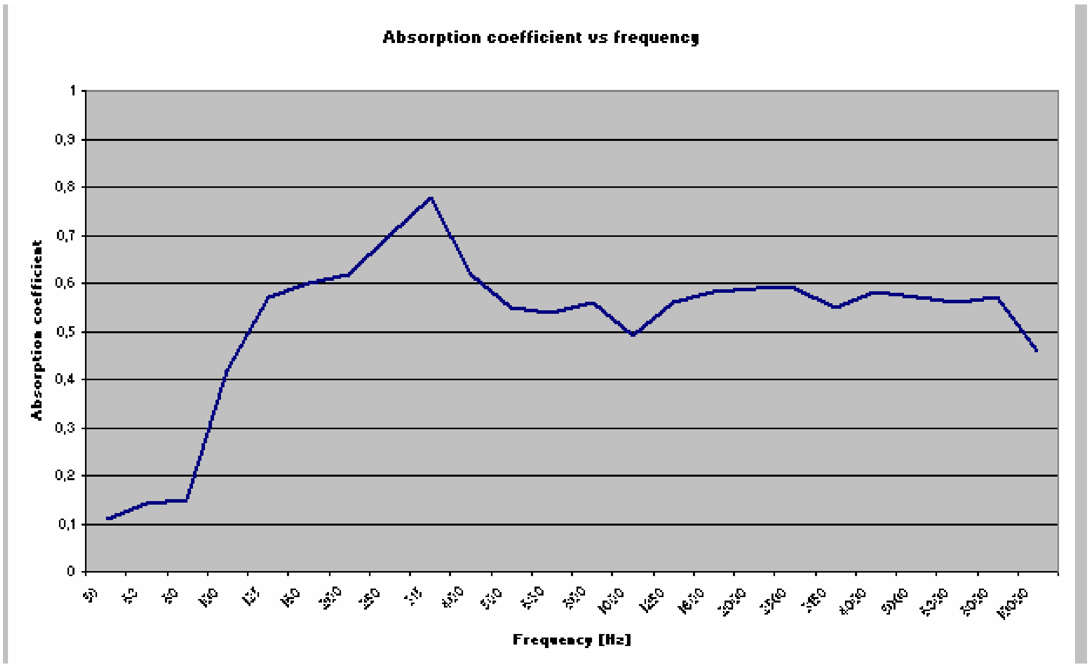

Figure 6 – Acoustic absorption spectrum of the perforated resonant absorbent panel.

4. CONCLUSIONS

This paper has illustrated the design of a system that allows to attenuate noise close to the wheel-track contact, for open sections of underground railway lines.

The absorption spectrum of the resonant absorbent system was measured using a lab prototype built for the purpose with the intensimetric method, using an innovative technique.

The proposed system allows to take mitigation actions whenever it is not possible to build ad-hoc noise barriers or total enclosures for aesthetic or structural reasons.

5. REFERENCES

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