Mitigation of vibrations due to vibroacoustic coupling in an environmental testing laboratory

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ABSTRACT: Implementation of large vibration exciters for the environmental testing of electronic equipments in test rooms may induce high acoustic pressures and heavy structural vibrations, which limits the accuracy of optical measurements carried out in the meantime.

This paper illustrates this with the example of a large test facility in which operators experienced high acoustic pressures and vibrations. The paper quantifies the physical parameters measured, and details the solutions found to solve the problem.

1. POSITION OF THE PROBLEM

1.1. Description of the existing situation

A large vibration exciter (90 000 Newtons rms) is installed in a large test room (14.4 x 9.0 x 5.5 m).

The exciter is fitted on a 40 tons concrete slab floating on soft springs with a vertical resonance frequency of 15 Hz (less was not wished since displacement of slab must be limited for optical tests).

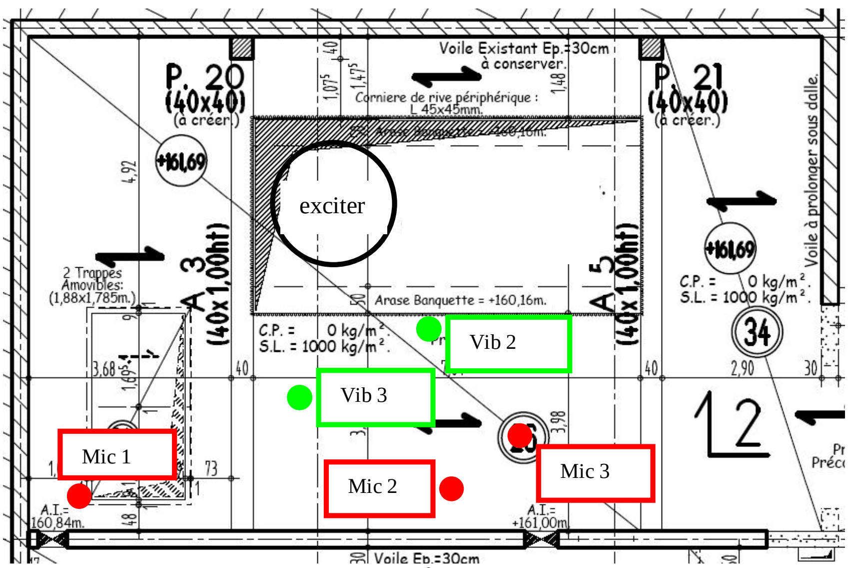


Figure 1. Layout of exciter test room

When sweeping from 5 Hz to 200 Hz with a constant exciter acceleration of 20g - and despite the existence of a conventional acoustically absorbing material in the room - very large acoustic pressures are observed at low frequencies in the test room, typically up to :

- 31 Pascals (124 dB SPL) at 19 Hz and
- 40 Pascals (126 dB SPL) at 23 Hz.

Since operators are supposed to carry out optical measurements during sweeping, these pressures – although not acoustically audible – are unbearable because they are felt as important body stresses.

Furthermore important vibrations of floor of the test room are encountered, which limit or prevent the capacity of carrying out such optical measurements. Also, floor vibrations of upstairs laboratories in the same buildings disturb control testing.

1.1. Experimental characterization of existing situation

Acoustic pressure and floor vibration were measured during exciter sweep (constant acceleration of table : 20 g).

Acoustic resonance effects are clearly visible on picture below, their amplitude naturally depends of microphone position in room, but the resonance frequencies can be accurately located at 11.9Hz, 18.8Hz, 22.2Hz, 23.0Hz, 35Hz, and 38 Hz for the most energetic acoustic modes.

Vertical displacement of floor exceeds 30 microns rms, which is excessive for carrying out optical measurements.

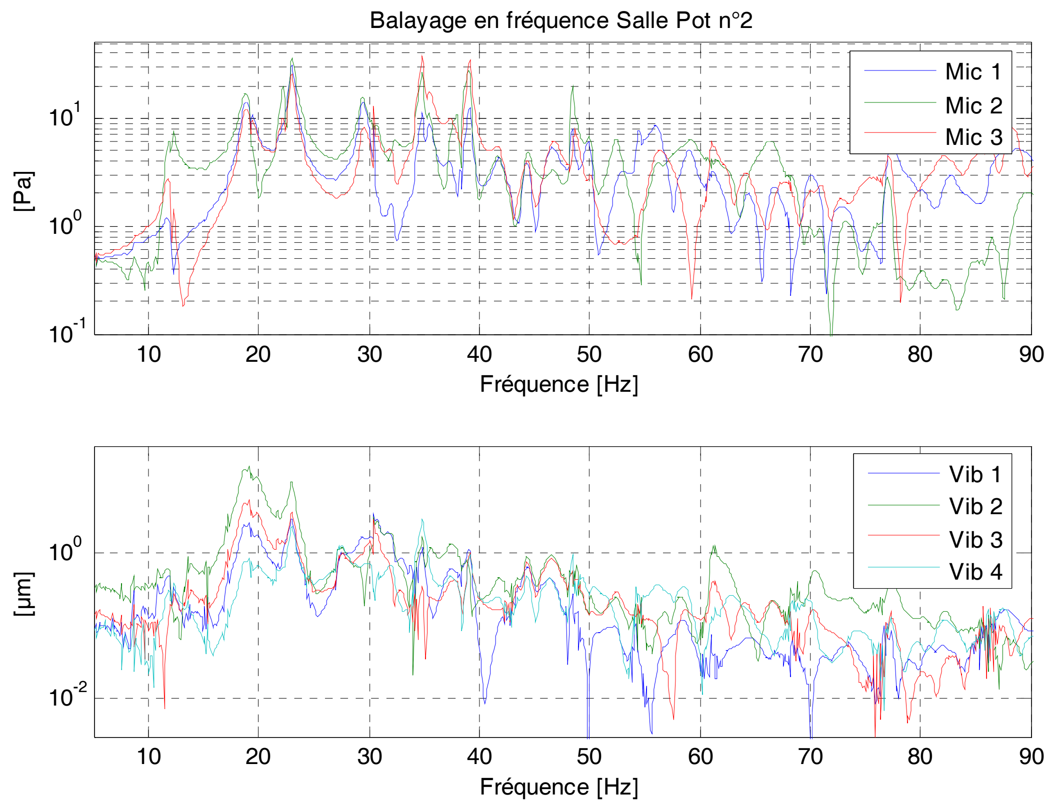


Figure 2. Acoustic pressure in room, and vertical displacement of floor during exciter sweep.

It may be quickly checked that the resonance frequencies correspond to the case of a parallelipipedic room of dimensions listed above. Acoustic mode shapes are showed on next page.

The case of mode 2 is particularly interesting, since it exhibits maximum acoustic pressure along the largest dimension of room. This applies a high pressure to the floor area located in front of exciter, where optical measurements are supposed to be carried out, close to sensors vib 2 and vib 3 in Figure1.

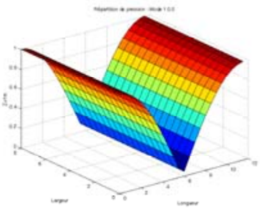
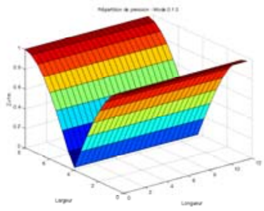
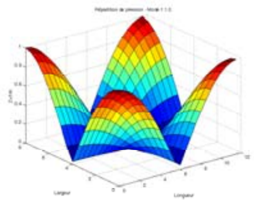
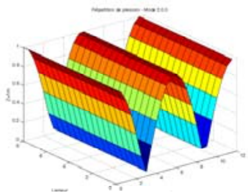
Mode	**Fréquence [Hz]**	**Rang n,m,p**	**Répartition de pression acoustique**
1	**11.9**	**1.0.0**	
2	**18.8**	**0.1.0**	
3	**22.2**	**1.1.0**	
4	**23.0**	**2.0.0**	

Figure 3. Acoustic mode shapes in test room (calculated).

This high acoustic pressure in turn induces structural vibrations in this floor area. It is thus interesting to measure the mechanical inertance of this floor area. Inertance is measured at point Vib 2 (see figures 1 and 2), and displayed in figure 4.

The inertance curve exhibits two close modes around 20 Hz, which magnifies the effect of the acoustic resonance mode at 18.8 Hz. A closer analysis of this floor area gives insight into the mode shape of these modes. The first mode at 18.3 Hz exhibits the mode shape of figure 5, with a damping of 1.2% of critical. This mode shape also concerns the floor of the command room which is adjacent.

A second mode shape is identified at 20.1 Hz, with almost the same mode shape but which concerns the test room floor mainly. Damping is 1.3 % of critical.

It is thus clear that when the exciter sweeps from 18 Hz to 20 Hz, a strong acoustic room resonance occurs at 18.8 Hz which exerts high acoustic pressure on floor area Vib2 – due to spatial coïncidence -

which itself exhibits low damped structural modes at 18.3 Hz and 20.1 Hz. The situation could even be worse is frequency coïncidence were better.

Since optical measuring is compromised, and since operators discomfort is excessive, it was decided to solve this problem using simultaneously acoustic and structural solutions.

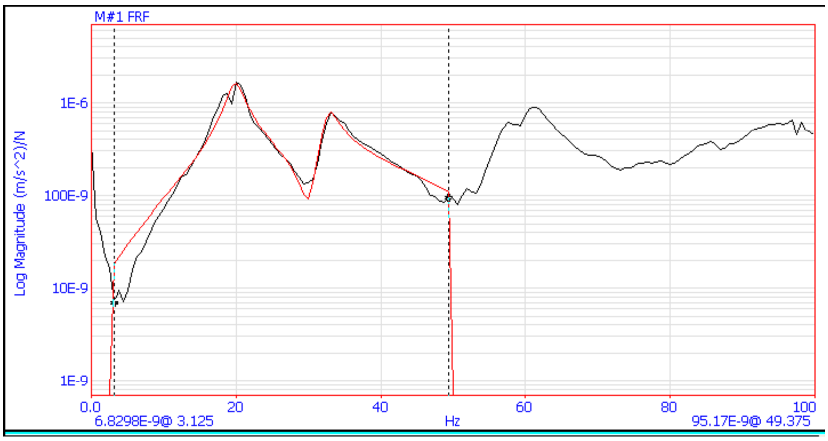


Figure 4. Vertical inertance of floor in area Vib2

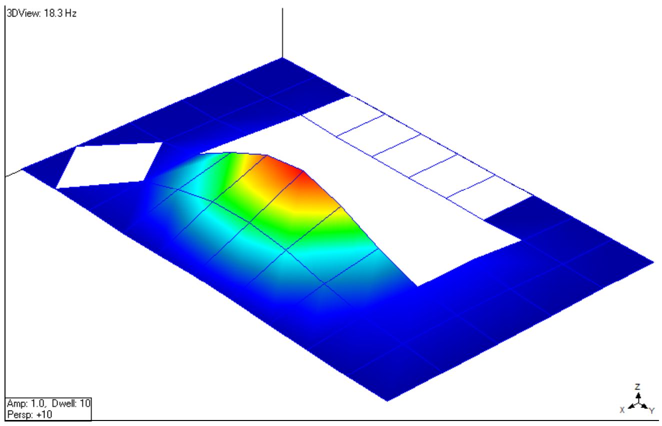


Figure 5. Mode shape of floor in area Vib2, frequency 18.3 Hz damping 1.2 %

2. CONCEPTION OF SOLUTIONS

1.1. Acoustical solutions

Stationary waves occur because the internal surface of room walls is acoustically very stiff. To limit stationary waves it was decided to install Helmholtz resonators tuned to modal frequencies, and to lay them on proper surfaces, depending on the type of mode to be damped.

Three modes were damped : mode 010 (18.8 Hz), mode 110 (22.2 Hz), mode 200 (23.0 Hz). The overall area of Helmhotz resonators is about 100 m2. The thickness of resonators is 0.4 m, which is acceptable compared to the large dimensions of room (14 m x 9 m). The purpose of this modification is to reduce the peak acoustic pressure at the above frequencies, in the areas concerned by these modes. Notice that conversely pressure will be increased at the points where it is presently at a minimum.

1.2. Structural solutions

The acceleration level of the floor is high due to spatial and spectral coincidence between acoustic and structural modes. It is thus necessary to change the structural resonance frequencies to avoid this coincidence. It is not appropriate to lower the frequency, since this would induce higher response from walking, among other drawbacks. Resonance frequency of first modes of floor should much preferably be increased by stiffening the floor, which in turn would output less response for a similar dynamic loading. Several stiffening solutions were simulated using finite element modeling.

Firstly an fem model of the existing structure was elaborated, and compared to experimental test results (see below).

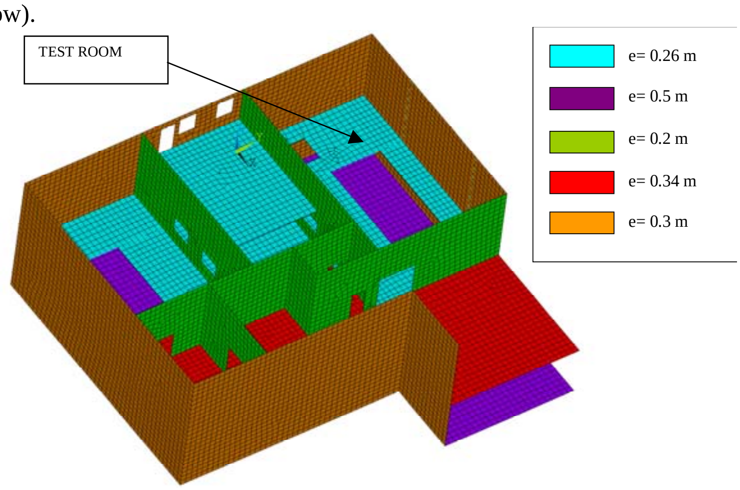


Figure 6. Finite element model (Ansys) of overall structure of laboratory.

Except for damping values, this model did not need updating since modal frequencies were close enough to those found by testing (less than 5% error up to 4th mode of floor).

A typical stiffening solution is displayed below.

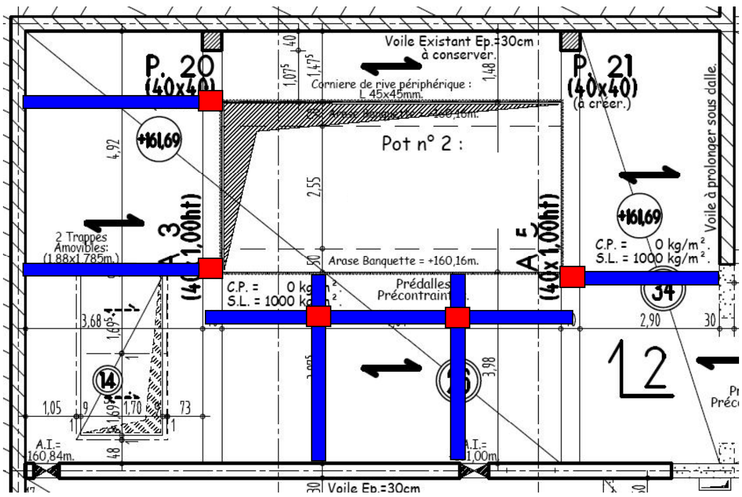


Figure 7. Typical stiffening solution for laboratory floor.

The stiffening solution is parameter based, with variable distance between central beams and columns, variable number of columns under test floor (1 or 2), use of steel or concrete beams of variable section.

The following solution with HEA400 steel beams was found to be a good compromise between efficiency and cost. It was supposed to raise first floor mode at 47 Hz - in the case of perfect contact (merged nodes) between existing concrete slab and steel structure. It was decided to realize this modification to the laboratory floor, displayed below. The largest beams are the existing concrete beams. The other beams are the reinforcement steel beams.

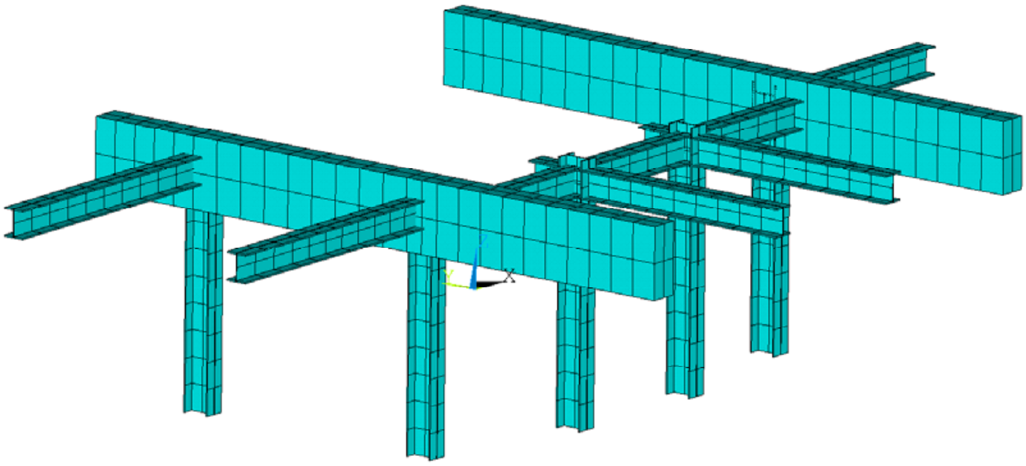


Figure 8. Accepted stiffening solution for laboratory floor.

3. STRUCTURAL STIFFENING IMPLEMENTATION

The accepted stiffening solution was carried out on site. The result being dependent on a good linking between steel and concrete, high performance epoxy adhesive was used to improve the mechanical connection.

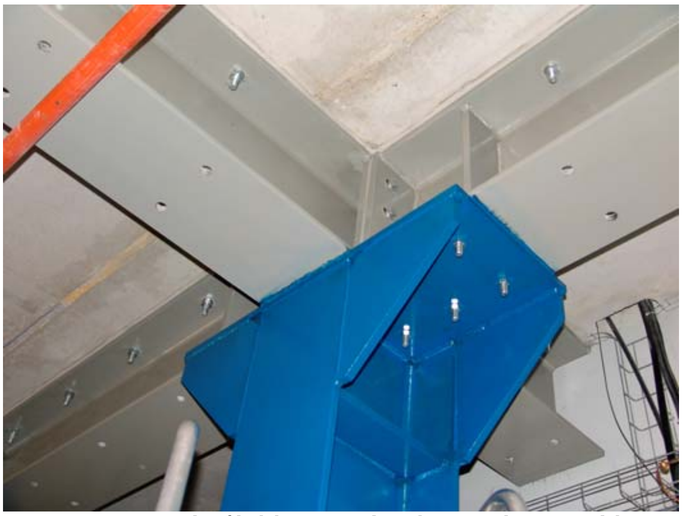


Figure 9. Example of link between colums beams and concrete slab.

4. DYNAMIC RESULTS

4.1. Acoustic results

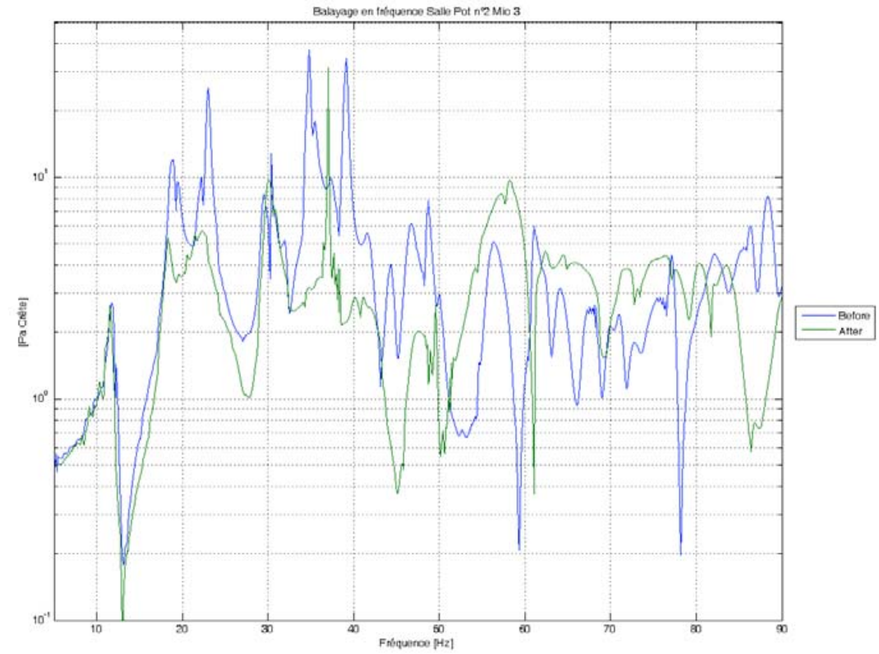


Figure 10. Acoustic pressure at mic 3 during sweep of exciter, before / after implementing Helmholtz resonators

As expected, the acoustic pressure in the testing area (microphone 3) is considerably reduced at the resonance frequencies concerned by the Helmholtz resonators. Some higher rank modes concerned by the spatial location of the Helmholtz resonators are also affected. The peak at 37 Hz is not a resonance, but a ventilating unit of the laboratory. A reduction of 6 to 12 dB is observed on the two first modes. The maximum rms acoustic pressure observed on all three microphones is now 10 Pascals instead of 37 Pascals.

4.1. Vibration results

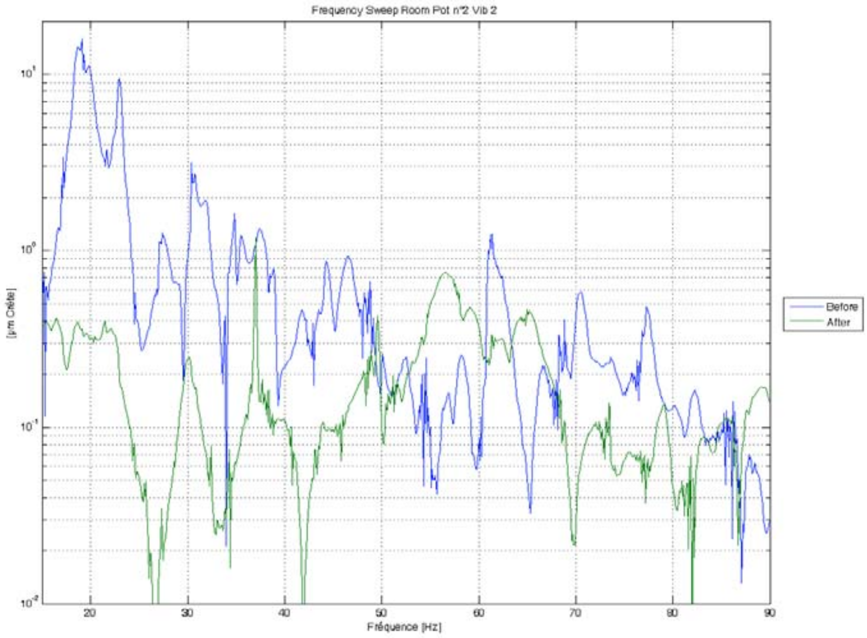


Figure 11. Vibration displacement at vib 2 during sweep of exciter, before / after implementing structural stiffening

The vertical dynamic displacement on the floor area reserved to optical testing (sensor vib2) is considerably reduced, due to the mitigation of acoustic pressure at the resonance frequencies, and due to the stiffening of floor structure. Now, the maximum displacement encountered during sweep is 0.4 micron rms in the frequency range of interest, instead of 15 microns

A vertical inertance measurement of floor showed that the first mode at 18.3 Hz was shifted upward at 41.0 Hz.

5. CONCLUSION

During frequency sweep in the environmental test room, the large vibrating surface of the exciter behaves as a loudspeaker, generating high acoustic pressures which in turn induce large displacements of test floor, whose modal frequencies in some cases coincide with room acoustic resonance frequencies.

Using Helmholtz resonators tuned to the first resonance frequencies of the room, the acoustic pressure during exciter sweep was reduced of about 6 to 12 dB depending on mode.

Using steel stiffening beams below floor brought the first modal frequency from 18.3 Hz to 41 Hz, and the maximum rms displacement during sweep was reduced from 15 microns to 0.4 microns.