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VIBRATION INSULATION OF TEST BENCHES FOR COMFORT AND FATIGUE ASSESSMENT OF CARS

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ABSTRACT

Vibration control assumes great importance in test bench design, especially when test frequencies vary in a very wide range.

The aim of the paper is to illustrate the studies carried out for the design of the insulating foundations of two big test benches to be built in the FIAT Research Centre (CRF) in Turin.

The benches, that will be located in a suitably modified existing building, are intended to be employed for comfort and fatigue tests on cars of different type and dimensions. Each bench is equipped with four actuators, able to act independently, producing load time histories whose frequency spectra can vary in a very general way between 0 and 200 Hz. The two benches, whose individual surface is about 20 m^2, differ mainly on the degrees of freedom of each actuator: in the first bench the load direction is vertical, while in the second one the loads can act in any direction.

To satisfy the wide set of design constraints a twin insulation system has been foreseen, using both pneumatic suspensions and elastomeric pads.

In the paper, the theoretical analyses carried out are illustrated and the results are discussed, justifying the design choices.

1. INTRODUCTION

In order to improve the knowledges concerning the human comfort and the behaviour of the vehicles subjected to random vibrations, the FIAT Company planned the execution of two test benches to be built in its Research Centre (CRF) in Turin, in such a way that experimental tests on cars can be performed under very general conditions.

The test rigs are located inside an existing building, where several different activities can take place, so that particular attention has been devoted to minimise the transmission of environmental noises and vibrations.

Each test bench is equipped with four independent actuators: in the first bench they can act only in the vertical direction while in the second one they can act in any direction.
In the design of this kind of test benches the study of the insulation system plays a very relevant role. In fact, depending on the particular test conditions, several different types of excitations can be used, reproducing narrow and broad band random vibrations with different nominal upper and lower cut-off frequencies, as well as fixed-frequency periodic vibrations, so that the insulation system must operate efficiently within a very wide range of working parameters.
In the following the theoretical studies carried out for the design of the very particular twin insulation system of the test benches are illustrated and the actual design choices are discussed.

2. DESIGN SPECIFICATIONS

The specifications for the test bench design result very severe, in fact it is required that the actuators can reproduce all kind of load histories, in particular simple harmonic histories with maximum frequencies consistent with the dynamic properties of the test apparatus, following performance curves like the one reported in figure 1.

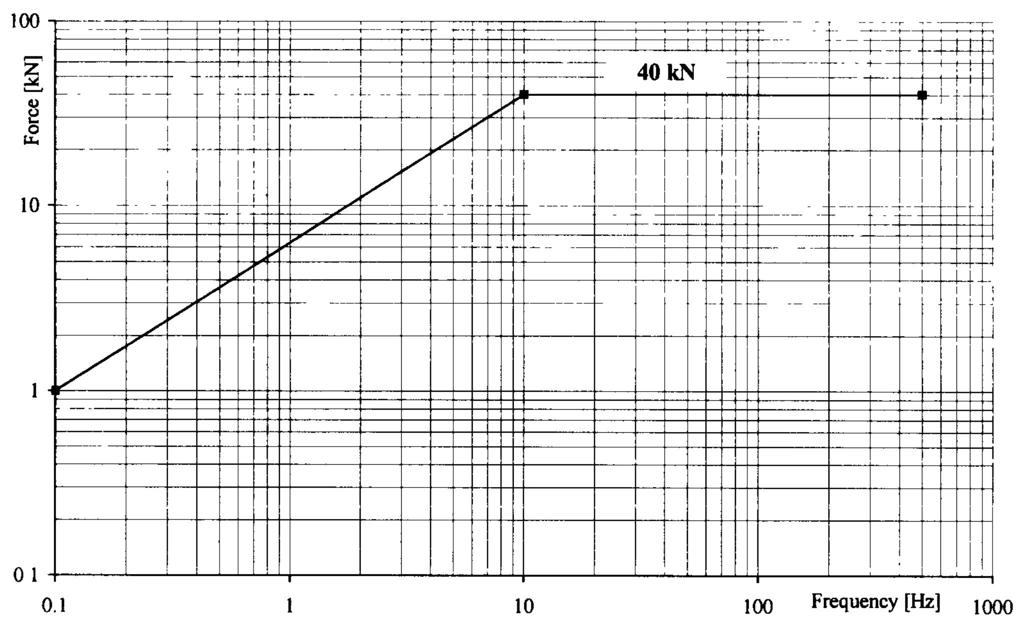

Fig 1 - Actuator performance curve

Besides, because the actuators must correspond to the wheels of the tested vehicle, their distance can vary between 1100 and 2100 mm transversally and 1500 and 3500 mm longitudinally, so that lower limits of 3300 and 4700 mm, respectively, result for the planar dimensions of the benches.
To satisfy the aforesaid design prescriptions three different topics have been considered in detail, concerning, respectively, the increase over the working limit of the natural frequency of the bench considered as elastic body, the vibration control of the bench considered as rigid body and, finally, the vibration control of the whole system made up of the suspended bench and of its foundation.

3. FREQUENCY ANALYSIS OF THE BENCH

The study is started with a wide parametric analysis, aimed to choose the material to be used for the reaction mass and the height of the bench itself, in order to maximise its fundamental frequency.

Of course, the dimensions of the bench are so big, that the total building cost is strongly dependent on the material used for the reaction mass. For this reason the choice has been limited to low cost materials, like, for example, steel or reinforced concrete, disregarding, in this first stage, more refined solutions based on hi-tech materials.

As known, the eigenfrequencies of an elastic homogeneous isotropic body are expressed by suitable functions of the propagation velocities c_1 and c_2 of longitudinal and transverse waves through the elastic medium, given, respectively, by

$$c_1 = \sqrt{\frac{E \cdot (1-\nu)}{(1+\nu) \cdot (1-2\nu) \cdot \rho}} \quad , \quad c_2 = c_1 \cdot \sqrt{\frac{1-2\nu}{2 \cdot (1-\nu)}} \tag{1}$$

where E is the Young's modulus, ν is the Poisson's ratio and ρ is the mass density of the material [1], [2]. These velocities result equal to 3526 m/sec and 2262 m/sec respectively for reinforced concrete, assuming $E=3\ 10^9$ daN/m^2 and ν equal to 0.15, and to 5943 m/sec and 3177 m/sec respectively for steel.

Clearly, because the fundamental frequency of an elastic body of assigned mass tends upward as the ratios between its characteristic dimensions tend to one, the optimal height of the reaction mass, whose planar dimensions are 3300 and 4700 mm, is 3300 mm, as result from diagrams of figure 2, in which are reported the first four frequencies of a free prismatic reinforced concrete mass, having the given planar dimensions and height varying between 330 and 6600 mm.

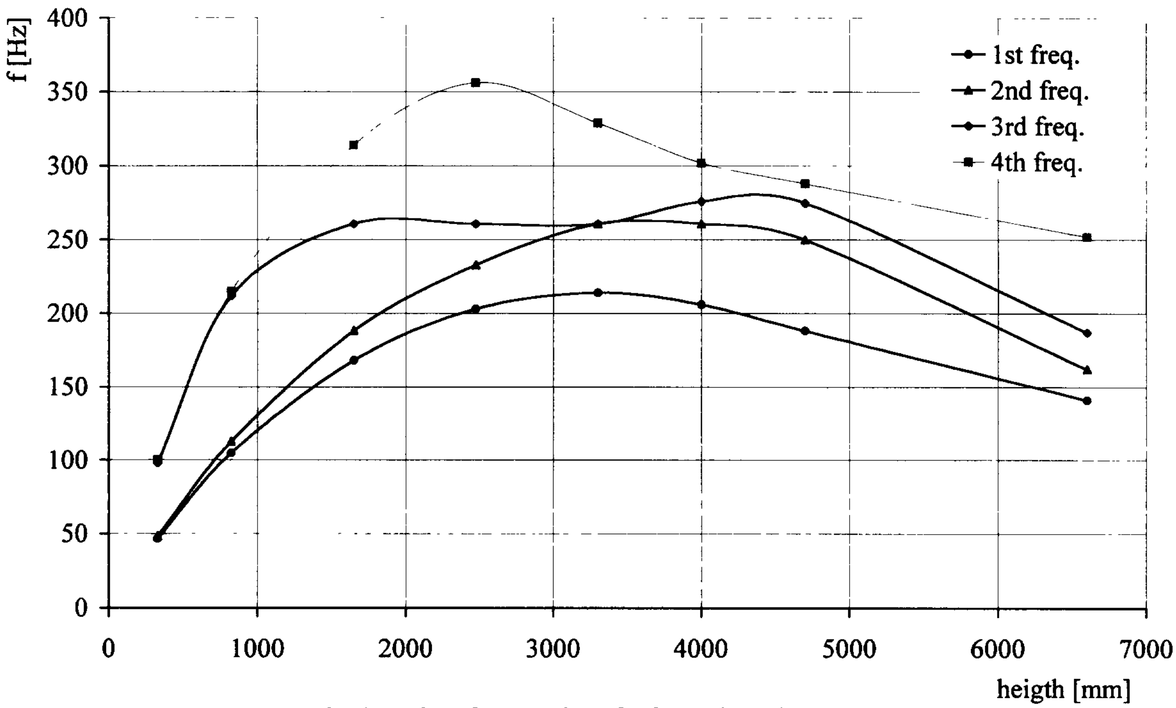

Fig. 2 - Eigenfrequencies of a free prismatic r. c. mass

The diagrams of figure 2 demonstrate that the fundamental frequency of a such reinforced concrete mass attains a maximum of 214 Hz, remaining however in the neighbourhood of 200

Hz when the height varies between 2400 and 4000 mm, so that 200 Hz can be considered as an upper limit for the frequency of harmonic excitations.

Furthermore, since the suspension system must be not only compliant, but also able to support the block, the total mass of the block itself cannot exceed a practical maximum.

On the basis of the aforesaid considerations, a 2400 mm height reinforced concrete reaction mass represents the best compromise in terms of cost and in terms of ratio between total mass and fundamental frequency; in fact a steel block having the same mass and the same planar dimensions (block height equal to 764 mm) has the fundamental frequency equal to 139 Hz, while the steel block height must be increased up to 1250 mm to attain a fundamental frequency of 200 Hz.

4. INSULATION SYSTEM OF THE BENCH

The preliminary study leads so to the definition of a reinforced concrete mass having a minimum height equal to 2400 mm and able to operate in the frequency range 0-200 Hz.

For practical reasons, in the upper part of the bench a longitudinal square channel, whose side is 600 mm, is foreseen, so that the total height of the block becomes 3000 mm (see figure 3).

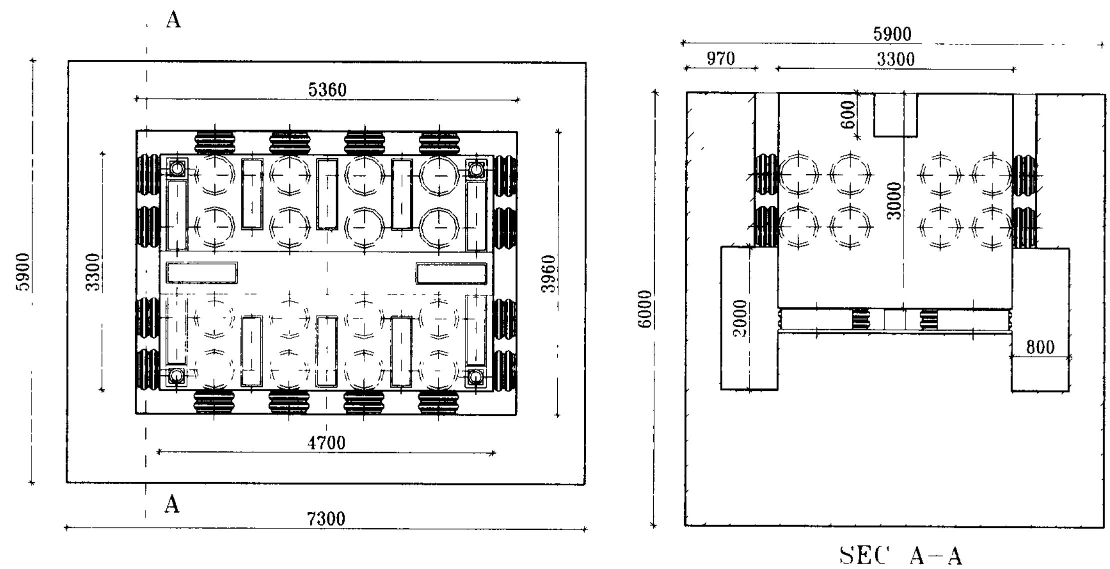

Fig. 3 – General scheme of the bench

As known, with the aforesaid design constraints, the optimal insulation system [3], allowing to obtain rigid body eigenfrequencies of the mass below 2 Hz, consists of air springs, which has been adopted, in fact, for the chief insulation system of the bench.

From the theoretical point of view, the problem has been studied according to the general theory of vibration of rigid body [3], [4] arranging the air springs in such a way that the coupling of modes is avoided whenever is possible and the eigenfrequencies result nearly coincident.

The final arrangement of the 16 air springs with convolutions is sketched out in figure 3. The axial stiffness of each isolator is about 850 daN/cm and the undamped eigenfrequencies vary from 1.24 Hz for the fundamental mode to 2.00 Hz for the 6th mode.

The characteristic parameters of the system, forces, displacements, velocities and accelerations, for any combination of exciting harmonic forces imposed by the actuators have been then evaluated through the solution of the differential equations of the motion of the rigid body. In figure 4, for example, are reported the maximum forces transmitted by the most heavily loaded vertical air spring, obtained in this way on the basis of the performance curve of figure 1: the two diagrams refer, respectively, to all the actuators acting in phase in the vertical direction and to two pairs of in phase actuators in opposition of phase.

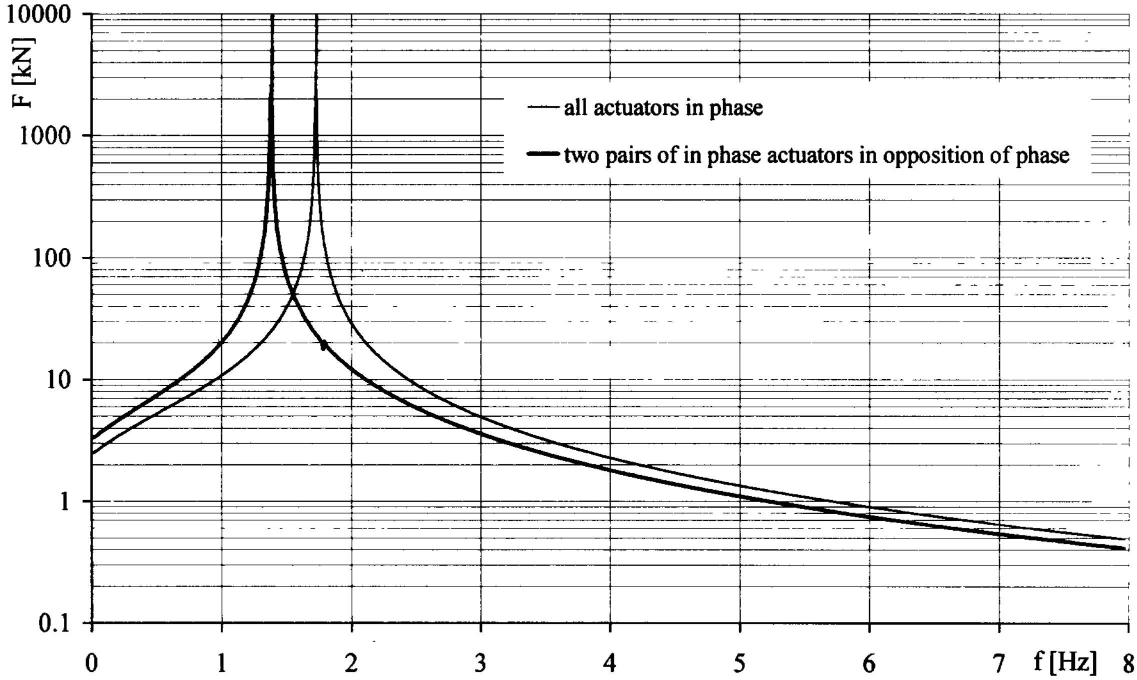

Fig. 4 – Force transmissibility for the heavily loaded vertical air spring

In the actual case, the standing wave effects [3] in the air springs is not relevant because the standing wave frequency is bigger than 600 Hz, beyond the operating frequencies of the bench.
It is clear that pneumatic suspension system operates very satisfactorily for forcing frequencies bigger than 3 Hz, while for frequency of harmonic excitations below this value it is necessary to foresee an alternative kind of suspension system, consisting of stiffer isolators.
The alternative system is composed by a set of 24 elastomeric pads of 250×500 mm having a reaction modulus of about 90 daN/cm^3, arranged as shown in figure 3. This system, designed to be active below 5 Hz, is also effective as security system for the chief suspension system, while the two systems can be interchanged simply inflating or deflating the air springs.
An additional independent lifting system, based on four hydraulic jacks mounted at the four corner of the mass (see figure 3), allows the usual maintenance of the bench.
When the operating system is the elastomeric suspension, the natural frequencies of the mass belong to the interval 17.75÷32.33 Hz and it is still possible to apply the general theory. In figure 5, are represented the diagrams, similar to those reported in figure 4, concerning with the maximum forces transmitted by the most heavily loaded vertical elastomeric pad.

5. STUDY OF THE BENCH FOUNDATION

In consequence of the heavy design prescriptions, detailed studies regarding the dynamic behaviour of the bench foundation and the interaction between the whole system and the soil has been performed, in order to estimate the actions transmitted in the surroundings during the tests.

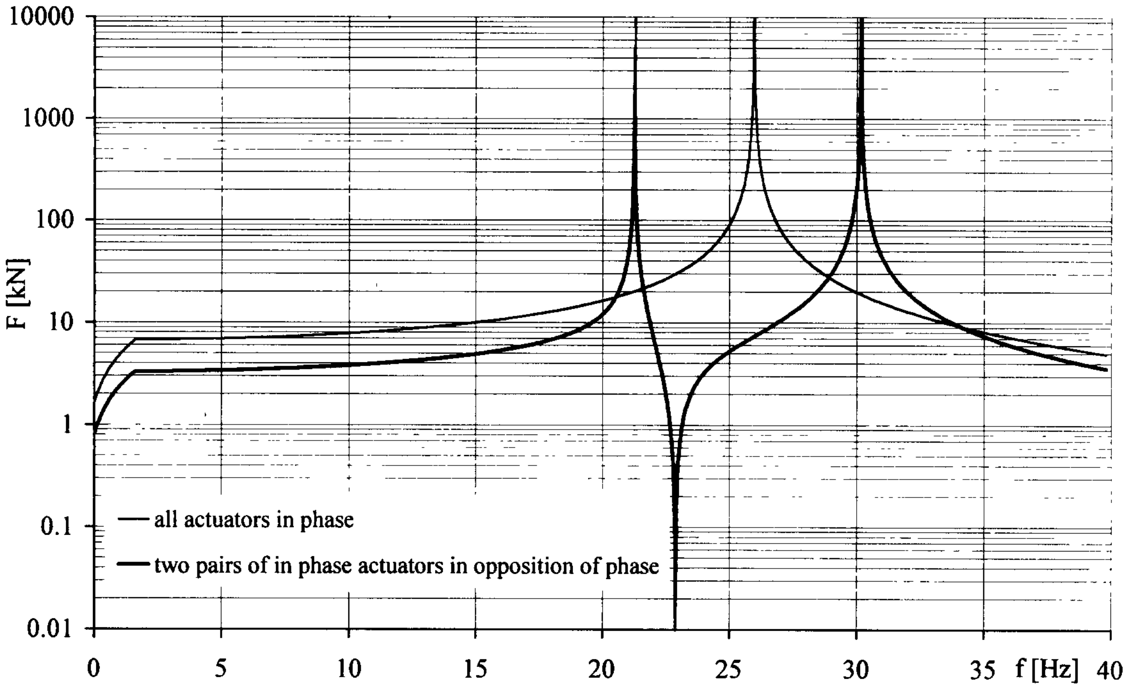

Fig. 5 – Force transmissibility for the heavily loaded vertical elastomeric pad

Preliminarily, in-situ investigations have been carried out, according to the literature [5], [6], to establish the relevant dynamic properties of the foundation soil.

On the basis of these tests, even if the characteristics of the soil, composed by sand and gravel, result very good, it has been decided to realise a pile group, in such a way to increase adequately the stiffness and the damping capacity of the ground.

In this final stage of the research the dynamic behaviour of the whole system, made up of the reaction mass, the twin suspension system, the foundation and the ground, has been investigated, using the COSMOS/M finite element program version 1.75. The reaction mass and the foundation have been modelled using the 8-nodes Solid elements of the COSMOS/M library, the air spring and the elastomeric pads have been modelled using 2-nodes Spring elements, while 1-node Spring elements have been used to simulate the soil. A partial view of the mesh of the FEM model is reported in figure 6.

In the FEM frequency analyses the first 40 natural frequencies of the whole have been evaluated, disregarding the damping of the material, for both types of suspensions system. Some relevant results are illustrated in the figures 7, 8 and 9. In particular, figures 7 and 8 refer to the first elastic mode shapes of the reaction mass and of the foundation, respectively, while figure 9 refers to the first rigid mode shape of the reaction mass on elastomeric suspension. It must be noted that the elastic mode shapes are not very much influenced by the suspension system, in fact the fundamental frequencies of the reaction mass and of the foundation are 187 Hz and 45 Hz, respectively, with air spring suspended mass and 187 Hz and 47 Hz with elastomeric pad suspended mass.

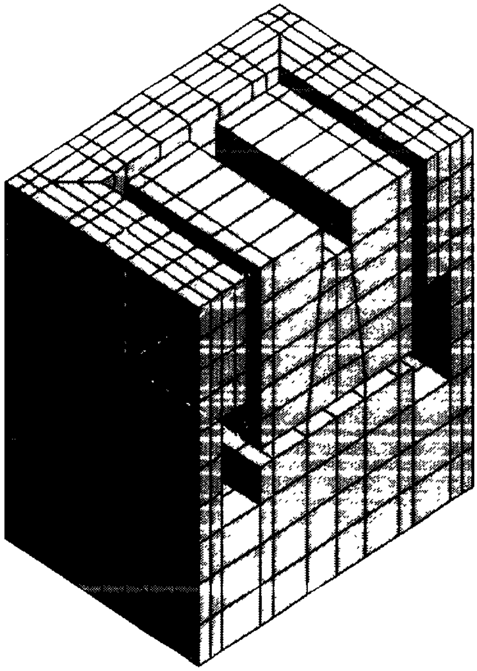

Fig. 6 – Section of the mesh of the model

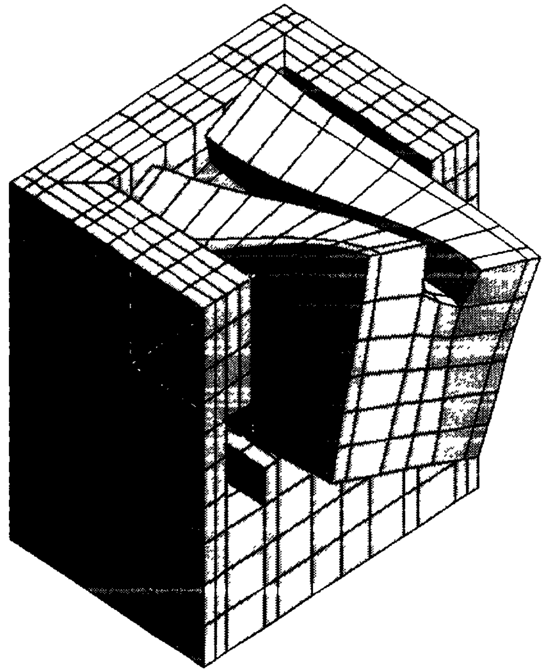

Fig. 7 – Plot of the 30th mode (air springs) (1st elastic mode of the mass) – f=187 Hz

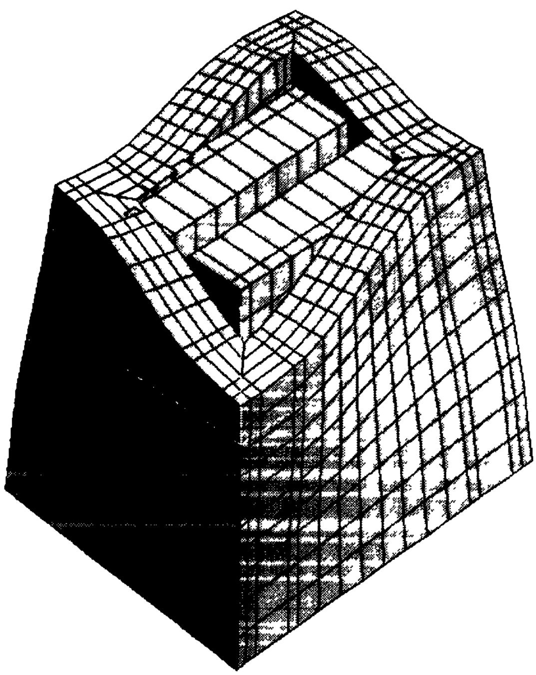

Fig. 8 – Plot of the 13th mode (air springs) (1st elastic mode of the foundation) - f=45 Hz

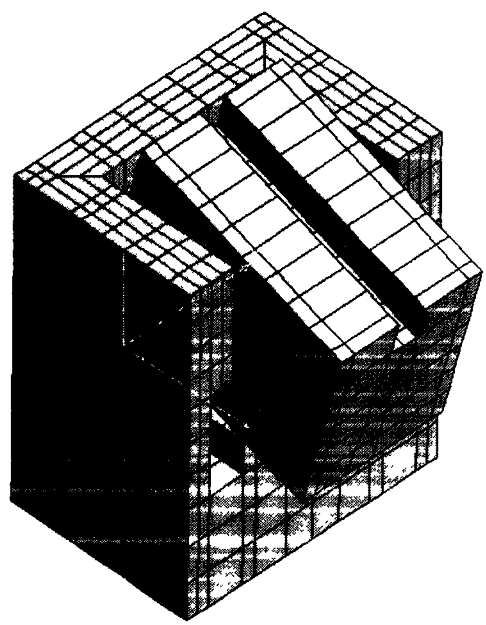

Fig. 9 – Plot of the 4th mode (elast. pads) (1st rigid mode of the mass) – f=16.8 Hz

Finally, the fundamental frequency of the whole system is equal to 12 Hz for air suspensions and 9.6 Hz for elastomeric suspensions.

6. CONCLUSIONS

The discussion of the results of the theoretical and numerical analysis performed till now validates the design choices and leads to very relevant conclusions.

The simplified analysis carried out in the preliminary phase, regarding the evaluation of the natural frequencies of the linear elastic reaction mass and of the suspended rigid block, is substantially confirmed by more refined investigations and it results a very useful tool for the preliminary design of benches.

It is possible, using air suspensions, to operate in the frequency range 3-160 Hz with absolutely arbitrary dynamic excitations.

The alternative elastomeric suspension system allows also the 0÷3 Hz frequency interval covering.

The natural frequencies of the foundation coffer, bigger than 45 Hz, allow to operate safely considering that, beyond this frequency, the transmitted forces are negligible and that, in any case, some structural damping can be taken into account.

When air is replaced by the stiffer elastomeric suspension, the fundamental frequency of the whole tends to reduce to 9.6 Hz, however out of the working range (0-3 Hz). Besides that, it must be noted that the effect of low frequency excitations on the ground is not particularly important, even disregarding the soil damping.

The proposed solution can be easily improved by using a very refined mix design and/or a suitable fibre reinforcement of the concrete, so obtaining an increase of its mechanical properties.

Further developments of the research, still in progress, concern the use, in combination with traditional building materials, of hi-damping materials to enlarge the frequency working range.

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