Vibration Isolation Support System for SCHENBERG Detector

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Abstract: We designed a mechanical isolation system for a spherical resonant gravitational wave detector we are building in Brazil. We have used the Finite Element Method to perform the dynamical analysis. The system is a multiple stage passive pendulum formed by cylinders joined by C springs and rods. Our results showed that the designed system could allow a 280 dB attenuation factor in the bandwidth, from 3.1 kHz to 3.2 kHz, where the SCHENBERG detector will be sensitive.

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1. Introduction

The Brazilian Graviton Project Group is presently constructing the "Mario SCHENBERG Detector" - a 0,65 m diameter spherical gravitational wave detector made of CuAl6%. We have designed the vibration isolation system for the SCHENBERG detector. The basic idea used in the system design was to construct a mechanical multi-stage low pass filter capable of filtering all the mechanical vibration noises. The dynamical behavior of the system was numerically simulated using the Finite Element Method[1] (FEM).

2 .The Isolation Support System Designed for SCHENBERG DETECTOR

With the SCHENBERG detector, we expect to obtain a spectral sensitivity of about 10^{-22}m/(Hz)$^{1/2}$, which means that is necessary to attenuate the seismic noise by a factor of 10^{-10} , or -200 dB, at least, in the spectral range of interest (3100Hz). So it is necessary to construct an isolation system capable of providing this level of attenuation. The classic way to obtain such high attenuation is to use a series of spring-mass modules in a multi-stage set-up[2]. As is well known, this type of structure works as a low-pass filter and so provides an attenuation to mechanical noises. This kind of isolation system is normally used both in interferometers and resonant antennas[3].

The figure 1 shows a schematics of the structure we are proposing: five cylinders joined together by six sets of C springs compose the system. The C springs connect the upper face of a cylinder to the bottom face of the immediately above one. The use of C shaped springs in isolation systems for gravitational waves detection was first proposed by AURIGA group.

The basic design constraints we used were: 1) the cylinders must be small enough to have the first internal resonance at a frequency higher and far from the detection frequency; and 2) to choose the geometrical parameters of the springs (internal and external radius and widths) in such a way to adjust the elastic characteristics in order to obtain a spectral window, as large as possible, around the detection frequency, where there are not internal resonances.

The top three cylinders and their C springs, as well as the resonant sphere itself, will be made of Cu-Al(6%) alloy (light gray in Figure 1). The two lower cylinders and tubes will be made by OFHC (Oxygen Free High purity Copper). At last we have the higher tube which will be Titanium made, since this material has higher yield stress values, allowing to use a tube with a smaller diameter; which decrease the heat flux between outside and inside the detector.

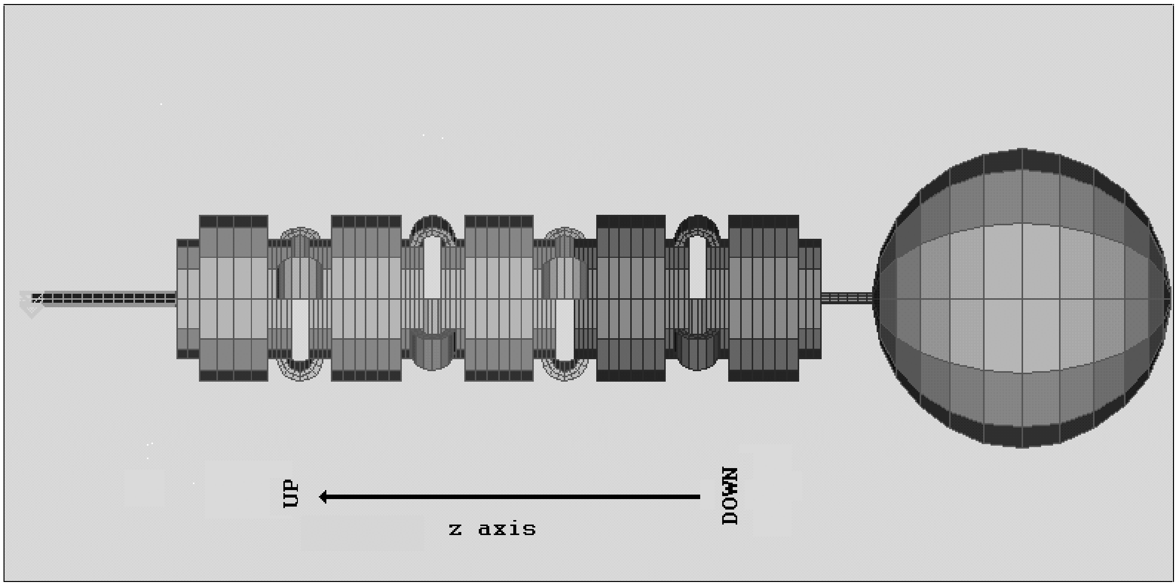

Figure 1 - Finite Elements Model of the SCHENBERG isolation system with the resonant mass.

Our numerical FEM model also includes the resonant sphere. The simulation of both the resonant sphere and the isolation system in the same model allows to analyze the frequency response directly in the surface of the sphere at the position where the transducers will be assembled.

3 . THE FINITE ELEMENT ANALYSIS: NUMERICAL SIMULATIONS

Using the above considerations we defined a first geometric set-up for the vibrational isolation system. We have numerically modeled the structure using a large number of finite elements in order to increase the FEM calculation accuracy. We also conceived a geometrical configuration as symmetrical as possible (see Fig. 1) to simplify the numerical solutions.

As a first step of the analysis we used the FEM to calculate the stress of the structure, considering the total weight (including, of course, the mass of the resonant sphere). It's a design requirement that the stress on the model must be from 10% to 20% of the yield stress, since the phonon production must be the least possible. The OFHC yield stress is 3×10^9 dyn/cm^2 and the Cu-Al6% yield stress is 1.3×10^9 dyn/cm^2 . Of course, the maximum stress in the structure occurs in the C spring middle at the top of the system. In the proposed model this maximum stress value is 2.5×10^8 dyn/cm^2, which corresponds to only 20% of the yield stress for the OFHC. In all other elements of the model, the stress was under this value, which means that the materials in all the structure are far enough from the non-linear region in way to prevent the occurrence of up-conversion phenomena by body stress relaxation.

In Figure 2 we show the calculated normal modes, using the Finite Element Method, for the proposed isolation system set-up, in a large spectral band (from 0 Hz to 4000Hz) that includes the detector characteristic frequency (3100Hz) . With this set-up we obtained a kind of spectral window, from 2837 Hz to 3478 Hz, where the structure has no internal resonances. This 640 Hz window is large enough to prevent the noise increasing by the internal modes resonances and so to allow us to expect a large attenuation factor around the detection frequency.

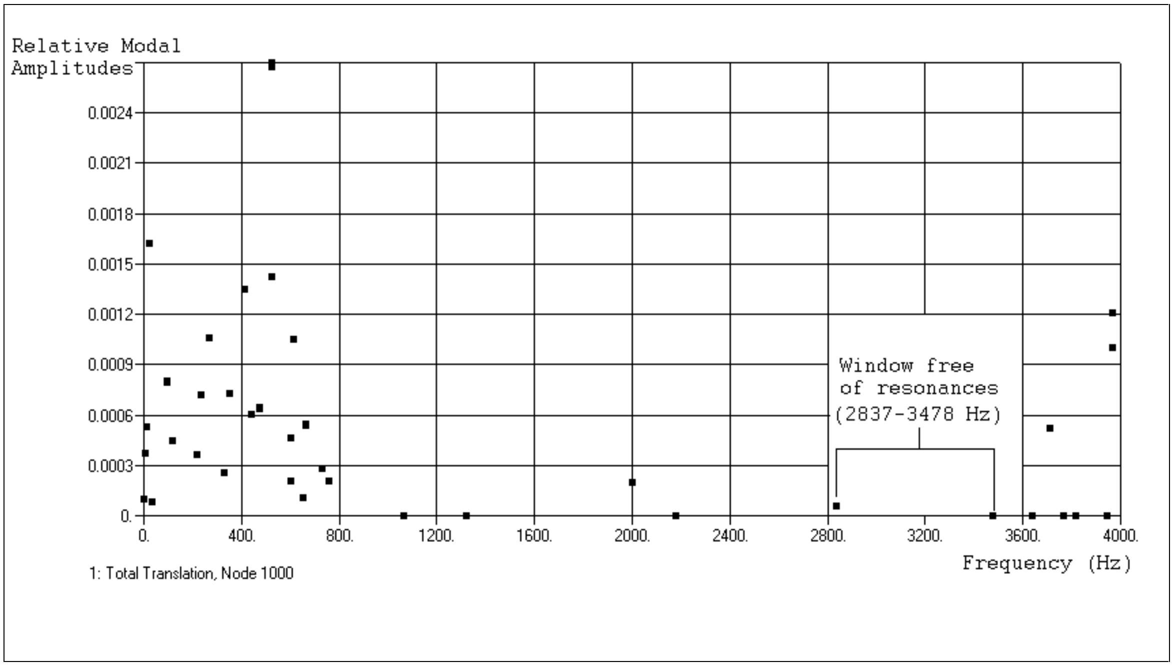

Figure 2 - Vibration Isolation System Normal Modes Frequencies.

4- DISCUSSION : DYNAMICAL ANALISYS AND THE FREQUENCY RESPONSE CURVES

Even if the proposed structure is not directly comparable with a simple multi-stage uncoupled pendulum, since we must to consider also the internal high frequency resonances of the rigid springs, we can expect to achieve a good attenuation level at frequencies far enough from the first modes. It is possible to calculate the frequency response solving numerically the dynamical equations of the structure, considering also a force of excitation applied to the structure.

The frequency response will be calculated by:

$$\frac{\{x(w)\}}{\{P(w)\}} = \frac{1}{\left[-w^2 M + K\right]}.$$

Where M and K are respectively the matrices for the mass distribution and elastic constants and $P_{(w)}$ is an excitation force (acceleration) applied to the structure. To perform our calculation, we have used a high acceleration amplitude signal, but we have imposed to $P_{(w)}$ a spectral dependence similar to the classical[4] seismic noise behavior given by $\alpha\omega^{-2}$, where α depends on the geographical coordinates on Earth surface.

It would be necessary to consider at least frequencies from 0Hz to 4000Hz, which means computing all amplitudes in the six degrees-of-freedom for each element of the isolation structure for each frequency step.

To solve this problem we used the direct method[5] which calculates "exactly" the numerical response. The Figure 3 shows the results of this calculation. The curve1 corresponds to the spectral response of the system in a node located near to the excitation point at the top of the structure. The curve 2 represents the frequency response in a node located at the surface of the resonant sphere. The transfer function to the system can be estimated comparing both curves. In curve 2 it is also possible to see the sphere resonances (torsional – at 2886 Hz - and radial quadrupolar at 3157 Hz) since we have used a large excitation signal in our calculations. The difference between the curves corresponds to the attenuation in each frequency. In the detection region (form 3100 Hz to 3200 Hz) we obtained about 280 dB which is sufficient to achieve the sensitivity level we are expecting to the SCHENBERG Detector.

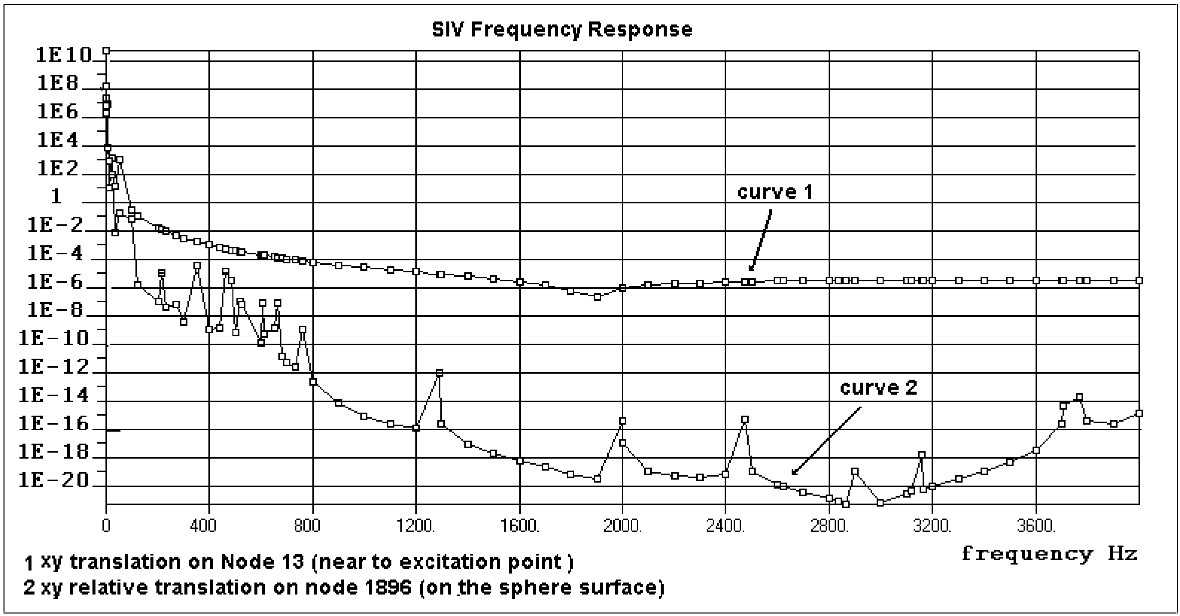

Figure 3 - Attenuation produced on the structure by the vibration isolation system

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