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# NUMERICAL AND EXPERIMENTAL STUDY FOR A BEAM SYSTEM WITH LOCAL UNILATERAL CONTACT MODELING SATELLITE SOLAR ARRAYS

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In collaboration with Thales Alenia Space, France

## 1 ABSTRACT

The mass reduction of satellite solar arrays results in significant panel flexibility, so possibly striking one another dynamically leading ultimately to structural damage. To prevent this, rubber snubbers are mounted at well chosen points of the structure and they act as one sided linear spring; as a negative consequence, the dynamic of these panels becomes nonlinear. The finite element approximation is used to solve partial differential equations governing the structural dynamic. The models are validated and adjusted with experiments done in the ISVR laboratory, Southampton university.

significantly if the imposed displacement is replaced by a periodic force excitation, the configuration used is easiest to be realized from a technical point of view as the rig is very simple to built [see figure 2]

The present study is to simulate the behavior of a beam which strikes a snubber under a periodic excitation. As the interest is to deal with the first three eigen frequencies, the beam is modeled by ten finite elements.

The results are presented and compared in the frequency domain, the FFT is applied to the predicted and the experimental displacement of the last node of the beam finite element. The mass effect of the force transducer is taken in account in the finite element code. Note that no treatment is done or used from the acquisition system as the problem is nonlinear, the transfer function and the other functions use linear assumptions. The time signal is taken and the processing is done using an external software (Scilab [8]). The numerical predictions compared to experimental results show very good agreement.

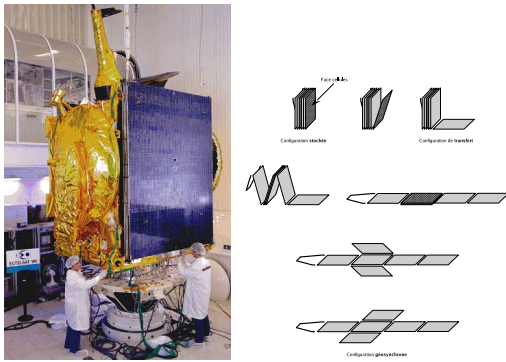


Figure 1: Solar array of the satellite under a test on a shaker

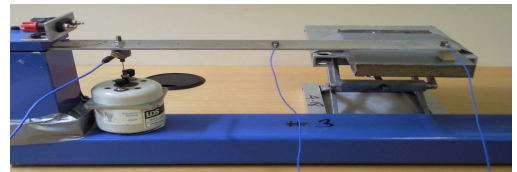
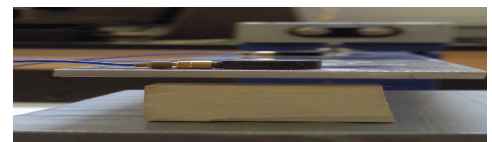


Figure 2: The rig used for the experiments: a linear clamped-free beam in contact with a rubber

## 2 INTRODUCTION

The study of the total dynamic behavior of solar arrays in a folded position with snubbers are so complicated, that to simplify, a solar array is modeled by a clamped-free Bernoulli beam with one-sided linear spring. This system is subjected to a periodic excitation force.

The real state of the problem is close to a beam with unilateral contact subjected to a periodic imposed displacement of the base, but the dynamical behavior of the system do not change



### 3 THE MODEL

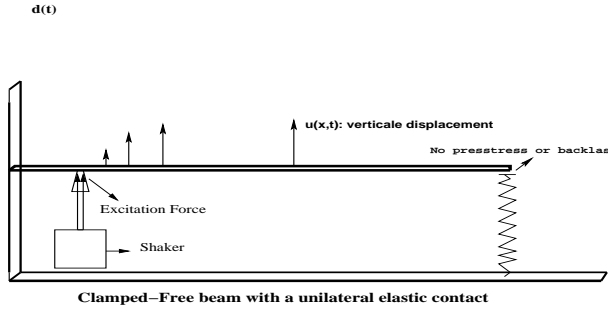


Figure 3: beam system with an unilateral spring under a periodic excitation

The beam motion with a snubber can be modeled as:

$$\rho S \ddot{u}(x, t) + EI u^{(iv)}(x, t) = F(t) \delta_{x_0} - k_r u(x_1, t)_- \quad (3.1)$$

$$u(0, t) = 0, \partial_x u(0, t) = 0$$

$$u(x, t)_- = \begin{cases} u(x, t) & \text{if } u \leq 0 \\ 0 & \text{if } u > 0 \end{cases} \quad (3.2)$$

The classical Hermite cubic finite element approximation is used, it yields an ordinary differential system in the form:

$$M \ddot{q} + Kq = -[k_r(q_{n_1})_-]e_{n_1} + F(t)e_{n_2} \quad (3.3)$$

Where  $M$  and  $K$  are respectively the mass and stiffness assembled matrices,  $q$  is the vector of degrees of freedom of the beam,  $q_i = (u_i, \partial_x u_i)$ ,  $i = 1, \dots, n$ , where  $n$  is the size of  $M$ ,  $n_1$  and  $n_2$  are the indices of the nodes where the spring and the excitation force meet the beam respectively. Numerical time integration was performed using ODE numerical integration for 'stiff' problems, package ODEPACK is called and it uses the BDF method.

The use of a small electrodynamic shaker yields a technical problem due to the reaction of the beam, the input force  $F(t)$  can not be a simple sine wave. To deal with this problem, a force transducer is fixed between the shaker and the beam. The numerical codes use the actual force time signal coming from the acquisition system which is periodic. Figure [5] shows the input force signals with its spectrum contents.

### 4 EXPERIMENTAL AND NUMERICAL RESULTS

Figures [2] and [3] give a clear idea of the experiment setup. In the table, all the parameters used in the simulations and the experiments are shown as well as the first three predicted eigen frequencies of the linear beam without the spring. The integration time is fixed at  $t = 1s$  for all the simulations and the experiment sequences which is five times the greater period of

the system.

Figure [7] shows the FFT of the numerical and the experimental displacements for  $32Hz$  and  $124Hz$ , the height of the peaks are normalized by the maximum; the predicted frequencies found are exactly the same measured for a large number of harmonics. However, a small shift in the height of these pikes appears from the fifth harmonic, this is maybe due to the low number of finite elements used to model the beam; some investigations are in progress to understand this aspect. Other tests with random excitations have shown good agreement. The effect of the unilateral contact is underlined, the input frequency is split into its all harmonics.

Figures [5] and [6] show the input excitation force, it is clear from the time signal and from its frequency content that this force is not a single sine wave, but it is periodic.

Figure [4] shows the predicted displacement for an excitation of  $32Hz$  and  $124Hz$ , the displacement is almost positive, this is due to the high stiffness of the spring but the time response is still periodic.

Note that the frequencies  $32Hz$  and  $124Hz$  are chosen and presented herein because they represent the first two nonlinear normal modes of the nonlinear normal mode, they are computed numerically by a numerical sweep test. Non linear normal modes  $NNM$  of a nonlinear differential systems are an extension of the well known linear normal modes, another interest of the authors with some papers in progress.

beam length	beam width	beam thickness	beam's Young modulus
0.35m	0.0385m	0.003m	$69 \times 10^9 N/m^2$
Spring stiffness	1 <sup>st</sup> linear eigen frequency	2 <sup>nd</sup> linear eigen frequency	3 <sup>rd</sup> linear eigen frequency
57.14 K N/m	19.97Hz	122.2Hz	318.8Hz

(4.1)

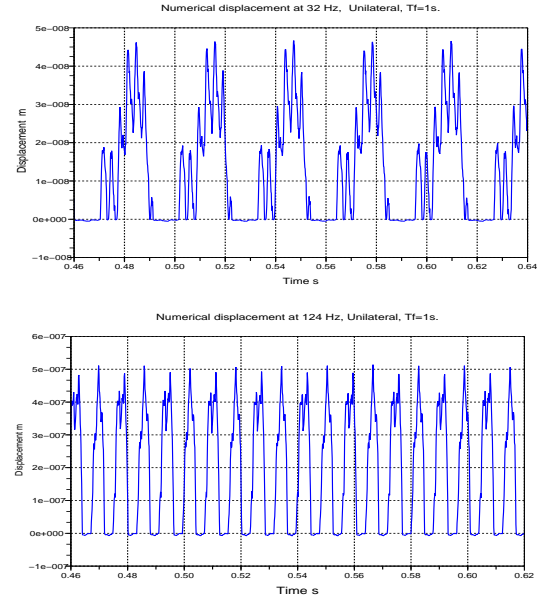


Figure 4: The predicted displacements for an excitation of  $32 Hz$  and  $124 Hz$ , the high stiffness of the rubber yields almost a positive displacement

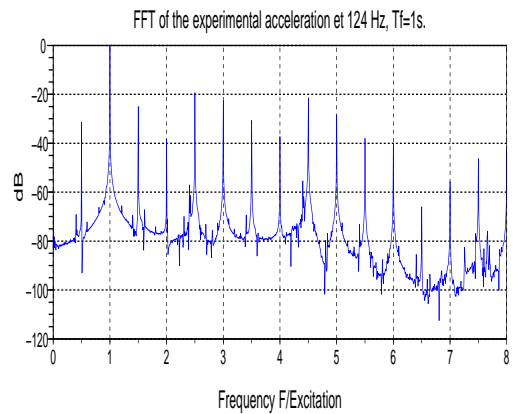
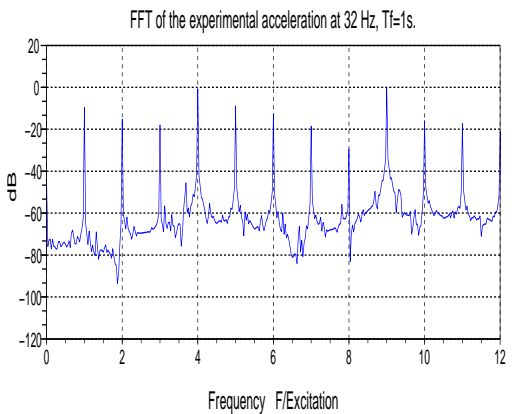
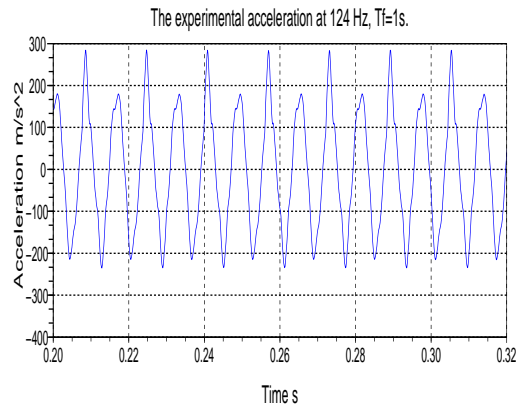
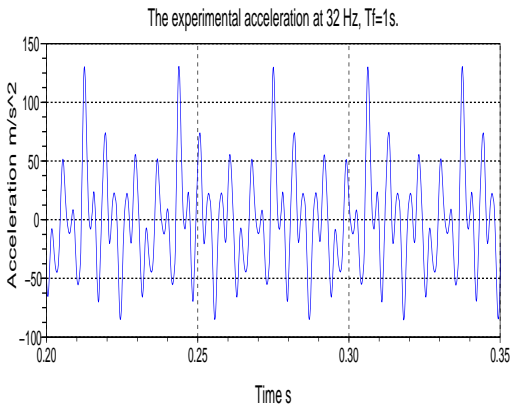
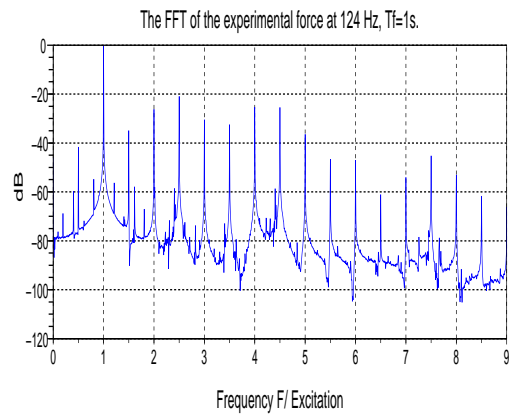
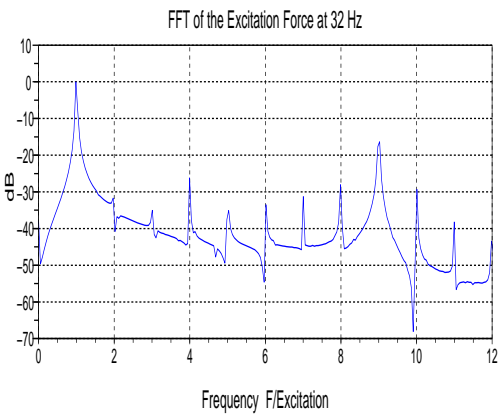
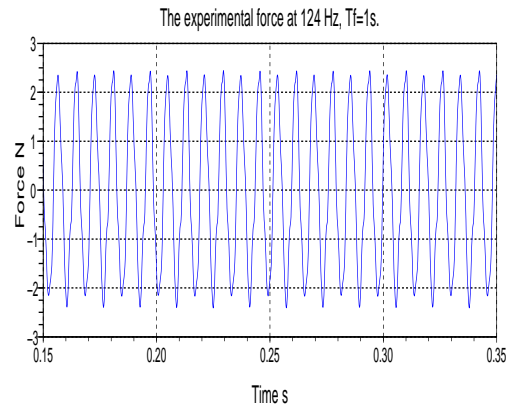
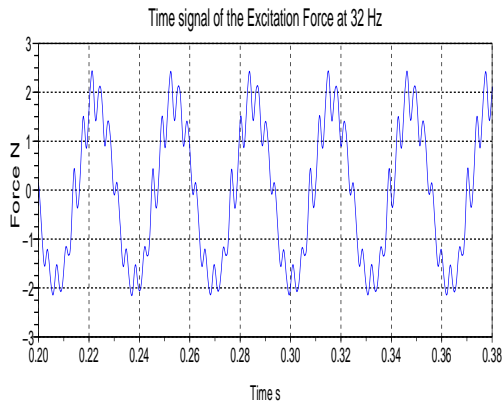


Figure 5: The time signal and its FFT of the input force and the experimental acceleration for an excitation of  $32\text{ Hz}$

Figure 6: The time signal and its FFT of the input force and the experimental acceleration for an excitation of  $124\text{ Hz}$

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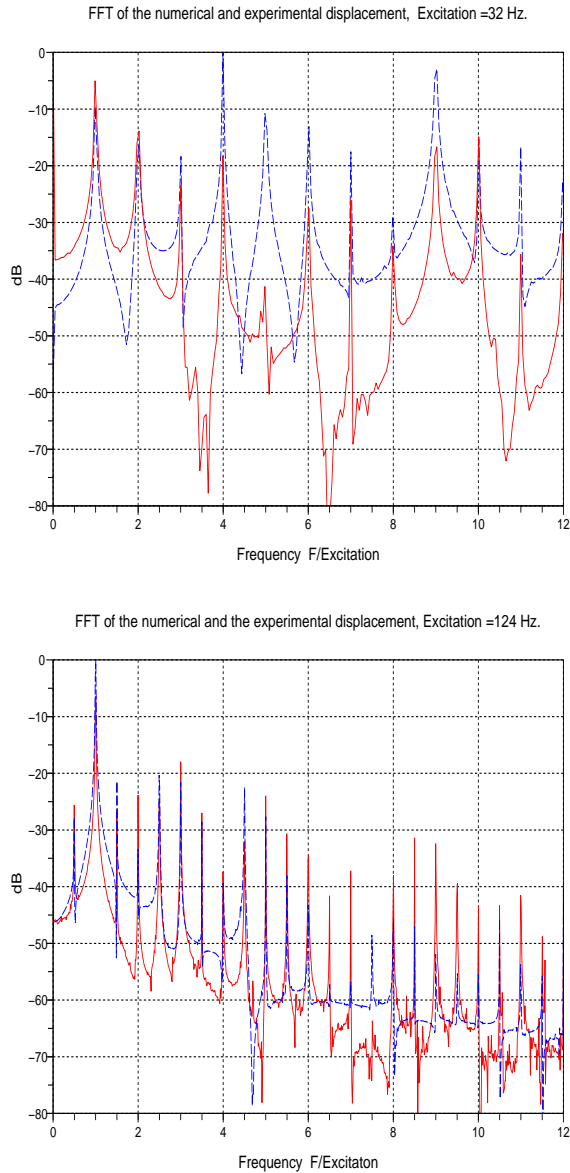


Figure 7: dashed line: FFT and the experimental displacements, solid line: FFT of the numerical displacements. Two different excitation:  $32\text{Hz}$  and  $124\text{Hz}$