VIBRATION BEHAVIOR OF A LOW-ORBIT SATELLITE SUBASSEMBLY

Sorin DRĂGHICI[1], Ioan PĂRĂUŞANU[2], Ştefan Dan PASTRAMĂ[3], Laurenţiu ZAHARIA[4], Anton HADĂR[5]

In this paper, the authors conducted a comparative experimental - numerical analysis on the vibration behavior of a satellite subset. The studied structure was a box housing electronic components. A platform for noise and vibration analysis was used to experimentally determine the natural frequencies of the box. Finite element analyses were undertaken for the same structure and the numerical values were compared with the experimental ones. Since the differences between the numerical and experimental results were less than 6%, it can be considered that the numerical model was validated.

Keywords: satellite, natural frequency, vibrations, aluminum, finite element.

1. Introduction

In recent decades, aerospace industry in general, and satellites in particularly have become integrant part of the living standard, so one almost cannot conceive modern living in their absence. Communications satellites are the ones that transmit television and mobile phones signals through the world. Networks of satellites help to locate exact position of objects and allow the use of a Global Positioning System (GPS). Meteorological satellites are the basic tool in the weather forecast. Satellites allow also expansion of scientific knowledge of humanity on Earth and the universe.

Low earth orbit (LEO) satellites are usually used for communication and earth imaging [1]. During their development, manufacturing and launch to the final operating position in space, satellites experience several types of mechanical, thermal and electromagnetic loads [2]. Several researchers have drawn their

[1] Assistant prof., Department of Strength of Materials, University POLITEHNICA of Bucharest, Romania, e-mail: sdrg79@yahoo.com;
[2] Prof., Department of Strength of Materials, University POLITEHNICA of Bucharest, Romania, e-mail: ioan.parausanu@upb.ro;
[3]Prof., Department of Strength of Materials, University POLITEHNICA of Bucharest, Romania, e-mail: stefan.pastrama@upb.ro
[4] Dr. Eng., AFER (Romanian Railway Authority) Bucharest, Romania, e-mail: rentizaharia@yahoo.com
[5]Prof., Department of Strength of Materials, University POLITEHNICA of Bucharest, Romania, e-mail: anton.hadar@upb.ro

interest to the study and evaluation of such loads. The thermal analysis of a small satellite orbiting around the Earth was studied by Pérez-Grande *et al.* [3] using direct integration of the heat balance equations of a two-node reduced model. As the thermal loads (solar radiation, albedo, etc.) are harmonic, the problem was solved by means of Fourier analysis methods. Cho and Rhee [2] studied the complex vibration characteristics of a satellite, using the Finite Element Method and experimental measurements. An extensive random vibration test campaign was conducted on the satellite structure using an electro dynamic shaker, and the obtained results were compared with the numerical values. Results for active vibration control predicted from experimental measurements are reported by Moshrefi-Torbati *et al.* [4] for a 4.5m long satellite boom consisting of 10 identical bays with equilateral triangular cross sections. The authors employed a genetic algorithm to find high-quality positions for three actuators on the structure that will achieve the greatest reductions in vibration transmission. The design, modeling and analysis of a satellite model used for remote sensing was presented by Israr [5]. The author performed static, modal and harmonic finite element analyses at the time of ground testing and launching phase. Oda *et al.* [6] [6] developed a system that uses a Complementary Metal-Oxide Semiconductor (CMOS) camera to measure the distortion and vibration of the solar array paddle from the Japan Aerospace Exploration Agency's earth observation satellite "GOSAT". The reason for this complex study was the lack of accuracy of the images taken when a satellite goes into or out from an eclipse, due to the deformation and vibration of the satellite's solar array paddle along with the instantaneous change in solar energy received by the satellite. Results of vibration tests on the Italian Space Agency satellite LARES and the trade-off analysis performed to select the best technology for reliable monitoring of the preload were reported by Paris [7].

The electronic box studied in this paper is one of the subassemblies of the PROBA 2 (PRoject for On Board Autonomy) satellite [8]. PROBA2 is the second satellite in the European Space Agency's series of PROBA low-cost satellites that are being used to validate new spacecraft technologies while also carrying scientific instruments [9]. PROBA2 is a small satellite (130 kg) which was launched on November 2, 2009, in a sun-synchronous low Earth orbit (altitude of 725 km).

Since the cost of launching an object on a LEO satellite has been estimated at about 20,000 $/kg [10], the authors have previously studied the possibility of obtaining a new, lighter geometry to replace the existing electronic housing box in the PROBA2 satellite, following studies regarding the behavior of the structure subjected to gravity and vibrations [11]. In [12], the mechanical behavior of the same structure subjected to thermal cycles occurring on orbit was analyzed.

2. Experimental and numerical analyses

To validate the finite element model presented in [11] and [12], the experimental determination of the dynamic characteristics for the proposed subassembly is necessary. For this purpose, a box was manufactured from an aluminum plate, respecting the overall dimensions of the proposed subassembly (Fig. 1).

The natural frequencies were determined for the subset, both experimentally and numerically, using the finite element method.

a) outside view b) inside view

Fig. 1 View of the experimental model

Fig. 2 Experimental set-up for natural frequencies measurement

The assembly was fixed on a concrete foundation with bolts having a diameter of 8 mm, simulating the mounting situation inside the satellite. The excitation was performed using an 8206 B&K modal hammer with steel head. For measuring the response, three 4514 B&K piezoelectric accelerometers were used (Fig. 2). Each accelerometer was mounted on one side of the structure.

The analysis of experimentally recorded data to obtain the frequency response curves was done using the specific software for dynamic analysis of the platform for noise and vibration measurement PULSE [13].

Accelerometers were mounted on three distinct faces of the subassembly, noted F1, F2, F3, as in Fig. 2. On each of these faces, excitation was done using a modal hammer and the response was measured.

Although nine frequency response curves were obtained, only those showing the natural frequency on the excited face were shown in Figs. 3-5.

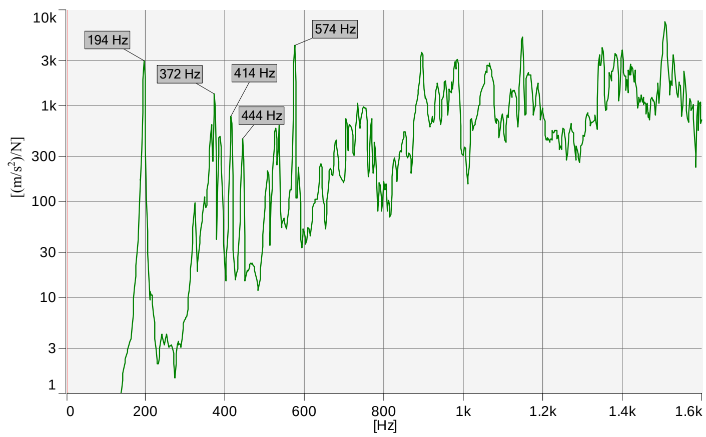


Fig. 3 The first five natural frequencies experimentally identified on Face 1

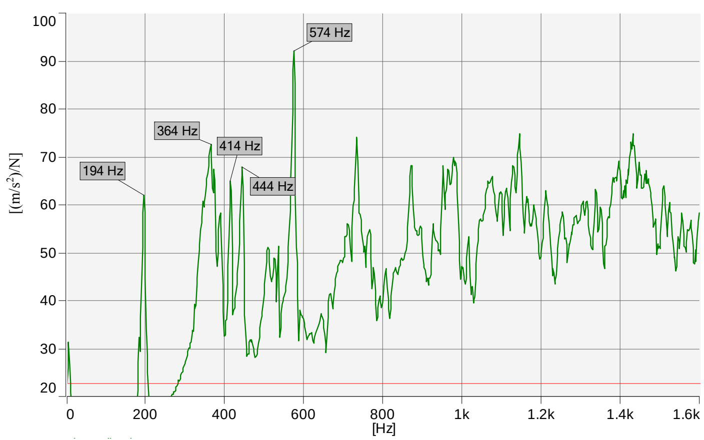


Fig. 4 The first five natural frequencies experimentally identified on Face 2

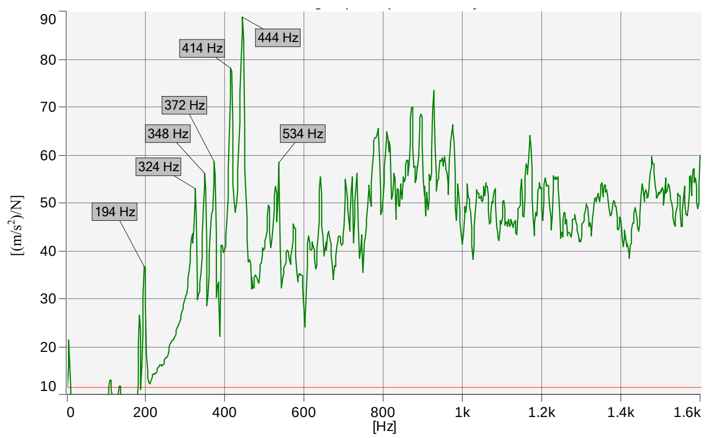
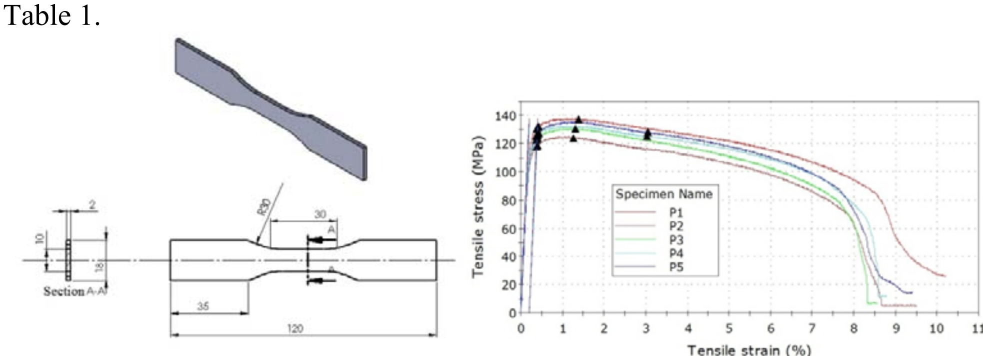


Fig. 5 The first five natural frequencies experimentally identified on Face 3

The values of the experimentally obtained eigenfrequencies (natural frequencies) were compared with those that resulted from a numerical analysis using the finite element method. In order to correctly model the analyzed structure, the real mechanical characteristics and elastic constants of the material were determined. For this, five standardized specimens SR EN ISO 6892-1:2010 (Fig. 6, a) were manufactured from the aluminum alloy used for the studied box and were subjected to a tensile test using an INSTRON 8801 universal testing machine. The obtained engineering stress-strain curves are depicted in Fig. 6, b. From these curves, the Young's modulus E, the 0.2% offset yield strength σ_y and ultimate tensile strength σ_u were determined. Experimental determination of Poisson's ratio was done using the strain gauge technique, with a set of five test specimens for each determination. Two strain gauges were glued on each test specimen, one on the longitudinal direction (the direction of the force) and the other on the transversal direction. All averaged experimental results are listed in Table 1.



a) b)

Fig. 6. Aluminum specimens subjected to tensile test and resulted engineering stress-strain curves

Table 1

The average elastic constants and mechanical characteristics of the material

E [MPa]	ν	σ_y [MPa]	σ_u [MPa]
67956.75	**0.32**	**125.64**	**128.21**

It should be mentioned that in this research the electronic components inside the box were not taken into account. During the experimental determinations, it was seen that the bolts used to fix the box on the foundation do not ensure a perfectly rigid fixation. That is why in the numerical analysis the structure was considered as leaning on elastic supports during excitation. Also, the box components were considered as riveted between each other, with a coefficient of friction aluminum-aluminum of 1.05 [14], to properly model the real structure. Preliminary numerical analyses conducted by the authors show that the friction coefficient plays an important role in the vibration behavior: if friction was not considered, other values of the eigenfrequencies than those presented here have been obtained.

For the analyzed model the number of nodes was 159018 and the number elements was 107333. The used element type was SOLID 187 [15].

Following the finite element analyses, the results for the first eigenfrequencies are shown in Fig. 7. Only the first elastic eigenfrequency is of interest for the proper functioning of the satellite at launching [11]. For a better comparison between the experimental and numerical results, the first elastic four eigenfrequencies were compared.

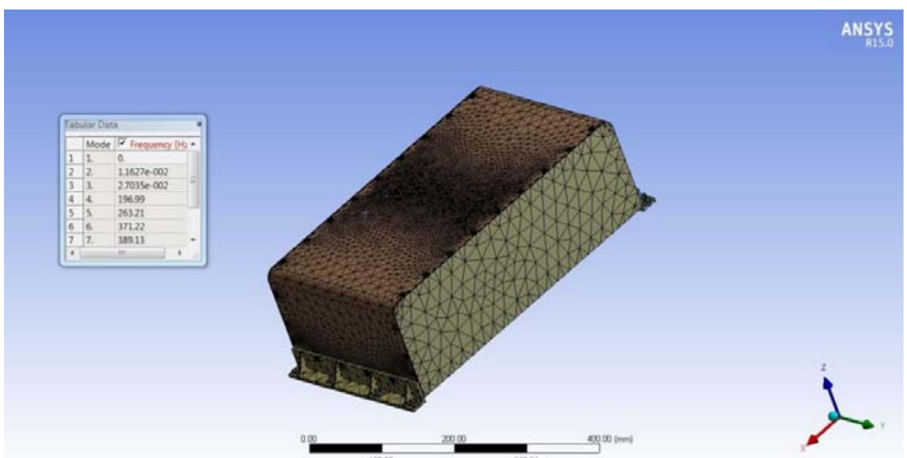


Fig. 7. The mesh and the first four natural frequencies of the subassembly

Since elastic supports were considered, the first three null vibration modes corresponding to the rigid body motion were neglected. Thus, the count of the natural frequency starts with the first non-zero value (which is actually the fourth one) whose deformed shape is shown in Fig. 8. The first elastic eigenmode appears on all sides as seen also in Fig. 8. The same conclusion was revealed in the experimental analysis.

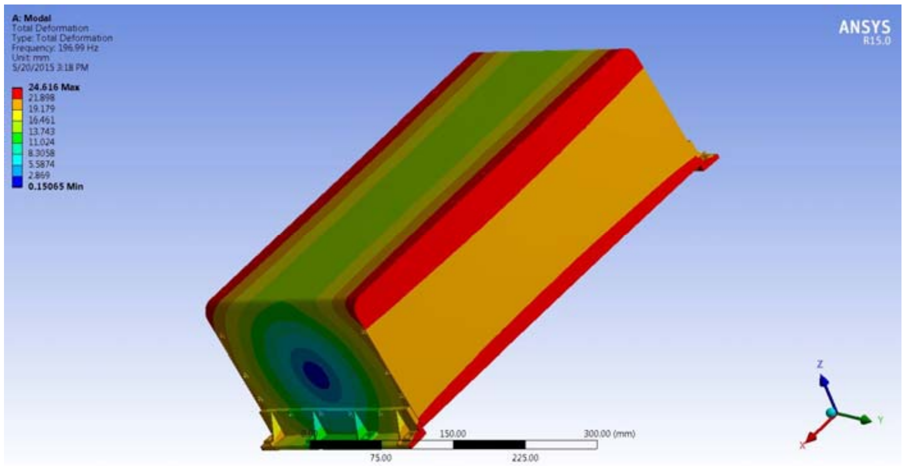


Fig. 8. The first elastic natural mode of the aluminum box

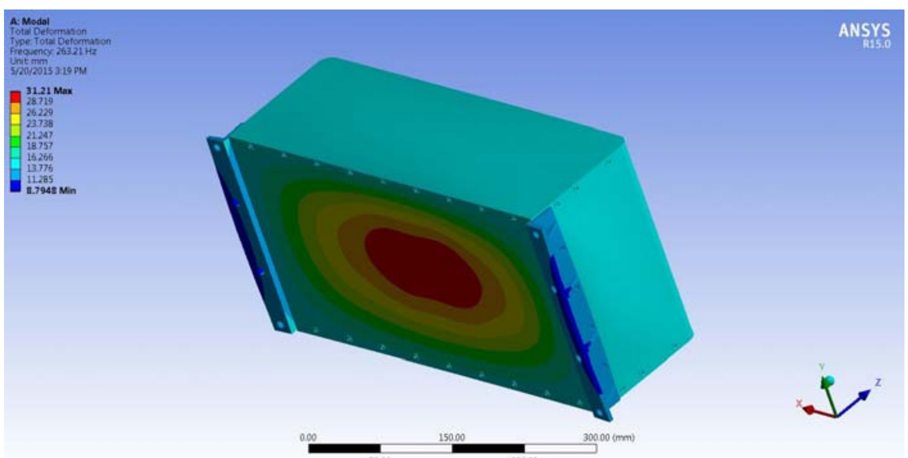


Fig. 9 The second elastic natural mode of the aluminum box

The second eigenmode found in the numerical analyses is a local one (Fig. 9), since it appears only on the lower face of the subassembly, a face on which an accelerometer was not mounted. That is why this value of the natural frequency could not be obtained experimentally. However, this mode is scarcely transmitted on top of the box, as it was emphasized in the small amplitude peak (between first and third eigenfrequency) which appears on the superior face F1 (Fig 3).

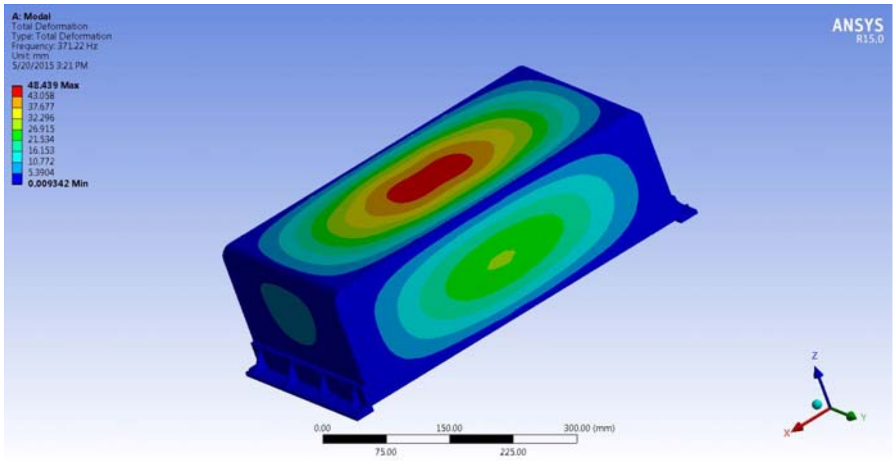


Fig. 10 The third elastic natural mode of the aluminum box

The third natural frequency of the subassembly can be found particularly on the upper face of the subassembly (Fig. 11), and less on F2; on F3 it is almost no detectable. Therefore, it appeared more often in experimental results on the faces F1 and F2 and only once on face F3.

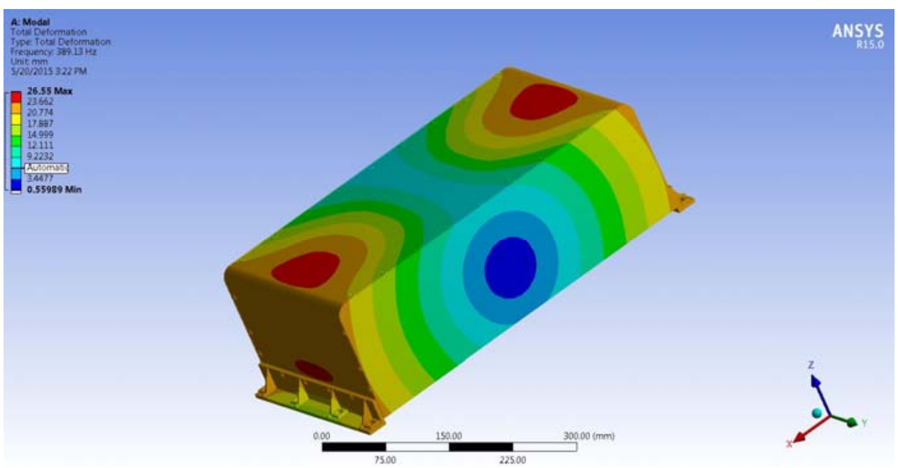


Fig. 11 The fourth elastic natural mode of the aluminum box

The fourth vibration mode is present on all sides of the box, having a maximum amplitude near the smaller sides of the face F1, as it can be seen in Fig. 11.

Following the numerical analyses, the natural frequencies were obtained and compared to the experimental ones. The comparative results are listed in Table 2. A very good agreement can be noticed, except for the second non-zero natural frequency where no experimental results are available due to the reasons explained above.

Table 2

Experimental and numerical eigenfrequencies of the studied structure

Mode	Experimental frequency [Hz]	Numerical frequency [Hz]	Relative error [%]
1	194	196.99	1.54
2	-	263.21	-
3	372	371.22	0.20
4	414	389.13	6.00

3. Conclusions

This paper presents a comparative analysis regarding the dynamic response of a subassembly from a LEO satellite. Both experimental determinations and finite element analyses were undertaken in order to assess the vibration behavior of the subassembly.

From the obtained results, one can conclude that there is a very good correlation between the experimental and numerical values of the natural frequencies. The differences do not exceed 6 % for any of the first four non-null natural frequencies. Regarding the first natural frequency, which is the one of greatest interest, the difference is only 1.54%.

Taking into account the obtained comparative results, one can conclude that the finite element model accurately predicts the vibration behavior from the point of view of the eigenfrequency of the subassembly. Thus, a similar model can be used to further investigate other aspects regarding the mechanical behavior of the subassembly as the behavior of the subassembly subjected to thermal cycles occurring on orbit or to gravity. Due to the different nature of the thermal analysis, a supplementary validation may be needed for a similar model.

Acknowledgement

The work has been funded by the Sectoral Operational Programme Human Resources Development 2007-2013 of the Ministry of European Funds through the Financial Agreement POSDRU/159/1.5/S/134398

R E F E R E N C E S

[1] *H. Khan, A. Israr*, Design, Modeling and Analysis of Low Earth Orbit Satellite, in The Second TSME International Conference on Mechanical Engineering, 19-21 October, 2011, Krabi, Thailand, Paper nr. AMM39

[2] *H.K. Cho, J. Rhee*, Vibration in a satellite structure with a laminate composite hybrid sandwich panel, in Composite Structures, **vol. 93**, 2011, pp. 2566–2574.

[3] *I. Pérez-Grande, A. Sanz-Andrés, C. Guerra, G. Alonso*, Analytical study of the thermal behaviour and stability of a small satellite, in Applied Thermal Engineering, **vol. 29**, 2009,pp. 2567–2573
[4] *M. Moshrefi-Torbati, A.J. Keane, S.J. Elliott, M.J. Brennan, D.K. Anthony, E. Rogers*, Active vibration control (AVC) of a satellite boom structure using optimally positioned stacked piezoelectric actuators, in Journal of Sound and Vibration, **vol. 292**, 2006, pp. 203–220.
[5] *A. Israr*, Vibration and Modal Analysis of Low Earth Orbit Satellite, in Shock and Vibration, vol. 2014, 2014, Article ID 740102,://dx.doi.org/10.1155/2014/740102.
[6] *M. Oda, A. Honda, S. Suzuki, Y. Hagiwara*, Vibration of Satellite Solar Array Paddle Caused by Thermal Shock When a Satellite Goes Through the Eclipse, in "Advances in Vibration Engineering and Structural Dynamics", Chapter 13, Francisco Beltran-Carbajal, Ed., ISBN 978-953-51-0845-0, 2012.
[7] *C. Paris,* Vibration tests on the preloaded LARES satellite and separation system, in Aerospace Science and Technology, vol. 42, 2015, pp. 470–476.
[8] *K. Gantois, F. Teston, O. Montenbruck, P. Vuilleumier, P. V. D. Braembussche, M. Markgraf, "*PROBA-2 - Mission and New Technologies Overview, in "Small Satellite Systems and Services", The 4S Symposium, Chia Laguna, Sardinia, Italy, Sept. 25-29, 2006.
[9] http://www.esa.int/Our_Activities/Space_Engineering_Technology/Proba_Missions/About_Proba-2
[10] *J. J. Eric, R.J. Hull, T.R. Anderl,* Laser hardened materials evaluation laboratory simulates space environment for advanced materials, space system testing, "Materials in Space: How Air Force research is Maintaining America's Advantage in Space, AMPTIAC Quarterly, **vol. 8**, nr. 1, 2004, pp. 31-34.
[11] *S. Drăghici, F. Baciu, R Voicu, A. Hadăr,* "Modeling the behavior at launching for a satellite's subassembly", Conference Proceedings of the Academy of Romanian Scientists PRODUCTICA Scientific Session **6**, No 1.
[12] *S. Drăghici, H.A. Petrescu, R Voicu, A. Hadăr, "*Behavior analysis for a subassembly in a LEO satellite" U.P.B. Sci. Bull., Series D – Mechanical Engineering (accepted for publication)
[13] http://www.bksv.com/Products/pulse-analyzer
[14] http://www.roymech.co.uk/Useful_Tables/Tribology/co_of_frict.htm
[15] * * *, ANSYS Mechanical ADPL Theory Reference, ANSYS, Inc., Canonsburg, PA, 15317, USA, 2013.