The numerical simulation of the heterostructures behaviour from dielectric materials and superhard at launch in space

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Abstract. During the space missions, the resistance structure of the launch vehicle as well as the transported space objects, dynamic loads are produced which have as their source the operation of the thrusters, the shocks generated by the switching moments of the regime and the vibrations. The intensity of dynamic stresses is appreciated through the magnitude of the accelerations. The accelerations of the structural elements occur in the launch phase of the space object. Mechanical shocks are short-term actions with rapid installation, accompanied by high-frequency vibrations and high amplitudes. In the paper, the heterostructure subjected to the analysis has the rectangular plate shape on which conductive layers and protective layers are deposited. On the modelled heterostructure with finite elements were analysed the behaviour in the shock regime, the spectral response and the dynamic response to the harmonic excitation in the base with acceleration intensity of 20 g. The objective of the numerical simulations followed the conditions of strength and rigidity and the optimization of the fastening mode of the heterostructure dynamically loaded. The guarantee of the accuracy of the numerically simulated solutions is supported in the paper with a convergence test. The results obtained during the work are very useful in the conceiving and design phase of the antennas with low PIM from dielectric and super hard materials under the conditions of the mechanical loading regime during launching in space.

1. Introduction

During the launch and then during the flight, when the rocket launching stages are separated and the spatial objects placed, different loads (deterministic, random) are developed in the resistance structure of the launch vehicle as well as in the spatial objects. Thus, the accelerations produced by vibration are the main source of loads generation and the mechanical shocks are short-term actions with fast installation, with spectrum of high-frequency and with large amplitudes of the accelerations. In the paper, the structure under analysis has the shape of a rectangular plate on which layers of conduction and protective layers are laid. The capacity of the spatial object components to properly function according to the launching and separation regime is checked by tests contained in the norms in force, which are detailed in the present paper. Related study of the numerical simulation of the heterostructure behaviour of dielectric and super hard materials, launch, separation and dispersion, is done on a small model, preceding a definitive solution, which will be defined in a future paper. The
The main objective of the paper is to establish and test a methodology for simulating the dynamic behaviour of some components (spatial objects) installed on space vehicles in order to conceiving and design them in conditions that ensure their integrity and functionality throughout the mission. Methodology of numerical simulation has been developed based on some preliminary assessments looking establishing the dynamic regimes and the loads produced in the structure components of the spatial vehicle during launching and on the basis of the vibrations particularities of the structures excited in the base by accelerations. Analytical methods of the dynamics of deformable solids provide useful results in a small number of applications, on simple models. In practice, the shape of the bodies, the complexity, the various conditions of the functioning regimes make analytical approaches unnecessary. Calling on numerical calculus is the solution to solving the most complex problems in field of physics. The numerical simulation of phenomena and the physical processes based on the finite element method has now become one of the most used research and design tools. In summary, the main steps which are going through for achieving the numerical simulation of a process are as follows:

- conceiving the physical model;
- realizing the discretized model;
- transposing the properties of the physical model and the operating conditions on the discretized model;
- establishing the type and conditions of analysis;
- uploading data to the computer and solving the problem numerically;
- post processing of the data and extraction of interest information from the solution database;
- presenting and the analyse of the solution.

The methodology of numerical simulation of heterostructure dynamics was applied to a physical model of the object represented in a reduced configuration. In the future, following the conclusions reached on the reduced configuration, the complexity of the physical model will be increased to the final, functional form.

2. The physical model discretized with finite elements

Finite element discretization introduces a certain degree of approximation. The more elements there are, the more accuracy precision increases (figure 1).

![Figure 1. Physical model of discretized multi-layered antenna with finite elements](image)

To ensure the required precision, numerical simulations of the multilayer antenna dynamics with low PIM are made on 3D models (figure 1) with hexahedral elements (bricks) with 8 nodes.

The accuracy of the meshing density solutions was checked in the convergence test on a simplified model (figure 2), by which it was validated the accepted configuration, according to table no. 1.
Figure 2. The meshing pattern of the support plate in the convergence test.

The meshing pattern accepted

Table 1. The accuracy of solutions based on mesh density.

| Mesh        | Maximum displacement [mm] | Maximum stress [N/mm$^2$] | Error [%] |
|-------------|---------------------------|---------------------------|-----------|
| 20x16x2     | 4.223                     | 381.4                     |           |
| 40x32x4     | 4.227                     | 389.0                     | 1.91      |
| 60x48x6     | 4.228                     | 391.7                     | 0.69      |
| 80x64x8     | 4.228                     | 393.0                     | 0.33      |

In the convergence test, the structure response to a static stress was followed, considering rigidity (displacements) and strength (maximum stress). Four meshes were tested, according to the table no. 1.

As an exact solution to the displacements in the problem, a combination of the bar type solution with one obtained on the cylindrical flexural model was chosen. From this calculation the arrow $f = 4.23\,\text{mm}$ resulted. Analyzing the convergence, especially on the column of the maximum tension (table no. 1), it is found that a justified mesh is the 60x48x6 hexahedral mesh which gives a relative error below 1%. The accepted mesh was transferred on the structure under analysis (multilayer antenna) figure 1. The basic mechanical properties of the hard materials are shown in figure 1.

The discretized model with the accepted mesh was subjected to a dynamic validation test, which consisted of comparing its own frequencies, on the structure (multilayered antenna) fixed in the three modes (supported on the points, simply supported and clamping), numerically simulated and analytically calculated. The results of the test are presented in Table no. 2, where it is found that the error between the results obtained by the two methods is less than 1%. In conclusion, it is expected that the model will also meet the accuracy requirements in dynamic issues where solutions are built by overlapping their own vibration modes. In the paper only the fastening mode of the multilayer antenna is presented - simply supported.

Table 2. Comparative Table of Own Frequencies.

| No. crt. | Simulated frequencies [Hz] | Calculated frequencies [Hz] | Error [%] |
|----------|-----------------------------|-----------------------------|-----------|
| 1        | 1700                        | 1704                        | 0.24      |
| 2        | 3691                        | 3699                        | 0.21      |
| 3        | 4815                        | 4821                        | 0.13      |
| 4        | 6792                        | 6816                        | 0.35      |
3. Conditions for the simulated numerical tests

The loads generated in the spatial objects during launch were specified in paragraph no. 1. The intensity of these requests depends on the transport vehicle, the flight program and its operating status, as well as the conditions for the installation in orbit. Loads in the spatial object components also occur after the dispersed, as a result of the deployment of photovoltaic panels and antennas.

For a mechanical or electronic component incorporated in the carrier vector or spatial object to be able to run the launch cycle, a series of tests must be passed, to meet the flight conditions. In this regard, the tests are tougher than the loads that occur at the launch. This is an additional reason for accepting testing conditions instead of launch conditions.

In the paper the numerical simulations for the following tests required by the norms in force were retained so:
- Vibration test according to MIL STD 750, TM 2056, Test A (20g, 10 - 2000 Hz);
- Mechanical shock test according to MIL STD 750 TM 2016, Test B ((in planes X1, Y1, Z1), 1500g, duration of 0.5 ms));

The numerical simulations in these tests were done on the complete model and watched, in addition to the dynamic response, in the case of vibrations and shocks, the stress state made in the antenna, in the peak loads moments. The three fastening modes set in the physical model, were preserved. In the simulations of the first two dynamic tests presented above, the principle of modal overlap was used, taking into account the first 12 own frequencies. The quality factor was established to the agreed value Q = 10.

4. Simulation of antenna behaviour at harmonic excitation in base with 20g amplitude

4.1. The modal analysis of the multilayer antenna

Each fixation mode was analysed separately, totally, being realised three numerical simulations on the base structure. For each fastening mode (supported on points, simply supported and clamping), for representation the first 8 modes of vibration were retained, which are transverse modes with normal displacements on the median plane of the plate.

In the paper is presented the modal analysis for simply supported mode of the analysed structure (figure 3). In this figure, there are presented their own shapes, at arbitrary scales that highlight the transverse displacements of the plate. In the central area, the fixing mode is schematized and the control modes required to continue the analysis are identified.

When analysing the representations for the three fastening modes, it is found that there are symmetrical vibration modes and antisymmetric modes that will respond dynamically differently depending on the type of excitation. As the fastening modes become more restrictive, their own frequencies increase and their own shapes become more regular.

When we're going through the dynamic analysis algorithm of a mechanical structure, the first step consists of modal analysis, by which determines a number of own modes that will later be the basis of the calculation principle of the overlapping effects. The more you many own modes are taken into account; the answer will be more complete.
4.2. Harmonic analysis of the multilayer antenna

Numerically simulated harmonic analysis followed the dynamic response of the structure, excited in the base, harmonic, in the frequency band 100 Hz ... 10 kHz. The excitation was a harmonic, transversal acceleration, distributed over the contour. The direction of the transverse excitation is consistent with the natural state of vibration of the structure found during the modal analysis. This type of uniform excitation on outline will only stimulate symmetric modes. In a future paper, the structure (multilayered antenna) on which the electronic components will be mounted, the situation will change, being necessary to investigate the dynamic properties on the three directions of the space.

The dynamic response of the structure to the base excitation under the set conditions is given by the resonance curves in the frequency band in which the numerical simulation was performed. As has been shown, because acceleration plays an important role in dynamic problems, it is recommended that the parameter represented in the resonance curve be the acceleration. The resonance curves were plotted in the control modes for the three chosen fixing modes. In the paper the resonance diagram for the simply supported base structure is presented (figure 4). The size represented is amplification (the amplification factor). A special note is required about the sensitivity of the one's own modes to uniform excitation. It is found that the fundamental mode is the most sensitive (amplification about 16.5 times). Antisymmetric modes do not react and the other upper symmetries are heavily damped.

4.3. The quality factor

All simulations for harmonic analysis were made under the condition of the prescribed quality factor, $Q = 10$. This factor has been fundamentally controlled and has been achieved by adjusting of the damping coefficients, considered proportional to rigidity.
Figure 4. Resonance chart for base structure (multilayered antenna) with simply supported. Uniform excitation in the base with normal harmonic acceleration on the median plane.

For the quality factor the definition that was used:

\[ Q = \frac{B}{f} \]  

where \( B \) is the bandwidth at power level 1/2 and \( f \) is the fundamental frequency.

In figure 5, the fundamental responses for the three fastening modes are reunited, with the procedure of calculating of the quality factor.

Figure 5. Conditioning the quality factor for fundamental modes.
The small differences that occur between the calculated values and the imposed value are of no practical significance, the error being less than 1%. These small differences come from the fact that the bandwidth resolution was reduced to one unit.

4.4. Dynamic response in stresses to harmonic excitation in base with the 20g acceleration

For the current configuration of the multilayer antenna, in order to capture the fundamental modes for the three fixing variants in the frequency band, an extension of up to 4 kHz was made. Excitation was made harmonic, based on the frequency of 100 Hz, with an acceleration whose amplitude was 20g. In the numerical simulation, the principle of modal overlap was applied using the first 12 vibration modes.

The results of the numerical calculation for the three fastening modes are displayed on joint graphical representations. Acceleration amplification, given by the resonance curves, is shown in figure 6, which in fact reproduces with small differences, the simulated situation for the base structure (figure 5), amplified 20 times.

![Resonance Diagrams](image)

**Figure 6.** The resonance diagrams at harmonic excitation in base with the 20g acceleration.

At resonance, peaks of the amplitudes of acceleration response are reached, depending on the fixing mode, between 300g and 350g. In the paper, only the simply supported structure (multilayer antenna) is presented. The moderate dynamic regime corresponding to this fastening mode produces stress states that do not endanger the strength capacity of the structure (figure 7). To ease the interpretation of the resistance condition, on figure 7 lists the strength characteristics of the materials. It is expected that for the antenna in the complete configuration the stress state will be change. For this reason, in a future paper, the numerically simulated vibration test must be retained as a main resistance criterion.

5. Numerical simulation of multilayer antenna behaviour at excitation in shock regime

In MIL STD 750 TM 2016, Test B, the testing of dynamic response is done by excitation in the base with a shock whose time function is in the form of a trapezoid (figure 8). The shock in base has the nature of an acceleration. The shock installation ramp is 0.2 ms. Duration of shock at maximum magnitude (1500g) is 0.5 ms, followed by reduction phase to zero of 0.2 ms.
The dynamic at shock response was tracked until the cancellation of the vibrations (max 16 ms) and is graphically represented in figure 9 for all three fixing modes. Analysing the figure 9, it can be noted the entrance of the plate of antenna in vibrating once time with the shock application. At the beginning there is a transient regime, differentiated according to the fastening mode, after which the vibrations are regularized, becoming damped vibrations, according to the quality factor $Q = 10$. The frequencies with which the vibrations are produced correspond, for each supporting mode, to the fundamental mode. This conclusion is better supported by comparing the vibration periods measured on the graphs in figure 9 with those established at the properties analysis. Comparison is made in table no. 3.

**Figure 7.** The dynamic response to the harmonic excitation in base with 20g acceleration. The maximum equivalent stress at the resonance for base structure with simply supported.

**Figure 8.** Excitation in the basis with trapezoid shock.

**Figure 9.** The dynamic response to shock.
Table 3. Comparison of the measured vibration periods.

|                     | Period [ms] | At the shock * | Modal analysis |
|---------------------|-------------|----------------|----------------|
| Supported on points | 0.85        | 0.848          |                |
| Simply supported    | 0.60        | 0.602          |                |
| Clamping            | 0.31        | 0.302          |                |

* Measured approximately on the graph from figure 9

The transient regime, which corresponds to the duration of the shock application, is very different. On the supported plate on points this lasts about half a period. For the other supporting modes, the alternations are produced, with a lower frequency on the simply supported plate and with a higher frequency at the clamped plate. Amplification is much less than that produced at the resonance. Thus at the shock, amplifications about 2 times are realised, while at the resonance (figure 5) the amplification factor takes values between 15 and 17.5.

The vibrations outlined in figure 9 further emphasize the effect of damping and its correlation with frequency. At higher frequencies, damping is faster due to its speed dependence, which in turn is proportional to the frequency.

In addition to the dynamic aspect of the movement, the numerical simulation of the shock aimed at addressing the internal mechanical state of the structure.

In the paper the results of the numerical simulations at shock for the simply supported structure are presented (figure 10). There are 4 representations on the figure:
- the deformed state with the mesh of the elements;
- transversal displacement field;
- the acceleration field;
- the field of equivalent stress Von Mises.

From the comparative analysis of the results obtained in the numerical simulations to the shock (for the three fastening modes) it was found that the mode of supporting on the points for the studied structure is not able to satisfy the conditions of stiffness and strength and the case of clamping is ideal. Since the realization of a "Perfect" clamping is very expensive, it is made with large masses and practically the theoretical clamping function is partially realized. Without losing too much from the safety, the use of simply supported plates may be the optimal solution for satisfying functional conditions at the shock, being simpler to constructive and less restrictive.

In figure 10, the equivalent stress field is more homogeneous and the resistance condition is satisfied with different safety factors depending on the material.

From the detail extracted of the equivalent stress field, it is found that the top layers of protection are the most strongly required, here at tensile, and in the opposite phase to the compression, with a stress of 150 N/mm². For the materials used in the protection layers, the tensile force safety coefficients in the centre of the plate are:
- for α-Al2O3 – 1.67;
- for ZrO2 – 1.65;
- for SiC – 2.06.

The fastening types for the multilayer antenna with low PIM were determined by the fact that both sides of there to be free, for functional reasons (circuits, electronic components, etc.). If it would give up to this condition, it would be possible a fastening of the antenna more securely by attaching (bonding) it to a wall of the satellite. In this configuration, the dynamic behaviour of the antenna would be different, with less mechanical effects than those set for the fastening with free sides.
6. Conclusion
The main objective of the paper is to establish and test a methodology for simulating the dynamic behaviour of components (spatial objects) installed on space vehicles in order to conceive and design them in conditions that ensure their integrity and functionality throughout the mission.

The loads produced during launch are deterministic, defined as time functions, and random loads that are defined statistically. These loads are static and dynamic.

In practice, the loads are determined and analysed using the measurement technique in flight and tests on models. Within this project, tests on models are considered because the research are at an early stage, and secondly, there are no in-flight testing possibilities.

The most important static and dynamic loads that acting at the launch are: dynamic high frequency loads (10 to 2000 Hz) and the mechanical shocks.

Accelerations are the motion parameters who interfere in generating the loads to which the structures in transport compartment of the launch vehicle are subjected.

Vibrations are the main source of load generation in the structures of transported space objects, which are on the one hand sources of excitation and on the other hand they enrol of the dynamic response of the structure.

Mechanical shocks are short-duration transient actions with fast installation, with high frequency spectrum and high amplitudes of the accelerations.

The capacity of the spatial object components to properly function to the loading regime at the launching and separation is checked by vibration and shock tests. Components that have successfully passed the test program are valid for the mission.

In the paper it was analysed the behaviour of the spatial object subjected to research defined by the standards in force.

The structure subjected to the analysis has a rectangular plate shape on which conductive layers and protective layers are deposited.

Modelling and the discretization of the multilayer antenna with finite elements introduces a certain degree of approximation. The more elements there are, the more accuracy precision increases (figure 1).

The accuracy of the meshing density solutions was checked in the convergence test on a simplified model (figure 2), by which it was validated the accepted configuration. In the convergence test, the
structure response to a static stress was followed, considering rigidity (displacements) and strength (maximum stress). Four meshes were tested, according to the table no. 1. Analyzing the convergence, it is found that a justified mesh is the 60x48x6 hexahedral mesh which gives a relative error below 1%. The accepted mesh was transferred on the structure under analysis (multilayer antenna) figure 1.

The discretized model with the accepted mesh was subjected to a dynamic validation test, which consisted of comparing its own frequencies, on the structure (multilayered antenna) fixed in the three modes (supported on the points, simply supported and clamping), numerically simulated and analytically calculated. The results of the test are presented in Table no. 2, where it is found that the error between the results obtained by the two methods is less than 1%. In conclusion, it is expected that the model will also meet the accuracy requirements in dynamic issues where solutions are built by overlapping their own vibration modes. In the paper only the fastening mode of the multilayer antenna is presented - simply supported.

That a mechanical or electronic component incorporated in its carrier vector or spatial object to be able to work the launch cycle, a series of tests must be passed, in which conditions are more severe than flight conditions. This is an additional reason for accepting testing conditions instead of launch conditions (vibrations test according to MIL STD 750, TM 2056 Test A and shocks test according to MIL STD 750 TM 2016, Test B, see paragraph 3).

When we're going through the dynamic analysis algorithm of a mechanical structure, the first step consists of modal analysis, by which determines a number of own modes that will later be the basis of the calculation principle of the overlapping effects. The more you many own modes are taken into account, the answer will be more complete. In the modal analysis three ways of fastening for the multilayer antenna were studied separately. For each fastening mode, the first 12 vibration modes were retained. In the paper the modal analysis for the simply supported analysed structure is presented (figure 3). On the figure, shows its own shapes at arbitrary scales which highlight the transverse movements of the plate. It is found that there are symmetrical vibration modes and antisymmetric modes that will respond dynamically differently depending on the type of excitation.

Numerically simulated harmonic analysis followed the dynamic response of the structure, excited in the base, harmonic, in the frequency band 100 Hz ... 10 kHz. The excitation was a harmonic, transversal acceleration, distributed over the contour. The dynamic response of the structure to the base excitation under the set conditions is given by the resonance curves in the frequency band in which the numerical simulation was performed (for the three chosen fastening modes). The paper presents the resonance diagram for the simply supported base structure (figure 4). It is observed that the fundamental mode is the most sensitive (amplification about 16.5 times). Antisymmetric modes do not react and the other superior symmetrical ones are heavily damped.

All numerical simulations for harmonic analysis were made under the conditions of the quality factor adjusted to the prescribed value $Q = 10$. In figure 5, the fundamental responses for the three fastening modes are reunited, with the procedure of calculating of the quality factor.

For the current configuration of the multilayer antenna, in order to capture the fundamental modes for the three fixing variants in the frequency band, an extension of up to 4 kHz was made. Excitation was made harmonic, based on the frequency of 100 Hz, with an acceleration whose amplitude was 20g. In the numerical simulation, the principle of modal overlap was applied using the first 12 vibration modes. The results of the numerical calculation for the three fastening modes are displayed on joint graphical representations. At resonance, peaks of the amplitudes of acceleration response are reached, depending on the fixing mode, between 300g and 350g. In the paper, only the simply supported structure (multilayer antenna) is presented (figure 7).

In MIL STD 750 TM 2016, Test B, the testing of dynamic response is done by excitation in the base with a shock whose time function is in the form of a trapezoid (figure 8).

The dynamic at shock response was tracked until the cancellation of the vibrations (max 16 ms) and is graphically represented in figure 9 for all three fixing modes. The figure shows the multilayer antenna input in vibration when applying the 1500g shock. At the beginning there is a transient regime, differentiated according to the fastening mode, after which the vibrations are regularized,
becoming damped vibrations. The frequencies with which the vibrations are produced correspond, for each supporting mode, to the fundamental mode. This conclusion is better supported by comparing the vibration periods measured on the graphs in figure 9 with those established at the properties analysis. Comparison is made in table no. 3.

Finally, the results of the numerical simulations at shock for the simply supported base structure are presented (Figure 10). From the comparative analysis of the results obtained in the numerical simulations to the shock (for the three fastening modes) it was found that the fixed on the points mode of the studied structure is not able to satisfy the conditions of stiffness and strength and the clamping case is ideal. Realising a "perfect" clamping is, however, very costly. Without losing too much from the safety, the use of simply supported plates may be the optimal solution for satisfying functional shock conditions, being simpler constructive and less restrictive.

The fastening types for the multilayer antenna with low PIM were determined by the fact that both sides of there to be free, for functional reasons (circuits, electronic components, etc.). If it would give up to this condition, it would be possible a fastening of the antenna more securely by attaching (bonding) it to a wall of the satellite. In this configuration, the dynamic behaviour of the antenna would be different, with less mechanical effects than those set for the fastening with free sides.

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