FINITE ELEMENT MODAL SYNTHESIS OF A SATELLITE STRUCTURE COMPOSED BY SANDWICH PLATES

Mônica Souza Martins, monicasm@ita.br
Airton Nabarrete, nabarret@ita.br
Ednardo Dantas de Oliveira, ednardodantas@gmail.com
Instituto Tecnológico de Aeronáutica. Praça Marechal Eduardo Gomes,50,SJC-SP, Brasil

Abstract. *This work presents the finite element analysis of a satellite structure which was designed using sandwich plates made of aluminum fa sheets and a foam core. A three layer finite element model was created for the sandwich plates including solid elements applied to the core and plate elements applied to the faces. The structure assembly was analyzed in two models. One of them considers the plates are joined by adhesives and other uses bolt joints. Displacement substructuring theory was applied to permit the static analysis of a small group of plates. After the analysis, this group was treated as a superelement that was assembled in the satellite model with other superelements. The method modal synthesis of components was used for extracting the natural frequencies of the whole satellite structure. For both analyses, static and dynamic, the results obtained with these substructuring methods were very close in comparison to the analysis for the whole structure. It was observed that the use of undercarriages presented several advantages in the analysis of plate joints as for instance the adhesive joint between two sandwich plates.*

Keywords*: substructuring, superelement, sandwich plate, modal analysis, satellite*

1. INTRODUCTION

The Itasat Satellite is a development program for designing and construction of a operational satellite that started in 2005 as a joint venture between “Instituto Tecnológico da Aeronáutica - ITA” and “Instituto Nacional de Pesquisas Espaciais – INPE” supported by “Agencia Espacial Brasileira – AEB”. Itasat was previously dimensioned to have a square horizontal surface with an edge length of 700 mm and 650 mm as height. Its structure is composed by internal walls that are used to set up the several subsystems needed during the mission (Fig. 1).

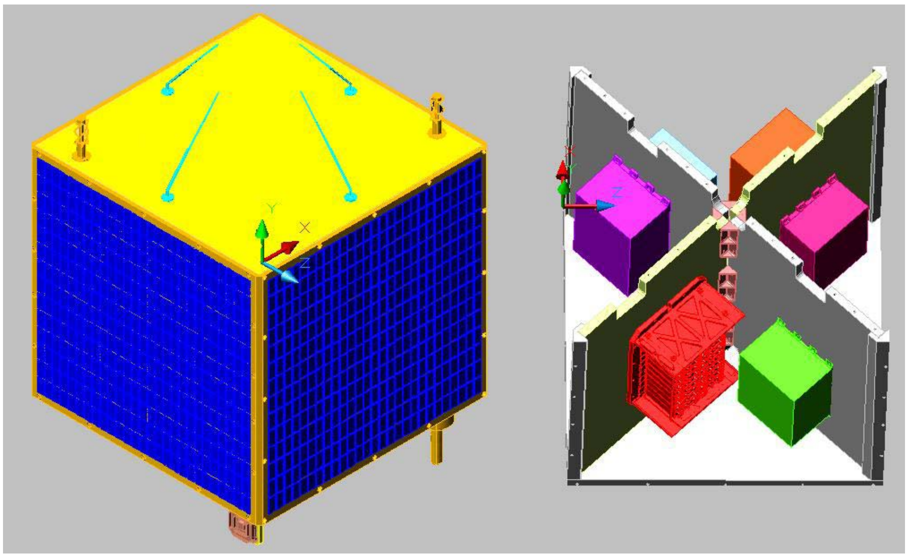

Figure 1. Itasat structural model: complete satellite view and disposal of panels and equipment

The panels that compose the base and top satellite surfaces as well as the internal walls were proposed as sandwich plates (Fig. 2). The dimensioning of these plates considers mainly dynamic loads during the satellite launching to the final orbit height. The procedures described in this paper show the use of modal analyses applied to substructures composed by sandwich plates.

The first works describing the sandwich plate behavior was made by Reissner (1948) and Mindlin (1951). Their works are named as first order theory. They assumed linear distribution for displacements in x and y directions (Fig. 2). Higher order theories have been proposed for sandwich structures. Reddy (1984) presented a high order theory to model the through-thickness shear deformation. Oskooei and Hansen (2000) developed a finite element where the sandwich stiffness was formulated by a high order theory. Based on this work, Nabarrete (2002) developed the mass and geometric stiffness formulation to improve the results for sandwich problems in dynamic and buckling analyses. In this work the sandwich faces are modeled according to Reissner-Mindlin theory while the core is modeled as a continuous tridimensional formulation.

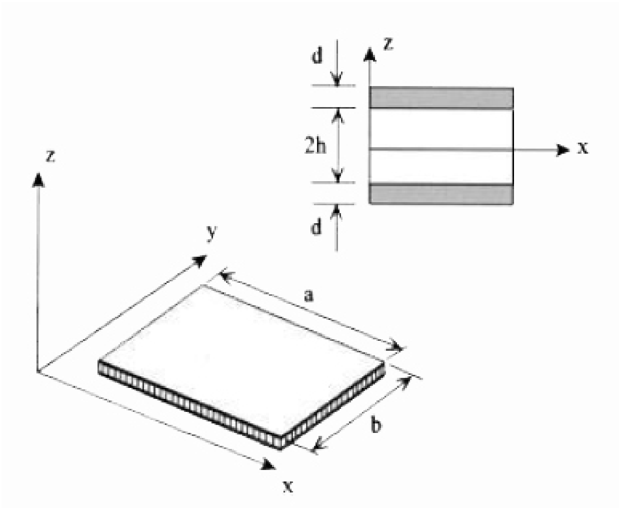

Figure 2. Sandwich plate dimensions

The finite element method was developed, allowing an accurate modeling of complex problems very close to real situations. Based on Nabarrete (2002) formulation for dynamic analysis, hybrid modeling could be adopted for sandwich plate analyses. Using a combination of linear shell elements representing the faces and linear solid elements representing the core, dynamic responses can be predicted with high precision. Although sandwich plates have high stiffness to bending moments they are susceptible to damage imposed by concentrated loads. For the same reason, joints between sandwich panels present a big challenge in design. The union of a sandwich border with the middle part of any other panel in angles like 0 to 90 degrees can be made by screws, adhesives and other elements applied to the joint. This specific problem identified as sandwich T joints was studied by Theotokoglou (1996, 1999), Turaga and Sun (2000). In these works, two failure modes were identified: (1) Delamination between the leg and base panel and; (2) shear failure of the core in base panel (Fig. 3). Zhou (2008) presented in his work results in performance for sandwich T joints submitted to static and dynamic loads. The tests results are being used to validate detailed numerical models capable of simulating the damage process.

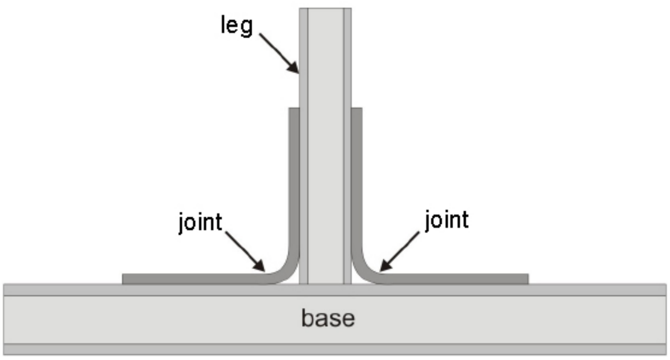

Figure 3. Sandwich T joint

Very often, structural analysts are faced with problems that require a lot of computational work and can exceed the capacity of the used systems. When these problems appear, the structure can be partitioned into several smaller pieces for analysis. The substructure models have been researched since 1960 and nowadays they are largely used in the solution of structures like airplanes, space vehicles and others. Przemieniecki (1986) and Azar (1972) treat the structural partitioning by displacement and force methods to solve static problems.

Dynamic analyses with substructuring can be evaluated by two methods: modal component synthesis (CMS) and frequency response function (FRF). According to Ewins (2000), CMS solves models created for finite element analyses or obtained by the experimental modal identification, while the FRF method is only applied to experimental models. CMS refers to any numerical process that synthesizes the normal modes of the complete structure using the modes of its individual components. This procedure was introduced by Hurty (1965) and basically represents a Rayleigh-Ritz procedure. Craig and Bampton (1968) formulated a method similar to Hurty one but used restricted and normal modes for the fixed interface. Craig and Chang (1976) showed that the difference between several modal synthesis methods is the description of the substructure generalized coordinates in different ways to force the compatibility of the substructure interfaces. Details of CMS were described by Craig (1981, 2000) and Araújo (1998). In this work, sandwich plate formulation was written under two hypotheses. The first one agrees with the first order theory of Reissner-Mindlin (1949, 1951) and it will be applied in sandwich plate faces. The second hypothesis, applied to model the sandwich core was proposed by Oskooei and Hansen (2000) and it is based in higher order theories. High order theories does not use shear correction factors because they have the transverse shear energy well estimated.

2. DYNAMIC ANALYSIS

The modal synthesis methods are based on the interface boundary conditions between substructures and the vectors used to obtain the vibration modes. There are four method variants: fixed interface methods, free interface methods, loaded interface methods and hybrid methods (Hurty, 1965 and Craig, 1981). This work considered the fixed interface method for normal modes and restriction static modes known as Craig-Bampton method (2000). The difference among several modal synthesis methods comes from different modal supergroups (Craig, 1981) in the description of the generalized coordinates and the compatibility of substructure interfaces. For any boundary conditions, the normal modes are enriched with static modes that provide the movements due to substructure joints in order to compose a group of eigenvectors that represent the synthesized movement. Figure 4 shows a beam model that is divided in components (r) and $(r+1)$. In each substructure, the coordinates can be divided in two groups: joint coordinates $\{u_j\}$ and internal coordinates $\{u_i\}$.

2.1 Generalized coordinates

In order to describe the undamped motion of a complete structural system represented in figure 4, a vector of generalized coordinates $\{\bar{q}\}$ is applied as,

$$[\mathrm{M}]\{\ddot{\bar{q}}\}+[\mathrm{K}]\{\bar{q}\}=\{\mathrm{F}\} \tag{1}$$

Each substructure has equations of motion written in terms of physical coordinates:

$$[M]^{(r)}\{\ddot{u}\}^{(r)}+[K]^{(r)}\{u\}^{(r)}=\{f\}^{(r)} \tag{2}$$

Partitioning Eq. (2) in terms of joint and internal displacement vectors:

$$\begin{bmatrix} M_{ii} & M_{ij} \\ M_{ji} & M_{jj} \end{bmatrix}^{(r)} \begin{Bmatrix} \ddot{u}_i \\ \ddot{u}_j \end{Bmatrix}^{(r)} + \begin{bmatrix} K_{ii} & K_{ij} \\ K_{ji} & K_{jj} \end{bmatrix}^{(r)} \begin{Bmatrix} u_i \\ u_j \end{Bmatrix}^{(r)} = \begin{Bmatrix} f_i \\ f_j \end{Bmatrix}^{(r)} \tag{3}$$

In modal analysis the physical coordinates $\{u\}$ of each substructure can be transformed to modal coordinates $\{p\}$ through the expression,

$$\{u\}^{(r)}=[\Psi]^{(r)}\{p\}^{(r)} \tag{4}$$

where $[\Psi]^{(r)}$ is the modal transformation for the substructure. Eq. (4) can be introduced in Eq. (2) for the substructure, as follows,

$$[\Psi]^T[M][\Psi]\{\ddot{p}\}+[\Psi]^T[K][\Psi]\{p\}=[\Psi]^T\{f\} \tag{5}$$

and for the substructure r,

$$[\mu]^{(r)}\{\ddot{p}\}^{(r)}+[\kappa]^{(r)}\{p\}^{(r)}=[\Psi]^{(r)^T}\{f\}^{(r)} \tag{6}$$

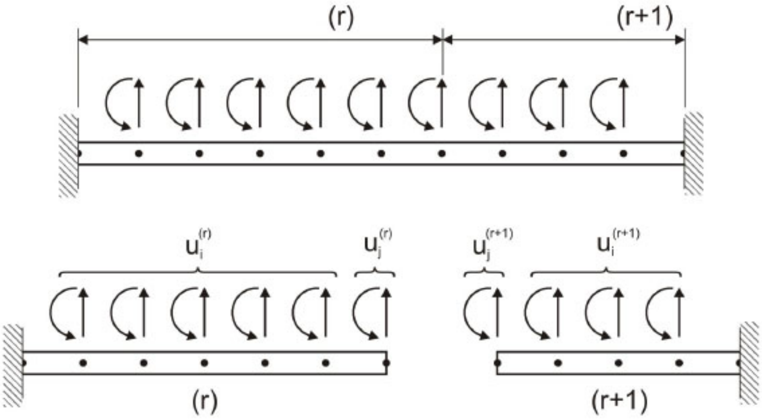

Figure 4. Substructuring: internal and joint nodes of a beam.

The matrices in Eq. (6) can be adapted to relate adjacent substructures in the form:

$$[\mu]^b \{\ddot{p}\}^b + [\kappa]^b \{p\}^b = [\Psi]^{bT} \{f\}^b \tag{7}$$

where,

$$[\mu]^b = \begin{bmatrix} [\mu]^{(r)} & 0 \\ 0 & [\mu]^{(r+1)} \end{bmatrix}, \quad [\kappa]^b = \begin{bmatrix} [\kappa]^{(r)} & 0 \\ 0 & [\kappa]^{(r+1)} \end{bmatrix}, \quad \{p\}^b = \begin{Bmatrix} \{p\}^{(r)} \\ \{p\}^{(r+1)} \end{Bmatrix}, \quad \{f\}^b = \begin{Bmatrix} \{f\}^{(r)} \\ \{f\}^{(r+1)} \end{Bmatrix} \tag{8}$$

Eq. (7) represents the equations of motion for the hole beam, or a group of two connected substructures. Boundary conditions are related for the joint degrees of freedom in the vector {p} and represent the physical process of joining. If on the interface between substructures (*r*) and (*r+1*) the displacements are the same, the first equation for boundary conditions is written as follows,

$$\{u_j\}^{(r)} = \{u_j\}^{(r+1)} \tag{9}$$

If k is a number of boundary conditions and m is the number of coordinates for $\{p\}$, there will rest a subset of linearly independent coordinates after the connection of both substructures. These generalized coordinates are represented by the vector $\{q\}$. The relation between coordinates $\{p\}$ and $\{q\}$ can be written using the $[S]$ matrix as follows:

$$\{p\} = [S]\{\overline{q}\} \tag{10}$$

The $[S]$ matrix depends on the interface relation for the coordinates.

Combining Eq. (10) and Eq. (7) and pre-multiplying all terms by the $[S]^T$ matrix it leads up to the equation:

$$[S]^T[\mu][S]\{\ddot{\overline{q}}\} + [S]^T[\kappa][S]\{q\} = [S]^T[\Psi]^T\{f\} \tag{11}$$

where the system matrixes are given by:

$$[\mathrm{M}] = [S]^T[\mu][S], \quad [\mathrm{K}] = [S]^T[\kappa][S], \quad \{\mathrm{F}\} = [S]^T[\Psi]^T\{f\} \tag{12}$$

The equilibrium on the interface is written by:

$$\{f_j\}^{(r)} + \{f_j\}^{(r+1)} = \{0\} \tag{13}$$

2.2 Craig-Bampton method

The Craig-Bampton method (CB) uses normal modes with fixed interfaces combined with restriction modes it is one of the most precise methods as CMS (Craig, 1981). In the above formulation for the beam, the normal modes with fixed interface are obtained by solving the eigenvalue problem for internal coordinates in Eq. (3).

$$\left(\left[K_{ii}\right]-\lambda_{fi}\left[M_{ii}\right]\right)\left\{\phi_{fi}\right\}=\{0\} \tag{14}$$

where λ_{fi} is the eigenvalue and $\{\phi_{fi}\}$is the eigenvector of the substructure with fixed interface. The eigenvectors are normalized and assembled in a matrix form as,

$$\left[\Phi_{fi}\right]^T\left[K_{ii}\right]\left[\Phi_{fi}\right]=\left[\Lambda_{fi}\right], \qquad \left[\Phi_{fi}\right]^T\left[M_{ii}\right]\left[\Phi_{fi}\right]=\left[I_{fi}\right] \tag{15}$$

Thus the normal modes of fixed interface are given by:

$$\left[\mathbf{\Phi}_f\right]=\begin{bmatrix}\mathbf{\Phi}_{fi}\\ \mathbf{0}\end{bmatrix} \tag{16}$$

Eq.(16) represents a complete set of fixed interface modes defined by one substructure, however, the physical coordinates of each substructure can be represented by a truncated set *k* of fixed interface normal modes plus a set of interface modal coordinates *j*,

$$\{u\}=\left[\mathbf{\Phi}_k\right]\{p_k\}+\left[\mathbf{\Psi}_j\right]\{p_k\} \tag{16}$$

Eq. (16) can also be written as:

$$\begin{Bmatrix}u_i\\ u_j\end{Bmatrix}=\begin{bmatrix}\mathbf{\Phi}_{ik} & \mathbf{\Psi}_{ij}\\ \mathbf{0}_{jk} & \mathbf{I}_{jj}\end{bmatrix}\begin{Bmatrix}p_k\\ p_j\end{Bmatrix} \tag{17}$$

Considering the (*r*) substructure,

$$\left[\mathbf{\Psi}\right]^{(r)}=\begin{bmatrix}\mathbf{\Phi}_{ik} & \mathbf{\Psi}_{ij}\\ \mathbf{0}_{jk} & \mathbf{I}_{jj}\end{bmatrix}=\left[\mathbf{\Phi}_k \quad \mathbf{\Psi}_j\right] \tag{18}$$

Using Eq. (17) the matrix that forms the complete structure is written,

$$\{u\}=\left[\mathbf{\Psi}\right]\{p\} \quad \Rightarrow \quad \begin{Bmatrix}u_i^{(r)}\\ u_j^{(r)}\\ u_i^{(r+1)}\\ u_j^{(r+1)}\end{Bmatrix}=\begin{bmatrix}\left[\mathbf{\Phi}_k \quad \mathbf{\Psi}_j\right]^{(r)} & \mathbf{0}\\ \mathbf{0} & \left[\mathbf{\Phi}_k \quad \mathbf{\Psi}_j\right]^{(r+1)}\end{bmatrix}\begin{Bmatrix}p_k^{(r)}\\ p_j^{(r)}\\ p_k^{(r+1)}\\ p_j^{(r+1)}\end{Bmatrix} \tag{19}$$

3. SATELLITE ANALYSIS

When studying substructure techniques we note the advantages of using them for large and complex structures like satellites. When partitioning the Itasat satellite the analysis can work with few degrees of freedom to evaluate component natural frequencies and vibration modes. As the following step, the analysis uses the component modes information to form the vibration modes and frequencies for the complex structure. This procedure allows the possibility of testing variations in the component configuration with time saving. It is possible to verify the influence of some materials in a certain component and after only join the best solution to the complete structure.

3.1 Joint analysis of Itasat structure

Sandwich structures have in general a plate configuration with thin faces separated by a thicker core. These plates are joined by screws, rivets or adhesives whose purpose must satisfy the project requirements of transferring loads. The Itasat internal panels are assembled as T joints with 90 degrees between two plates. The T joints for sandwich plate union are analyzed in two main configurations: glued and screwed. Figure 5 shows the sketches of a glued T joint.

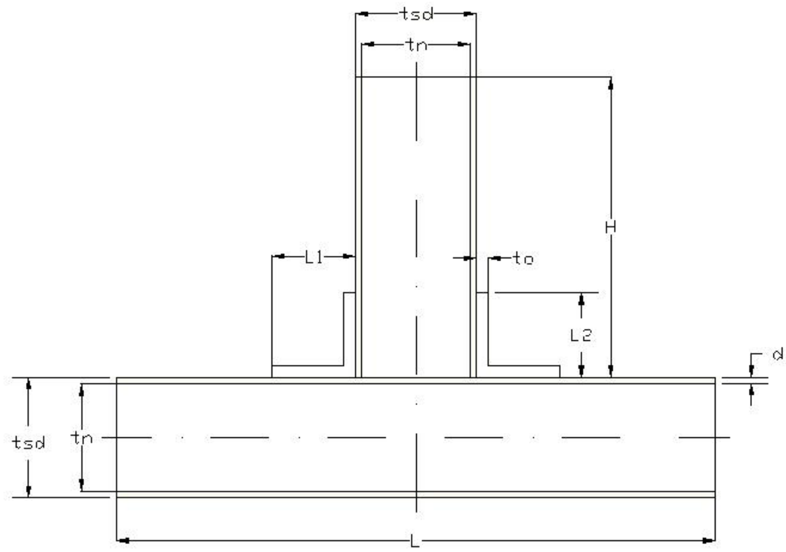

Figure 5. Sketch of a glued T joint

T joint dimensions are given by Tab.1. Table 2 shows the materials used in both cases and lastly the material properties are listed in Tab. 3 (Hexcell, 2006). The analyses consider the joints are fixed in opposite base edges. Some L beams are assembled near the edge of sandwich plates to turn the joint more effective in stiffness. These short beams are equally spaced by a distance of 50mm.

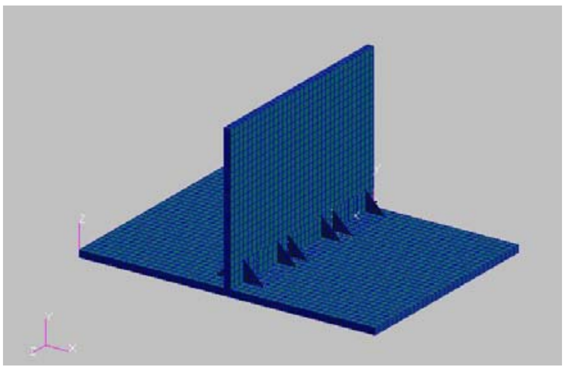

Figure 6. Glued T joint numerical model

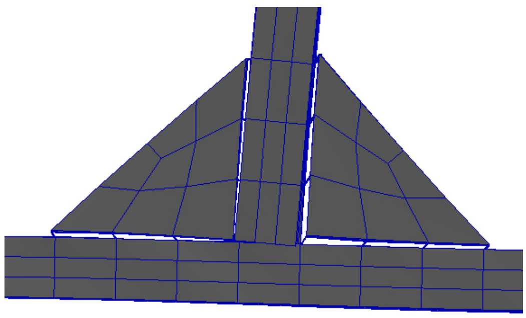

Figure 7. Glued T joint first vibration mode

3.2 Substructuring analysis of the satellite

The satellite has been divided in two substructures. The first one considers external, superior and inferior panels. The other substructure is composed by internal panels, reinforcements made as L short beams and assembled with adhesive glue. Stiffness and mass matrices for external panels are obtained by Nastran (MSC, 2001) analysis that uses the Craig-Bampton method. The influence of these matrices in the residual element (internal panel) was analyzed previously. The connection between the internal and external panels is also made by L short beams. In this analysis, the inferior panel of the satellite is fixed in the nodes of the launcher flange.

Both substructures are connected by common nodes and these nodes are considered as joint nodes.

Table 1. T joint dimensions

Description	Symbol	Measure [mm]
Total width	-	650
Total Length	L	905
Height	H	442.5
Screw diameter	d_p	6
Sandwich thickness	t_{sd}	22
Core thickness	t_n	20
L-beam thickness	t_0	2

3.3 Glued T joint modal analysis

Figure 6 shows the glued T joint numerical model. Sandwich thin faces are modeled with four node plate elements (CQUAD4) assembled in one layer. The core is modeled as hexahedron solid elements (CHEX8) assembled in three layers. The L-beam is modeled using CQUAD4 elements and the adhesive is modeled as one-dimensional spring elements (CELAS1) that connect L-beam and sandwich nodes.

Table 2. T joint materials

Description	Material
Faces	Aluminum 7001-O
Core	HexWeb CRIII 5052 1/16 Micro-Cell
L-beam	Aluminum 7001-O
Adhesive	Plexus MA310
Screw	Steel

The T joint detail in Fig. 7, shows the adhesive behavior by the visualization of the first vibration mode. There are places in the adhesive that are subjected to traction and compression as the T joint legs vibrates.

3.4 Screwed T joint modal analysis

The model of this T joint has regions with aluminum inserts for fixing the screws. This model uses RBE2 elements to model the screw head and beam elements CBAR for the screw body (MSC, 2004).

Table 3. Material properties

Property	Faces	Core	L-Beam	Adhesive	Screw
Density ρ [kg/m^3]	2,840	100	2,840	1,000	7,800
Longitudinal modulus E [GPa]	71		71	1.2	200
Longit. modulus $E_{11} = E_{22} = E_{33}$ [MPa]		1.9			
Poisson ratio ν	0.33	0.40	0.33	0.40	0.30
Shear modulus G_{12} [MPa]		275			
Shear modulus $G_{13} = G_{23}$ [MPa]		620			

Table 4. Itasat natural frequency numerical results

Mode	Substructure model [Hz]	Complete model [Hz]	Difference [%]
1	224.7	224.5	0.08
2	261.9	261.9	0
3	278.9	278.9	0
4	288.6	288.4	0.07
5	289.8	289.6	0.07
6	333.1	333.0	0.03
7	427.3	426.7	0.14
8	472.1	470.5	0.34

Table 4 shows the natural frequencies obtained for complete and substructure Itasat models. On the first vibration mode occurs the bending in the internal panels and torsion in the external panels. On the second mode (Fig. 8) in internal and external panels occurs bending and on the third mode presents bending in the external components and shear in the internal ones as Fig. 9 shows.

3.5 Study of internal panel materials

This analysis compares various materials in the composition of the satellite internal panels. Table 5 shows the material specifications. Configuration *A* refers to previous analysis. In configuration *B* the aluminum will be changed by laminated epoxy-carbon to the faces. In configuration *C* the core will be formed by rigid foam besides aluminum. These material properties used for comparison can be seen in Tab. 6.

It is not possible to say the vibration modes from configurations *A*, *B* and *C* are equivalent to each other just because the frequency values are similar. In Fig. 10 it is possible to see the difference between the vibration modes. The frequencies are arranged in ascending order but it might be changed modes between the configurations.

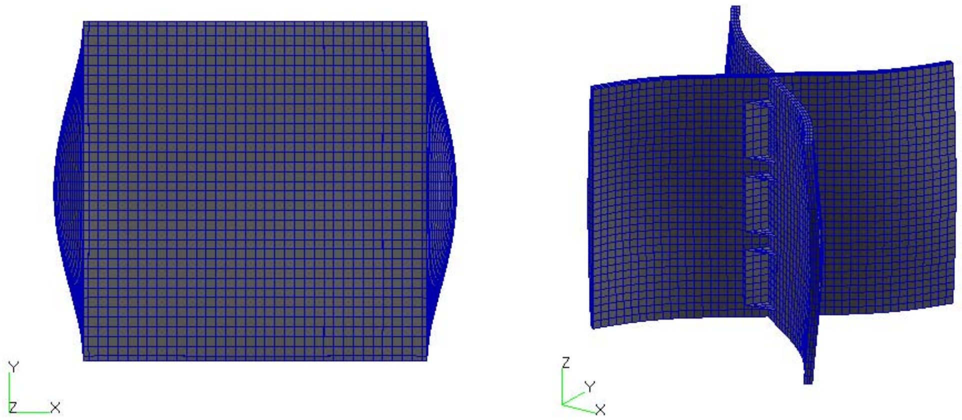

Figure 8. Vibration modes of the external set and T joints for frequency of 261.9 Hz

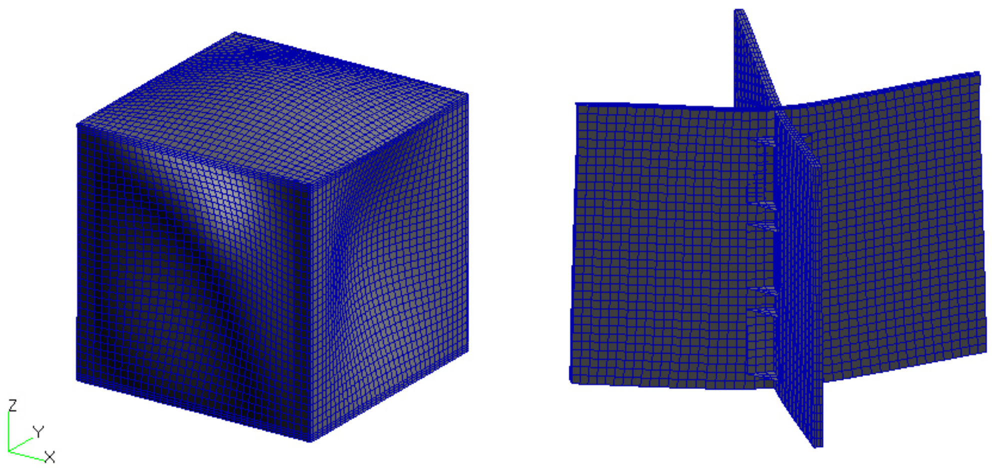

Figure 9. Vibration modes of the external set and T joints for frequency of 278.9 Hz

Table 5. New internal panel materials and configurations for Itasat

Description	***B* configuration**	***C* configuration**
Face	Carbon-Epoxy AS4/3501-6	Aluminum 7001-O
Core	HexWeb CRIII 5052 1/16 Micro-Cell	Divinycell H-60

Table 6. Material properties for new configuration

	Face in Carbon-Epoxy	**Core in Rigid Foam**
Density ρ [kg/m^3]	1600	60
Longitudinal Modulus E [MPa]		53
Longit. Modulus E_1 [GPa]	147	
Longit. Modulus $E_2 = E_3$ [GPa]	10.3	
Poisson ratio ν_{12}	0.27	0.32
Poisson ratio ν_{23}	0.54	
Poisson ratio ν_{13}	0.27	
Shear modulus $G_{12} = G_{13}$ [GPa]	7	
Shear modulus G_{23} [GPa]	3.7	

Table 7. Natural frequencies for configurations *A*, *B* and *C* of complete models

Mode	**Configuration *A* [Hz]**	**Configuration *B* [Hz]**	**Configuration *C* [Hz]**
1	224.7	234.3	272.7
2	261.9	268.2	279.8
3	278.9	270.8	300.0
4	288.6	289.7	300.7
5	289.8	291.0	319.0

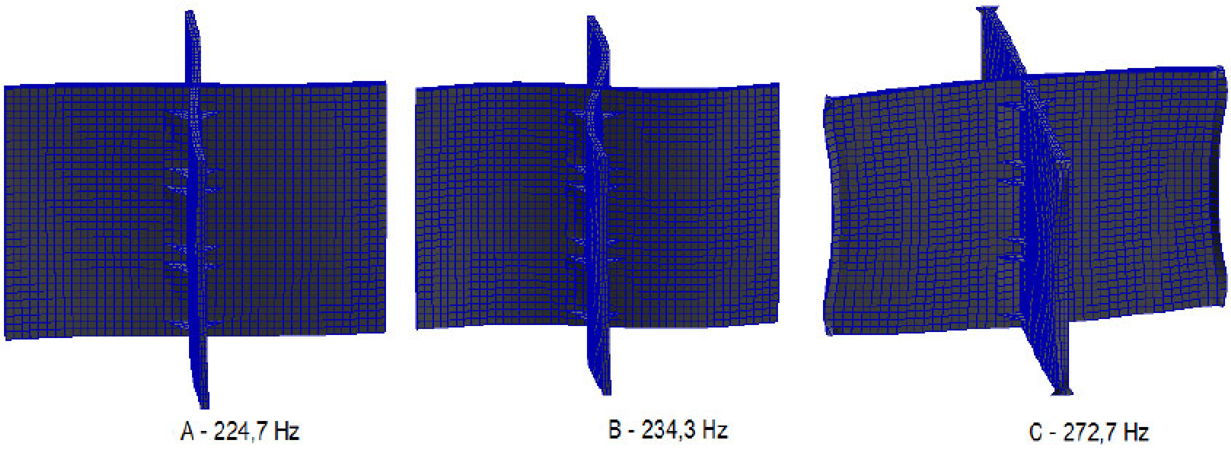

Figure 10. Vibration of the internal panels relative to the first mode in Table 9

4. FINAL CONSIDERATIONS

This paper developed a numerical model representing the Itasat satellite made by sandwich plates and using finite element methodology. Some considerations must be made when analyzing sandwich plates. It is necessary to build an appropriate model whose elements sizes and layer numbers employed for the solid core element must be correctly represented. The mesh obtained for the complete model has too many degrees of freedom and it requires extensive computational resources. If model simplifications are applied the precision of solution can be affected, otherwise simulations are long and very expensive.

When making the Itasat numerical model a lot of assumptions were made. The electronic devices as well as cables and antennas were not considered in the analyses. In this work, only the sandwich plates were investigated. All connections among sandwich plates were considered as T joints. Glued and screwed T joints were analyzed. The analyzed structure with glued joints had stiffness 45% higher and mass 40% lower then the structure formed by screwed joints. The frequency results for the two cases of connections were not conclusive since the tension on the panels were not analyzed. These results only bring attention to natural frequency differences when changing the joint connections.

Despite of results herein, the analysis shows that when changing the T joint models it implies differences in degrees of freedom for the complete Itasat model making it more practical to use partial analyses.

Table 6 shows the great results when comparing the satellite natural frequencies obtained using the complete and partitioned satellite. The small differences between results are explained due the complexity of the structure and the large number of degrees of freedom used to model the two substructures connections. The obtained error is bigger when more degrees of freedom for joints are used. In this case since the complete structure has more then 150000 degrees of freedom, the use in the joints 1% of this quantity leads to small errors. When observing Tab. 7 it can be seen that natural frequencies for A and B configurations are similar but different from natural frequencies for C configuration.

Another interesting point is how long the analysis takes. Using substructuring method the running time was 50% lower in comparison with the same for analyzing the complete structure. In some few cases of substructuring the analysis was 20 times faster. The running time is one of the most relevant advantages for substructuring use.

5. ACKNOWLEDGEMENTS

This research was supported by Brazilian Aerospace Agency (AEB).

6. REFERENCES

Araújo, C.A., "Modelagem de sistemas dinâmicos através da síntese modal de componentes", PhD thesis – Universidade Estadual de Campinas, 1998.

Azar, J.J., "Matrix structural analysis", New York: Pergamon Press, 1972, 217p.

Craig, R.R. and Bampton, M.C.C., 1968, "Coupling of substructuring for dynamic analysis", v.6, n.7, pp. 1313-1319.

Craig, R.R. and Chang, C-J., 1976, "A review of substructure coupling methods for dynamic analysis", NASA CP-2001, v.2, pp. 393-408.

Craig, R.R., "Structural dynamics: an introduction to computer methods", New York: Wiley, 1981, 527p.

Craig, R.R., 2000, "Coupling of substructures for dynamic analysis: an overview", Collection of Technical Papers - AIAA/ASME/ASCE/HS/ASC Structures, v.5, pp.3-14.

Ewins, D., "Modal testing: theory, practice and application", Hertfordshire: Research Studies Press, 2000, 562p.

Hexcel, "Hexweb cr-III micro-cell product data", São José dos Campos, 2006, Disponível em <www.hexcel.com>.

Hurty, W.C., 1965, "Dynamic analysis of structural systems using component modes", AIAA Journal, v.3, n.4, pp.678-685.

Mindlin, R.D., 1951, "Influence of rotary inertia and shear on flexural motions of isotropic elastic plate", Journal of Applied Mechanics, v.18, n.1, pp.31-38.

MSC.Software, "MSC Nastran reference manual", Santa Ana: The Macneal-Schwendler Corporation, 2004.

MSC.Software, "MSC Nastran superelements user's guide", Santa Ana, 2001.

Nabarrete, A., "A three layer quasi 3D finite element model for structural analysis of sandwich plates", PhD thesis - Instituto Tecnológico de Aeronáutica, 2002.

Oskooei, S. and Hansen, J.S., 2000, "A higher order finite element for sandwich plates", AIAA Journal, v.38, n.3, pp.525-533.

Przemieniecki, J.S., "Theory of matrix sructural analysis", New York: McGraw-Hill, 1986.

Reddy, J.N., 1984, "A simple higher-order theory for laminated composite plates", Journal of Applied Mechanics, v.51, n.3, pp. 745-752.

Reissner, E., 1948, "Finite deflection of sandwich plates", Journal of Aerospaces Sciences, v.15, n.7, pp. 435-440.

Theotokoglou, E.E. and Moan, T., 1996, "Experimental and numerical study of composite t-joints", Journal of Composite Materials, v.30, n.2, pp. 190-209.

Theotokoglou, E.E., 1999, "Study of the numerical fracture mechanics analysis of composite t-joints", Journal of Reinforced Plastics and Composites, v.18, n.3, pp. 215-223.

Turaga, U.V.R.S. and Sun, C.T., 2000, "Failure modes and load transfer in sandwich t-joints", Journal of Sandwich Structure Materials, v.2, n.3, 225-245.

Zhou, D.W., 2008, "Numerical simulation of sandwich t-joints under dynamic loading", Composites Part B: Engineering, v.39, n.6, pp. 973-985.

7. Responsibility notice

The authors are the only responsible for the printed material included in this paper.