ABSTRACT

The Brazilian Multi Mission Platform (MMP) is a modern satellite architectural concept that shall provide support for several of low Earth orbit missions whose attitude control subsystem includes a three-axis stabilized platform carrying different payload modules. This paper presents the dynamic analysis of the Brazilian MMP solar array generator (SAG) deployment. The objective of the study is to analyze the shock spectrum due to the SAG latch up. The transient response or time history was verified at typical hinges, yoke, solar array drive assembly, solar cells and at the sandwich panel. The finite element technique has been used to obtain the numerical results with the software package MSC.Nastran. The problem is an initial value problem whose initial conditions were derived from the SAG deployment mechanism simulation. The results of the study have shown that the satellite solar array response fulfills the requirements associated with the latch up shock spectrum.

INTRODUCTION

The power supply of most satellites orbiting the Earth come from photovoltaic solar cells bonded to either solar panels or the satellite body. In many cases the area covered by solar cells must be large when compared to the satellite body dimensions. Thus, solar panels or appendages must be used to provide the large areas due to power requirements. Because of space limitations of the launcher, a satellite must occupy the smallest volume possible. It means that the solar panels and appendages must be retracted or closed during launching. As soon as the satellite reaches its orbit the solar panels are deployed to their full extend. The deployment mechanism must be designed to ensure that there is sufficient potential energy stored in the retracted configuration such that friction forces that arise during deployment are overcome and the panel
fully deploys. On the other hand, too much potential energy initially stored would result in large final angular velocities that could damage the solar cells, the panels, hinges, equipments, etc.

The MMP SAG comprises two wings attached to opposite sides of the satellite main body. Each wing has three rectangular sandwich panels and a yoke; three pairs of hinges are used to connect the panels one another and another hinge connects the yoke to the satellite body. A FEM model of one wing was built to analyze the MMP SAG structure and to guarantee compliance with the project requirements.

The wings are connected to the satellite main body by a mechanism designated as SADA (Solar Array Drive Assembly) which consists of a step motor that controls the SAG orientation in the deployed configuration.

The present work focuses on the SAG deployment and post latch up dynamic analyses. A simplified model is proposed to simulate the deployment dynamics that considers only rigid bodies. This model leads inherently to a nonlinear dynamic problem numerically solved by a C code specially implemented for that purpose. On the other hand, the transient analysis after latch-up is conducted with the commercial FE package MSC.Nastran.

MODEL DESCRIPTION

The solar array generator has three panels connected by hinges. The innermost panel is connected to the satellite body through a truss-like structure; the yoke. Each hinge has a pre-stressed torsional spring such that a driving torque favors deployment. Additionally, the connection between yoke and the bapta contains four pre-stressed torsional springs. Figure 1 depicts the mechanism.

In order to establish the basic kinematics of the panels, auxiliary coordinate systems are introduced as shown in Fig. 2. The coordinate of a point on each panel is given in the global coordinate system xyz in terms of \( \xi_1, \xi_2, \xi_3 \) and \( \theta \). Also, the panels have width \( 2a \) and height \( h \) while the yoke has length \( b \).

The moment of inertia of panel \( i \) about a transverse axis passing through its center of mass is:

\[
I_i = \frac{m_i a^2}{3}
\]

where \( m_i \) is the mass of panel \( i \) and \( a \) is the semi-width. The position of points on panels are given in terms of local coordinates as in equation (2).

\[
x_i = \left[ b + (2i - 2)a + \xi_j \right] \sin \theta
\]

\[
y_i = (b - a) \cos \theta + (-1)^i (\xi_j - a) \cos \theta
\]

\[
z_i = z_i
\]
where $0 \leq \xi_i \leq 2a$ for all $i$. Notice that $a$, $\xi_i$ and $z_i$ are independent of time but $\theta$ is time dependent. Consequently, there is no variation along the $z$ axis with time. The velocity components of a point on panel $i$ is obtained when equations (2) are differentiated with respect to time to yield

$$
\dot{x}_i = \dot{\theta}(2(i-1)a + b + \xi_i) \cos \theta
$$

$$
\dot{y}_i = -\dot{\theta}(b - a) + (-1)^i (\xi_i - a) \sin \theta
$$

$$
\dot{z}_i = 0
$$

The kinetic energy of the solar panel assembly is given by the kinetic energy of the yoke plus the kinetic energy of each panel. The yoke kinetic energy depends on the angular velocity $\dot{\theta}$ and the moment of inertia about the $z$ axis.

$$
T_{\text{yoke}} = \frac{1}{2} \dot{\theta}^2 I_{\text{yoke},zz}
$$

The kinetic energy of panel $i$ is given in equation (5) with the aid of equation (3).

$$
T_{\text{panel}_i} = \frac{2a}{2} \int_0^{2a} \left( \dot{x}_i^2 + \dot{y}_i^2 + \dot{z}_i^2 \right) \rho_i h_i \, d\xi_i =
$$

$$
\frac{m_i \dot{\theta}^2}{2} \left[ 4a i (a i + b - a) \cos^2 \theta + (b - a)^2 + \frac{a^2}{3} \right]
$$

where $h_i$ is the height and $\rho_i$ is the mass density per unit area of panel $i$. Introduction of equation (1) into equation (5) yields the expression for the kinetic energy of panel $i$.

$$
T_{\text{panel}_i} =
$$

$$
\frac{1}{2} \dot{\theta}^2 \left[ I_i + (b - a)^2 m_i + 4ai(a i + b - a)m_i \cos^2 \theta \right]
$$

The potential energy due to the pre-stressed torsional springs is derived as follows. An arbitrary number of springs are assumed between yoke and bapta. These are subjected to a pre-stress angle $\varphi_{yb}$ that favors the solar panel mechanism deployment. The potential energy is then

$$
V_{yb} = \frac{n_{yb}}{2} k_{yb} (\varphi_{yb} - \dot{\theta})^2
$$

where $n_{yb}$ is the number of springs between yoke and bapta, and $k_{yb}$ is the spring constant. The summation of the potential energies of the torsional springs present in the hinges is given in equation (8).

$$
V_h = \frac{n}{2} n_h k_h (\varphi_h - 2\theta)^2
$$

where $n$ is the number of panels, $n_h$ is the number of hinge springs per panel, $k_h$ is the spring constant and $\varphi_h$ is the pre-stress angle. Notice that, if there is one spring per hinge and two hinges per panel, then $n_h = 2$.

The energy generated by the solar cells mounted on the solar panels is transmitted through power cables that extend from one panel to the other, and from the innermost solar panel to the yoke where they are connected to the bapta. Additionally there may be cables for signal transmission. The design of the power plant requires several distinct electric circuits what, in turn, requires several power and signal cables. Hence, it is expected that the presence of these cables interfere with the deployment dynamics.

Over the region where the power cables extend from one panel to the other the power cables are aligned with the hinge axis such that, during deployment, the cables will produce torque about the same axes as the torsional springs in the hinges. The cables are mounted on the panels ensuring that the torque produced favors deployment over most of the time the panels unfold.
These power and signal cables are pre-stressed when the solar panels are retracted such that they act as springs during deployment. Moreover, cables are often made out of viscoelastic material what motivates the use of a viscoelastic constitutive law. The torques applied by the power and signal cables are designed to favor the panel deployment and can be modeled by:

\[
\begin{align*}
T_{\text{power}} &= k_{p,c}(\phi_{p,c} - 2\theta) + 2\dot{\phi}_{p,c} \\
T_{\text{signal}} &= k_{s,c}(\phi_{s,c} - 2\theta) + 2\dot{\phi}_{s,c}
\end{align*}
\]  

(9)

where \(k_{p,c}\) is the spring constant, \(\phi_{p,c}\) is the pre-stress angle and \(c_{p}\) is the viscoelastic constant of the power cables while \(k_{s,c}, \phi_{s,c}\) and \(c_{s,c}\) are the respective counterparts for the signal cables. These coefficients may be estimated experimentally by drawing graphs of \(T_{\text{cable}}\) versus time, and \(\theta\) versus time admitting that \(\phi_{\text{cable}}\) is known. In practical applications \(c_{\text{cable}}\) is expected to be negligible for the range of \(\theta\) but it is considered in the model.

If there is no friction, equations (4), (5), (7) and (8) can be used to derive the governing dynamic equations of the mechanism. However, friction forces acting on the hinges are considerable and have a significant effect on the overall dynamics. These forces are partially accounted for through two parameters: (i) a constant friction torque and (ii) a friction parameter \(\zeta \in [0,1]\) that reduces the effectiveness of the torsional springs (Palerosi, 1997). The friction torques are also assumed to have a constant component per bearing such that Thus, the torques acting in between yoke and bapta, \(T_{yb}\) and hinges, \(T_{h}\) are given by

\[
\begin{align*}
T_{yb,\text{friction}} &= \zeta k_{yb}(\phi_{yb} - \theta) + n_{yb} T_{yb,\text{cte}} \\
T_{h,\text{friction}} &= \zeta k_{h}(\phi_{h} - 2\theta) + n_{h} T_{h,\text{cte}}
\end{align*}
\]

(10)

where \(T_{yb,\text{cte}}\) is the constant component of the friction torque in between yoke and bapta, \(T_{h,\text{cte}}\) is the constant component of the friction torque in the hinges, \(n_{yb}\) is the number of bearings between yoke and bapta, and \(n_{h,\text{b}}\) is the number of bearings in one hinge. Governing equations are directly obtained considering the Lagrangian \(L\) of the mechanical system

\[
L = T_{\text{yoke}} + \sum_{i=1}^{n} T_{i} - \frac{n_{yb}}{2} k_{yb}(\phi_{yb} - \theta)^2 - n_{h} k_{h}(\phi_{h} - 2\theta)^2
\]

(11)

Notice that \(T_i\) given in equation (6) is a function of both \(\theta\) and \(\dot{\theta}\). Taking the first variation of \(L\) and considering Hamilton’s principle yields

\[
\begin{align*}
&\sum_{i=1}^{n} I_i + \sum_{i=1}^{n} m_i (b - a)^2 + 4a \cos^2 \theta \sum_{i=1}^{n} m_i (ai - b + a) \dot{\theta} - \\
&2a \dot{\theta}^2 \sin(2\theta) \sum_{i=1}^{n} m_i (ai - b + a) + \\
&n_{yb} k_{yb}(\theta - \phi_{yb}) + 2nn_{h} k_{h}(2\theta - \phi_{h}) = \\
&- n_{yb} T_{yb,\text{cte}} - 2nn_{h} T_{h,\text{cte}} + T_{\text{power}} + T_{\text{signal}}
\end{align*}
\]

(12)

Equation (12) is generic, in the sense that an arbitrary number of panels can be considered, it is nonlinear in \(\theta\), and it does not admit closed form solution. Thus, it must be solved numerically. Even for small angles \(\theta\) the second term in equation (12) would contain a quadratic term in \(\dot{\theta}\), which prevents any kind of analytic, simplified solution. Equation (12) is nonlinear in \(\theta\) and, therefore, must be numerically solved. The method selected is the Hughes, Hilber & Taylor \(\alpha\)-method [3] where a linear acceleration is assumed.

If a driving torque is to be exerted by the torsional springs then the pre-stress angles \(\phi_{yb}\) and \(\phi_{h}\) must be positive. Negative pre-stress
angles would result in negative torques applied to the mechanism. The constant component of the friction torques given in equation (10) are positive and, according to equation (12), work against the panel deployment. The sign of the power and signal cable torques depends on the cable pre-stress angles $\phi_{p,c}$ and $\phi_{s,c}$.

Observe that there are \( n+1 \) axes of power and signal cables and, hence, the cable torque designated by \( T_{\text{power}} \) and \( T_{\text{signal}} \) must be understood as the summation over all cables involved. It means that constants \( k_{p,c}, \phi_{p,c}, c_{c,p}, k_{s,c}, \phi_{s,c} \) and \( c_{s,p} \) must be obtained for each cable.

Figure 3 shows the MMP SAG design performed on CATIA software and the respective FE model. The MMP SAG finite element model (one wing) in the stowed configuration has a total of 8034 elements and 6011 nodes. In deployed configuration, the SAG finite element model has a total of 7065 elements and 5583 nodes (hold-down base and some DOF springs were not included). In the deployed configuration, only the yoke tip was constrained in the six degrees of freedom. Non-structural mass corresponding to solar cells, electrical cables and adhesive film used on sandwich panel assembly was included in the models.

It must be emphasized that the transient analysis begins after the complete SAG deployment. The initial conditions applied correspond to prescribed velocities obtained from the deployment dynamic analysis of the mechanism. These prescribed velocities were applied normal to the panel plane. Figure 4 shows the SAG deployment and the distributed velocity vectors in the beginning of the transient simulation (end of deployment).

<table>
<thead>
<tr>
<th>CATIA Design</th>
<th>FE Model</th>
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<tbody>
<tr>
<td><img src="a" alt="Figure 3" /></td>
<td><img src="b" alt="Figure 3" /></td>
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<tr>
<td><img src="c" alt="Figure 3" /></td>
<td><img src="d" alt="Figure 3" /></td>
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<tr>
<td><img src="e" alt="Figure 3" /></td>
<td><img src="f" alt="Figure 3" /></td>
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Figure 3: MMP CATIA Design and FE model. It can be seen SAG assembled on satellite (a), SAG panels on the stowed configuration (b) and (c), typical hinge connection detail (d), SAG deployed configuration (e) and hinge detail in the deployed configuration (f).
The MSC.Nastran Transient Response analysis and Response Spectrum Analysis were used considering prescribed initial velocities [1]. Through the transient response analysis it is possible to predict the response of the structure subjected to time-varying excitation and/or specified initial conditions. The structure response is also a time domain function.

Response spectrum is a post processing of the transient response. It converts the response from time domain to frequency domain using the fast Fourier transform.

The direct transient method was adopted [1] with damping of the structure (transient response) of 3% (structural damping). This damping was converted into equivalent viscous damping at the deployed SAG first natural frequency (0.64 Hz).

Additionally, an oscillation damping coefficient of 5% was considered. This damping defines the damping to be used when computing the shock response spectra from the time histories. The total simulation time was 0.8 seconds.

**MMP SHOCK REQUIREMENTS AND COMPONENTS FAILURE MODES**

The MMP equipment shock requirement is based on a half sine acceleration wave of 50 g applied on the three axes, one at a time, during 6 to 10 milliseconds. The failure modes of the most important SAG components were obtained from Ref. [2] and are listed in Table 1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Failure mode</th>
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<tbody>
<tr>
<td>Honeycomb core</td>
<td>Transverse shear</td>
</tr>
<tr>
<td>Face sheet</td>
<td>Intracell buckling</td>
</tr>
<tr>
<td>Typical and yoke hinge</td>
<td>Internal spherical bearing and shaft, attachment lug and pin</td>
</tr>
<tr>
<td>Solar cell</td>
<td>Bending crack</td>
</tr>
<tr>
<td>SADA</td>
<td>Radial shear</td>
</tr>
<tr>
<td>Yoke laminate arm</td>
<td>Tension fail</td>
</tr>
</tbody>
</table>

Table 1: SAG components failure mode description

Shock requirements are applicable to the SADA and sun sensor. The criteria of maximum stress
over time are used to compute margins of safety for honeycomb core, face sheets and yoke. For the solar cells the criterion is based on the maximum curvature over time whereas the requirement for the hinges is based on the maximum forces over time.

RESULTS

Table 2 shows the minimum margins of safety of SAG components due to transient response or time history behavior. These margins were computed comparing the maximum stresses, forces or curvatures provided by the transient response analysis with allowable values of each component or results based on the shock spectrum (SADA and sun sensor) as described in the previous section. Factors of safety of 1.5 were applied, according to MMP specifications.

<table>
<thead>
<tr>
<th>Component</th>
<th>Minimum MS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Honeycomb core 3/8&quot;</td>
<td>11.4</td>
</tr>
<tr>
<td>Honeycomb core 1/4&quot;</td>
<td>17.4</td>
</tr>
<tr>
<td>Face sheet on core 3/8&quot;</td>
<td>9.9</td>
</tr>
<tr>
<td>Face sheet on core 1/4&quot;</td>
<td>18.6</td>
</tr>
<tr>
<td>Hinge spherical bearing</td>
<td>7.5</td>
</tr>
<tr>
<td>Hinge internal shaft</td>
<td>3.6</td>
</tr>
<tr>
<td>Hinge attachment lug</td>
<td>64.0</td>
</tr>
<tr>
<td>Hinge attachment pin</td>
<td>15.0</td>
</tr>
<tr>
<td>Solar cell</td>
<td>10.7</td>
</tr>
<tr>
<td>SADA</td>
<td>0.11</td>
</tr>
<tr>
<td>Yoke laminate arm</td>
<td>&gt;100</td>
</tr>
</tbody>
</table>

Table 2: Margins of safety for the SAG latch up

The following figures show both transient response and shock spectrum graphs for each component listed above. Figures 6 and 8 show the time histories for the SADA and the sun sensor, respectively. From Figs. 7 and 9 it can be seen that shock spectrum (derived from Figs. 6 and 8 respectively) provided by SAG latch up shock analysis (red curve) is lower than MMP specifications (blue and green curves). It means that SAG latch up shock spectrum fulfills MMP equipment requirements.

![Figure 6: SADA force time history](image)

![Figure 7: SADA shock spectrum](image)

![Figure 8: Sun sensor acceleration time history](image)
CONCLUSIONS

A few simplifications have been adopted in the model for the deployment dynamic analysis:

- The synchronism cables are infinitely rigid;
- the solar panel are treated as rigid bodies for the dynamic deployment analysis;
- there is no movement along the $z$ direction;
- asymmetries on the panels due to devices mounted on them can be neglected.

Despite those simplifications the model delivers very accurate results provided a proper calibration of parameters prior to simulation.

It can be seen that all margins of safety are positive and also all shock spectrum response are lower than specified MMP requirements (except for very high frequencies). Therefore, the present design qualifies for the experimental testing campaign.

Table 2 can be a good starting point for improving the SAG design. It can be seen that the lowest margin of safety is associated with the SADA. However, several margins of safety lie above 10.0 meaning that possible structural mass reduction could be achieved (specially in the yoke arm).
REFERENCES

