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APPLICATION OF SENSITIVITY ANALYSIS FOR OPTIMIZATION OF A SATELLITE STRUCTURE

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ABSTRACT

An application of sensitivity analysis in the optimization process of a complex satellite structure is presented. The optimization problem posed is the minimization of the lowest natural longitudinal frequency of the structure (up to the minimum allowed by the launcher), subject to constraints on the natural lateral frequencies. The need for reducing the lowest longitudinal natural frequency was evidenced by the numerical and experimental results which showed that the levels of acceleration on the electronic components were excessively high, because that frequency was very close to the natural frequency of the printed circuit boards. This type of problem is very hard to deal with by a trial-and-error procedure, but it is amenable to design sensitivity analysis, which is used in two stages. In the first stage, a design sensitivity analysis is performed with the whole set of design variables. Then, the design space is reduced by elimination of all the design variables corresponding to sensitivities below a specified threshold. In the second stage, the reduced design space is used to perform the optimization procedure. With this strategy, a significant reduction of the lowest natural longitudinal frequency was attained.

1. INTRODUCTION

Design sensitivity analysis has been used in the solution of small academic problems for over thirty years¹. In the early days, the only numerical technique used to perform structural design sensitivity analysis was the finite difference method. Although this method is very easy to implement, it is unbearably expensive when applied to complex structures. For example, use of central difference to compute the sensitivities of a given response functional with respect to n design variables requires $2n$ additional analyses. Besides the high costs, the precision of the method is not always easy to control due to two important sources of error: the truncation error and the condition error.

In recent years, very efficient design sensitivity analysis formulations (the Direct Differentiation Formulations and the Adjoint Variable Formulations) have been developed for a large class of structural problems²⁻⁶. Several of these formulations are now implemented in commercial finite element analysis software, making real world applications to complex structural systems possible. In general, the additional cost of the sensitivity

Figure 1: Exploded view of the structure

analysis with these new formulations is only a fraction of the structural analysis cost.

Here we apply sensitivity analysis in the optimum design of the Brazilian Scientific Satellite (SACI-1). SACI-1 is a low earth orbit spacecraft for scientific purposes. It will fly in a circular orbit inclined about 98° with respect to the Earth's equatorial plane, at an approximate altitude of 750 km. The on-board instruments will be used to study the plasma bubbles in the ionosphere; to measure the intensity of the terrestrial airglow emission; to study the geomagnetic phenomena; and to measure fluxes, spectra and composition, temporal and spatial variations of ions from C to Ne.

The shape of the spacecraft's mechanical structure is a 600x400x400 mm parallelepiped, weighting about 65 kg. The structure is divided into four parts: main body, sensor bay, solar panels and adapter cone (Figure 1). The main body consists of a pack of 9 ribbed aluminium frames of different

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thickness', stacked horizontally and connected by 12 stud bolts. The sensor bay is assembled on top of the platform and has six ribbed plates forming a box in which most of the payload is assembled. The satellite has four deployable panels; each measuring 440x570 mm and supporting a payload sensor. Three of the panels are covered with solar arrays. The adapter cone is the interface between the satellite and the launcher⁷.

The function of the structure is to fulfil strength and stiffness requirements and provide a suitable environment for the satellite instruments. A finite element model (FEM) was developed based on the design of the structure and used for its structural analysis and redesign. Another purpose of this model was to predict the vibration levels on the most delicate electronic components. Since-burst, sinusoidal, and random vibration tests were performed and the results were used to adjust the numerical model.

The finite element analysis performed with the adjusted numerical model indicated that the levels of acceleration on the electronic equipment were exceedingly high. It was found that the lowest longitudinal natural frequency of the structure was very close to the fundamental frequency of the frames where the printed circuit boards were assembled. However, it was significantly above the minimum allowed by the launcher's specification. The lowest lateral natural frequencies were twice as large as the minimum allowable and, on the other hand, the total mass of the structure was at the maximum allowable.

The main task of the structural analysis and design team was changing the original design in order to reduce the lowest longitudinal natural frequency of the satellite structure, while reducing its structural mass. Making changes to a complex satellite structure, based on a trial-and-error procedure is highly ineffective and time consuming. Fortunately, with the new design sensitivity analysis formulations, it is possible to approach those complex problems in a very systematic way.

In this paper, we show how design sensitivity analysis is used in the solution of the problem described above. The paper is organized as follows. In Section 2, we describe the original numerical model and the finite element analysis. In Section 3, we define the optimization problem and show, in a systematic way, how design sensitivity analysis was used to make more effective the design changes in the optimization process. Finally, in Section 4, we present some conclusions.

2. FINITE ELEMENT ANALYSIS

The SACI-1 structure was analysed using the Finite Element code MSC/NASTRAN⁸. The model has 5,251 nodes and 5,838 elements (bar, plate and rigid elements) (Figure 2). All instruments were modelled as concentrated masses at their CG positions, considering their moments of inertia. The components of the electronic boards were modelled as distributed non-structural masses.

Figure 2: SACI1 Finite element model.

The model, adjusted by the test results, was used to perform the dynamic analysis of the structure. First, a modal analysis was performed. Then, modal superposition techniques were used to reduce the computational costs of obtaining the dynamic response of the model. Determining the effective mass associated with each vibration mode⁹, we neglected the contributions of all modes whose frequencies were above 600 Hz. The two first lateral and the first longitudinal vibration modes are presented in figures 3 to 5, and, in Table 1, the corresponding frequencies are shown in comparison to those obtained on the lab tests¹⁰.

Table 1: Numerical and test results of the natural frequencies.

Vibration Mode	MSC/Nastran Results (Hz)	Test Results (Hz)
1	51	51
2	52	51
5	162	165

A thorough examination of the fifth mode revealed that an amplification of displacements occurred at the central region of all the ribbed aluminum frames where the printed circuit boards were assembled (figures 6 and 7). A typical aluminum frame was constructed and tested at the laboratory (Figure 8) to determine the first four natural frequencies in the z-axis¹¹. The results of the test are shown in Figure 9 and on Table 2. This

showed that the natural frequency of 162 Hz associated with the fifth mode was close to the first natural frequency of 182 Hz (in the z direction) of the ribbed frames with the printed circuit boards assembled on them. This is an unacceptable situation since that might seriously damage the electronic circuitry during launching.

Figure 3: First vibration mode (original structure)

Figure 4: Second vibration mode (original structure)

Figure 5: Fifth vibration mode (original structure)

Figure 6: Fifth mode vibration of Orcas ribbed frame

Figure 7: Fifth mode vibration of Plasma ribbed frame

Figure 8: Low-level sine sweep test of a typical frame

Figure 9: Low-level sine sweep test results

Table 2: Natural frequencies of a typical frame from low-level sine sweep test

Number	Frequency (Hz)	Acceleration ($\times g$)
1	181.87	1.445
2	394.72	11.529
3	1,038.05	4.439
4	1,283.09	9.751

In the next section, we describe the original optimization problem we needed to solve and how design sensitivity analysis was used effectively in the process.

3. DESIGN OPTIMIZATION

3.1 Description of the problem

The problem consists of modifying the dimensions of the structural members (maintaining the topology of the structure unchanged) in order to reduce the lowest longitudinal natural frequency to minimize the amplifications described in Section 2 and shown in figures 5 to 7. The structure is modeled with 5,838 elements (3,997 shell elements and 1,841 beam elements) which have material and section properties that can be considered in the design variables set. Fortunately, many elements have the same material and section properties, which reduces drastically the design space. In this problem, changing the material properties is not an option and, therefore, another significant reduction of the design space is achieved. After these considerations, the design space was reduced to a total of 173 design variables, divided into three groups: 120 beam element sectional dimensions, 35 shell element thickness', and 18 design variables controlling the shapes of the mechanical interface ribs (Figure 10).

Figure 10: Mechanical interface modulus

Defined the design space, some of the questions we might ask are:

- Which design variables, out of the set of 173 design variables, should be changed in order to reduce the fifth natural frequency of the structure?
- How big should be those changes?
- Which variables should be increased and which should be decreased?
- How much the other natural frequencies will be affected by those changes?
- What restrictions on the response of the structure would be violated by the design changes?
- Will the fifth mode of the modified structure still be a longitudinal vibration mode?

For a complex structure like this, it is very difficult and time consuming to answer all of these questions based on trial and error, even for an experienced analyst.

3.2 Reduction of the design space

3.2.1 Design sensitivity analysis —

Design sensitivity analysis is a very important tool, which helps to answer some of the questions listed in Section 3.1. With the sensitivity information, we can further reduce the design space, as we show in the remainder of this subsection.

Let λ_5 , the fifth natural frequency of the initial design, be the response measure in which we are interested, and let X be the set of design variables defined as

$$X = \{x_i \in \mathbb{R} \mid 1 \leq i \leq \text{NDV}\}. \quad (1)$$

where NDV is the number of design variables. We call $F: X \rightarrow \mathbb{R}$ the implicit mapping of a point in design space to the fifth natural frequency. Thus

$$\lambda_5 = F(X). \quad (2)$$

The mapping F is implicit with respect to X because the dependence of λ_5 on the design variables is established through the governing differential equation of the problem. Hence, design sensitivity analysis seeks to determine the changes in λ_5 for small variations on each design variable. We can represent this as

$$\delta\lambda_5 = \nabla_X F(X) \cdot \delta X, \quad (3)$$

where $\delta\lambda_5$ is the variation of λ_5 induced by the variation δX on the set of design variables and $\nabla_X F(X)$ is the sensitivity array.

The sensitivity analysis results are shown in tables 3 through 5. Based on these results, we eliminate from the design space:

- 1) Each design variable whose sensitivity term has a negative sign, since we are interested in design changes that reduce λ_5 without increasing the mass of the structure (these design variables are shaded on tables 3 through 5 with a 20% gray level);
- 2) Each remaining design variable whose sensitivity term is less than the threshold of $1.E+05$ (these design variables are shown on the tables with white font on a black background).

The set of design variables dropped from 173 to 89 design variables (14 shape variables, 22 plate thickness variables and 53 beam sectional dimensions)

3.2.2 Design restrictions —

As the next step to further reduce the design space, we examine the remaining 89 design variables to see if there are any design restrictions which will not allow them to vary.

The design variables PS2011, PS7002, and PS9000 are the thickness' of external plates and cannot be less than 3mm since they have to shield the interior components of the satellite. Thus, they are

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eliminated from the design space (these variables are marked with an asterisk on Table 4).

All the beams, related to the design variables of Table 5, are part of the frames where the electronic components were assembled. From the results of the low-level sine sweep test shown in Figure 9 and Table 2, we decided to eliminate the remaining design variables of Table 5. The idea is to leave the natural frequency of the frames unchanged so that uncoupling with the lowest longitudinal frequency of the satellite's structure may occur by changing the other design variables. With this decision the final design space is composed of only 33 design variables.

Table 3: Sensitivities of λ_3 with respect to the ribs shape design variables

Numb	Name	Init. value (mm)	Sensitivity
1	Z1197	29.0	4.31E+06
2	Z1658	35.0	9.58E+05
3	Z1182	33.0	3.48E+06
4	Z1169	33.0	3.32E+06
5	Z1659	35.0	8.26E+05
6	Z1145	29.0	3.82E+06
7	Z1209	16.1	6.85E+05
8	Z1206	16.1	5.32E+05
9	Z1185	16.1	-2.25E+05
10	Z1192	16.1	-1.59E+05
11	Z1167	16.1	6.05E+05
12	Z1179	16.1	4.76E+05
13	Z1174	16.1	4.81E+05
14	Z1166	16.1	5.51E+05
15	Z1154	16.1	-1.49E+05
16	Z1153	16.1	-1.53E+05
17	Z1151	16.1	3.28E+05
18	Z1148	16.1	5.86E+05

Table 4: Sensitivities of λ_3 with respect to the plate thickness design variables

Numb	Name	Init. value (mm)	Sensitivity
1	PS1001	2.00	2.07E+05
2	PS1002	3.00	9.36E+06
3	PS1005	1.50	1.00E+07
4	PS1006	1.30	1.90E+06
5	PS1010	1.90	3.09E+05
6	PS1011	5.20	8.98E+05
7	PS1012	3.00	3.31E+06
8	PS1013	3.00	1.03E+07
9	PS1014	3.00	5.92E+06
10	PS1015	3.00	8.79E+06
11	PS1016	3.00	8.34E+06
12	PS1017	3.00	5.85E+06
13	PS1018	3.00	0.00E+00
*14	PS2011	3.00	1.33E+07
15	PS2012	2.00	6.68E+06
16	PS2019	2.00	2.41E+06

17	PS2020	2.00	4.50E+06
18	PS2021	2.00	6.72E+06
19	PS3003	2.00	9.66E+04
20	PS4003	2.00	1.70E+04
21	PS5002	2.00	-9.66E+04
22	PS6010	3.00	-7.24E+04
23	PS6011	2.30	-9.06E+04
*24	PS7002	3.00	1.28E+05
25	PS8002	3.00	-2.85E+05
26	PS8011	3.00	-2.39E+05
*27	PS9000	3.00	3.30E+06
28	PS9005	3.00	1.41E+06
29	PS9007	3.00	1.89E+04
30	PS9008	3.00	-7.10E+04
31	PS9009	3.00	1.21E+04
32	PS9010	3.00	8.86E+05
33	PS9011	3.00	-2.48E+04
34	PS9012	3.00	1.33E+05
35	PS9013	3.00	0.00E+00

Table 5: Sensitivities of λ_3 with respect to the beam cross sectional dimensions

Numb	Name	Init. value (mm)	Sensitivity
*1	B2006	10.00	5.26E+05
*2	H2006	5.00	6.72E+05
3	B2007	7.10	2.04E+04
4	H2007	5.00	2.89E+04
*5	B2008	12.00	1.63E+05
*6	H2008	5.00	1.71E+05
*7	B2009	17.50	1.95E+05
*8	H2009	5.00	3.81E+05
*9	B2014	10.00	1.30E+05
*10	H2014	5.00	2.08E+05
*11	B2015	10.00	1.87E+05
*12	H2015	5.00	9.92E+05
*13	B2016	10.00	1.35E+05
*14	H2016	5.00	2.27E+05
*15	B2017	10.00	1.62E+05
*16	H2017	5.00	2.08E+05
17	B2018	10.00	5.83E+04
*18	H2018	5.00	2.21E+05
*19	B3001	2.00	1.83E+05
*20	H3001	11.50	1.24E+05
21	B3002	8.00	4.13E+04
*22	H3002	1.00	1.04E+05
23	B3004	2.00	3.17E+04
24	H3004	11.50	3.23E+04
25	B3005	2.00	3.36E+04
26	H3005	11.50	3.64E+04
27	B3006	2.00	-4.15E+04
28	H3006	11.50	-7.40E+03
*29	B4000	2.00	1.07E+05
*30	H4000	11.50	1.25E+05
31	B4001	8.00	5.29E+03
32	H4001	1.00	-5.58E+04
33	B4004	2.00	-5.31E+04
34	H4004	11.50	2.04E+04

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35	B4005	2.00	-3.25E+04
36	H4005	11.50	3.17E+04
37	B4006	2.00	-1.28E+05
38	H4006	11.50	-2.06E+04
*39	B5000	2.00	1.14E+05
*40	H5000	11.50	1.32E+05
41	B5001	8.00	-6.37E+03
42	H5001	1.00	-1.44E+05
43	B5004	2.00	-4.44E+04
44	H5004	11.50	2.39E+04
45	B5005	2.00	-2.45E+04
46	H5005	11.50	3.50E+04
47	B5006	2.00	-1.31E+05
48	H5006	11.50	-2.11E+04
49	B6000	3.00	-7.73E+04
50	H6000	16.00	5.49E+03
51	B6012	3.00	4.42E+03
52	H6012	16.00	2.96E+04
53	B6013	3.00	-2.78E+04
54	H6013	16.00	1.62E+04
*55	B6014	3.00	2.02E+05
56	H6014	16.00	5.11E+04
57	B6015	3.00	-8.47E+03
58	H6015	16.00	6.31E+02
59	B6016	3.00	-2.24E+04
60	H6016	16.00	5.02E+03
61	B6017	3.00	1.71E+04
62	H6017	16.00	3.01E+04
63	B6018	3.00	8.41E+03
64	H6018	16.00	1.22E+04
65	B6019	3.00	2.93E+04
66	H6019	16.00	4.26E+03
67	B6020	3.00	-4.01E+04
68	H6020	16.00	-9.47E+03
69	B6021	3.00	-4.89E+04
70	H6021	16.00	-8.76E+03
*71	B7000	2.00	4.89E+06
*72	H7000	11.50	2.14E+06
73	B7001	8.00	-1.29E+04
74	H7001	1.00	-1.29E+05
*75	B7004	2.00	2.13E+06
*76	H7004	11.50	9.90E+05
*77	B7005	2.00	2.10E+06
*78	H7005	11.50	9.71E+05
79	B7006	2.00	1.69E+04
80	H7006	11.50	-1.68E+03
*81	B8000	2.00	1.37E+07
*82	H8000	11.50	5.62E+06
83	B8001	8.00	-1.39E+04
84	H8001	1.00	-1.82E+05
85	B8005	8.00	-1.82E+04
86	H8005	1.00	-2.15E+05
*87	B8006	2.00	3.27E+06
*88	H8006	11.50	1.37E+06
*89	B8007	2.00	2.62E+06
*90	H8007	11.50	1.20E+06
*91	B8008	2.00	2.56E+06
*92	H8008	11.50	1.16E+06

93	B8009	2.00	2.87E+03
94	H8009	11.50	3.71E+03
95	B8010	2.00	3.04E+04
96	H8010	11.50	-7.27E+01
*97	B8012	2.00	3.32E+06
*98	H8012	11.50	1.25E+06
99	B9001	8.00	-9.57E+03
100	H9001	1.00	-7.96E+04
*101	B9002	3.00	1.80E+06
*102	H9002	4.00	3.86E+06
103	B9006	8.20	7.13E+04
*104	H9006	1.00	2.60E+05
*105	B9014	3.00	5.31E+05
*106	H9014	4.00	6.85E+05
*107	B9015	3.00	1.80E+06
*108	H9015	4.00	4.00E+06
*109	B9016	3.00	9.38E+05
*110	H9016	4.00	2.12E+06
*111	B9017	3.00	5.07E+05
*112	H9017	4.00	8.00E+05
*113	B9018	3.00	8.25E+05
*114	H9018	4.00	1.66E+06
115	B9020	8.00	-3.28E+03
116	H9020	1.00	-2.73E+04
117	B9021	8.00	-2.05E+03
118	H9021	1.00	-2.32E+04
119	B9022	8.20	6.56E+04
*120	H9022	1.00	2.11E+05

3.3 Optimization with reduced space

After the reduction of the design space from a total of 173 to only 33 design variables, we could perform a much cheaper design optimization procedure. In this section we state the final optimization problem we submitted to the solution 200 of MSC-NASTRAN and analyze the obtained results.

The optimization problem we are interested to solve can be stated as follows:

$$\begin{aligned}
 &\text{Minimize } \lambda_5 \\
 &\text{Subject to} \\
 &\quad \lambda_1 \text{ \& } \lambda_2 \geq 15,791 \text{ (20Hz).} \\
 &\quad \lambda_3, \lambda_4 \text{ \& } \lambda_5 \geq 98,696 \text{ (50Hz).} \\
 &\quad X_{\min} \leq X \leq X_{\max}
 \end{aligned} \tag{4}$$

It is important to notice that, as the design changes with the optimization steps, the natural frequencies and corresponding vibration modes of the new design may not correspond to the vibration modes of the previous design. Thus, the objective function, which is λ_5 (the frequency corresponding to the first longitudinal vibration mode) for the starting design, may have to change, say to λ_3 , a few optimization steps latter. To avoid having to stop the optimization process to check for any changes in vibration mode, we changed the objective function to be the sum of the third, fourth, and fifth natural

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frequencies. Thus, the final optimization problem is stated as follows:

$$\left. \begin{array}{l} \text{Minimize } (\lambda_3 + \lambda_4 + \lambda_5) \\ \text{Subject to} \\ \lambda_1 \text{ \& } \lambda_2 \geq 15,791 \text{ (20Hz).} \\ \lambda_3, \lambda_4 \text{ \& } \lambda_5 \geq 98,696 \text{ (50Hz).} \\ X_{\min} \leq X \leq X_{\max} \end{array} \right\} \quad (5)$$

The results of the optimization problem of equation (5) obtained by solution 200 of MSC-NASTRAN with the method of feasible directions are shown in figures 11 through 15 and in tables 6 through 8. The first natural longitudinal frequency was reduced from 162Hz of the original design to the minimum of 88Hz in five optimization steps. In the optimum design, the first two natural vibration modes were still lateral vibration modes, but the first natural longitudinal vibration mode became the third vibration mode.

Figure 11: First vibration mode (optimized structure)

Figure 12: Second vibration mode (optimized structure)

Figure 13: Third vibration mode (optimized structure)

Figure 14: Third mode vibration of Orcas ribbed frame (optimized structure)

Figure 15: Third mode vibration of Plasma ribbed frame (optimized structure)

Table 6: Comparison of natural frequencies of the original .vs. the optimized structure

Vibration Mode	Original structure (Hz)	Optimized structure (Hz)
Lateral x direction	51	31
Lateral y direction	52	35
Longitudinal	162	88

Table 7: Initial and final values of shape design variables

Design variable	Initial value (cm)	Final value (mm)
Z1197	2.90	2.00
Z1658	3.50	2.00
Z1182	3.30	2.00
Z1169	3.30	2.00
Z1659	3.50	2.00
Z1145	2.90	2.00
Z1209	1.61	2.00
Z1206	1.61	2.00
Z1167	1.61	2.00
Z1179	1.61	2.00
Z1174	1.61	2.00
Z1166	1.61	2.00
Z1151	1.61	2.00
Z1148	1.61	2.00

Table 8: Initial and final values of the plate thickness design variables

Design variable	Initial value (mm)	Final value (mm)
PS1001	2.00	3.00
PS1002	3.00	3.00
PS1005	1.50	2.99
PS1006	1.30	3.00
PS1010	1.90	3.00
PS1011	5.20	5.20
PS1012	3.00	3.00
PS1013	3.00	3.00
PS1014	3.00	3.00
PS1015	3.00	3.00
PS1016	3.00	3.00
PS1017	3.00	3.00
PS2012	2.00	2.00
PS2019	2.00	2.00
PS2020	2.00	2.00
PS2021	2.00	2.00
PS9005	3.00	3.00
PS9010	3.00	3.00
PS9012	3.00	3.00

4. CONCLUSIONS

The efficient design sensitivity analysis formulations implemented in commercial structural analysis software are extremely helpful in the optimization process of complex structures.

Regardless of the optimization strategy adopted (be it a gradient method or not), use of design sensitivity information prior to the optimization process is recommended.

In this paper we used the design sensitivities of the fifth eigenvalue in order to select a subset of the design variables space to be used in the optimization process. With design sensitivity information only, we were able to reduce the original design variables set from 173 to a subset of only 89 design variables. First, we eliminated all the design variables whose sensitivity terms had negative signs, since this implies that a reduction of the functional requires an increase on the design variables, which, in our case, would increase the mass of the structure. Then, we eliminated all the design variables below a selected threshold value (in our case 10^{-5}). Further reduction of the design space was achieved by examining the remaining set of design variables to exclude those, for which some design restrictions would apply. The final design space was then reduced to only 33 design variables.

With the reduced set of design variables the optimization problem was formulated and solved by the method of feasible directions of MSC-NASTRAN in only five optimization steps. The lowest longitudinal frequency was minimized from the

original value of 162Hz to the optimized value of 88Hz. Without violating the lower boundary requirements of the frequencies, the first two global natural frequencies were also decreased from 51Hz and 52Hz to 31Hz and 35Hz, respectively (see Table 6)

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