STRUCTURAL OPTIMIZATION OF REINFORCED PANELS USING CATIA V5

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Abstract. *The aim of this work is to employ CATIA V5 for solving structural optimization of reinforced panels under a lower bound constraint on the first natural frequency of vibration. At first, the CAD modeled panels are presented with the parameters defining the design variables. The panels are discretized using thick shell triangular finite elements. The nonlinear optimization problems are then solved by the Simulated Annealing and the Conjugate Gradient, which are the optimization algorithms inside the Product Engineering Optimizer (PEO) of CATIA. Finally, the results obtained are discussed aiming to evaluate the CATIA performance in this type of engineering application.*

Keywords: *CATIA V5, finite element method, structural dynamics, structural optimization*

1. INTRODUCTION

Sheet metal panels reinforced by blisters and/or grooves are commonly used in several and varied engineering applications, from automobiles to tin cans. These reinforcements aim to provide more stiffness to the structural components. Their layout can be set intuitively, but this rarely will result in optimal structural characteristics. Structural optimization techniques are in this case perhaps the best way to design these components.

In this paper, the structural characteristics under study are the natural frequencies of the reinforced panels. The optimization problems are stated such that the structural mass is minimized a under lower bound constraint on the fundamental natural frequency. This is a very common problem that industry frequently has to deal with. Several parameters that define the shape and position of the reinforcements are defined as design variables.

CATIA is proving to be a reasonable structural optimization tool. In the recent Hernandes et al. (2007), some interesting structural optimization problems were studied and good results were reported, motivating the authors to keep studying other kinds of structural optimization applications. CATIA has the power for CAD modeling, structural analysis and optimization fully integrated.

The computational cost involved in solving the eigenproblem related to complex structural finite element analysis is usually high. This is the case of the models used in this work, with thousands of shell finite elements. Therefore this is an opportunity to evaluate how CATIA deals with optimizations depending on eigenproblems of considerable size.

2. PANELS UNDER STUDY

Two panels are discretized with finite elements and optimized.

2.1 Panel 1

The Panel 1 is a steel panel reinforced with blisters. Parts like this are commonly seen in automobiles, home utilities like refrigerators and stoves, etc. In the Fig. 1 it can be seen the panel depicted in 2D views together with the design variables used in the respective optimization problem as well as the boundary conditions used in the structural analysis. Several CATIA parameters are used as design variables. The a_i $(i = 1, 2, 3, 4)$ control the lengths of the blister shaped reinforcements and the b_i their widths, while the c_i control their depths. The panel thickness which is uniform and the same everywhere is controlled by parameter t.

With u_i being the displacement in the i-direction, the boundary conditions are $u_z = 0$ in the edges where $x = 0$, $y = 0$, $x = 495\ mm$ and $y = 350\ mm$. At the point $(x = 495\ mm, y = 350\ mm)$, $u_x = 0$ and at the point $(x = 495\ mm, y = 0)$, $u_x = u_y = 0$. These two last boundary conditions prevent the part from having rigid body natural modes. Figure 2 shows an isometric view of the initial design.

2.2 Panel 2

The Panel 2 is an aluminum panel reinforced with grooves of trapezoidal shape as seen in Fig. 3. Reinforcements like these are seen in several sheet metal panel composed parts like those used in many automotive roofs, ship containers,

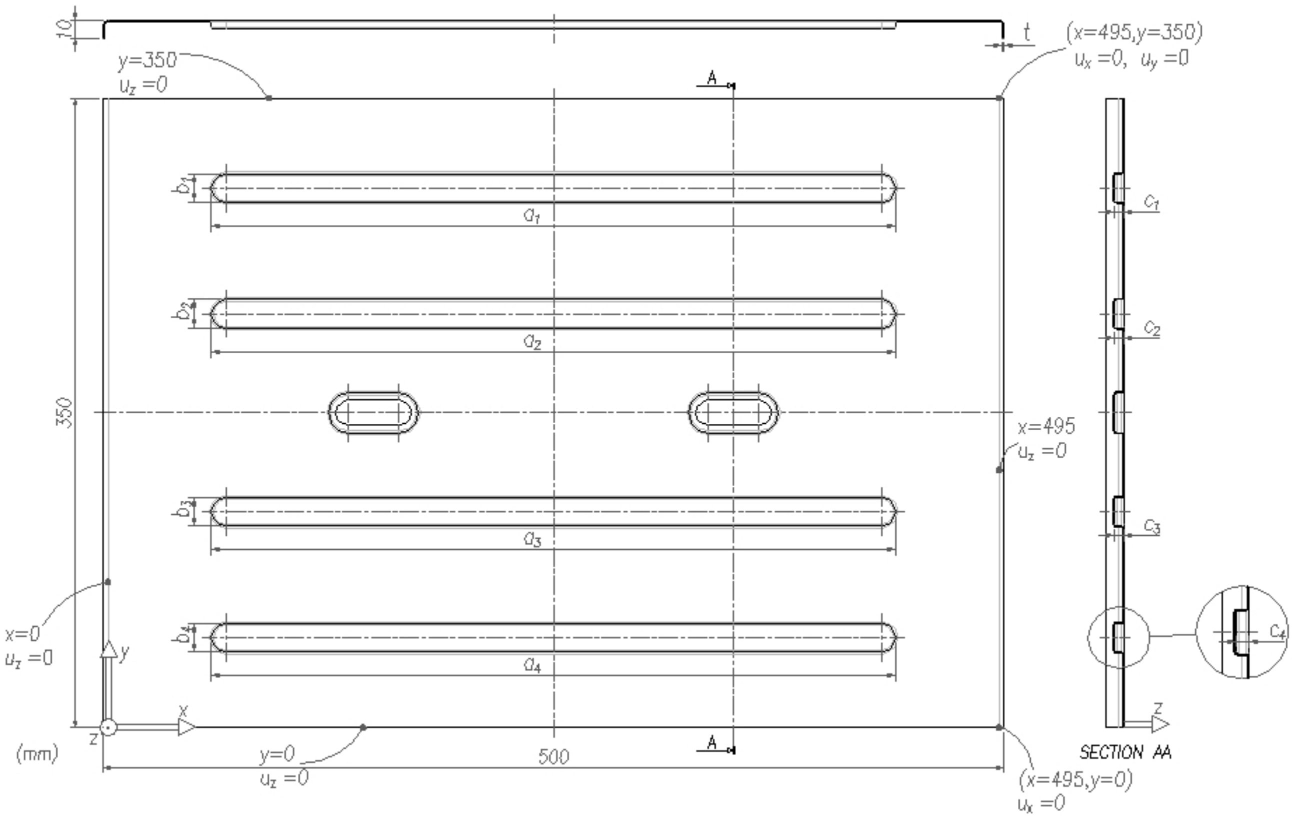

Figure 1. Panel 1, 2D views with design variables depicted and boundary conditions used in the structural analysis.

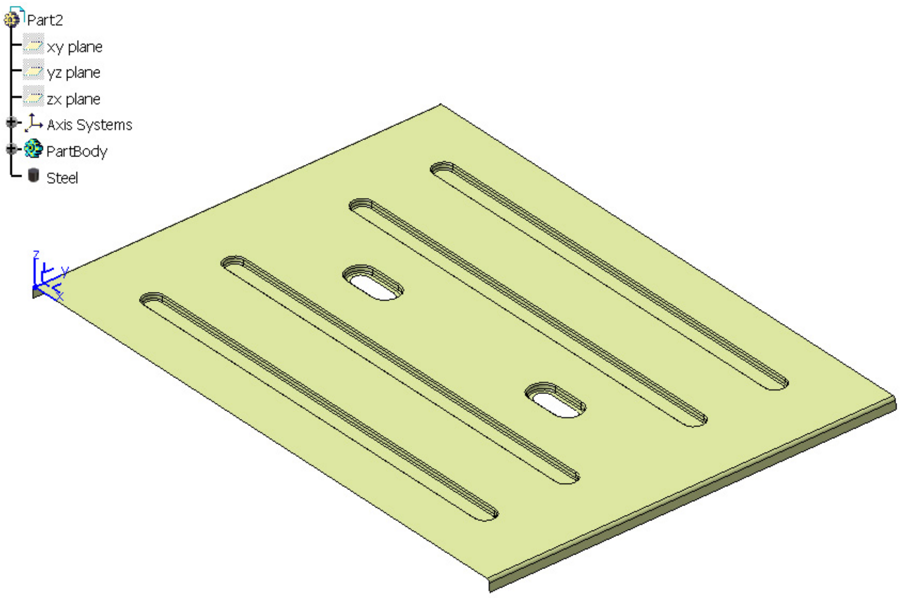

Figure 2. Panel 1, isometric view.

truck boxes and so on. Thus, this is an interesting optimization problem to study due to its great applicability.

Figure 3 shows the CATIA parameters used as design variables in the optimization, all of them related to the cross sectional characteristics of the panel. The dimensions a_i and b_i $(i = 1, 2, 3)$ define the bases of the trapezoidal reinforcements and the dimensions c_i their depths. The parameters d_1 and d_3 are design variables that define the position of the two lateral reinforcements in the cross section. The central reinforcement is kept in a fixed position. The thickness of the panel is also a design variable defined by the parameter t.

In the same Fig. 3 the boundary conditions for the structural analysis are shown. On the edges $x = 0$ and $x =$

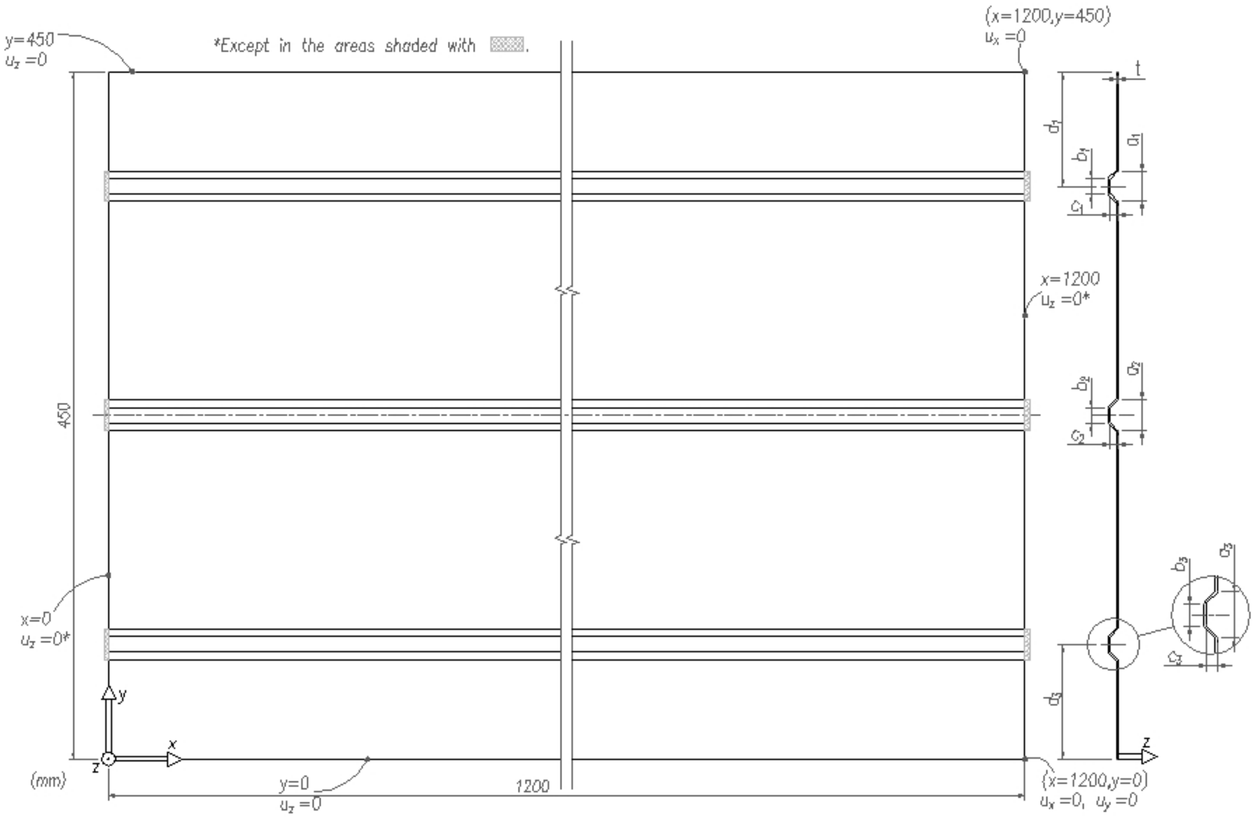

Figure 3. Panel 2, 2D views with design variables depicted and boundary conditions used in the structural analysis.

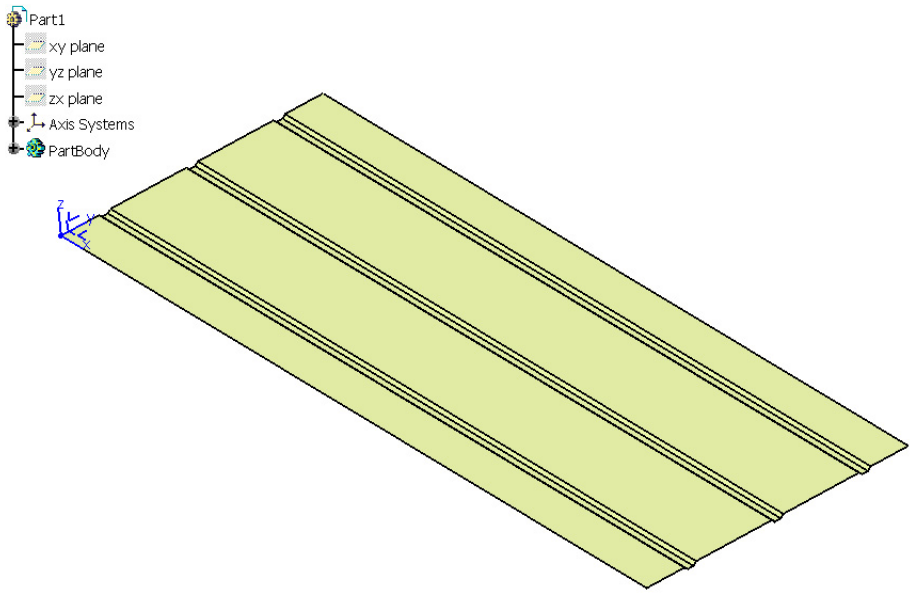

Figure 4. Panel 2, isometric view.

1200 mm, the displacement $u_z = 0$, but this time there are some free regions over the edges, which are shaded as indicated in Fig. 3. The edges $y = 0$ and $y = 450$ mm have $u_z = 0$. Again to avoid rigid body modes, it is imposed $u_x = 0$,$u_y = 0$ at the corner $(x = 1200\ mm, y = 0\ mm)$ and $u_x = 0$ at the corner $(x = 1200\ mm, y = 450\ mm)$. Figure 4 shows an isometric view of the initial design.

3. MESHES UTILIZED IN THE STRUCTURAL ANALYSIS

Typical finite element meshes employed in the natural frequencies analysis are shown in Fig. 5 and Fig. 6. For the Panel 1, the mesh size[1] used in all cases is $3\ mm$, perhaps too small, resulting in a very refined mesh as seen in Fig. 5, with approximately 63,700 elements. This mesh has the following quality parameters according to CATIA: 88.6% of elements of a good quality and 11.4% of elements of a poor quality. For the Panel 2 the mesh size was defined as $10\ mm$, resulting in a less refined mesh, though considered still effective. The mesh quality parameters are 79.3% of good quality elements and 20.7% of poor quality, with the total number of elements being about 11,000.

Both panels were discretized using meshes of linear triangular thick shell finite elements, available in the CATIA finite elements library.

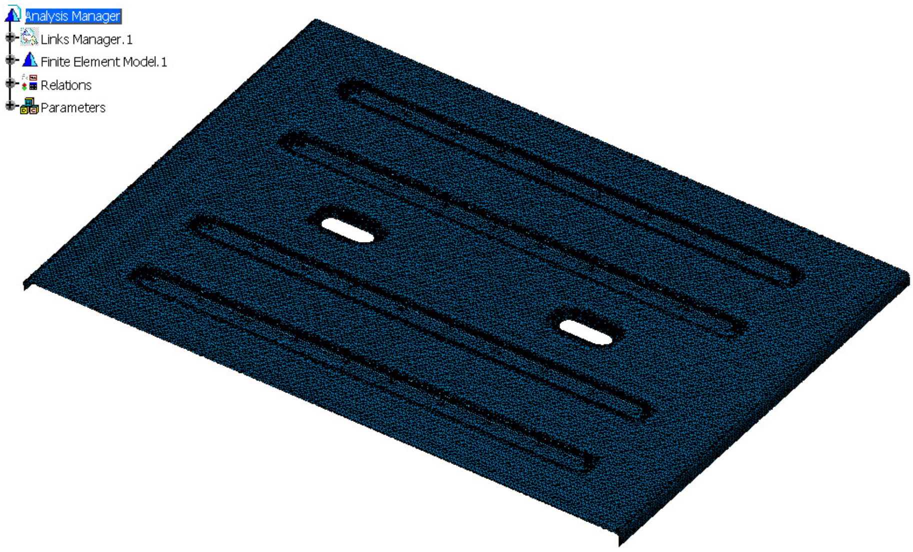

Figure 5. Mesh utilized in the natural frequencies analysis of the Panel 1.

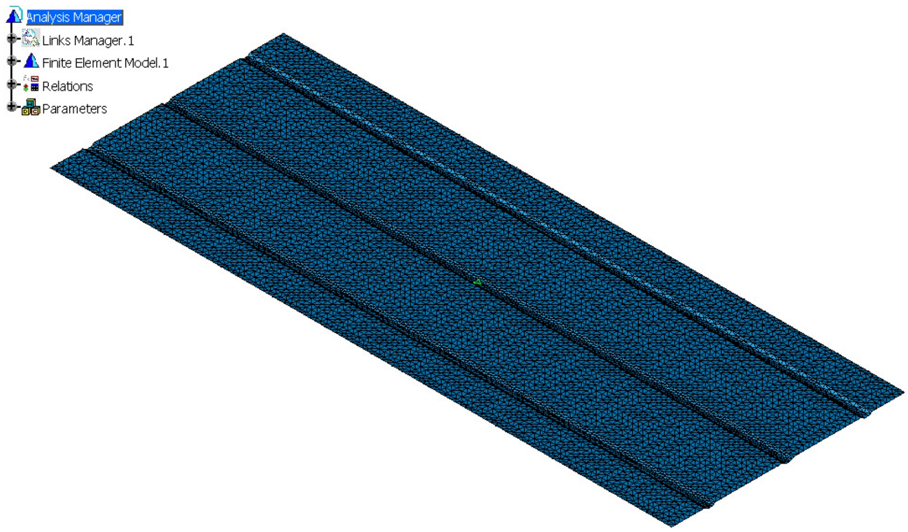

Figure 6. Mesh utilized in the natural frequencies analysis of the Panel 2.

4. OPTIMIZATION PROBLEMS

The structural optimization problems to be solved are the following:

[1] "...*size* is the general size of the longest edge of the finite elements used..." (GAS Docs, 2005)

4.1 Panel 1

Minimize:

$$M(a_1, a_2, a_3, b_1, b_2, b_3, c_1, c_2, c_3, t) \quad (1)$$

Subject to:

$$g_1 = 1 - \frac{\omega_1}{10Hz} \leq 0 \quad (2)$$

Where the objective function M is the mass of the structure; a_1, a_2, a_3, b_1, b_2, b_3, c_1, c_2, c_3, t are the design variables; g_1 is the imposed constraint with ω_1 being the first natural frequency of the structure, coming from the solution of the well-known free vibrations eigenproblem in Eq. 3.

$$[K]\{\phi_i\} = \omega_i^2 [M]\{\phi_i\} \quad (3)$$

Where $[K]$ and $[M]$ are respectively the structural stiffness and mass matrices of the panel, both of order n. The ω_i and the $\{\phi_i\}$ are respectively the i-th natural frequency and the i-th vibration mode shape of the panel, with $i = (1, 2, ..., n)$. The literature is plenty of information about this eigenproblem (Bathe, 1996).

4.2 Panel 2

Minimize:

$$M(a_1, a_2, a_3, b_1, b_2, b_3, c_1, c_2, c_3, d_1, d_3, t) \quad (4)$$

Subject to:

$$g_1 = 1 - \frac{\omega_1}{12Hz} \leq 0 \quad (5)$$

Again, the objective function M is the structural mass; a_1, a_2, a_3, b_1, b_2, b_3, c_1, c_2, c_3, d_1, d_3, t are the design variables; and g_1 is also the lower bound natural frequency constraint for ω_1 , that comes from the solution of an eigenproblem similar to the one in Eq. 3.

5. OPTIMIZATION METHODS EMPLOYED

The optimization problems will be approached by two well-known optimization algorithms, the Simulated Annealing and the Conjugate Gradient (henceforth SA and CG) available in CATIA.

The SA is a zero order method (no need of derivatives of the objective and constraints) that aims to reduce gradually the objective function based in the aleatory generation of a set of design variables and its acceptance or not by a probabilistic choice. Theoretically, this method is able to find global minimum points of functions.

The CG is a method that performs a directional search for points that minimize the the objective function. The directions of search are composed by conjugating gradients of the function being minimized, so this is a first order method, needing first order derivatives of the objective function and constraints. For more details about both methods refer to Haftka and Gürdal (1992).

6. RESULTS

6.1 Panel 1

There are four optimization subcases of Panel 1 with the results obtained presented in Tab. 1. The first subcase has thirteen independent design variables and was solved with the SA algorithm. The results of this run are identified in Tab. 1 as SA$_{ns}$, below the label Final Results. For the second subcase the same problem was ran with the CG method, with results shown in the column labeled with CG$_{ns}$. For the third and fourth subcases geometric symmetry conditions of the reinforcements shapes were imposed to the design variables, such that $a_3 = a_2$, $a_4 = a_1$, $b_3 = b_2$, $b_4 = b_1$, $c_3 = c_2$ and $c_4 = c_1$, reducing the number of independent design variables to seven. These additional runs were labeled as SA$_{sim}$ and CG$_{sim}$, respectively. In the end, the subscript ns means *non-symmetric* and the sim means *symmetric*.

We can see from the results of Tab. 1 that the four runs led to similar final optimal weights for the panel. All the runs had the same initial design whose mass and first natural frequency were $M = 0.760\ Kg$ and $\omega_1 = 26.837\ Hz$, respectively. The mass reduction for all the four runs was equivalent, around 60%. The optimal thickness reached its lower bound ($t = 0.2\ mm$) practically in all the cases, however the variables defining the shape of the reinforcements have considerably distinct values. The optimal shapes obtained in all subcases are illustrated in Fig. 7, from where it can be seen that the resulting designs are quite different.

Table 1. Optimization data and results for runs with Panel 1.

Design (mm)								
Variable	Initial	Range Bounds		Step	Final Results			
		Lower	Upper		SA$_{ns}$	CG$_{ns}$	SA$_{sim}$	CG$_{sim}$
a_1	380	100	420	1	140.453	331.961	102.002	376.250
a_2	380	100	420	1	376.914	420.000	413.465	370.348
a_3	380	100	420	1	333.510	271.249	$= a_2$	$= a_2$
a_4	380	100	420	1	418.171	307.587	$= a_1$	$= a_1$
b_1	16	10	35	0.1	18.315	12.424	35.000	16.781
b_2	16	10	35	0.1	11.052	11.046	10.000	14.623
b_3	16	10	35	0.1	23.089	15.076	$= b_2$	$= b_2$
b_4	16	10	35	0.1	17.340	11.545	$= b_1$	$= b_1$
c_1	5	3	10	0.1	4.809	3.296	3.000	3.636
c_2	5	3	10	0.1	3.000	4.088	3.000	3.415
c_3	5	3	10	0.1	3.000	3.123	$= c_2$	$= c_2$
c_4	5	3	10	0.1	3.000	3.664	$= c_1$	$= c_1$
t	0.5	0.2	1	0.03	0.200	0.201	0.200	0.200
Data								
Parameter	Initial				Final Results			
					SA$_{ns}$	CG$_{ns}$	SA$_{sim}$	CG$_{sim}$
g_1	-1.683				-0.125	-0.182	-0.435	-0.128
$M(Kg)$	0.760				0.294 (-61.4%)	0.297 (-60.9%)	0.291 (-61.7%)	0.297 (-60.9%)
$w_1(Hz)$	26.837				11.249	11.816	14.350	11.278
Total Iterations					200			
Iter. for Convergence					173	194	162	61

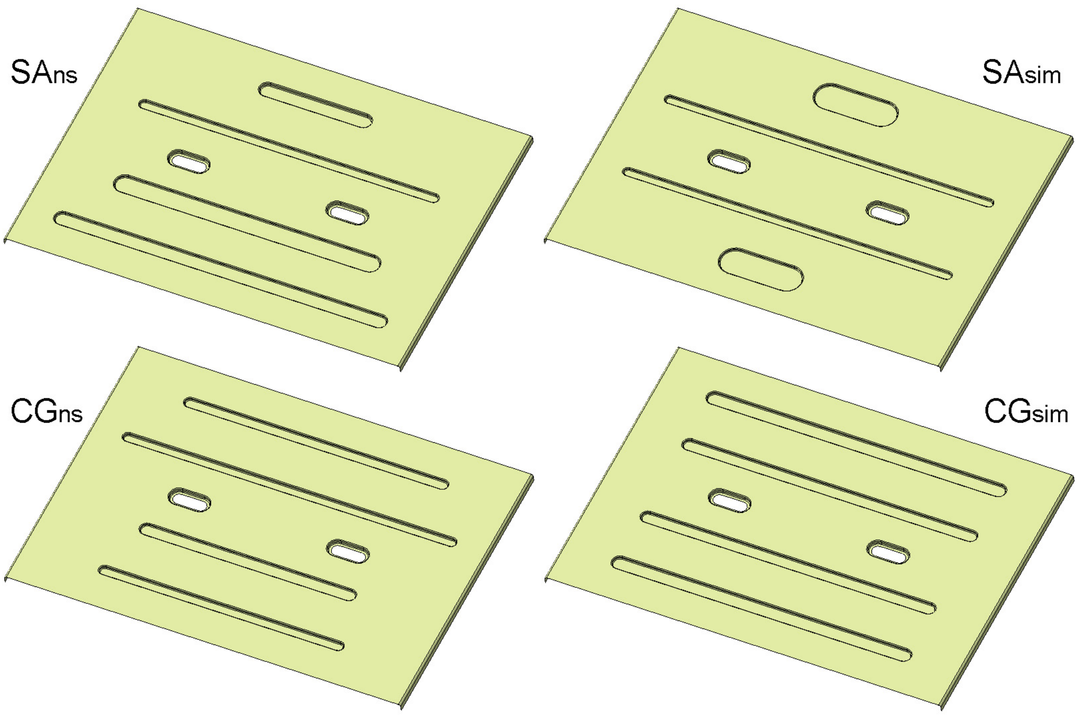

Figure 7. Optimized designs obtained for Panel 1.

Surprisingly the natural frequency constraint is not active for any of the four subcases, however it is closer to become active for the subcases SA$_{ns}$ and CG$_{sim}$. For the subcase SA$_{sim}$ the natural frequency is $\omega_1 = 14.350\ Hz$, corresponding to 43% of feasibility and far from being active.

For all the subcases the same limit of 200 iterations was allowed but not consumed entirely. The symmetric subcase CG$_{sim}$ was less costly in terms of iterations used for convergence (only 61) and perhaps is the best run in terms of optimal design. The run CG$_{ns}$ used 192 iterations to converge to similar results in weight and frequency.

It should be mentioned that analyzing the Panel 1 using the optimal values of the design variables in Tab. 1 for the subcase CG$_{sim}$ but modifying two variables such that $a_1 = 357.4\ mm$ and $a_2 = 351.8\ mm$ (corresponding to 95% of their optimal values) the results obtained are $g_1 = -0.0123$ ($\omega_1 = 10.123\ Hz$) and $M = 0.296\ Kg$. However, the PEO was not able to find such a design that is better than the provided result.

6.2 Panel 2

Table 2. Optimization data and results for runs with Panel 2.

Design (mm)						
Variable	Initial	Range Bounds		Step	Final Results	
		Lower	Upper		SA$_{sim}$	CG$_{sim}$
a_1	20	15	25	0.1	24.618	21.315
a_2	20	15	25	0.1	19.765	21.315
a_3	20	15	25	0.1	$= a_1$	$= a_1$
b_1	10	5	15	0.1	12.445	8.685
b_2	10	5	15	0.1	8.972	8.685
b_3	10	5	15	0.1	$= b_1$	$= b_1$
c_1	5	4	15	0.1	4.000	4.000
c_2	5	4	15	0.1	5.770	4.000
c_3	5	4	15	0.1	$= c_1$	$= c_1$
d_1	75	50	150	0.5	130.095	73.867
d_3	75	50	150	0.5	$= d_1$	$= d_1$
t	0.8	0.4	1	0.01	0.400	0.721
Data						
Parameter	Initial				Final Results	
					SA$_{sim}$	CG$_{sim}$
g_1	-0.187				-0.001	-0.014
$M(Kg)$	1.203				0.598 (-50.3%)	1.072 (-10.9%)
$w_1(Hz)$	14.243				12.008	12.163
Total Iterations					200	
Iter. for convergence					158	36

The Tab. 2 shows the results and data for the two optimization subcases performed with the Panel 2. This time, the design symmetry was imposed from the beginning, by doing $a_3 = a_1$, $c_3 = c_1$ and $d_3 = d_1$. There are ten independent design variables. The subcases were ran respectively with SA and CG methods and these runs were labeled as SA$_{sim}$ and CG$_{sim}$, as is seen in Tab. 2.

Both problems started with $M = 1.203\ Kg$ and $\omega_1 = 14.243\ Hz$. The run SA$_{sim}$ resulted a final mass $M = 0.598\ Kg$, while the run CG$_{sim}$ led to a final mass $M = 1.072\ Kg$, with reductions of 50.3% and 10.9%, respectively. The results obtained for the SA method were superior than for the CG method, that probably found a local optima. In both cases the frequency constraint is active, resulting in $\omega_1 = 12.008\ Hz$ for SA$_{sim}$ and $\omega_1 = 12.163\ Hz$ for CG$_{sim}$ with respectively, $g_1 = -0.001$ and $g_1 = -0.014$.

The Figure 8 illustrates the optimal designs obtained.

It should be mentioned that a finer mesh (with size $5\ mm$) was tried for the CG subcase, trying to avoid eventual problems with finite differences gradient calculations that could exist with the finite element mesh used so far. However the mass converged to practically the same design. That is another indicative that this solution is perhaps a local minimum where the CG got trapped, while the SA, being a global minimum finder, could avoid.

7. CONCLUSIONS

The results obtained with the simulations showed that PEO could lead to encouraging optimal design results for Panel 2 with the SA optimizer, while the CG result led to a much heavier optimal, though with the good characteristic of an active frequency constraint. Remains to be seen whether this CG solution is a local optimum or just a failure of the PEO. However, for the Panel 1 the frequency constraint was not active in any of the optimal solutions, but nearly active in three of the four subcases. The authors suspect that this is due to the presence of variables defining the longitudinal lengths of reinforcements in Panel 1, since Panel 2 had better results with variables defining only thickness and cross section dimensions, but not reinforcement lengths. These issues are to be clarified in future continuing experiments with CATIA

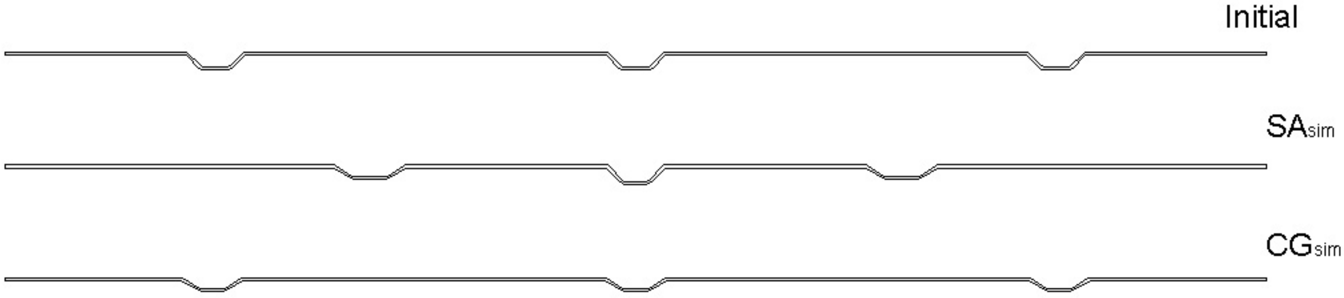

Figure 8. Optimized designs obtained for Panel 2.

in similar panel optimization problems.

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