OPTIMAL DESIGN OF A SIMULATOR MODULE FRAME

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ABSTRACT

The support frame of a simulator module for the entertainment industry is designed to meet stiffness and strength conditions while minimizing the weight. As such, it represents a straightforward design optimization task with the novelty associated with the nonconventional vehicle that is being designed. A simple frame structure is used for the design. MSC/NASTRAN's ability to synthesize property values allows considerable generality in the specification of cross-sectional dimensions. This paper presents the design concept, the loading conditions, and the design constraints. The MSC/NASTRAN model is discussed, and the results of the the optimization task are given. The initial arbitrary design was infeasible, with a 33 percent weight increase necessary to achieve the final optimal design.

INTRODUCTION

A simulator module for the entertainment industry is shown in Figure 1. The module is approximately 8 feet long and 5 feet in diameter at its base. Intended for a single occupant, it is supported at the rear on a hub that allows the module to roll freely. The hub, in turn, is supported on an articulated arm moved by hydraulic actuators, that translates the module in plunge (vertical translation) and pitch. The interior of the module resembles the cockpit of a high performance fighter-bomber, with a sidestick controller, a throttle, and buttons for arming and launching ordnance. Visual cues are provided by a video camera, installed below the seat, that projects a simulated cockpit view on a screen in front of the occupant. The screen video and simulator motions, including sound effects and external disturbances or threats (such as gusts or flak), are provided and controlled by a Silicon Graphics computer. The simulator is designed to produce significant vertical accelerations, so the reality of the cues experienced by the occupant is limited only by the sophistication of the software and the need to simulate forces in six degrees of freedom with only three. The prototype simulator is set up to simulate an FA-18 mission, starting from launch from the deck of a carrier. The mission is to find and destroy "enemy" ground installations just onshore. The video display replicates the view out of the cockpit plus the standard head-up display. Flak is encountered as the target is approached. Various skill levels that control the response of the simulator to pilot inputs can be selected in advance. Other missions, including air-to-air combat, and different vehicles can be simulated with software changes.

Figure 1. The Prototype Magic Edge Flight Simulator.

The internal frame that supports the simulator and the occupant must meet design criteria that are very familiar to anyone in the aerospace industry. It must be light, since unnecessary weight penalizes performance. It must be stiff in order to preserve the quality of the video path. Since it will be used by the public, it must have the strength margins dictated by the State building codes. There are a number of loading conditions, since the module can accelerate significantly at any point within its movement envelope. Meeting these criteria has all the ingredients of an interesting optimal design problem and that is the subject of this paper. To be precise, we shall examine the redesign of the prototype frame.

PROBLEM DEFINITION

An exploded view of the prototype frame is given in Figure 2. The frame loads are transmitted to the hub through bearings at the rear hex(1) and the front hex(3). Reinforcements referred to as "arrows" (2) are located where the trusses (4) attach to the hex weldment. The frame is completed by a shear panel located above the video path and below the occupant's seat. In the redesign, we will be concerned with the labeled components. They are all cut from sheet steel, so the design variables are the widths of the individual members. Two different sheet metal thicknesses are used: 0.625 inches for the trusses and hex structure and 0.75 inches for the arrows.

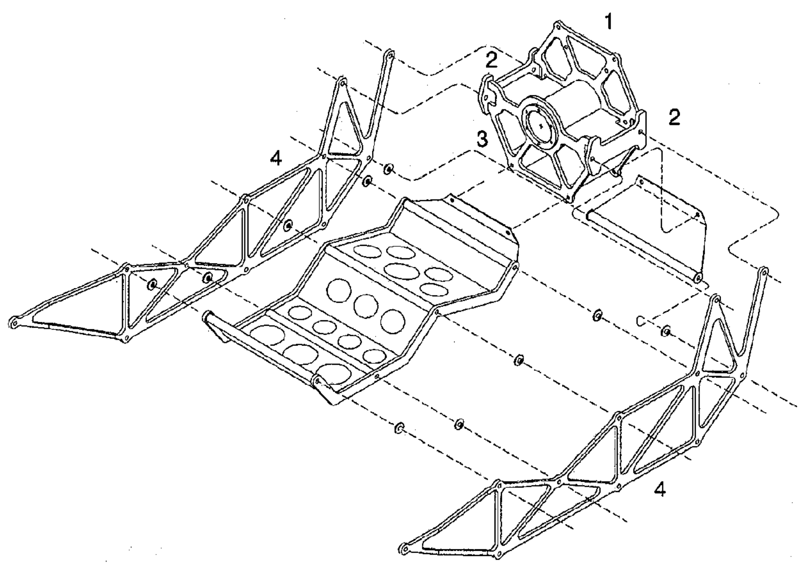

Figure 2. Exploded View of the Simulator Frame.

The original structural analysis was performed using NISA, Version 93. A preliminary design was analyzed for a single load condition and then resized, where necessary, while assuming that the internal loads would remain constant. When a re-analysis was performed and additional load cases were added, it was found that the resized structure did not meet the stringent strength requirements and further resizing was required. Clearly, optimization procedures would provide a solution to the design task. The optimization capability in NISA was not available, and attempts to apply ASTROS, Version 10 were not successful due to a the lack of a general beam sizing capability in that code. The optimization capability of Version 68 of MSC/NASTRAN became available and was used for this study.

If the finite element model had used solid elements to represent the truss cross–sections, the design task could have been posed as one of finding the optimal shape optimization. However, standard one-dimensional elements were used so that a sizing optimization task was appropriate. With the sheet metal thicknesses designated as h_f, the bar properties and stress recovery points are designed using the following relations in terms of the design variables, which are the widths of the bars:

$$Area_i = h_f x_i$$

$$I_{1i} = h_f x_i^3/12.0$$

$$I_{2i} = h_f^3 x_i/12.0$$

$$J_i = 0.3(h_f x_i)^3/(x_i^2 + h_{f_2})$$

$$C_{1i} = 0.5x_i$$

$$D_{1i} = -0.5x_i$$

$$E_{1i} = 0.5x_i$$

$$F_{1i} = -0.5x_i$$

The motion envelope comprises 5 feet of vertical translation, pitch from 45 degrees nose up to 24 degrees nose down and continuous roll through 360 degrees. The acceleration limits are 2.25 g's for roll from 0 degrees to 45 degrees, 1.5 g's for roll to 90 degrees, and 1.0 g for roll to 180 for any pitch angle. The occupant is assumed to weigh 250 pounds. Since the frame structure includes weldments, the stress limits are rather severe for a frame constructed entirely of steel: 2500 psi in tension or compression and 1250 psi for shear. These stress limits were incorporated into the optimal design problem as constraints of 2500 psi for the von Mises stresses based on the loading conditions given in Table 1.

Table 1. Frame Loading Conditions.

ACCELERATION (G)	ROLL (DEG)	PITCH (DEG)
2.25	0	0
2.25	45	0
1.50	90	0
2.25	45	45

The frame finite element model is illustrated in Figure 3. The model is made up of beams and quadrilateral plate elements. There are 86 grids and 179 elements. The open square symbols at some of the grids identify points where lumped inertias representing the nonstructural portion of the model (shell, occupant, equipment, etc.) were placed. The model is constrained at the nodes where the bearing fits over the hub, so hub and arm flexibility are ignored. There are 23 design variables: 17 for the trusses, 3 for the rear hex, 1 for the arrows, and 2 for the front hex. Video path rigidity was enforced by constraining the deflections at the tips of the frames to lie within spheres of 0.050 inch radius. Upper and lower bound side constraints were also imposed on the design variables. Frame bilateral symmetry was enforced through design variable linking.

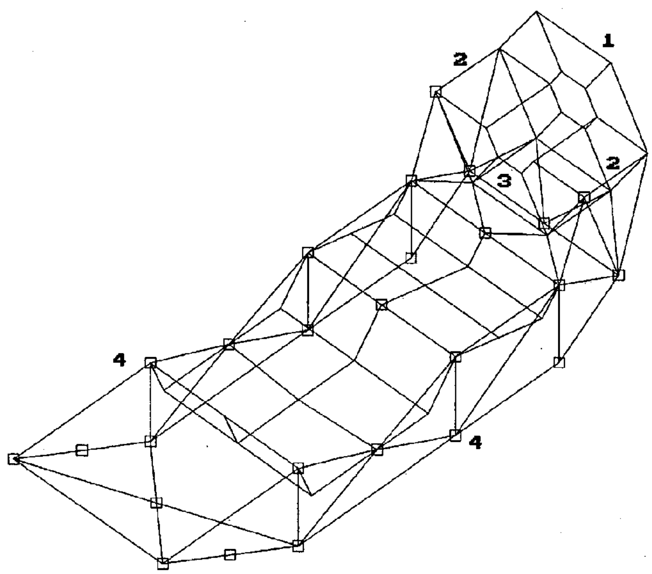

Figure 3. Finite Element Model.

ANALYSIS

Version 68 of MSC/NASTRAN was used to perform the design studies. The only challenge proved to be the specification of the upper bounds on the design variable values. The initial design was infeasible and no feasible design was obtainable with the initial design variable constraints. After adjusting the upper bounds, the optimal design entailed a weight increase of 118 pounds, which represents a 33% weight increase over the initial, arbitrary design which has stresses that are as much as 250% higher than the allowable value. Figure 4 depicts the design history for the objective function and the maximum constraint value, indicating that the design task entailed overcoming the stress limits for the first eight design cycles, following which, the solution converged rapidly to its final design. The thickness of the members in the front hex and the truss members connecting to the front hex were substantially increased, as were certain members near the middle of the trusses where the occupant seat is supported. The thicknesses of the arrows and the upper members of the rear hex were decreased. Note that the weight data given above do not include the inactive and nonstructural components, which add some 620 pounds to the system.

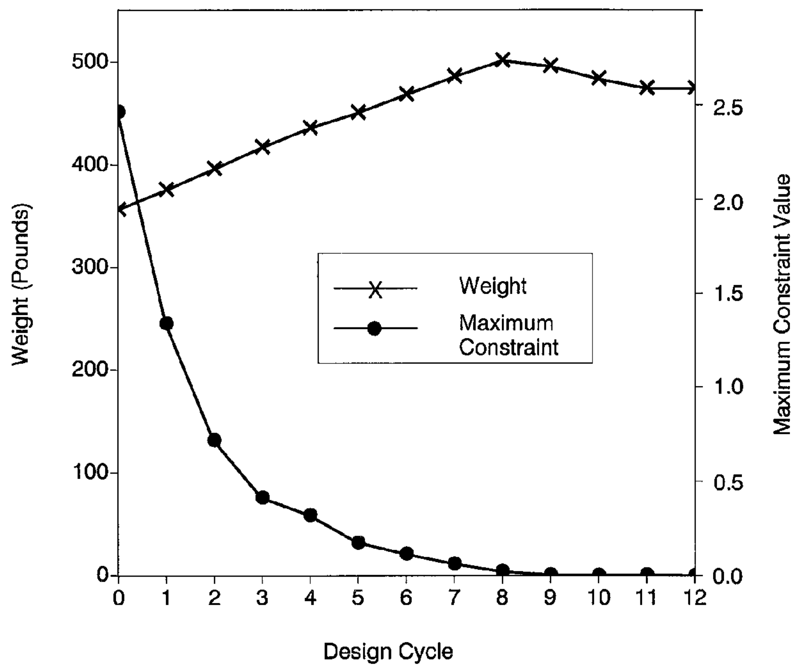

Figure 4. Design Cycle History.

DISCUSSION

In some respects, these results serve as a graphic reminder of the often counterintuitive paths that formal optimization procedures take. A simple stress ratioing procedure would not have been successful in this case since modifications in the beam section properties produce changes in the load paths. Although this design cannot be viewed as a finished product, it has certainly served to identify the areas of the frame that need attention. A more detailed finite element analysis that replaced the simple beam structure with a model using plate and solid elements was subsequently used in the frame redesign for production. This analysis confirmed our identification of the problem areas.

ACKNOWLEDGMENT

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