Engineering Notes

Dynamic Characterization of an Inflatable Concentrator for Solar Thermal Propulsion

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Introduction

SOLAR thermal propulsion has received considerable attention in recent years as an economical means of enabling low-thrust orbital transfer and perhaps deep space missions. The basic concept behind solar thermal propulsion is to utilize solar energy as a means of heating a working fluid (propellant) to provide thrust at increased specific impulse. As described in Ref. 1, thrust is produced by expanding the heated propellant through a nozzle. No combustion occurs, and the thrust level is low. For this reason, solar thermal propulsive systems are mainly applicable for orbital transfer vehicles and relatively long-term space missions.

Inflatable structures, which have been investigated in detail in recent years, have characteristics that are particularly advantageous for solar thermal propulsion systems. First, inflatables are extremely lightweight, which makes them an ideal match for use with low-thrust solar thermal rockets, where vehicle weight is at a premium. An obvious second advantage is onorbit deployability and related space savings in the launch configuration.

A solar thermal upper stage has been envisioned as an orbital transfer vehicle that could be developed and flown in the near future. Two inflatable/rigidizable parabolic collectors could be rotated and gymballed for focusing sunlight into an absorber cavity (Fig. 1). The Shooting Star Experiment, which also included an inflatable solar concentrator structure in its design, was conceived as a precursor mission for a future solar thermal upper stage.

The purpose of this Technical Note is to describe dynamic characterization of a prototype inflatable solar concentrator that was developed for the Shooting Star Experiment. Modeling and test activities were complicated by the nonlinear nature of the inflatable, with modulus varying as a function of frequency and level of excitation. This work is highly significant because of the interest in inflatable structures for space applications and because of the difficulty in accurately modeling such systems.

Modal Testing of the Inflatable Concentrator

Modal tests were performed for the prototype concentrator assembly in atmospheric conditions.[2] As shown in Fig. 2, the prototype concentrator in the test configuration was attached to an aluminum plate at the top of the structure, which represented the mounting

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fixture for the solar-thermal engine. Three tapered struts were attached to the aluminum plate, with diameter varying from 17.4 cm (6.8 in.) at the top to 10.2 cm (4 in.) at the strut/torus intersections. The struts were approximately 1.83 m (72 in.) in length and were constructed of Kapton® (100) HN film with an average thickness of 51 μm (2 mil). The torus had 15.2 cm (6 in.) diameter cross section and outside overall diameter of 1.83 m (72 in.), with film thickness of 46 μm (1.8 mil). It was constructed from Kapton 300-JP film. The lens was simulated with a polyethylene sheet attached to the inner edge or flange of the torus.

The system was hung from three bungee cords for a free–free test. Excitation was provided with a shaker attached to the support plate at the top of the inflatable assembly, and lightweight accelerometers

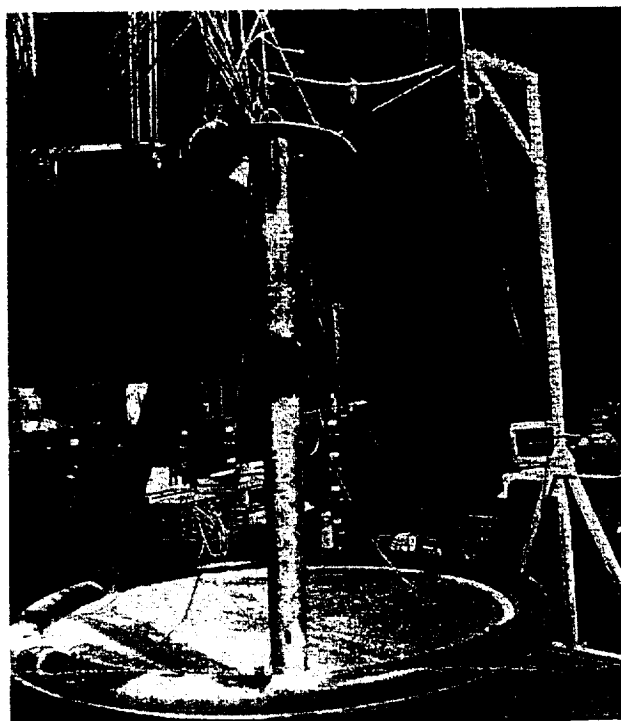Fig. 1 Solar-thermal upper stage concept utilizing inflatable concentrators.

Fig. 2 Shooting star experiment prototype concentrator in modal test configuration.

mounted on the film surface were used for response measurements. Modal tests were run at three different inflation pressures, 1.72, 3.45, and 6.89 kPa (0.25, 0.50, and 1.0 psig). Mode shapes, frequencies, and damping characteristics are described in detail in Ref. 2.

Dynamic Modeling and Comparison to Test

Geometric-nonlinear pressurized shell models were developed for comparison to test results. Considerable effort was made to include air pressure realistically in the model because it was known that pressure and the pretensioning of the films had significant impact on dynamics of the structure. A two-step approach was taken using the MSC/NASTRAN finite element code. In the first step, a nonlinear static pressurization resulted in an updated stiffness matrix that accounted for the large deformations of the film. This updated stiffness matrix was then imported into an eigensolution procedure. Reference 3 describes this modeling approach in more detail.

An effort was made to include as much detail as possible in the models to match closely the test article. Varying thickness and modulus of the glued joints, and masses of the accelerometers, were incorporated. Also, flanges created by the overlapping material due to the joining of the torus halves were included in some of the models. These refinements were done because it was found during development of a simplified beam model that including these seemingly minor details had the potential to change the behavior of the model significantly. Figure 3 shows the concentrator full assembly finite element model.

A building-block approach was used to verify the shell models by creating parts of the concentrator assembly model and comparing those component models with previous test results or other models. Initially, a suspended system model consisting of three pressurized struts, support system bungees, and the aluminum support plate was developed. The tapered strut modes and frequencies were compared to test data and a model for a nontapered 15.2-cm- (6-in.-) diam strut. The comparison provided confidence in the accuracy of the three-strut assembly model.
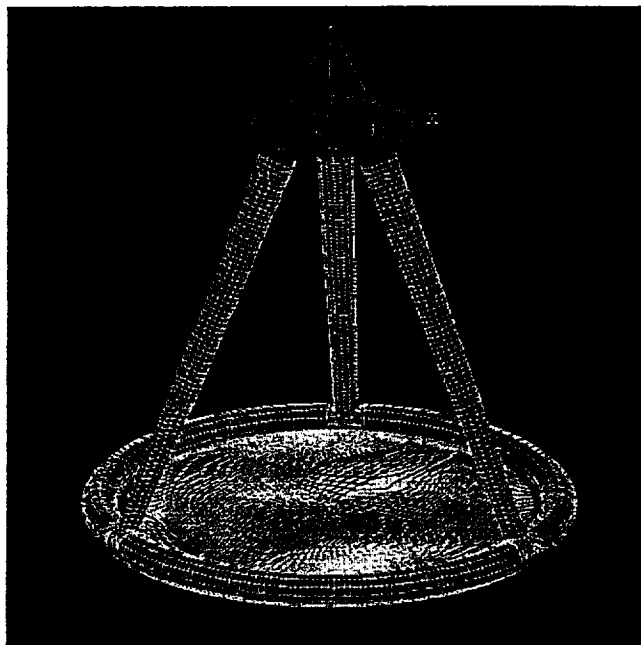

Fig. 3 Full model of solar concentrator assembly with lens and joining flanges.

Shell models of the torus element independent of the support struts and lens simulator were also compared to existing University of Kentucky test data and models.[4] Initially, a torus model without the joining flanges was developed to avoid localized flange modes. The mass of the flanges was distributed on the torus, but flange stiffness was not taken into account. Torus film properties for modulus, Poisson's ratio, density, and thickness, were $E = 2.55$ GPa (370 ksi), $\nu = 0.34$, $\rho = 1.42$ g/cm^3 (0.0513 lb/in.^{3}), and $t = 46$ μm (1.8 mil), respectively. Internal pressure was 5.52 kPa (0.8 psig). Figure 4 shows the first in-plane bending mode as determined by visual comparison with the test data of Ref. 4. As shown in Table 1, the model in-plane modes were very close to test results, but the out-of-plane frequencies were considerably high.

Because of the likelihood that the frequency discrepancies for the simplified torus model were largely due to the missing flange stiffnesses, a torus model including these joints was compared to the test data. Joints were represented with shell elements of the appropriate thickness and with material properties based on averages of film and epoxy properties. As shown in Table 1, the model with flange joints had frequencies higher than test for both in-plane and out-of-plane bending, but the model did capture the behavior of inplane modes having higher frequency than out-of-plane modes. The great difficulty with such a model is that numerous local modes of the joint flanges obscure the torus global modes of interest. Table 1 also shows frequency results for the University of Kentucky torus model, and Ref. 4 provides details of the torus geometry, including the joining flanges. Note that mass of internal air was not accounted for in any of the torus models listed in Table 1.

Pressurization and modal analyses of a simplified torus/strut assembly model (Fig. 5) without the joint flanges and lens simulator were also accomplished. Film properties for the struts were the same as for the torus film described earlier in this section, with the exception that the film thickness was 51 μm (2 mil). Torus film properties for the assembly model were $E = 2.79$ GPa (405 ksi), $\nu = 0.34$, $\rho = 1.4$ g/cm^3 (0.0506 lb/in.^{3}), and $t = 46$ μm (1.8 mil). Internal air pressure for the assembly was 3.45 kPa (0.5 psig). Accelerometers were represented by concentrated masses. Frequency results for the assembly model were comparable for some modes to the full-assembly modal test, as shown in Table 2. It is recognized that the lens and flanges contribute significant mass and stiffness to the concentrator assembly (Figs. 2 and 3). However, the stiffness effect was primarily from the flanges because the lens simulator was attached very loosely in the test configuration. Close examination of the concentrator assembly model results in Table 2 and Fig. 5 shows that the inflated struts and the suspension system were modeled well, but the torus model is obviously less accurate due to omission of the joint flanges and lens simulator. These results and those in Table 1 point out the importance of including details such as joints, adhesive layers, and even loosely fitting membrane components in the models of inflatable structures. Furthermore, the mass of internal air was not accounted for in any of the models, and better results should be expected if it were included.

Note that considerable difficulty was encountered in modeling the suspension system and its interaction with the inflatable structure. For this reason, it is recommended that constrained-boundary tests be done for structures of this type when possible. A better test configuration for the concentrator described in this Note would have the aluminum support plate fixed to the floor or a stiff overhead fixture. Finally, a better design for the torus would eliminate the outer (free-edge) joining flanges. The highly flexible, free-edge flanges added considerable difficulty in identifying the global modes for the model, due to large numbers of local flange modes.

Table 1 Comparison of analytical and test frequencies for torus element, 5.52 kPa (0.8 psig)

Mode description	Ref. 6 test frequency, Hz	Model (no flanges) frequency, Hz	Model (flanges) frequency, Hz	Ref. 6 model frequency, Hz
First in-plane	15.80	16.02	24.17	20.10
First out-of-plane	13.00	20.91	17.82	14.30
Second in-plane	40.10	38.96	——	47.50
Second out-of-plane	30.60	44.10	——	40.10

Table 2 Comparison of simplified model and test frequencies for concentrator assembly, 3.45 kPa (0.5 psig)

| Major mode description	Test frequency, Hz	Model (no lens/flanges) frequency, Hz
First pendulum and weak strut bending	1.48	1.41
Second pendulum and weak strut bending	4.74	4.35
Torsion	6.29	4.26, 8.43
First coupled strut and torus bending	8.42	16.17
Second coupled strut and torus bending	9.66	19.47
First coupled strut bending and plate pitching	10.39	10.56
Second coupled strut bending and plate pitching	15.23	10.60
Pure strut bending	29.66	30.94

See revised Fig. 4 on the enclosed CD.
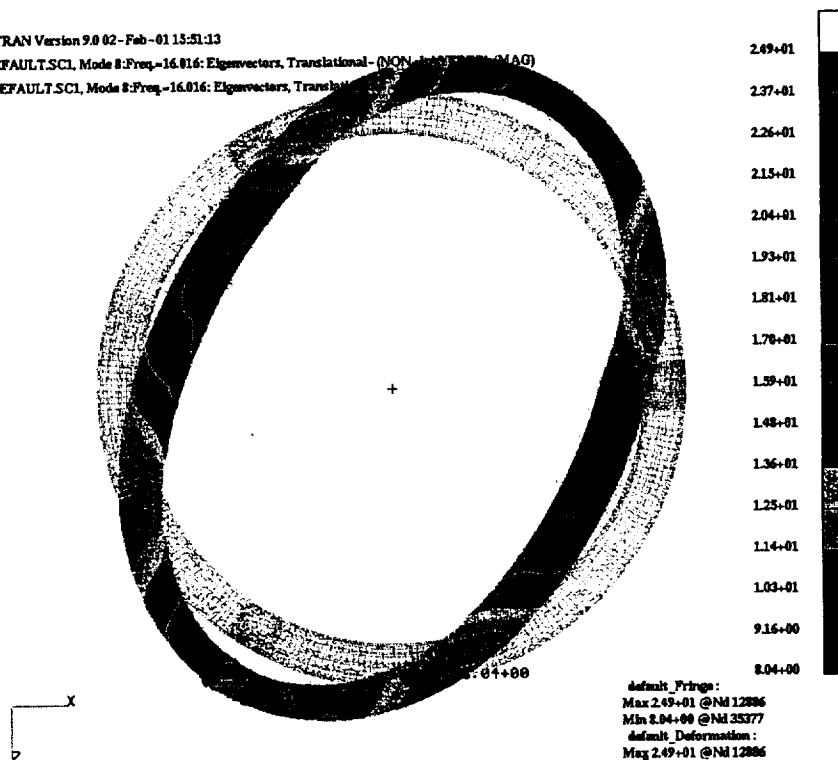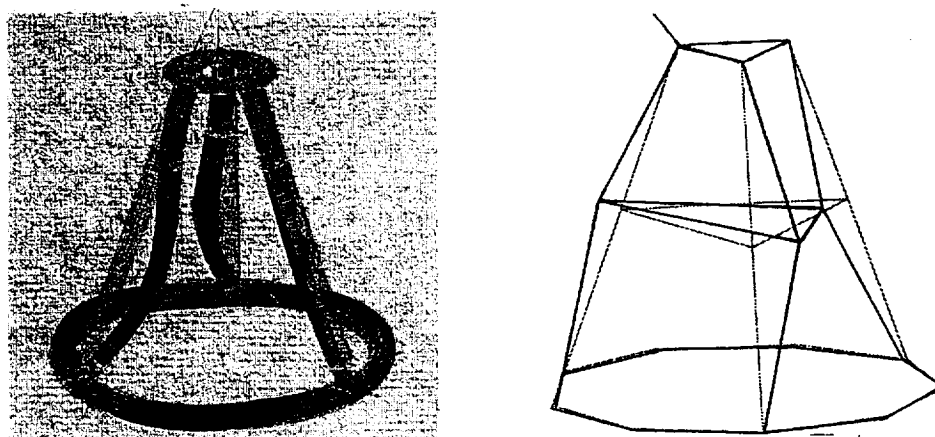
Fig. 4 Fundamental in-plane mode of torus.

Fig. 5 Comparison of strut bending modes for simplified assembly model (30.94 Hz) and test (29.66 Hz).

Conclusions

Solar-thermal propulsion is a concept for producing thrust sufficient for orbital transfers and requires innovative, lightweight structures. This Note presents a description of an inflatable concentrator that consists of a torus, lens simulator, and three tapered struts. Modal testing was discussed for characterization and verification of the solar concentrator assembly. Finite element shell models of the concentrator were developed using a two-step nonlinear approach, and results were compared to test data. Reasonable model-to-test agreement was achieved for the torus, and results for the concentrator assembly were comparable to the test for several modes.

References

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[2]Lassiter, J., "Shooting Star Experiment Prototype Inflatable Strut/Torus Assembly Modal Survey," NASA Marshall Space Flight Center, Rept. ED73(97-69), Huntsville, AL, June 1997.

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