AIAA JOURNAL
Vol. 44, No. 11, November 2006

Topology Optimization for an Evolutionary Design of a Thermal Protection System

Ravi C. Penmetsa* and Ramana V. Grandhi†
Wright State University, Dayton, Ohio 45435
and
Mark Haney‡
Air Vehicles Directorate, Wright–Patterson Air Force Base, Ohio 45433

DOI: 10.2514/1.15906

An optimal thermal protection system design for a spacecraft operating in extreme environments of thermal and acoustic loading is of significant importance for today's military space missions. These military space missions tend to push the envelope of the spacecraft's capabilities to the extreme, requiring robust performance of all key systems to assure a successful mission. Because thermal protection system protects the entire spacecraft, its survival from these extreme conditions is critical to the safety of the mission. The design criteria for the thermal and acoustic loading conditions tend to be conflicting. To reduce thermal stress, free boundaries that allow for expansion of the thermal protection system are desirable. However, the random sound pressure level fluctuations associated with, for example, engine noise during ascent, subject the thermal protection system to a wide-band random excitation. Therefore, if the thermal protection system possesses low frequency modes, damaging strains could result. To limit the magnitude of the strains, higher frequency (stiffer) designs with fixity at the boundary are desirable. Therefore, when designing the thermal protection system within these two simultaneous operating environments, the safe operating region is often severely restricted. This research aims at evolving a design that satisfies these two design requirements.

Introduction

REENTRY is a process that a spacecraft must undergo to safely reenter the Earth's atmosphere from a low Earth orbit (LEO). The magnitude of velocity in LEO is approximately 17,000 mph: much too fast for a modern space vehicle without an active cooling system to move through dense air and survive. When a space vehicle reenters the atmosphere from LEO, it hits the extreme fringes of the atmosphere broadside, using friction (or drag) to slow the vehicle. The descent of the space vehicle brings it deeper into the thicker atmosphere, increasing the vehicle's rate of deceleration as well as the amount of heat that is generated. One purpose of reentry is to dissipate most of the kinetic energy as heat to safely land the space vehicle. There are, however, limitations to the rate at which this energy conversion can take place. These limitations are imposed by the thermal protection system's (TPS) ability to maintain the maximum operating temperatures below the allowable values. This constraint places severe limitations on the path that can be taken by a space operational vehicle (SOV), such as a space shuttle or a reusable launch vehicle (RLV). For example, the space shuttle's path is considered as an extremely benign trajectory, limiting its cross-range and military usefulness.

For a space vehicle, the thermal protection system is as crucial as avionics, propulsion, and structure. The question of "how much heat can the vehicle tolerate?" dictates the vehicle's maximum speed at any place in the atmosphere, and that in turn dictates the point in LEO where the deorbit burn is initiated. In many ways, a vehicle's TPS is more limiting than its fuel constraints, its structural strength, or its engine's maximum thrust. The vehicle and the crew may be able to survive 6 g deceleration, but if the limit of the TPS is 1.5 g, this more restrictive constraint must be imposed to ensure survivability of the vehicle and the crew. Therefore, the selection of a TPS is integral to the design of a spacecraft.

A TPS (Fig. 1) is, in itself, an entire system that consists of metallic/composite plates, insulation, fasteners, and supports. It completely protects the space vehicle. Generally, a TPS consists of an insulating cover that attaches to the vehicle's load-bearing structure either directly or with joints. The cover can be either flexible blankets or shaped tiles. However, at certain key locations on the vehicle, particularly the leading edges, heat and pressure become so intense that tiles would provide insufficient protection. In these places, a strong, very-high-temperature-resistant composite is integrated directly into the aerodynamic structure. These hot surfaces transmit lift to the fuselage and wings exactly as a traditional leading edge would in addition to taking the brunt of the heat flux during reentry. A TPS is used to actively cool and/or passively shield the primary structure of a spacecraft or high-speed aircraft from temperature extremes that would otherwise prohibit operation. Figure 1 is representative of insulating TPS panels used to protect the primary structure of an SOV. Excluding leading edges, most current TPS concepts are parasitic in nature (i.e., they do not add significant structural integrity to the structure); therefore, they must be lightweight so as not to adversely impact the mission requirements.

A major design consideration is the thermal stresses that result from large temperature gradients through the thickness of the TPS. A "floating" design, which allows for expansion, can dramatically reduce the load due to thermal stress. However, the combination of using lightweight constructions and "loose fitting" attachments makes the designs susceptible to acoustic excitation. Hence, the feasible design space can be very small or nonexistent for certain classes of material systems. Each SOV concept consists of multiple types of TPS. Different environments are encountered on different parts of the vehicle, with windward leading edges being the most severe and the leeward skin being the most benign. The goal is to use the lightest protection covering that will survive the environment. Every pound of TPS decreases the payload of the vehicle, thereby increasing the cost of payload delivery. However, the military uses of an SOV are more demanding than that of the space shuttle. Quicker

Received 7 February 2005; revision received 8 May 2006; accepted for publication 2 June 2006.

*Assistant Professor, Department of Mechanical and Materials Engineering; ravi.penmetsa@wright.edu. Member AIAA.

†University Professor, Department of Mechanical and Materials Engineering; ramana.grandhi@wright.edu. Fellow AIAA.

‡Aerospace Engineer, Structural Sciences Center; Mark.Haney@wpafb.af.mil.

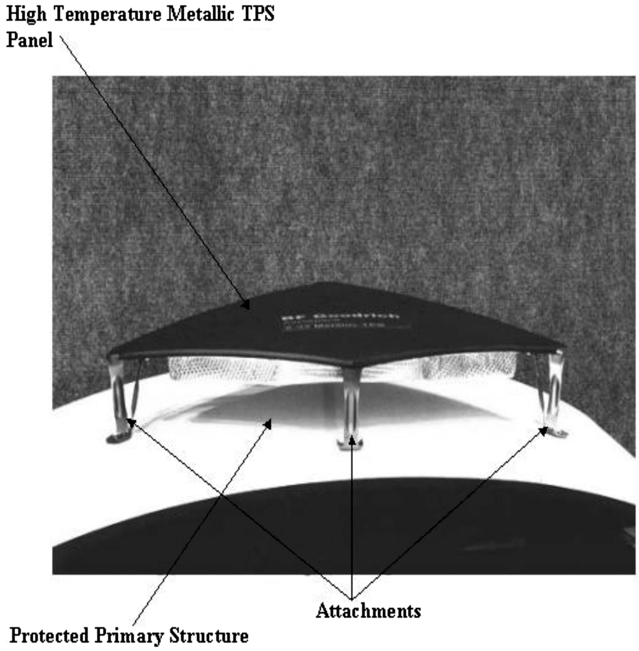

Fig. 1 Thermal protection system (courtesy of Cast et al. [8]).

turnaround times, all-weather operation, and much larger cross-range are just a few of the challenges that must be confronted. TPS is one of the key technology areas that must be developed to reach these goals.

Various material systems are employed in the manufacture of thermal protection systems. Much attention has been given to ceramic-matrix composites (CMC) in recent years due to their high temperature characteristics as well as their capacity for dissipating internal energy as heat. But metallic TPS, due to its well-characterized material properties and durability, is still useful in low to moderately severe environments. One challenge that is ever-present when dealing with new material systems is the uncertainty about their characteristics. Properties can vary widely across different vendors and even different batches of production. If these promising materials are to be successfully employed, the inherent variability in material properties, geometric sizes, boundary conditions, operating loads, etc., must be quantified and controlled.

In this work, the TPS design is posed as an optimization problem to satisfy the stress and frequency requirements for the system. This optimization problem is a multifold process, integrating topology design and sizing of the TPS. The topology design itself is a multidisciplinary optimization process integrating the heat transfer analysis in addition to the static and normal mode analyses. Topology optimization has been used by many researchers [1–4] to determine the structural configuration that meets a predefined criterion. This scheme is capable of generating completely new and innovative designs that are inconceivable by the designer. These approaches can be divided into two categories: one is where various elements are added and removed in each iteration, and the other is where the densities of the elements are modified to obtain a final configuration. Topology optimization routine in the commercial program GENESIS [5] uses the latter approach, where the density of each of the elements is modeled as a design variable.

In topology optimization, the material density of each element is used as a design variable to minimize the objective function, subject to the behavior constraints. An element that has a density value below a threshold would be considered removable from the final structural configuration without lowering its performance. At the end of the topology optimization process, these elements with low density will be removed and a solid model is created based on the information provided by the remaining elements. The density information obtained from the topology optimization iteration is used to scale the material properties in order to simulate the removal of material from the baseline model. Using these modified properties, a heat transfer analysis is performed to determine the new temperature distribution. This iterative process of material removal and temperature redistribution is carried out until a converged solution for the optimum density distribution is obtained. This process is automated by combining a commercial program, GENESIS, and software developed by the authors. The in-house software module is required to update the thermal properties of the material to simulate material removal and to obtain the new temperature distribution. A new model is then generated for the topology design with material properties modified based on the updated temperature distribution. These iterations can be performed automatically by linking GENESIS and the in-house software.

The final result of the topology optimization suggests the shape of the structure that would satisfy the constraints and minimize or maximize the objective. This shape can be used as an initial configuration in the sizing design to obtain the final TPS. In this study, once the topology was obtained, a parametric study was performed using the modeling package EDS-IDEAS [6]. A model that was constrained based on key dimensions was constructed, and the effect of these key dimensions on the stress and frequency of the TPS was investigated. This investigation provided insight into the TPS model and how its behavior differed based on some key structural dimensions. This paper presents a design process for topology optimization and 3-D parametric study of TPS geometry by integrating heat transfer, static, and normal mode analyses. The following sections detail the temperature range and material properties used in the research. Geometric details and boundary conditions for the TPS model used in topology optimization are also discussed.

Identification of a Suitable TPS

The selection of a TPS for a specific region on a spacecraft begins with the identification of the thermal environment acting in that given area. Different regions of the spacecraft experience different thermal responses, as shown in Fig. 2. Based on the severity of the identified thermal loading, different TPS configurations are employed. In this work, acreage TPS similar to Fig. 1 was chosen as the test bed for the demonstration of the design tools. Leading edges and nose sections are tightly coupled to the thermal environment through their shapes. The aerodynamic heating and shock interaction are strong functions of radius of curvature of the leading edges. So, use of these hot structures would introduce further complexity. However, this will be an area for further work.

The design of a TPS involves identifying the temperature distribution within the TPS at various times during the reentry phase. This temperature distribution greatly affects the TPS stress distribution, which, like frequency, is one of the major design constraints. Based on this temperature profile, a suitable material is identified to handle the high thermal stresses that are induced during this stage. The temperature profile used for the initial model was

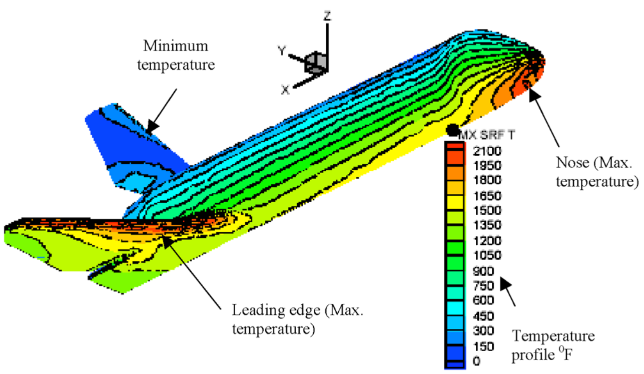

Fig. 2 Thermal profile on a NASA space shuttle.

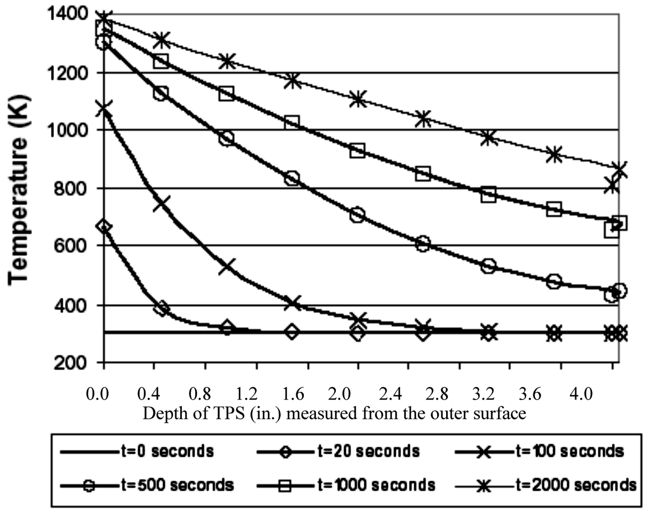

Fig. 3 Sample temperature distribution through the TPS thickness [5].

obtained from the literature [7] and is discussed in the following section.

Temperature Profile Within the TPS

The material selection for a TPS [9,10] depends on the thermal gradients it experiences during its operational life. The temperature gradient experienced by a TPS depends on how long the heat flux is applied and the thermal diffusivity. Cowart and Olds [7] present various temperature distribution profiles for the space shuttle throughout its TPS thickness for different time steps. Estimation of stress in the TPS due to this temperature distribution is a transient problem if one has to incorporate the time-varying data. However, in this research, for the first iteration a temperature profile that corresponds to the maximum temperature gradient is selected. Because this is the only temperature distribution data available in the public literature, it was used for demonstrating the methodology. For remaining iterations the thermal properties of the model are modified using the density information, and thermal analysis is performed using the top and bottom temperatures from the selected profile. The preprocessor used in this process has the capability to model any temperature distribution, making it possible to evaluate multiple configurations with little effort. Therefore, from Fig. 3, the temperature profile in which the maximum temperature is 1100 K (827°C) and the minimum temperature is 300 K (27°C) was selected, which corresponds to 100 s of time since entering the atmosphere.

Table 1 Thermal properties of Inconel

Temperature, °F	Thermal conductivity, $\mathrm{Btu \cdot in/ft^2 \cdot h \cdot {}^\circ F}$	Coefficient of expansion, 10^{-6} in/in/°F
73	64.3	——
200	73.1	7.22
400	87.8	7.57
600	102.8	7.84
800	117.2	8.09
1000	130.8	8.28
1200	143.2	8.60
2400	154.6	9.02
1600	165.2	9.38
1800	175.5	——
2000	186.3	——
2100	192.2	——

Table 2 Modulus of elasticity of Inconel

Temperature, °F	Modulus of elasticity, tensile, 10^6 psi
70	2805
200	28.2
400	27.3
600	26.6
800	25.6
1000	24.8
1400	23.2
1500	22.5

Material Properties

In this research, Inconel alloy 693 is used for the TPS. Table 1 provides details about the mechanical and thermal properties of this alloy. Thermal conductivity, the coefficient of thermal expansion, and Young's Modulus are considered as functions of temperature [11,12]. The mechanical properties of alloy 693, shown in Tables 1 and 2, were taken from [11], and the failure stress limits of alloy 625 were used ([12]) due to lack of data for Inconel alloy 693. The aforementioned temperature-dependent properties were used in both topology and sizing optimization models. A scaling factor equal to the ratio of individual element density to the maximum density value is used to scale the material properties during intermediate iterations of topology optimization. This procedure was employed to simulate the material removal in each of the iterations. The GENESIS topology optimizer models the volume fraction of each of the elements as a design variable and uses this to scale the element densities. The final output of the process is a distribution of densities of all the elements. Details about the relationship between the

Distance	Temperature (K)
0	1100
1	800
1.5	700
2	600
2.5	500
3	450
4	400
4.5	350
5	325
6	310
7	305
8	300
9	300
10	300
11	300
12	300

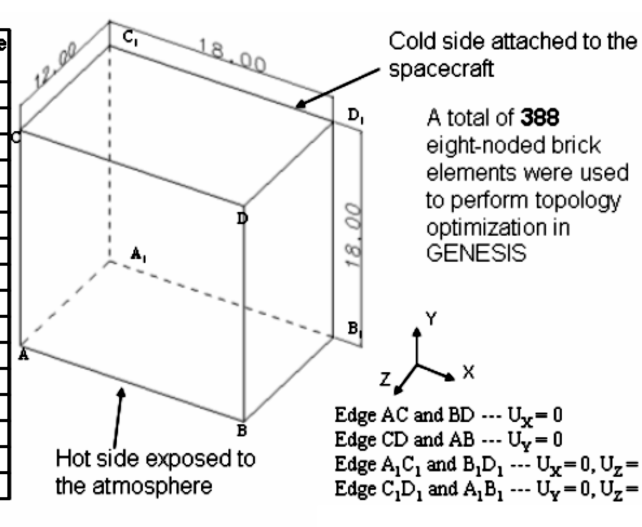

Fig. 4 Geometric model used for designing TPS.

material properties and the design variable in topology optimization are shown in Eq. (1) [5],

$$E(X) = E_0\, B + E_0(1 - B)X^A \qquad \rho(X) = \rho_0 X$$
$$t_{\min} \le X \le 1.0 \tag{1}$$

where $E(X)$ is Young's modulus, E_0 is the initial Young's modulus (this is the value in material card), $\rho(X)$ is density, ρ_0 is initial density, X is a topology design variable, A is a value supplied by the user, B is a parameter representing $E_{\min}/E_0$ where $E_{\min}$ is the minimum allowable value of Young's modulus, and $t_{\min}$ is the minimum value of the topology design variable.

Modeling and Boundary Conditions

The design of a TPS involves the evolution of a shape for the outer layer, which is exposed to maximum temperatures, and the supports that attach the TPS to the vehicle structure. The TPS also has to be constrained to avoid excitation at certain frequencies due to acoustic loading. A TPS panel with a first natural frequency greater than 1000 Hz would be ideal. This constraint would place the response outside the range of the random white noise excitations. To obtain this shape, a block of material representing the TPS is considered, as shown in Fig. 4. This block was selected to represent the volume of material enclosing the 18 × 18 in square TPS panel plus the support structure. This block is used in topology design to remove redundant material and obtain a shape for the support structure and the top panel that would satisfy the design requirements. Therefore, the final design would be contained within the volume of this block, and it would be a fraction of the initial material chosen.

The boundary conditions shown in Fig. 4 for the edges represent the fixed degrees of motion of the TPS in certain directions. For example, the edges AC and BD cannot expand into the neighboring panels; therefore, U_x is fixed for those two edges. Similarly, edges CD and AB cannot expand along the Y-direction; therefore, the U_y for these two edges is fixed. Similar boundary conditions are applied for the cold side of the TPS. These boundary conditions do not represent the actual operating conditions of the TPS, but in the preliminary design study they provide a good starting point.

During topology optimization, the shape of the TPS evolves as a density distribution in the initial block. Higher values of density indicate the amount of material to be retained, and lower values indicate the material to be removed. The topology optimization used is the density-based approach, in which the mechanical and thermal properties of all the elements are scaled using the density as shown earlier.

The optimization process modifies the volume fraction of these elements until all the constraints are satisfied and the objective function has converged. This final design has a density distribution that indicates the amount of alloy removed from the block. Any element that has a density value lower than the predetermined threshold is removed from the design process. Thus, geometry for the TPS support configuration that satisfies the required design constraints is evolved. This TPS configuration available as a density distribution will be converted into a 3-D solid model to perform a parametric study.

Design Approach

In this paper, a solid block of Inconel alloy is used to determine a configuration of the TPS that would be suitable for extreme conditions. The size of the block was selected after examining the shape of the existing TPS configurations in the literature [13–17]. The dimensions of the block of material are shown in Fig. 4. The material properties for this block were defined depending on the temperature profile through the thickness. GENESIS uses the density method [5] to remove the material to satisfy the design constraints, such as strain energy or frequency, while minimizing the objective. It also has the capability to model multiple objectives. These features would be used to explore various possibilities as discussed in the later sections. The density values obtained from the topology optimization were used to further scale the material properties to simulate the material removed during the iterations. The procedure for designing a TPS is shown in Fig. 5.

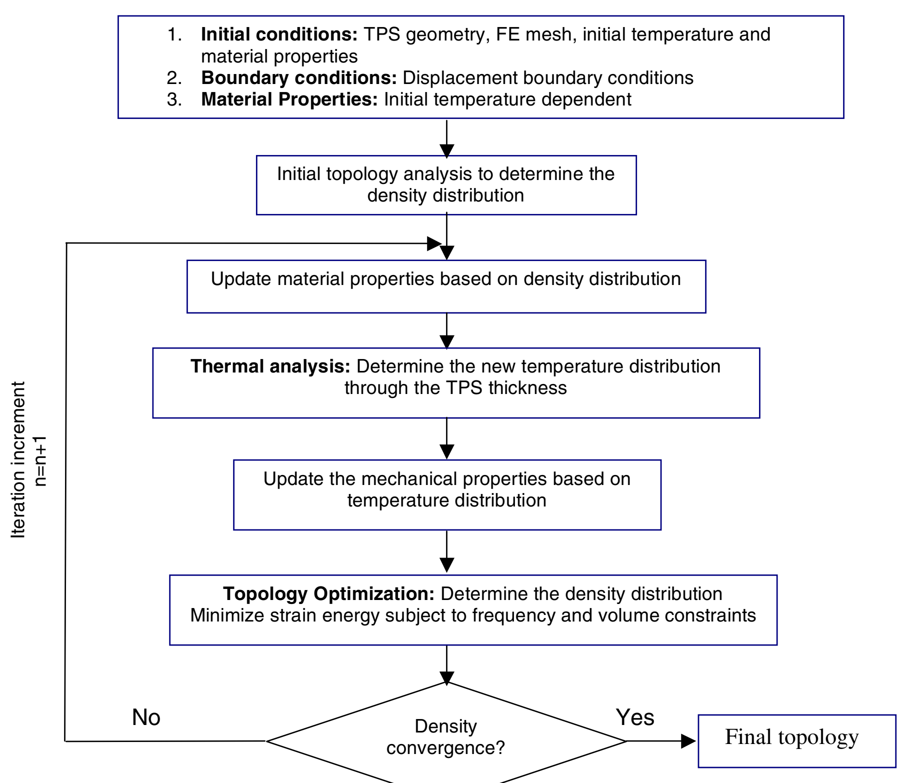

Fig. 5 TPS design: integration of thermal analysis and topology optimization.

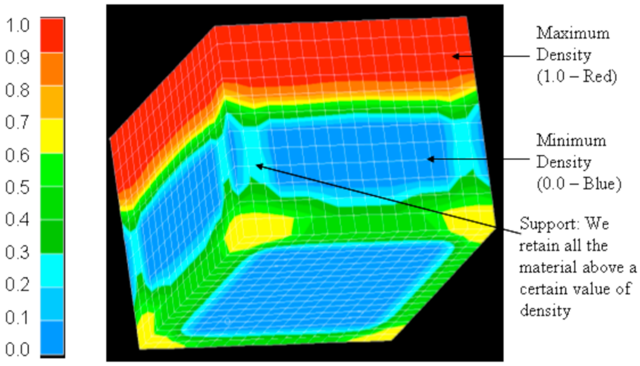

Fig. 6 Topology based on strain energy minimization.

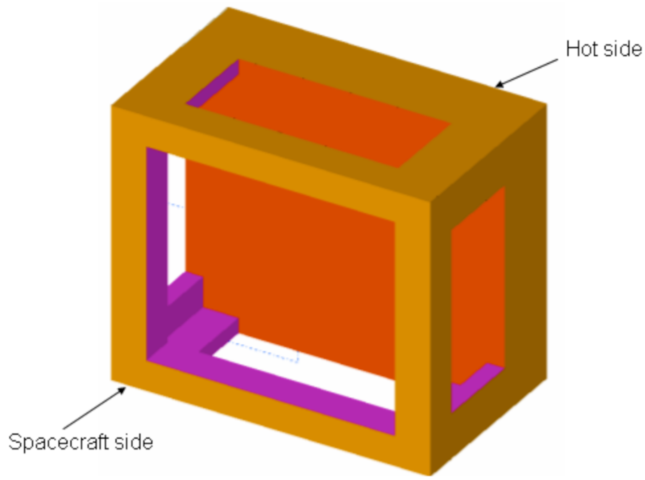

Fig. 7 3-D model of the TPS obtained in Case 1.

Usually, the optimum design obtained from GENESIS is not readily usable for manufacturing, but requires further refinement and engineering judgment. This is due to the rough edges and uneven surfaces created during the optimization process that controls each element of the finite element mesh individually. A 3-D model that is obtained using the results from the topology optimization process needs to be analyzed and modified to meet the design requirements. This 3-D model would seldom satisfy all the requirements, because the designer removes or adds material while interpreting the topology results that would make the component functional and manufacturable. This leads to constraint violations that can be satisfied by performing a parametric study. Therefore, parametric study is employed in this research to adjust the critical dimensions of the final 3-D model to satisfy the constraints.

This model was used with the current topology optimization procedure (with integrated thermal analysis) to obtain the final shape of a TPS, subject to certain operating conditions. Initially, a minimum strain energy design and a design with fundamental frequency greater than 1000 Hz were explored. Based on the results from these two cases, a combined analysis with minimum strain energy and a fundamental frequency of 1000 Hz was obtained. From studying the results of the three cases, it was determined that a combined objective function would result in a design that can be directly transferred to 3-D sizing optimization. Therefore, four different cases were studied to determine which shape of the structure to use in the parametric study. These four cases are as follows.

Case 1: Minimize the Strain Energy Density Subjected to a Defined Volume Reduction

For this case, the topology optimization problem is set up so that the maximum volume to be removed is 60%. However, for every 10% in volume reduction, the material properties of the model are scaled based on the densities at that iteration. This is accomplished by linking the GENESIS optimizer to a program that calculates the material properties based on the densities. Once the volume constraint is satisfied, the final design would have the minimum strain energy. Strain energy density is equal to 2.619×10^6 lb/in^2 for the configuration when 60% of the volume is removed. From Fig. 6, we can see that the design has a thick layer of material on the hot side of the TPS to withstand the high temperatures. This layer attaches to the aircraft structure using four supports, one on each corner. The figure also shows a frame connecting all four posts. The material in the center of the TPS has minimum density, which can be ignored while transferring the model to the sizing optimization problem. Because a postprocessor that directly converts the topology optimization result into a 3-D model is unavailable, IDEAS was used to model a 3-D geometry of the TPS (Fig. 7). To obtain the 3-D model from the topology optimization result, all of the elements that have a density value greater than 0.25 were considered. The 3-D model is selected for further analysis, and it would be modified in the sizing optimization stage. Therefore, a model that closely represents the optimum design would be acceptable for sizing optimization. Because frequency constraint was not included in this case, this design cannot be directly used in the parametric study.

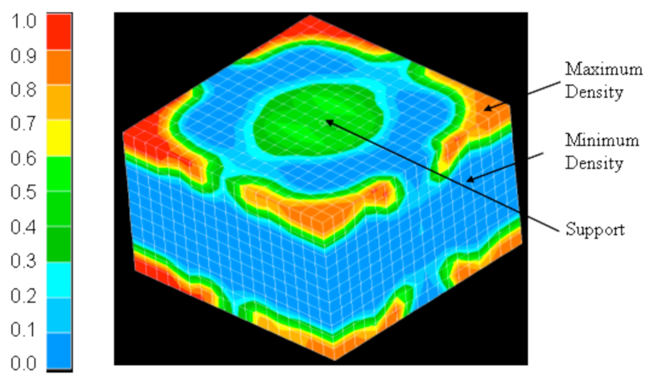

Fig. 8 Topology based on frequency constraint.

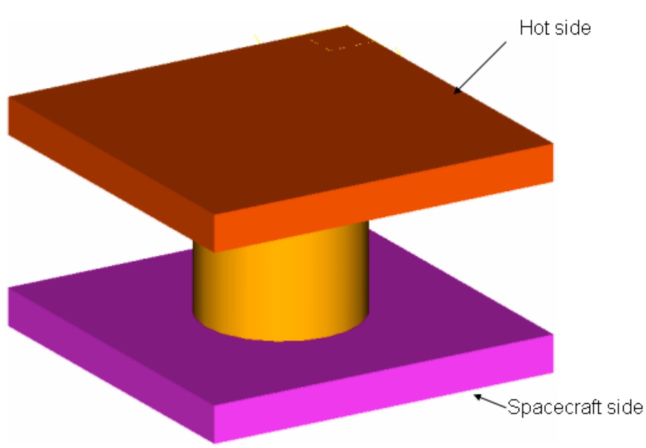

Fig. 9 3-D model of the TPS obtained from Case 2.

Case 2: Minimize the Volume Subjected to Fundamental Frequency Constraint

This second case deals with minimizing the volume to satisfy the frequency constraint. A fundamental frequency of 1000 Hz is used as the constraint to ensure that the structure is not excited due to low frequency acoustic loading. Figure 8 shows the final shape of the TPS to be used for further analysis. The initial mass of the structure is equal to 1166 lb, and the frequency is equal to 656 Hz. At the optimum point, the frequency constraint is 2515 Hz, and the mass of the structure is reduced to 218 lb.

Figure 8 shows that the TPS must have a thick support in the center with a frame on the top and bottom attached to this support to satisfy the required conditions. While creating a 3-D model, additional

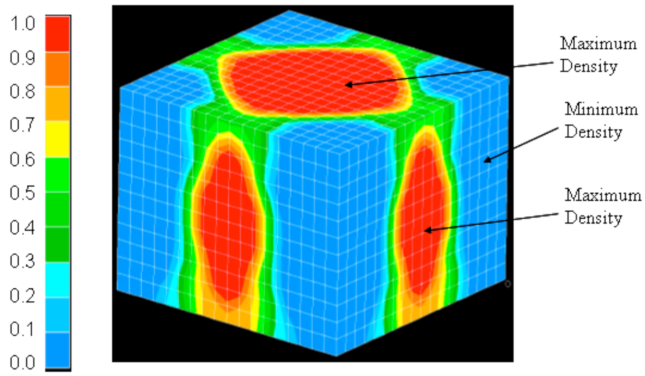

Fig. 10 Topology based on stress and frequency constraints.

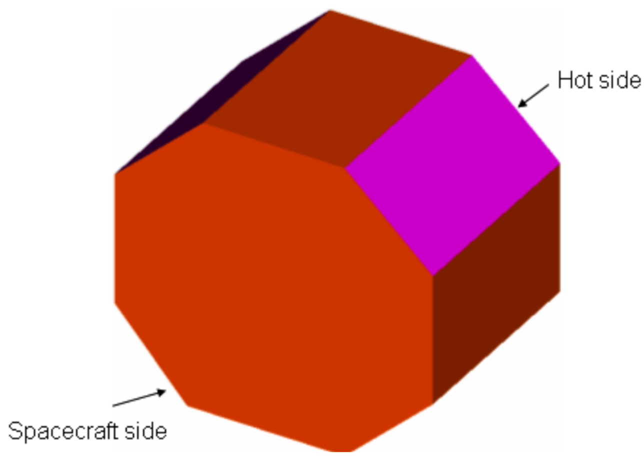

Fig. 11 3-D model for Case 3.

material is added on the "hot side" of the TPS to avoid direct contact of the internal structure with external temperatures. The additional platelike structure on the spacecraft side would provide the additional stiffness required to satisfy the constraint. Clearly, the design intent of this case is opposite to the one obtained in Case 1. In Case 1 there were side supports with no central supports; however, in this case we have one central support and no side supports (Fig. 9). Obviously, these two solutions cannot be used individually to perform any further design studies to obtain a feasible design for the TPS. If they are to be used individually, we would have two different design directions based on each of the initial designs. Therefore, Cases 3 and 4 are investigated. These additional cases would provide a design that would be sized to arrive at a final optimum TPS configuration.

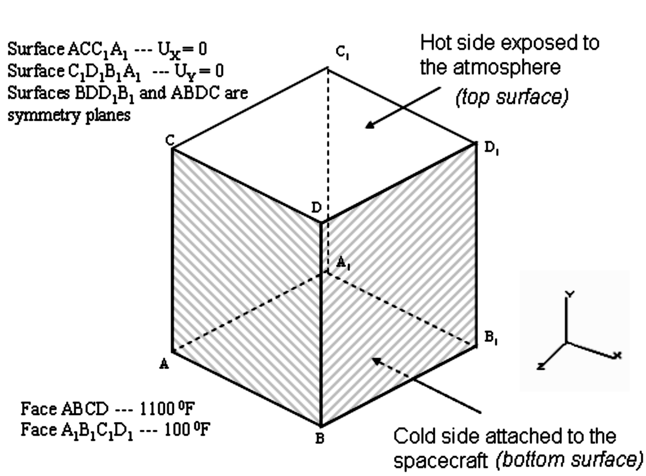

Fig. 12 Boundary conditions for Case 4.

Table 3 Initial and optimum values for Case 4

Objectives	Initial value	Final value
Frequency	656 Hz	1178 Hz
Strain energy density, psi	5.19×10^4	1.80×10^6

Case 3: Minimize the Strain Energy Subjected to a Constraint on Fundamental Frequency

In this third case, the strain energy is minimized with a constraint on fundamental frequency. This case was investigated to study the effects of both strain energy and frequency on the TPS shape. The shape from this analysis does not provide any additional information beyond the information already obtained from the first two cases. Figure 10 shows the final TPS shape. This shape shows that a solid central support is required, just as in Case 2. However, the framelike structures seen in Cases 1 and 2 are not seen in this case. Moreover, the reason this design is closer to the one in Case 2 is because the frequency constraint was the dominant factor. Frequency at the start of the optimization process was 656 Hz, and, at the optimum point, it was 2260 Hz, with strain energy equal to 1.33×10^6 lb/in^2.

This design proposes a large volume of material, a base design (Fig. 11) that does not provide a design direction that can be explored. The design obtained in this case was computationally feasible but physically not possible due to limitations on the weight of huge blocks of material on the spacecraft. This leads to the final case that would be a multiple-objective function design.

Case 4: Multiple Objectives: Strain Energy and Frequency with a Constraint on Volume of Material Removed

In this final case, the strain energy and frequency are combined in the objective function using the normalization scheme in GENESIS, which is as follows:

$$\text{Objective} = \frac{\text{Strain energy}}{\text{Strain energy in first iteration}} - \frac{\text{Frequency}}{\text{Frequency in first iteration}}$$

In this formulation the strain energy is minimized and the frequency is maximized. The optimization problem is to minimize this combined objective function with a constraint to remove at least

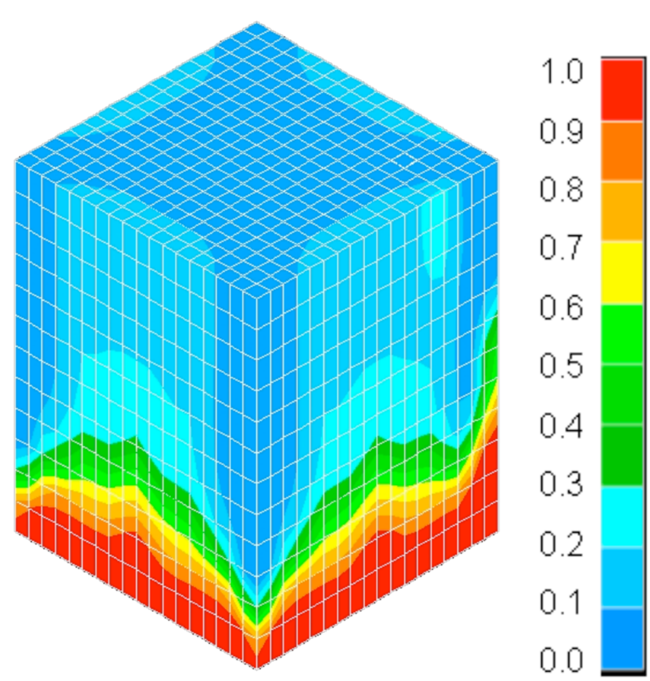

Fig. 13 Topology optimization results for Case 4.

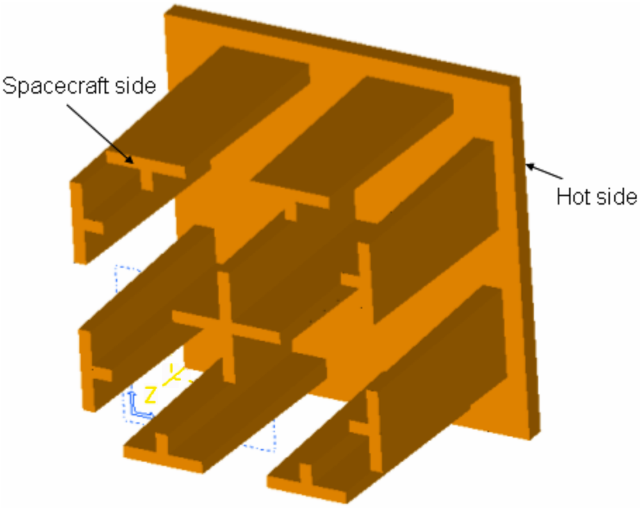

Fig. 14 3-D model of Case 4.

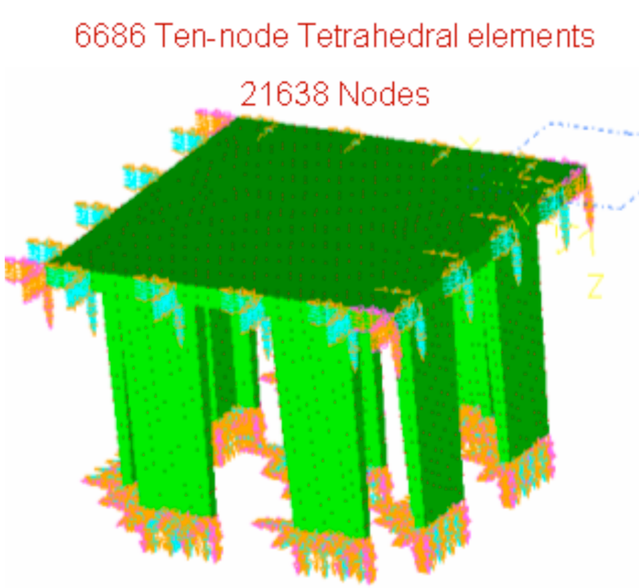

Fig. 15 Finite element model of the TPS (parametric study).

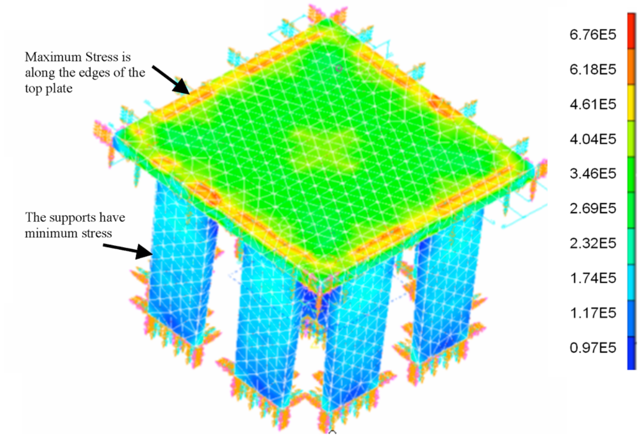

Fig. 16 Von Misses stress distribution (psi).

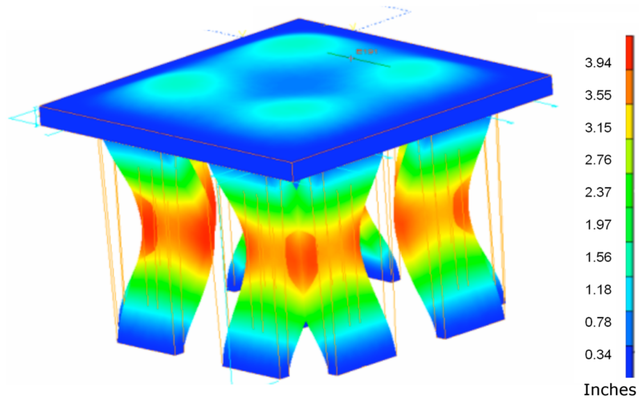

Fig. 17 First mode shape of the TPS.

60% of the alloy. The model used is a quarter block with symmetric boundary conditions. The reason for using this quarter model is to obtain more geometrical information about the central support as shown in Fig. 12.

This design evolved a TPS shape as shown in Fig. 13. This shape has both the central and side supports. Because the objective is a combination of strain energy and frequency, the results are similar to those of Cases 1 and 2. The initial and final values of the strain energy and frequency are shown in Table 3.

Using the results from the topology optimization, a 3-D model (Fig. 14) was constructed for performing a parametric study to determine the variation of frequency with respect to key dimensions of the supports. The shapes of the supports are modeled based on the density information. As mentioned earlier, this modeling task is not a direct translation of the topology results, but involves recreation of the 3-D model in IDEAS. Further analysis is done to determine the variation of frequency with respect to dimensional changes of the TPS. The dimensions of the initial T-section of all the side supports are modeled as design variables in the sizing optimization process. The central cross support was modeled based on the quarter model symmetric shape from the topology design.

The finite element model (Fig. 15) for the TPS consists of 6686 10-noded tetrahedral elements with 21,638 nodes. The support is assumed to be clamped to the spacecraft, and all the degrees of freedom of the side surfaces of the top panel are fixed. Figure 16 shows the stress distribution of the TPS with maximum stress (5.76×10^5 psi) near the edges of the top layer, which is subjected to maximum temperature. This is because only the edges of the top plate are constrained from expanding into the surrounding TPS panels.

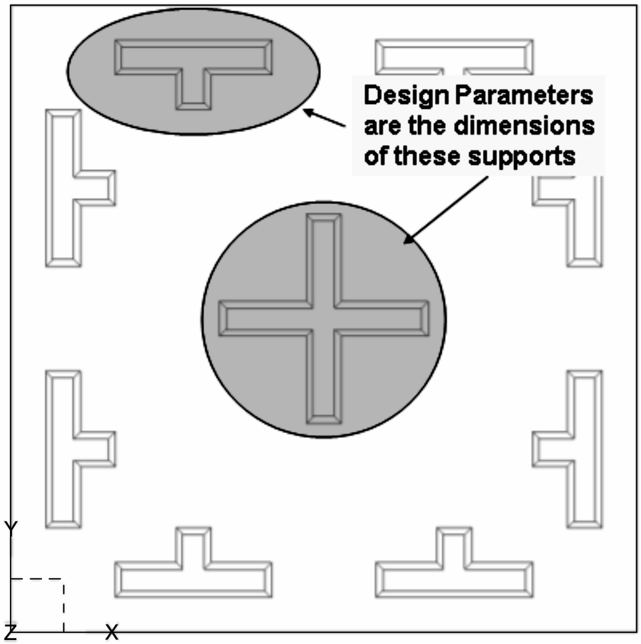

Fig. 18 Design parameters used for parametric study.

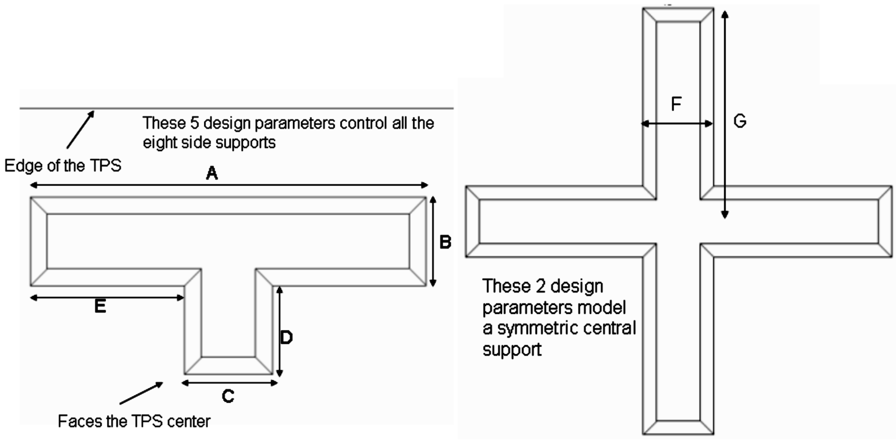

Fig. 19 Design dimensions controlling the side and central supports.

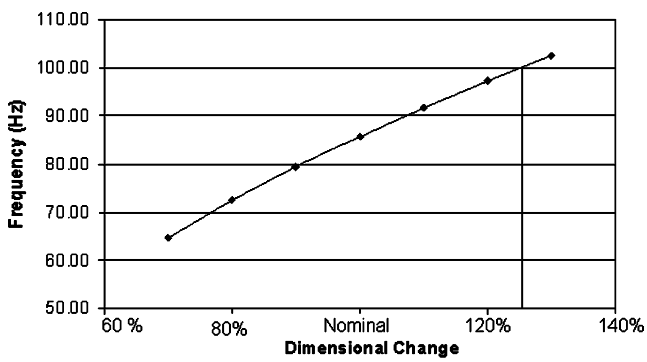

Fig. 20 Frequency variation with respect to dimensional change (A–G).

Currently, this constraint does not allow for any expansion; however, a certain amount of expansion is required to reduce the thermal stresses, and this can be done by introducing springs around the edges. The stiffness of these springs would be modeled as design variables to obtain an optimum configuration. The stiffness of these springs would suggest using an appropriate filler material between the two plates. This is a solution that is being explored in this research, and depending on the availability of the filler material this constraint can be modified. Figure 17 shows the fundamental mode shape of the TPS model that was selected for sizing optimization. It can be seen that only the supports are excited in this mode, and their design plays an important role in controlling the value of fundamental natural frequency.

Figure 18 shows the two supports that are used to define the design parameters. The dimensions of one side support are linked to the rest of the seven side supports. Five design variables (dimensions of the supports) are used to model the side support, and two design variables are used to model the symmetric central support.

The dimensions of the side supports (Fig. 19) are linked to reduce the total number of design variables and study the effect of these supports on the frequency. Moreover, considering the manufacturability of these supports, it would be more economical if they had the same dimensions.

Dimensions A, B, C, D, E, F, and G are perturbed using the IDEAS parametric optimization scheme to determine the parameter values that satisfy the frequency constraint. This perturbation would modify the cross section of the supports by chancing the flange and web thicknesses of the T-Section. Because frequency is the dominant constraint in the topology optimization, it is used to determine the TPS geometry that satisfies the 1000 Hz limit. The variation of frequency with percentage dimensional change in all the design dimensions is presented in Fig. 20. This plot shows a linear variation of frequency and the dimensions have to be 125% of the nominal values to satisfy the frequency constraint.

This design procedure has given insight into the possible shape of the supports that would satisfy the design constraints. Even though this is not a unique solution and the designer can interpret the topology solution in a different manner, the solution presented is feasible and a potential candidate for further study. Further study can involve more disciplines and more failure criteria.

Summary

This research has presented a methodology to couple frequency and stress analyses to evolve a topology that incorporates the effect of change in temperatures during the strain energy calculations and the effect of change in densities in the heat conduction process. Once a topology is obtained, a suitable 3-D model is developed using engineering judgment to perform the sizing optimization to get the final configuration of the TPS. During this process, springs are modeled to determine if they can evolve the boundary conditions that are required to obtain a design that satisfies all the design constraints. These springs are expected to identify filler material that needs to be incorporated between the plates.

Using the results from the topology optimization and the sizing optimization, a TPS configuration is evolved. Once this configuration is available, further design would involve identifying means to impose the boundary conditions that were used during the design process. This is a very important task because, as discussed earlier, the stress in the structure is most sensitive to the boundary conditions applied at the edges of the top surface. Identification of these fixtures would involve contact analysis using packages such as ABAQUS [18] (a commercial finite element analysis package) to model expansion of the TPS and stresses developed during this process.

Once the entire design process is completed and a final TPS configuration has been obtained, sensitivity analysis could be performed to determine the most significant parameters. Because all of the parameters would have slight variations due to manufacturing tolerances or operating conditions, they can be modeled as uncertain parameters, and a probabilistic study can be performed on this design. Uncertainty quantification is an important task that provides the designers insight into the most significant parameters and failure mechanisms. This analysis would provide information about how safe the final TPS would be, subject to uncertainties in the critical parameters.

Operational uncertainties could also be modeled into the uncertainty analysis, such as the probability of impact of debris of a certain size. The size of the debris and the possibility of impact could be modeled as uncertain phenomena to determine the reliability of the TPS subject to these types of uncertainties, as well as parametric variations. However, this type of analysis would involve complex integration of nonlinear contact mechanics, static analysis, normal mode analysis, and uncertainty quantification into a single framework.

Acknowledgments

This research work has been supported partly by the U.S. Air Force through the contract F33615-98-D-3210, D.O. 48, and partly by Wright State University Office of Research and Sponsored Programs.

References

[1] Annicchiarico, W., and Cerrolaza, M., "An Evolutionary Optimization Approach for the Shape Optimization of General Boundary Elements," *Electronic Journal of Boundary Elements*, Vol. BETEQ2001, No. 2, 2002, pp. 251–266.

[2] Jog, C. S., Haber, R. B., and Bendsoe, M. P., "Topology Design with Optimized, Self-Adaptive Materials," *International Journal for Numerical Methods in Engineering*, Vol. 37, April 1994, pp. 1323–1350.

[3] Bendsoe, M. P., "Optimal Shape Design as a Material Distribution Problem," *Structural Optimization*, Vol. 1, No. 1, 1989, pp. 193–202.

[4] Swan, C. C., and Kosaka, I., "Reuss and Voigh Mixing Rules for Variable Topology Material Layout: Linear Elasticity,"*International Journal for Numerical Methods in Engineering*, Vol. 40, Jan. 1997, pp. 3033–3057.

[5] GENESIS Structural Optimization Software, Ver. 6.0 User's Manual, Vanderplaats Research and Development, Colorado Springs, CO, Jan. 2000.

[6] Shih, R., *Parametric Modeling with I-DEAS 9*, Schroff Development Corporation Publications, Mission, KS, 2002.

[7] Cowart, K. K., and Olds, J. R., "Integrating Aeroheating and TPS into Conceptual RLV Design," AIAA Paper 99-4806, Nov. 1999.

[8] Cast, J., Amatore, D., and Lindeke, R., "X-33's Innovative Metallic Thermal Shield 'Ready for Flight'," NASA news release 99-022, 1999.

[9] Meyerson, R., Emde, W., Wallace, D., and Barber, D., "Development of a Reusable Thermal Protection System for the K-1 Orbital Vehicle," IAF Paper 99-1.6.04, Oct. 1999.

[10] Kunihiko, O., "Thermal Analysis of the Thermal Protection System for Re-entry Vehicle," *Computer Methods in Applied Mechanics and Engineering*, Vol. 151, Nos. 3–4, Jan. 1998, pp. 301–310.

[11] "INCONEL Alloy 693—Excellent Resistance to Metal Dusting and High Temperature Corrosion," Special Metals, www.specialmetals.com/products/inconelalloy693.htm [cited Sept. 2006].

[12] *MIL-HDBK-5G, Military Handbook on Metallic Materials and Elements for Aerospace Vehicle Structures*, U.S. Dept. of Defense, 1998, Chap. 6.

[13] Poteet, C. C., Abu-Khajeel, H., and Hsu, S-Y, "Preliminary Thermal-Mechanical Sizing of Metallic TPS: Process Development and Sensitivity Studies," AIAA Paper 2002-0505, Jan. 2002.

[14] Blosser, M. L., "Investigation of Fundamental Modeling and Thermal Performance Issues for a Metallic Thermal Protection System," AIAA Paper 2002-0503, Jan. 2002.

[15] Chen, R., and Blosser, M. L., "Metallic TPS Panel Flutter Study," AIAA Paper 2002-0501, Jan. 2002.

[16] Poteet, C. C., and Blosser, M. L., "Improving Metallic Thermal Protection System Hypervelocity Impact Resistance Through Design of Experiments Approach," AIAA Paper 2002-0912, Jan. 2002.

[17] Amundsen, R. M., Dec, J. A., Mitcheltree, R. A., Lindell, M. C., and Dillman, R. A., "Preliminary Thermal Analysis of a Mars Sample Return Earth Entry Vehicle," AIAA Paper 2000-2584, June 2000.

[18] ABAQUS Theory Manual, Ver. 6.4.1, Hibbitt, Karlsson, and Sorensen, Inc., Pawtucket, RI, 2003.

A. Messac
Associate Editor