NASA Technical Memorandum 102343

An Integrated Methodology for Optimizing Structural Composite Damping

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Prepared for the
1989 Winter Annual Meeting
of the American Society of Mechanical Engineers
San Francisco, California, December 10-15, 1989

NASA

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ABSTRACT

A method is presented for tailoring plate and shell composite structures for optimal forced damped dynamic response. The damping of specific vibration modes is optimized with respect to dynamic performance criteria including placement of natural frequencies and minimization of resonance amplitudes. The structural composite damping is synthesized from the properties of the constituent materials, laminate parameters, and structural geometry based on a specialty finite element. Application studies include the optimization of laminated composite beams and composite shells with fiber volume ratios and ply angles as design variables. The results illustrate the significance of damping tailoring to the dynamic performance of composite structures, and the effectiveness of the method in optimizing the structural dynamic response.

INTRODUCTION

Fiber composite materials are broadly utilized in light-weight structures, as they readily provide superior specific modulus and strength. In addition to stiffness and strength, polymer-matrix composite materials provide higher material damping than most metals because of their "viscoelastic" matrix and heterogeneity. High specific stiffness and strength are sufficient conditions for improved static performance, but they do not always ensure improved dynamic performance. Passive structural damping is also a crucial dynamic property in vibration and sound control, as it generally improves resonance phenomena, settling times, and fatigue life. Composite materials are primarily targeted for structures requiring good dynamic performance, such as engine, aircraft, and space structures. Therefore, the inherent damping capacity of composites becomes a significant design factor, making polymer-matrix fiber composites even more attractive as structural materials.

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Research on the damping capacity of composite materials, laminates, and beams [1-6], has shown that laminate damping is highly tailorable with respect to constitutive properties, volume fractions, and ply orientation angles. The same work suggests that composite structures should be tailored for optimal combinations of damping and stiffness in order to obtain improved dynamic performance. To the authors' best knowledge, formal methods for tailoring general plate/shell composite structures for optimal damping and optimal damped dynamic response are not presently available. Research has been reported on the tailoring of plate/shell composite structures for optimum static and/or undamped dynamic performance. Composite plates have been optimized [7-10] based on static performance criteria. Composite plates have been also optimized either with a constraint on the first natural frequency [11], or with performance criteria based on the undamped frequency response [12]. Refs. [13,14] present the optimal tailoring of composite beams and links for maximum damping capacity based on static constraints. Most reported methods can produce designs with improved integrity and stiffness, having natural frequencies within desirable ranges. The inclusion of the structural composite damping into the present method provides a new dimension regarding the optimal tailoring of composite structures, as trade-offs between damping, weight, stiffness, and placement of natural frequencies are now possible. Furthermore, to the authors' best knowledge, damping micromechanics have not been included so far into the optimal design of composite structures, and in most reported cases the analysis starts from the laminate level. The present method incorporates such a damping micromechanics theory, hence, it provides the capability for tailoring also the constituent materials and fiber volume ratios.

This paper presents an integrated computational methodology for simulating the dynamic response of composite structures, and optimizing structural damping in conjunction to other static and dynamic design criteria. The method is targeted for composite structures under forced excitation. The dynamic performance criteria are based on modal damping capacities, resonance dynamic amplitudes, and natural frequencies. The method can produce designs having natural frequencies inside the desirable frequency domain, and also minimized resonance amplitudes. The optimal tailoring includes tailoring of the basic composite material(s), and tailoring of the laminate configuration. The modeling of composite damping and other elastic properties is based on micromechanics and laminate theories [4,5,15]. The global damping capacity of composite structures is simulated based on a specialty finite element. Finite element damping matrices have been developed, based on previous theories, for a triangular plate element. The optimal design problem is formulated, and solved with the feasible-directions non- linear programing algorithm. The method is demonstrated by applying it to a composite beam and a composite cylindrical shell.

METHOD

This section summarizes the methodology. The method includes: (1) simulation of the structural composite damping, (2) simulation of the damped frequency response of composite structures, and (3) formulation and solution of the optimal design problem.

Structural Composite Damping

Local Laminate Damping

Approximate micromechanics equations are utilized, which have been derived based on hysteretic damping. On-axis specific damping capacities (SDC's) are represented in terms of elastic and dissipative properties of the fibers and matrix, interface properties, temperature, and moisture. Off-axis SDC's, ie. the SDC's of a ply loaded at an off-axis angle, are related to the on-axis SDC's with proper transformations, and subsequently, the specific damping capacity (SDC) of the composite laminate is synthesized. The laminate SDC is related to constituent properties, volume fractions, interface properties, hygrothermal parameters, fiber orientation, laminate configuration, and local deformation. The micromechanics and laminate damping theories are presented in refs. [4,5]. Additional micromechanics and laminate theories [15] are utilized for other mechanical properties of the composite material.

Structural Damping

The laminate damping capacity is a local structural property. The global damping capacity of the structure at a given deformation shape would be the integrated action of local laminate damping over the structural volume. For plate and shell structures, the structural SDC ψ_s would be the ratio of the integrals of the local laminate damping energy δW_L and local strain energy W_L respectively, over the structural area A,

$$\psi_s = \frac{\int_A \delta W_L dA}{\int_A W_L dA} \tag{1}$$

Hence, the local laminate damping can only provide a rough estimate of the global structural damping of simple structural components subjected to forced vibration loading conditions. In case of complex structures or structural components, global structural damping is recommended. Numerical integration of laminate damping over the structural volume is performed based on finite element discretization. The developed procedure is summarized in the following paragraphs.

The local laminate damping and strain energies are first integrated over the volume of the finite element based on numerical quadrature. The finite element damping and stiffness matrices, $[C_e]$ and $[K_e]$ respectively, are obtained from the laminate damping matrices ($[A_d]$, $[C_d]$, $[D_d]$)[5] and stiffness matrices ($[A_s]$, $[C_s]$, $[D_s]$) as follows:

$$[C_e] = \int_{A_e} [B]^T \begin{bmatrix} [A_d] & [C_d] \\ [C_d] & [D_d] \end{bmatrix} [B] dA_e \tag{2}$$

$$[K_e] = \int_{A_e} [B]^T \begin{bmatrix} [A_s] & [C_s] \\ [C_s] & [D_s] \end{bmatrix} [B] dA_e \tag{3}$$

where [B] is the strain shape function matrix, and A_e the element area. After the integration, the damping and strain energies of the finite element, δW_e and W_e, are directly related to the respective element matrices and nodal displacements $\mathbf{u_e}$.

$$\delta W_e = \frac{1}{2} \mathbf{u_e}^T [C_e] \mathbf{u_e} \tag{4}$$

$$W_e = \frac{1}{2} \mathbf{u_e}^T [K_e] \mathbf{u_e} \tag{5}$$

Summation of the damping and strain energies of the individual elements provides the global damping and strain energies of the structure, δW_s and W_s respectively, for the specific deformation state.

$$\delta W_s = \sum_{i=1}^{nel} \delta W_{e,i} \tag{6}$$

$$W_s = \sum_{i=1}^{nel} W_{e,i} \tag{7}$$

The SDC of the structure at a specific deformation state is the ratio of damping and stored strain energies.

$$\psi_s = \frac{\delta W_s}{W_s} \tag{8}$$

In order to establish a coherent measure regarding the damping capacity of a structure, specification of representative deformation shapes is required. The mode shapes of the structure are the natural choice for such representative deformation shapes, since any small vibrational deformation can be expressed as a linear combination of mode shapes. The structural damping associated with an individual mode shape is defined as modal structural damping.

The previously described structural damping theory has been incorporated into a triangular plate element with three nodes, six degrees of freedom per node (3 deflections u_x, u_y and u_z; and 3 rotations ϕ_x, ϕ_y and ϕ_z), linear shape functions for the in-plane (membrane) deformations, and cubic shape functions for the out-of-plane (flexural) deformations. The same element is also utilized for modal analysis and simulation of the dynamic response.

Structural Dynamic Response

Main emphasis is focused on the forced (frequency) response of composite structures. Using finite element discretization, the dynamic response of the composite structure may be approximated by the following system of n discrete dynamic equations,

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F(t)\} \tag{9}$$

where [M], [C], and [K] are the mass, damping and stiffness matrices respectively, and $\{u\}$ is the vector of the n discretized degrees of freedom.

A typical excitation force $\{F(t)\}$ would involve: surface tractions, body forces, and hygrothermal forces. In the present study, the effects of body forces and hygrothermal forces have been neglected. The study is limited to structures operating in room temperature dry environment. Hygrothermal effects may be significant for the design of composite structures, and the theory has provisions to model them. However, the design problem is complicated beyond the scope of the current paper, because the damping and stiffness of the composite depend also on temperature and moisture variations [5].

The dynamic system in eq. (9) is transformed to the $m \times m$ modal space through the linear transformation:

$$\{u\} = [\Phi]\{q\} \tag{10}$$

where q is the modal displacement vector, and $[\Phi]$ is ensembled from the first m normalized undamped eigenvectors of eq. (9). Assuming proportional damping, the transformation yields the reduced uncoupled dynamic system,

$$[m]\{\ddot{q}\} + [c]\{\dot{q}\} + [k]\{q\} = \{f(t)\} \tag{11}$$

where,

$$[m] = [\Phi]^T[M][\Phi]$$

$$[c] = [\Phi]^T[C][\Phi]$$

$$[k] = [\Phi]^T[K][\Phi]$$

$$\{f(t)\} = [\Phi]^T\{F(t)\}$$

The damping matrix [c] is formulated from the first m modal SDC's in accordance to eqs. (2-8).

Forced Dynamic Response

The frequency response of the j-th nodal displacement would be,

$$u_j(\omega) = \sum_{k=1}^{m} \Phi_{jk} q_k(\omega) \tag{12}$$

where $q_k(\omega)$ is the frequency response of the k-th mode in modal space. In general, $u_j(\omega)$ would be a complex number describing both amplitude and phase. The dynamic amplitude would be:

$$U_j(\omega) = ||u_j(\omega)|| \tag{13}$$

The resonance amplitude of the k-th mode, would be:

$$U_{jk}^r = ||u_j(\omega_{d,k})|| \tag{14}$$

where $\omega_{d,k}$ is the damped natural frequency of the k-th mode. The resonance amplitudes U_{jk}^r are used as dynamic performance measures.

Optimal Design

The structural damping methodology aims to optimize the forced response of composite structures. The objective is to tailor the composite materials and the laminate configurations for optimal combination of stiffness and damping such that selected natural frequencies are placed within desirable frequency ranges, and selected resonance peaks are minimized. Compared to other methods which only ensure proper placement of natural frequencies, the present method is far more powerful, as it drives all resonance frequencies in the feasible frequency domain, and concurrently minimizes the resonance peaks of selected modes at selected structural sites. Hence, the present methodology is even applicable when placement of all resonant frequencies into the desirable frequency domain is infeasible. The later problem is frequently encountered in design and is further complicated by the fact that the frequency bounds are usually stiff, in that, we have not control on them during the design of a given structural component. The present method readily includes the capacity to identify the most critical natural frequencies within the infeasible frequency range and to minimize their resonance amplitudes by increasing the respective modal damping values. The optimization problem is formulated in non-linear programming form, as described bellow.

Design Vector: The design vector may include: (1) the ply orientation angles θ_i of each sublaminate (group of $\pm\theta$ angle plies), and (2) the fiber volume ratios (FVR's) $k_{f,i}$ of each sublaminate.

The structural dimensions are not altered. The effect of shape optimization on the damped dynamic response of composite structures and its interaction with material tailoring will be addressed in a future study.

Objective Function: In general, the method provides the flexibility to minimize selected resonance amplitudes, based on the particular design requirements. The following objective function will minimize the maximum resonance amplitude of the first m vibration modes for q_d displacements:

$$min(max\{U^r_{jn}(\theta_i, k_{f,i})\}) \qquad j = 1, ..., q_d, \quad n = 1, ..., m \tag{15}$$

Constraints: The minimization of the objective function in Eq. (15) is subject to the following constraints:

Upper and lower bounds on design variables:

$$-90.0^o \leq \theta_i \leq 90.0^o \tag{16}$$

$$0.0 \leq k_{f,i} \leq 0.70 \tag{17}$$

Upper bounds $U^{s,U}_j$ on q_s static displacements u^s_j:

$$u^s_j \leq U^{s,U}_j \qquad j = 1, ..., q_s \tag{18}$$

Frequency constraints described in general by k upper and lower bounds, Ω^U_j and Ω^L_j respectively, on m natural frequencies:

$$\omega_n \leq \Omega^U_j, \qquad \omega_n \geq \Omega^L_j \tag{19}$$

$$n = 1, ..., m \;\; and \;\; j = 1, ..., k$$

Upper bounds $U^{r,U}_{jn}$ on m resonance amplitudes for q_d displacements :

$$U^r_{jn} \leq U^{r,U}_{jn} \qquad j = 1, ..., q_d, \;\; and \;\; n = 1, ..., m \tag{20}$$

It is pointed out that the objective function (15) and constraints (18) and (20) may include either deflections or rotations. Stress failure constraints are not included in the present paper, but they will be included in future work.

Optimization Algorithm

As already mentioned, the optimization problem has been formulated in non- linear programming form and is solved with the method of feasible directions. The feasible directions algorithm is a primal optimization method, performing a direct search in the feasible design space based on first order sensitivity for the objective function and active constraints. Primal methods are more suitable for non-linear programing problems which have computationally expensive constraints, since they typically require fewer constraint evaluations than do the penalty transformation methods. The feasible directions method used in this study incorporated active constraint set and line-search strategies for improved computational efficiency.

APPLICATION STUDIES

Assumptions

The present section presents applications of the previously described method on the two composite structures shown in Fig. 1:

(1) a cantilever laminated composite beam, 6in (152.4mm) long, 1in (25.4mm) wide, and 0.2in (5.08mm) uniform thickness, and

(2) a cantilever cylindrical laminated composite shell, 16in (406.4mm) long, 16in (406.4mm) wide, of 10in (254mm) radius, and uniform 0.2in (5.08mm) thickness.

Both structures were modeled with 80 triangular plate elements. The basic composite material system for each case was HM-S (high modulus surface treated) graphite fiber in an epoxy matrix. The mechanical properties of the fiber and matrix are shown in Table 1. Each ply is 0.01in (0.254mm) thick (equal to 2 preimpregnated tapes of 0.005in), hence, the composite laminate has 20 plies through the thickness. The applications were limited to symmetric laminate configurations, shown in Fig. 2, consisting by either one or three sublaminates at each symmetric side. As seen in Fig. 2, each sublaminate is a set of regular $\pm\theta$ angle-plies. The fiber orientation angle and the fiber volume ratio (FVR) of each sublaminate were considered as design variables. The effects of multiple sublaminates with different fiber orientation angles and FVR's are investigated with the following 4 laminates incorporating:

(a) one sublaminate of fixed 50% FVR with the ply angle as design variable.

(b) one sublaminate with both the ply angle and FVR as design variables.

(c) three sublaminates of fixed 50% FVR with their ply angles as design variables.

(d) three sublaminates with the ply angles and FVR's as design variables.

Case 1: Beams

The assumed loading conditions of the beam were a static load of 5 lb/in (876 N/m) applied along the free edge, and a cyclic load of 0.1 lb/in (17.5 N/m) also applied along the free edge. The design objective was to minimize the maximum resonance (z-axis) amplitude of the first five modes at the middle of the tip. The first natural frequency was constrained to be less than 350 Hz. An upper bound of 0.050in (1.27mm) was imposed on the z-axis static deflection (u_z) of the free edge.

For comparison purposes, an additional optimization study was performed for one sublaminate with varying ply orientation angle and FVR without optimizing damping. In the later case, the maximum static deflection of the beam was minimized subject to an identical frequency constraint. The performance criteria did not include any damped resonance amplitudes, hence, the effect of composite damping on the design was neglected.

The optimum designs for each of the previously described subcases are presented in Table 2. The same table presents the reference design of a 50% FVR unidirectional composite beam. The unidirectional beam was selected as a reference design, because is known to exhibit the maximum static bending stiffness. As seen in Table 2, the first mode has the higher resonance amplitude, consequently, this resonance peak was minimized. The present method has produced optimum designs (Cases 1a, 1b, 1c, and 1d) having lower resonance amplitudes for the bending modes by factor of 2, compared to the unidirectional design. In addition, the unidirectional design violates the frequency constraint, while all optimum designs have a first natural frequency less than 350 Hz. The trade-off for the improvements in dynamic performance was a reduction in static stiffness.

The superiority of the current method to other methods which do not optimize damping is also demonstrated in Table 2. The optimum design without considering damping in the performance criteria satisfies the frequency constraint marginally, but has a higher first resonance amplitude than the reference design. In contrast, all optimum designs produced by the present method have reduced first resonance amplitudes by factor of 2, and have first natural frequencies substantially below the upper bound constraint.

Additional comparison of the optimum designs produced by the current methodology for various laminate configurations indicates that the optimization of FVR has produced an additional 11% improvement in the objective function. This seems a rather insignificant improvement compared to the higher manufacturing costs related with the production of composites with customized FVR. Laminates with multiple sublaminate systems have virtually produced no additional reductions in the objective function.

The frequency response functions (FRF's) of the optimum design for Case 1d (three sublaminates with varying fiber volume), and the reference design are shown in Fig. 3. The z-axis dynamic deflection at the center of the tip induced by the previously described cyclic load is plotted as a function of frequency. Clearly, the optimum design has a better frequency response.

Case 2: Open Cylindrical Shells

The current section covers applications on a more complicated structure, such as the cylindrical laminated composite shell shown in Fig. 1b. The assumed loading conditions are: (1) a uniform static force of 0.31 lb/in (54.3 N/m) applied along the free-edge in the z-direction, and (2) a uniform cyclic force of 0.0625 lb/in (10.9 N/m) also applied along the free-edge in the z-direction. The objective function was set to minimize the maximum z-axis resonance amplitude of the first 5 natural frequencies at 3 sites (the middle and two corners of the free-edge). The dynamic deflections of both corners were incorporated into the design criteria, in addition to the deflections of the midpoint, in order to take into account the bending in transverse direction (y-axis) and the material coupling between bending and torsion. Frequency constraints were imposed on the first 5 natural frequencies such that:

$$\omega_n \leq 150Hz, \quad or \quad \omega_n \geq 350Hz, \qquad n = 1, ..., 5 \tag{21}$$

The z-axis static deflections along the free-edge were restricted to be less that 0.050in (1.27mm). The z-axis resonance amplitudes of the first five modes at the three sites were also restricted to be less than 0.050in (1.27mm). The reference design was a unidirectional shell $(0)_{20}$ of 50% FVR.

The resultant optimum designs for the 4 different laminate configurations are shown in Table 3. Table 3 also shows the maximum z-axis static deflection along the tip, the z-axis resonance dynamic amplitudes at the site with the maximum resonance peak, the natural frequencies, and the modal SDC's. The unidirectional reference design is also shown in Table 3. The reference design violates the static displacement constraint and the upper frequency bound of 350 Hz, since the 3rd and 4th natural frequencies are less than 350 Hz. All optimum designs have produced significant improvements to all performance measures, as they are within the feasible design space and exhibit superior dynamic and static stiffness. Contrary to Case 1 (beam), the consideration of multiple sublaminates and varying FVR's had a definite impact on the resultant optimal designs.

Subcase 2a: The optimum design for the simplest laminate configuration has produced a 39% decrease in the maximum resonance amplitude, in addition to a 59% decrease in static deflection, and has natural frequencies within the feasible frequency domain. The resultant optimum design is a ± 23.3 degrees angle-ply symmetric laminate.

Subcase 2b: The introduction of the FVR into the design vector has produced an optimum design of ∓ 25.8 degrees angle-plies with 0.62 FVR. With respect to the reference design, the optimum design has reduced the maximum resonance peak by 50%, the maximum static deflection by 69% and satisfies all frequency constraints. The optimum design has reduced the modal SDC's of the shell, hence, the additional improvements were mostly accomplished by the increased stiffness due to the higher FVR.

Subcase 2c: Multiple sublaminates produced further reductions in the objective function. In contrast to the beam in Case 1, the open cylindrical shell is a three-dimensional structure, therefore, multiple sublaminate systems are expected to provide better tailoring capacity. Indeed, the consideration of three sublaminates with equal and fixed fiber volume ratios has produced an optimum design with 53% reduction in objective function, 66% reduction in static deflections, and natural frequencies within the feasible frequency domain.

Subcase 2d: Three sublaminates with varying FVR's have produced the best improvement in the objective function, as the resultant optimum design exhibits a 64% reduction in the maximum resonance amplitude, decreased static compliance by 63%, and satisfies all frequency constraints. Interestingly, sublaminate 2 has been reduced to a passive damping layer. The outer sublaminate with ∓ 26.5 degrees angle-plies and 0.691 fiber volume ratio provides most of the stiffness. Sublaminate 2 with 0.010 FVR is virtually pure matrix and provides most of the damping. The inner sublaminate 3 with ∓ 12.8 degrees angle-plies and 0.245 FVR provides both in-plane stiffness and damping. This optimum design produced a constrained layer damping structure. The high first modal SDC indicates that the reduction in the dynamic amplitude was mostly accomplished due to increased damping.

In all previous subcases, both maximum dynamic and static deflections occurred at the corners of the free-edge. In contrast to the unidirectional design, all optimal designs have asymmetric bending modes due to coupling between torsion and bending. The mode shapes of the first four modes for the reference and optimum (subcase 2d) designs are shown in Figs. 4 and 5 respectively. The first mode in both designs is the first transverse bending mode. The second bending and twisting modes of the optimum design have been switched. Coupling between torsion and flexure was observed in the modes of the optimum designs.

The FRF's of the reference and optimum (Subcase 2d) designs are shown in Fig. 6. The z-axis dynamic deflections of the corner with the maximum resonance amplitude are plotted. The superior frequency response of the optimal design is apparent.

SUMMARY

An integrated formal method for the tailoring of structural composite damping was described. The method is based on static and dynamic performance criteria and its primary objective is optimization of the forced damped dynamic response of the candidate composite structure. The damping capacities of individual modes were optimized such that selected resonance amplitudes were minimized subject to constraints on static and dynamic deflections, and natural frequencies. The method incorporates unified micromechanics and laminate theories for composite damping and other mechanical properties, therefore, can be utilized for the simultaneous tailoring of basic composite systems and laminate configurations. The simulation of structural composite damping is based on finite-element analysis, for this reason, the method is applicable to a wide array of plate and shell composite structures.

Applications of the method included: (1) the structural tailoring of a composite beam, and (2) the structural tailoring of a composite shell. The effect of laminates with multiple sublaminate systems on the optimal designs was also investigated. The more important conclusions are summarized in the following paragraphs.

1. Both application cases have illustrated the effectiveness of the method. In both cases the tailoring of structural composite damping produced optimum designs with superior dynamic performance.

2. Tailoring of composite structures based only on static performance criteria and/or frequency constraints, and neglecting the effects of composite damping, may produce designs with poor dynamic performance.

3. The optimum tailoring of composite beams with laminates of one or multiple angle-ply sublaminates produced optimum designs with virtually equivalent dynamic performance. Compared to a unidirectional beam, the optimum designs have reduced the maximum resonance amplitudes by factor of 2, and have natural frequencies in the feasible frequency domain.

4. In the structural tailoring of the composite shell, laminates with multiple angle-plied

sublaminates and variable fiber content produced optimum designs with significantly improved dynamic performance. In view of the attained improvements, any higher costs that may result in the fabrication of more complex laminate configurations seem justified. With respect to a unidirectional shell, the optimum designs exhibited lower resonance amplitudes by factor of 2.76, lower static compliance by factor of 2.70, and natural frequencies within the feasible frequency domain.

5. The optimization of shells with multiple sublaminates of variable fiber volume ratio produced laminates with passive damping layers of pure matrix. These laminate configurations have provided the best dynamic performance of all other optimum designs, and resemble a constrained layer damping structure.

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Table 1. Mechanical properties of HM-S/epoxy system.

Epoxy	HM-S Graphite	50% HM-S/Epoxy(ref. [6])
E_m= 0.500 Mpsi (3.45 GPa)	E_{f11} = 55.0 Mpsi (379.3 GPa)	ψ_{l11} = 0.45 %
G_m = 0.185 Mpsi (1.27 GPa)	E_{f22} = 0.9 Mpsi (6.2 GPa)	ψ_{l22} = 4.22 %
ψ_{mn} = 10.30 %	G_{f12} = 1.1 Mpsi (7.6 GPa)	ψ_{l12} = 7.05 %
ψ_{ms} = 11.75 %	ν_{f12} = 0.20	
	ψ_{f11} = 0.4 %	
	ψ_{f22} = 0.4 %	
	ψ_{f12} = 0.4 %	

Table 2. Optimum designs for the beam (Case 1).

	Unidirectional Design	Optimum Designs (Case 1a)	(Case 1b)	(Case 1c)	(Case 1d)	(w/o damping optimization)
Ply Angles, degrees						
θ_1	0.0	26.69	26.09	26.46	25.97	5.66
θ_2	0.0	–	–	26.60	26.18	–
θ_3	0.0	–	–	0.02	29.99	–
Fiber volume ratios						
k_{f1}	0.50	–	0.700	–	0.700	0.408
k_{f2}	0.50	–	–	–	0.700	–
k_{f3}	0.50	–	–	–	0.495	–
Max. z-axis Static deflection (tip), (10^{-3} in)						
u_z^s	19.99	48.94	34.56	47.54	34.49	25.56
Resonance z-axis Amplitudes (tip), (10^{-3} in)						
U^r_{z1}	426.9	235.1	209.4	237.3	209.3	481.9
U^r_{z2}	0.0	0.0	0.0	0.3	0.0	0.1
U^r_{z3}	0.0	5.4	4.8	5.5	4.8	0.0
U^r_{z4}	9.3	0.1	0.1	0.1	0.1	10.2
U^r_{z5}	0.0	0.7	0.6	0.7	0.5	0.0
Natural Frequencies, (Hz)						
ω_1	383.2	254.1	289.2	257.5	292.2	350.0
ω_2	1088.7	1369.5	1530.9	1558.6	1492.5	1219.9
ω_3	1565.1	1587.1	1802.2	1608.0	1820.7	1577.7
ω_4	2395.5	2945.6	3272.5	2924.0	3293.6	2193.2
ω_5	3854.0	5072.6	5770.2	5130.9	5828.2	4167.6
Modal SDC's, (%)						
ψ_1	0.57	2.55	2.02	2.45	2.01	0.65
ψ_2	5.57	3.85	3.12	2.28	3.28	4.15
ψ_3	3.43	2.69	2.15	2.60	2.14	2.05
ψ_4	0.66	0.83	0.70	0.84	0.70	0.73
ψ_5	4.54	2.19	1.74	2.13	1.74	3.68

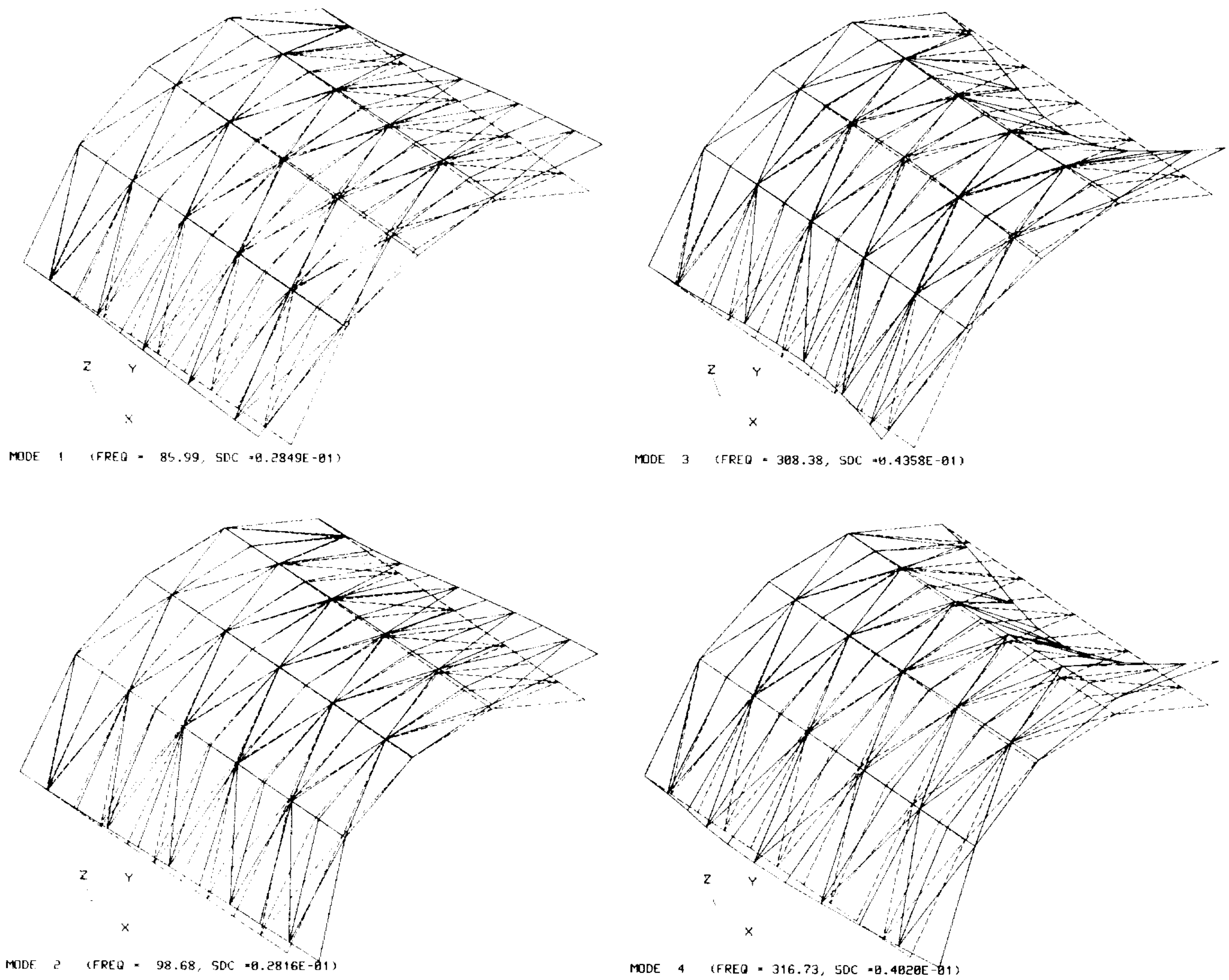
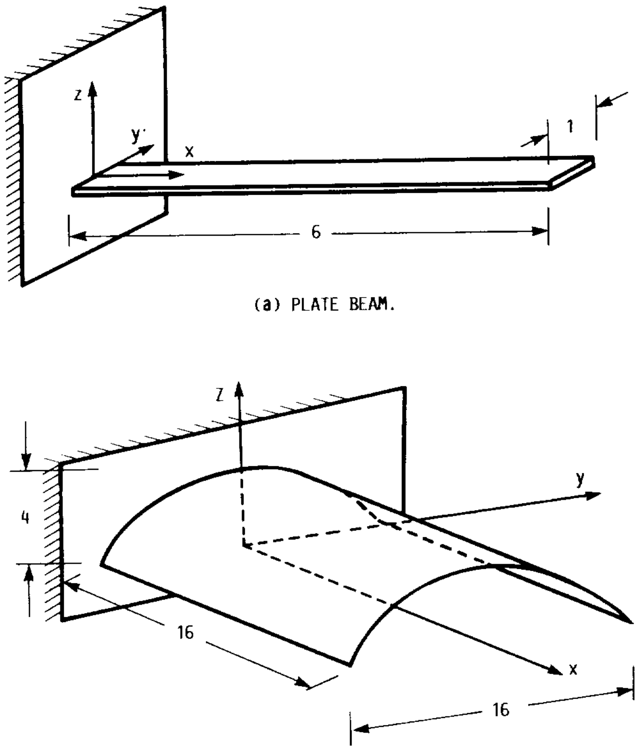


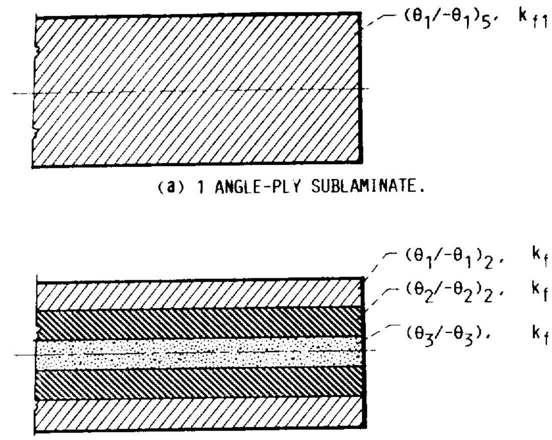
FIGURE 4. - MODE SHAPES OF THE UNIDIRECTIONAL COMPOSITE SHELL (CASE 2).



(a) PLATE BEAM.

(b) OPEN CYLINDRICAL SHELL.

FIGURE 1. - CANDIDATE COMPOSITE STRUCTURES. DIMENSIONS ARE IN INCHES.



(a) 1 ANGLE-PLY SUBLAMINATE.

(b) 3 ANGLE-PLY SUBLAMINATES.

FIGURE 2. - CANDIDATE LAMINATE CONFIGURATIONS.

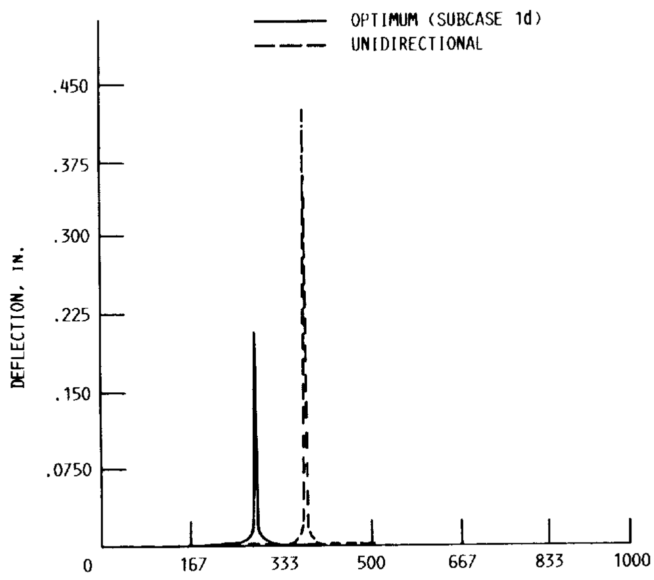


FIGURE 3. - FREQUENCY RESPONSE FUNCTION OF THE COMPOSITE BEAM (CASE 1). UNIDIRECTIONAL AND OPTIMUM (SUBCASE 1d) DESIGNS.

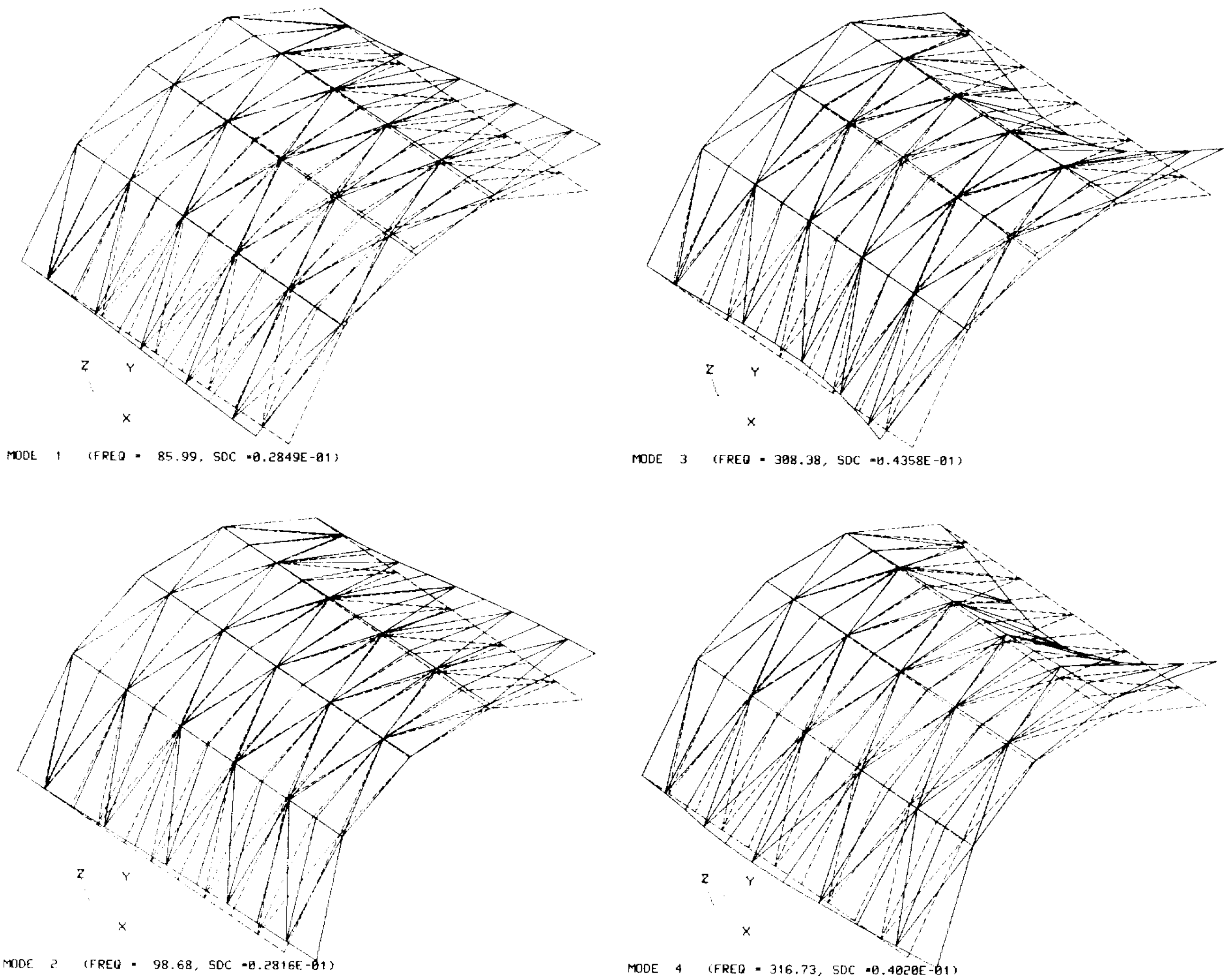


FIGURE 4. - MODE SHAPES OF THE UNIDIRECTIONAL COMPOSITE SHELL (CASE 2).

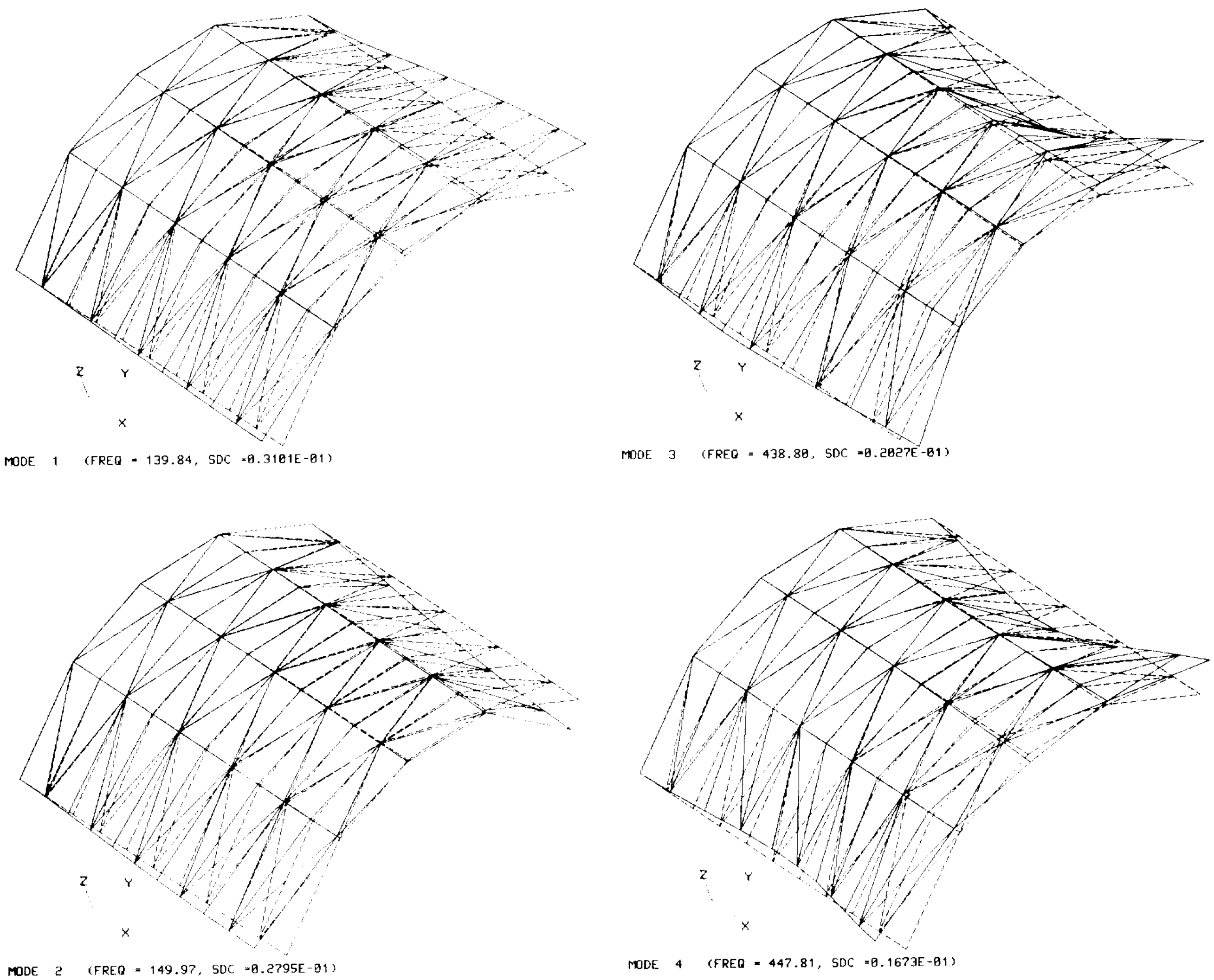


FIGURE 5. - MODE SHAPES OF THE OPTIMIZED (SUBCASE 2d) COMPOSITE SHELL.

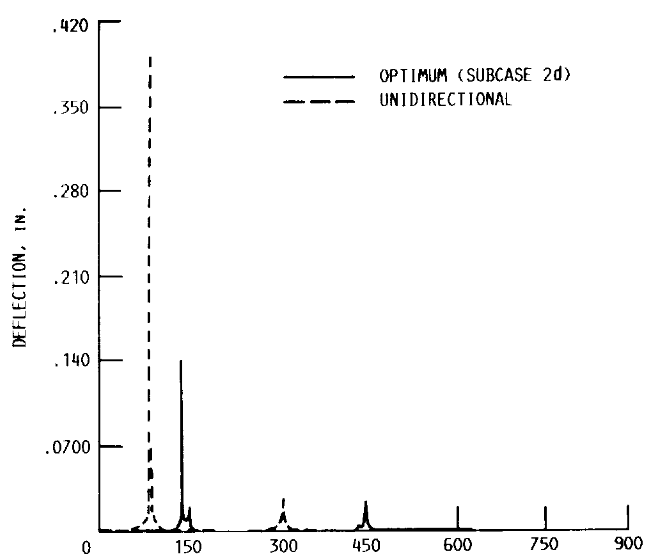


FIGURE 6. - FREQUENCY RESPONSE OF THE COMPOSITE SHELL AT THE SITE OF MAXIMUM RESONANCE AMPLITUDE. UNIDIRECTIONAL AND OPTIMUM (SUBCASE 2d) DESIGNS.

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National Aeronautics and Space Administration

Report Documentation Page

1. Report No.
NASA TM-102343

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle
An Integrated Methodology for Optimizing Structural Composite Damping

5. Report Date

6. Performing Organization Code

7. Author(s)
D.A. Saravanos and C.C. Chamis

8. Performing Organization Report No.
E-5051

10. Work Unit No.
505-63-11

9. Performing Organization Name and Address
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135-3191

11. Contract or Grant No.

13. Type of Report and Period Covered
Technical Memorandum

12. Sponsoring Agency Name and Address
National Aeronautics and Space Administration
Washington, D.C. 20546-0001

14. Sponsoring Agency Code

15. Supplementary Notes
Prepared for the 1989 Winter Annual Meeting of the American Society of Mechanical Engineers, San Francisco, California, December 10-15, 1989. D.A. Saravanos, National Research Council—NASA Research Associate.

16. Abstract

A method is presented for tailoring plate and shell composite structures for optimal forced damped dynamic response. The damping of specific vibration modes is optimized with respect to dynamic performance criteria including placement of natural frequencies and minimization of resonance amplitudes. The structural composite damping is synthesized from the properties of the constituent materials, laminate parameters, and structural geometry based on a specialty finite element. Application studies include the optimization of laminated composite beams and composite shells with fiber volume ratios and ply angles as design variables. The results illustrate the significance of damping tailoring to the dynamic performance of composite structures, and the effectiveness of the method in optimizing the structural dynamic response.

17. Key Words (Suggested by Author(s))
Composite structures; Damping optimization; Off-axis damping; Laminate damping; Structural damping; Forced dynamic response; Optimal design method

18. Distribution Statement
Unclassified – Unlimited
Subject Category 24

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No of pages	22. Price*
Unclassified	Unclassified	24	A03

NASA FORM 1626 OCT 86

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Space Administration

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