Traveling Wave Control for Large Spacecraft Structures

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This paper introduces the point of view that elastic deformations in large spacecraft structures may be aptly viewed in terms of propagating disturbances. Since the main topic of this paper is the control concepts, which result from such a viewpoint, the required structural dynamic description in terms of travelling disturbances is described only briefly, with reference to previously published works. The active control of these structures is approached from the point of view of actively modifying the natural disturbance propagation paths. Elastic energy is shunted into unimportant portions of the structure or is absorbed by an active "wave absorber." Several computational examples demonstrate the remarkable theoretical performance achievable by propagation-based controllers. Some discussion of practical difficulties in the implementation of such controllers is included.

Introduction

OVER the past five years, a trend has developed to divide the problem of control of flexible structures conceptually in two. It has been recognized[1] that several characteristics of the dynamics of flexible structures (uncertainty in the model, and the very large number of lightly damped structural modes participating in the response), make a direct approach to full-state estimation and feedback impractical. It is too easy to destabilize lightly damped and poorly modelled structural modes. The modes in greatest danger of being destabilized are typically those at, or just beyond, the bandwidth of such a "full state" controller.

One approach to alleviate these difficulties is damping augmentation of the structure, both by passive[2] and active[3] means. Active damping augmentation has been termed[4] "low-authority" control, and has been achieved by direct feedback of local velocity to a colocated force. If this is done at several locations, the feedback gain matrix generally can be full; as long as it is non-negative definite one is theoretically guaranteed stability of all flexible modes (with ideal sensors and actuators).[5] This basic idea also has been extended to the use of non-colocated sensors and actuators.[6] Such a low-authority controller can be thought of as an inner loop in the control structure. The outer loop is termed the "high-authority" controller and is designed with reference to the dynamics of the damped structure.[1]

It is not clear that damping ratios of flexible modes are the only measures of merit for a low-authority controller. The approach taken to low-authority control in this paper is only related to the procedures described above in that the objectives are similar; low authority control should modify the structural dynamics to ease the task of the high authority controller. The differences begin with the viewpoint taken towards the response of large spacecraft structures to typical disturbances. The prevalent approach is to describe the elastic response as a sum of the responses of many structural modes. Although this is a well-established technique, and has the advantage of broad applicability and availability of well-developed computational packages, it is not the only possible view, and in many cases may not be the best choice.

An alternative to modal analysis is to describe the structural response in terms of elastic disturbances that propagate through the structure. This alternative becomes ever more attractive as the external forces become spatially localized, the forcing frequencies become large relative to structural natural frequencies, and the structure becomes topologically simple. All three conditions are met by proposed large spacecraft structures to a greater degree than by aircraft, or most terrestrial structures. With these considerations as background, a disturbance-propagation approach to the structural dynamics of a restricted class of large spacecraft structures was developed over the past few years.[7,8,9] This naturally has lead to the development of the intuitive control concepts that are the subject of this paper.

The control ideas described here all modify the disturbance-propagation properties of the structure to which they are applied. Disturbances may be shunted into unimportant portions of the structure, to be dissipated there, or they may be very effectively dissipated locally by a controller based upon travelling wave concepts. A small relevant body of literature exists. Vaughan[10] has proposed an infinite dimensional controller, to be applied to one end of a beam in bending, that would imitate the behavior of a semi-inifite continuation of that beam, thus cancelling all reflections. Lallman[11] considered extensional strains of a space tether. He showed that a particular controller design, capable of controlling a low-order modal model of the elastic tether, lead to instabilities when the tether was modelled as an infinite-dimensional wave guide with propagation lags. The proposed "fix" for this was a passive damper, the design of which was based upon modification of the reflection coefficient at one end of the tether. Direct measurement of structure borne sound intensity is becoming possible due to recent theoretical[12,13] and experimental[12,14] developments. This is a related field, differentiated primarily by a several order of magnitude increase in frequencies and decrease in physical dimensions.

Disturbance Propagation Dynamics

References 7,8, and 9 describe the dynamics of structural networks in terms of travelling disturbances. The treatment is restricted to networks of slender structural members, which are connected to each other at junctions. The junctions can include flexible bodies, whose dynamics are described by a system of ordinary differential equations, quite possibly derived via the finite element method. Members are described by their governing partial differential equations or, in the case of trusswork members, by a modified Floquet theory, based upon the transfer matrix of a single trusswork bay.

Disturbances propagate along the members and transmit and reflect at junctions and discontinuities. The analysis makes extensive use of the concept of a travelling wave mode of a member. This must be briefly described.

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Travelling Wave Modes on Structural Members

The dynamics of any slender structural member are conveniently described in terms of its cross-sectional state vector, $y(x,\omega)$, typically containing pairs of cross-sectional deflections and related internal forces. There exist frequency-dependent wave-mode eigenvectors, $y_j(\omega)$, which represent combinations of cross-sectional state variables which will travel with unchanging relative magnitude along the member. Each wavemode eigenvector exhibits a particular propagation behavior, usually expressed in terms of its propagation coefficient $\gamma_j(\omega)=\alpha_j(\omega)+ik_j(\omega)$, where $i=\sqrt{-1}$, $\alpha_j(\omega)$ is the attenuation coefficient, and $k_j(\omega)$ is the wave number. These wave modes travel independently of one another within the member, each has the form $y_j(\omega)e^{\gamma_j(\omega)x}$. The wave modes come in pairs, corresponding to travel in opposite directions. A member model will support as many pairs of wave modes as it has deflection variables at each cross section.

The speed of propagation of each wave mode is given by the corresponding phase velocity; $c_p=\omega/k$. If this velocity is frequency dependent, the propagation is said to be dispersive; signals distort as they travel along the member. (A milder form of dispersion may arise due to frequency-dependent dissipation.) Most structural members, with the notable exception of cables in tension, rods in compression and torsion, and other situations described by the simple wave equation, are highly dispersive. An important form of dispersion arises within trusswork members. These members exhibit frequency ranges in which they are effectively opaque; signals at these frequencies do not propagate along the member but are spatially attenuated. These "stopping bands" occur near the resonant frequencies of local degrees of freedom within a single bay. Figure 1 shows the propagation coefficient of a periodic model of torsion deformation of a particular trusswork beam. The stopping band near $\omega=40$ rad/s is due to the excitation of such local degrees of freedom.

The resonant frequencies of such internal degrees of freedom may be rather freely chosen by the structural designer, possibly without penalty. Thus a structural member may serve as a passive vibration-isolation element, while still fulfilling its primary structural duties.

Scattering of Travelling Wave Modes at Structural Junctions

The following discussion makes reference to Fig. 2, which represents a "generic" structural junction. The nomenclature is taken from the field of microwave circuit analysis; the equivalent of Fig. 2 can be found in most basic textbooks in that field. In this structural application, the junction body β may be a flexible body.

Each member j supports incoming and outgoing wave modes, whose amplitudes are grouped in the vectors $\boldsymbol{a}_j$ (incoming), and $\boldsymbol{b}_j$ (outgoing). The member boundary conditions, and the dynamics of the (possibly flexible) body β can be transformed[7,8,9] into an equation which describes the behavior of the member wave modes at that junction;

$$_\beta \boldsymbol{b}={}_\beta \boldsymbol{S}(\omega)\,{}_\beta \boldsymbol{a}+{}_\beta \boldsymbol{B}_b^{-1}(\omega)\,{}_\beta \boldsymbol{f}_{\mathrm{EXT}}(\omega) \qquad (1)$$

where $_\beta \boldsymbol{b}(\omega)$ is the composite vector of all outgoing wave-mode amplitudes, $_\beta \boldsymbol{a}(\omega)$ is the vector of incoming wave-mode amplitudes, $_\beta \boldsymbol{S}(\omega)$ is the junction scattering matrix, and $_\beta \boldsymbol{B}_b^{-1}(\omega)$ is a matrix describing the generation of outgoing wave modes by the vector of applied external forces $_\beta \boldsymbol{f}_{\mathrm{EXT}}(\omega)$. The matrices $_\beta \boldsymbol{S}(\omega)$ and $_\beta \boldsymbol{B}_b^{-1}(\omega)$ describe the junction dynamics in wave-mode variables. They depend upon the transmission characteristics of the attached members, and upon the dynamics of the body β. Both these matrices are, in general, frequency dependent.

Figure 3 shows the frequency dependence of the scattering matrix of a "T" junction of a continuum model of a trusswork beam in torsion and a Timoshenko model of the same beam in bending. These continuum models were suggested by Noor and Andersen,[15] as were the actual parameter values used here. Such models are of limited applicability, since they do not show the banded filtering behavior of trusswork beams mentioned in the preceding section. They are used in this paper because a better alternative[8] has not yet been fully developed. The torsion model admits one wave mode in each direction, while the Timoshenko model admits a "bending" and a "shear" wave mode in each direction. Thus the scattering matrix is four by four. Only the lower triangle is shown, since with proper normalization of wave-mode eigenvectors, the scattering matrix is both symmetric and, in the absence of dissipation within body β, unitary. Most of the frequency dependence of these scattering coefficients is due to the propagation behavior of the Timoshenko bending model[8]; one of the mode pairs (a_3 and b_3) will not propagate at frequencies below 500 rad/s.

Network Dynamics

Descriptions of member dynamics in terms of wave modes and junction descriptions, as in Eq. (1), can be combined into a global dynamic description of an entire structural network in both the frequency and the time domain.[8,9] In the time domain, disturbances are synthesized in terms of piece-wise constant segments, whose progress as they distort and travel about the network is calculated with the aid of repeated convolution of component impulse responses.[8,9] The technique is computationally very intensive but also very accurate for short times.

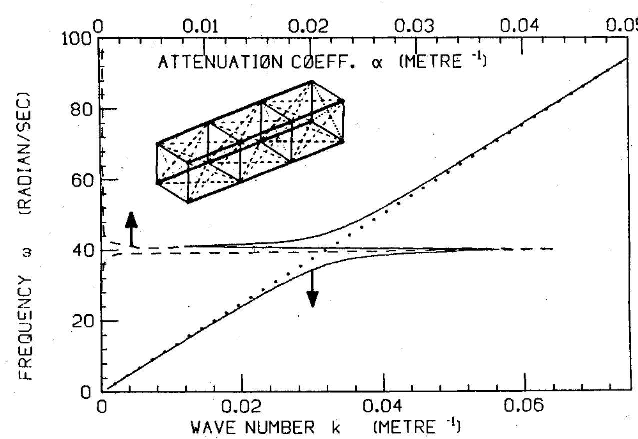

Fig. 1 Dispersion curves of a periodic torsion model based upon an equivalent continuum model of the pictured lattice beam. The continuum model was developed in Ref. 15. The discontinuity at $\omega=40$ rad/s is due to resonance of local degrees of freedom in the periodic model.

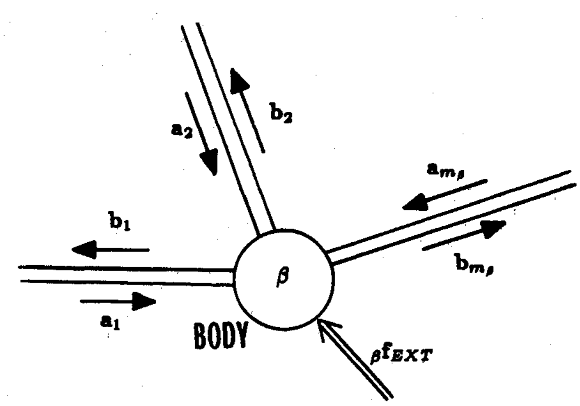

Fig. 2 The generic junction. The junction can include a flexible body and can be connected to many members. Each member, j, transmits incoming wave modes, a_j, and outgoing wave modes, b_j. External forces are grouped in the vector $_\beta f_{\mathrm{EXT}}$. The notation is standard in microwave circuit analysis.

Intuitive Control Synthesis

Viewing the structural response in terms of disturbance packets bouncing around the structure leads to a unique approach to the control of such structures. The objective is to modify the propagation characteristics of selected parts of the structure to achieve the control goals. As yet, no theory has been developed for justifying the chosen modifications; these choices must be made intuitively. Equation (1) is the basis upon which it is possible to propose active controllers that modify the propagation characteristics of junctions. Disturbances may be shunted into unimportant portions of the structure, there to be dissipated by active or passive methods. A junction may act as a universal wave absorber, preventing all reflections. If the "junction" is the termination of a single member, such a perfect wave-absorbing controller is termed a "matched termination" in the microwave literature.

Naturally, there are limitations to what can be achieved. The external (control) forces which achieve no outgoing wave modes ${}_\beta \boldsymbol{b}=0$ can be extracted from Eq. (1)

$$ {}_\beta f_{\text{EXT}}(\omega) = -{}_\beta \boldsymbol{B}_b(\omega)\, {}_\beta \boldsymbol{S}(\omega)\, {}_\beta \boldsymbol{a}(\omega) \tag{2} $$

which is a feed-forward law relating control forces to junction dynamics $[{}_\beta \boldsymbol{B}_b(\omega)$ and ${}_\beta \boldsymbol{S}(\omega)]$ and to the amplitudes of incoming wave modes ${}_\beta \boldsymbol{a}$. Junction modelling errors will limit the performance of such feed-forward control. A further limitation is that incoming wave-mode amplitudes cannot be measured directly, they must be inferred from measurements of physical variables. It will not always be possible or practical to measure the entire cross-sectional state vector of every member attached to a given junction, as would be required[8] to infer ${}_\beta \boldsymbol{a}$. Approximations for wave-mode amplitudes of beams, based upon partial measurements of the cross-sectional state vector at one point[12] and upon measurement of only deflection at a few closely spaced points[13] have been published. Such approximations will be possible for other members, but this is another effect which will prevent perfect behavior.

Equation (2) implies that one must use as many control forces ${}_\beta f_{\text{EXT}}(\omega)$ as there are outgoing wave modes. This will not usually be possible. At least part of the boundary conditions which describe the connection of several members at a junction will specify compatibility of displacements. Thus, the corresponding "force" is really a relative displacement. Such relative displacements are potential controls; however, a large amplitude or low-frequency actuation of such a control will imply effective disconnection of the structure. Thus there will be a frequency/amplitude limitation to such wave control, even if "full" actuation is used. If a fewer number of actuators are used than wave-modes depart an active junction, choices must be made about how to modify the open-loop scattering behavior. A controller which sets only part of the outgoing wave modes to zero would modify the junction dynamics so that elastic disturbances would be received from all directions, but would depart in only some directions.

Whether all or only part of the outgoing wave modes shall be cancelled, Eq. (1) together with a relation between incoming wave-mode aplitudes ${}_\beta \boldsymbol{a}(\omega)$ and measured cross-sectional state variables ${}_\beta y(\omega)$, yields a compensator of the form[8]

$$ {}_\beta f_{\text{EXT}}(\omega) = {}_\beta \boldsymbol{C}(\omega)\, {}_\beta y(\omega) \tag{3} $$

The compensator transfer functions ${}_\beta \boldsymbol{C}(\omega)$ are complex, transcendental functions of frequency. The implementation of this controller may not be trivial. One approach is to approximate the behavior with a finite-dimensional filter. Another approach would convolve the inverse Fourier transform of ${}_\beta \boldsymbol{C}(\omega)$ with the measurements

$$ {}_\beta f_{\text{EXT}}(t) = {}_\beta \boldsymbol{C}(t) * {}_\beta y(t) \tag{4} $$

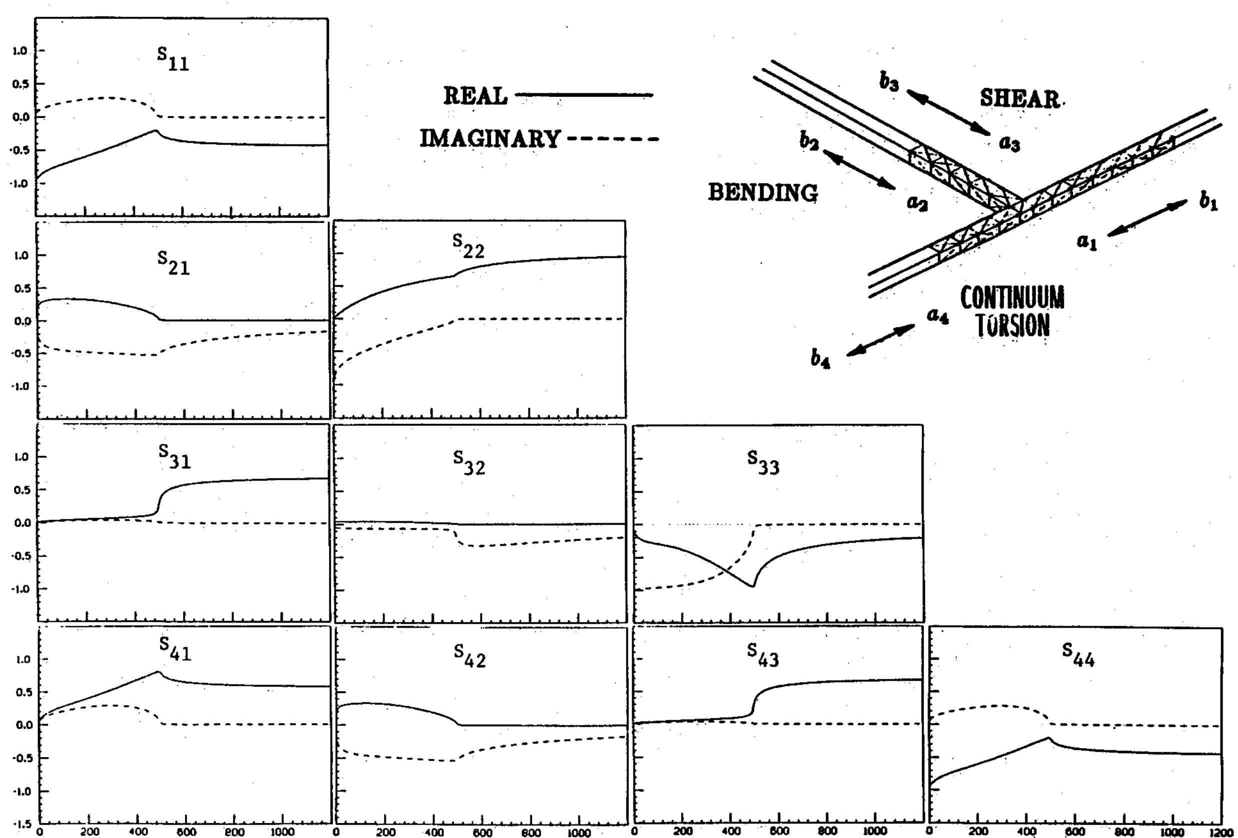

Fig. 3 The scattering matrix of the "T" junction of a beam modelled in torsion only, and a beam modelled in bending, with Timoshenko beam theory. The continuum models and parameter values employed are described in Ref. 15. Note that the scattering matrix is symmetric and frequency dependent.

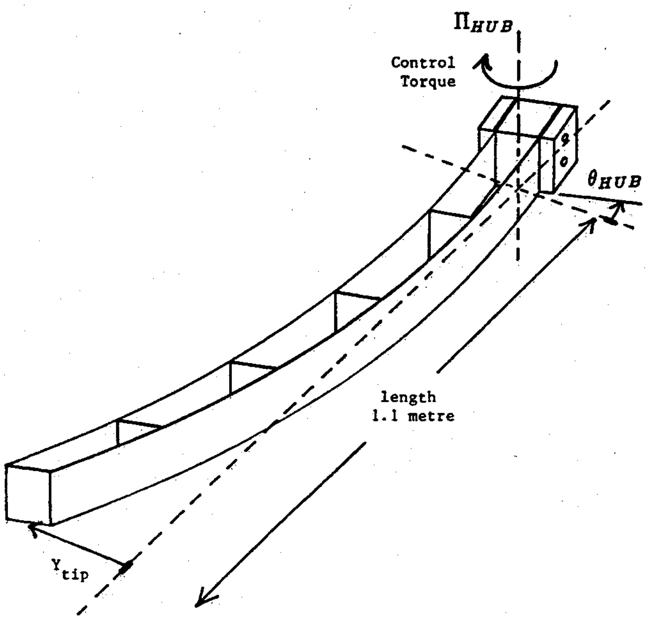

Fig. 4 A sketch of a one-link flexible manipulator in a laboratory at Stanford University.[16] The control problem is to use hub torque to perform slew maneuvers of the tip position. The beam has five flexible modes below 18 Hz.

Such convolution would take time. It is possible to create convolution kernels ${}_{\beta}C(t)$ with zero value for some initial time (thus providing calcuation time), by measuring the incoming disturbances some distance "upstream" of the junction. The travel time of the disturbance could then be used to prepare for it. This situation (it being advantageous to use separated actuators and sensors) is relatively unique in structural control.

Examples of Disturbance Propagation Controllers

The foregoing general discussion is perhaps best augmented by illustrative examples.

Wave Absorption in a Timoshenko Beam

Only the active termination of a pinned end of the Timoshenko beam is considered. Active termination of a free end is treated in Ref. 8. Both are relevant to a single-link flexible manipulator at Stanford University. Relevance also may be found in the problem of attitude control of a spacecraft consisting of a central rigid body with flexible appendages. Figure 4 is a sketch of the Stanford beam; further details should be sought in Ref. 16. The beam parameters chosen[8] for a Timoshenko model of the beam are: mass density, $\rho A = 0.425$ kg/m; rotary inertia density, $\rho I = 2.78 \times 10^{-4}$ kg-m; bending stiffness, $EI = 0.555$ N-m²; shear stiffness, $GA_s = 30.7$ N; and hub rotary inertia, $I_o = 0.0115$ kg-m². The beam has a length of 1.1 m. The control problem posed for the Stanford manipulator is to use a torque applied to the hub to perform slew maneuvers of the tip position. The problem is made interesting by the existence of many low-frequency structural modes and by the low damping inherent in the structure. Both characteristics are relevant to the problem of control of large spacecraft structures.

It is possible to view the dynamics of this structure in terms of travelling waves. The Timoshenko beam model admits two deflections at each cross section: face rotation and lateral deflection. Thus there will be two wave modes travelling in each direction. These two modes are conventionally termed the bending and shear modes, according to the dominant entry in the corresponding wave-mode eigenvector. For the beam

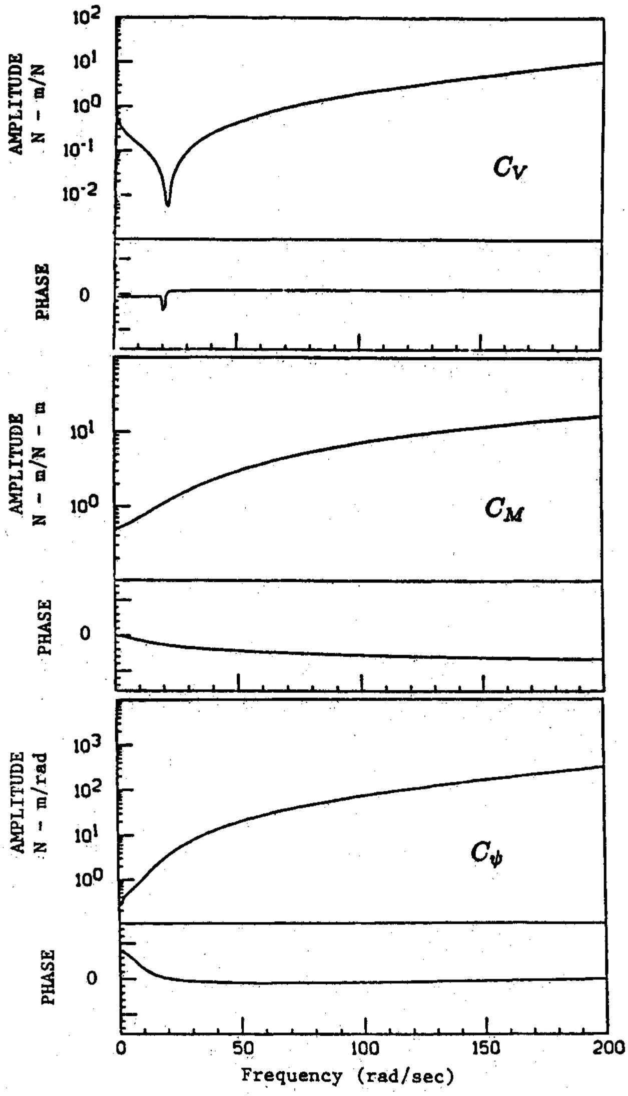

Fig. 5 Infinite dimensional, infinite bandwidth compensators for the Stanford beam. If these compensators are used to feed back hub beam variables to hub torque, as in Eq. (5), the bending traveling wave mode of the Timoshenko beam model, departing the hub, is set to zero. A resulting closed-loop transfer function is shown in Figure 6.

parameters stated above, the shear mode is a "near field" below $\omega = \sqrt{GA_s/\rho I} = 332$ rad/s. Such a near field propagates no energy, and decays exponentially with distance along the member. Its propagation coefficient is pure real.

Note that an essential hub boundary condition is that there be no lateral deflection; the hub is pinned. Thus, although there are two outgoing wave modes, only one torque is available to effect hub reflection control. For this example, Eq. (3) becomes

$$\Pi_{\rm HUB} = C_{\psi}(\omega)\psi_{\rm HUB} + C_M(\omega)M_{\rm HUB} + C_V(\omega)V_{\rm HUB} \qquad (5)$$

where $\Pi_{\rm HUB}$ is the control torque to be applied to the hub, in response to measured hub face rotation, $\psi_{\rm HUB}$, hub bending moment, $M_{\rm HUB}$, and hub shear force, $V_{\rm HUB}$. The three compensators, $C_{\psi}(\omega)$, $C_M(\omega)$, and $C_V(\omega)$ that set the outgoing bending mode to zero are plotted in Fig. 5. The frequency range $\omega < 100$ rad/s includes the first five flexible open loop modes. Note that all three of these compensators are infinite bandwidth and infinite dimensional. It is not practical to derive explicit algebraic expressions for these compensators. They have been calculated for a set of finely spaced frequencies, and plotted.

The compensation scheme of Eq. (5) requires measurement of hub rotation, internal root bending moment, and internal root shear force. Of these, the root shear force is practically inaccessible to measurement. If one is willing to measure and feed back both hub angle and root bending moment, good results can be obtained with

$$\Pi_{HUB} = C_M(\omega)M_{HUB} - C_\psi(\omega)\theta_{HUB} \tag{6}$$

Where the hub boundary condition is $\theta_{HUB} = (V_{HUB}/GA_s) - \psi_{HUB} \approx -\psi_{HUB}$.

Leaving aside, for the moment, questions of realizability, it is instructive to consider the closed-loop behavior of the beam controlled by these compensators. Figure 6 compares open and closed loop responses of tip position to additional hub torque. These are transcendental transfer functions, calculated with exact solutions of the Timoshenko beam equations.[8] The effectiveness of the reflection-cancelling controller of Eq. (5) is remarkable; all resonances have disappeared. Although a root locus has not been calculated for this example, a similar example[8] suggests that all poles have been moved to $s = -\infty + i\omega$. The response of the tip is now little more than a multiple integration, with a lag of approximately 130 ms. A lag of this duration has been observed experimentally.[16] This is also the high-frequency limit of the travel time of the bending mode in the Timoshenko beam model. The cancellation of hub reflections has exposed this essential behavior of the beam by removing the obscuring "noise" of resonances.

Figure 6 also gives the closed-loop performance of the compensator of Eq. (6). Note that the structural modes are beginning to reappear; evidently, some reflection is occuring at the hub. Ref. 8 gives further variations of this theme; feedback of only hub angle and only root-bending moment. The response becomes successively more like the open-loop behavior.

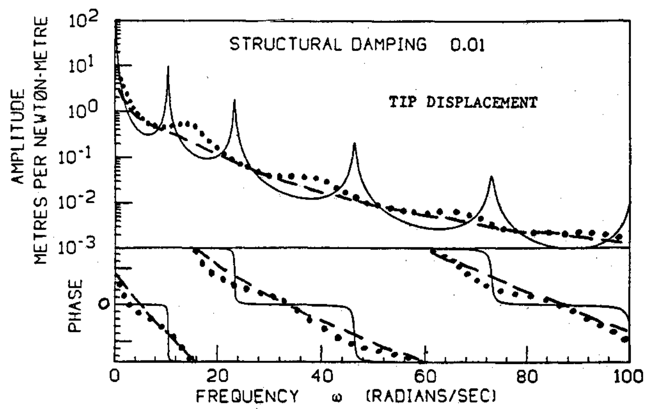

Fig. 6 Open-loop (—) and closed-loop response of the tip position of the Stanford beam to hub torque, as calculated with a Timoshenko beam model. The closed-loop response with the compensator of Eq. (5) (---) is little more than a lagged, multiple integration. All the poles have been moved to infinity. The closed-loop response with the compensation of Eq. (6) (.....) is also shown.

A question which has not been addressed in this example is the realization of the compensators $C_\psi(\omega)$, $C_M(\omega)$, and $C_V(\omega)$. At low frequencies, it appears possible to approximate the compensator behavior well with simple analog circuits. It is suspected that the inevitable "rolloff" of such approximations and of any finite bandwidth actuator may reduce the stability of flexible modes in this rolloff region. Analysis of these questions is the topic of current work.

Disturbance Shunting in a Hypothetical Large Spacecraft Structure

Figure 7 introduces a hypothetical structure which was proposed for the development of disturbance propagation concepts.[8] The members are all trusswork members whose dimensions and equivalent continuum models were introduced by Noor and Andersen.[15] To keep the problem simple, not all degrees of freedom are considered here. With the degrees of freedom described in the caption to Fig. 7, this structure has 14 wave modes departing five junctions. External forcing is provided by a torque applied to the free end of one member, which is modelled only in torsion. Thus there is only one wave mode departing from the driven point. It scatters at the first junction, generating four wave modes departing this junction. Two of these travel along member E (where Timoshenko bending theory is used), and one travels along each of members A and B (where only torsion is modelled).

The open-loop response of this structure, measured at three points, is given in Fig. 8. These are transcendental transfer functions and represent exact solutions of the governing system of partial differential equations.[8] Assumed structural hysteretic damping of 1% prevents infinite response at resonance.

As an arbitrary design exercise, a compensator has been calculated which prevents waves from departing junction 4 in the direction of the driven point; that is, with reference to Fig. 3, $b_1 = 0$. The control force to accomplish this was arbitrarily chosen to be an external moment applied to junction 4.

Once again, Eq. (1), together with a bit of algebra, yields a compensator of the form

$$\Pi_{EXT} = C_{\theta_A}\theta_A + C_{\tau_A}\tau_A + C_{W_E}w_E + C_{\psi E}\psi_E + C_{M_E}M_E + C_{V_E}V_E + C_{\theta_B}\theta_B + C_{\tau_B} \tag{7}$$

where θ (rotation) and τ (torque) are the cross-sectional state variables of the two torsion members at the junction, and w (lateral deflection), ψ (face rotation), M (bending moment), and V (shear force) are the four cross-sectional state variables of the bending member at the junction. Subscripts denote to which member the variable corresponds. Note that all eight local cross-sectional state variables are used by this compensator. The open-loop scattering matrix of this junction (Fig. 3) shows that all arriving wave modes must be countered. One of the eight compensators of Eq. (6) is displayed in Fig. 9. Note that once again, these compensators are infinite both dimensionally and in bandwidth.

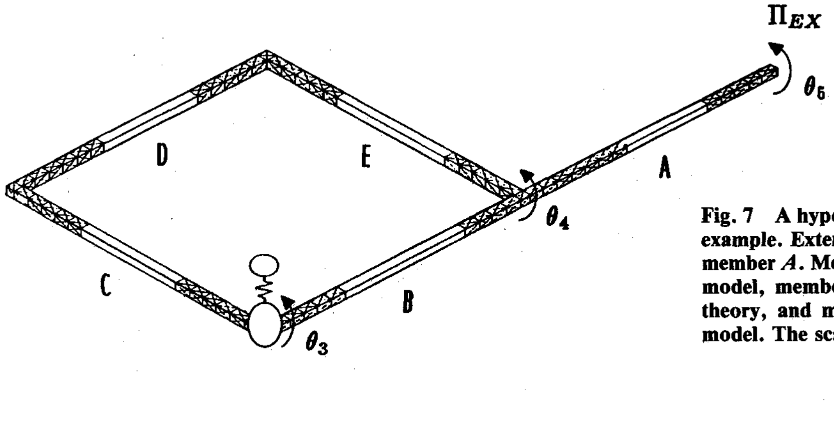

Fig. 7 A hypothetical large spacecraft structural segment, used as an example. External excitation is applied as a torque to the free end of member A. Members A and B are modelled with a continuum torsion model, members C and E are modelled with Timoshenko bending theory, and member D is modelled with a simple periodic torsion model. The scattering behaviour of junction 4 is presented in Fig. 3.

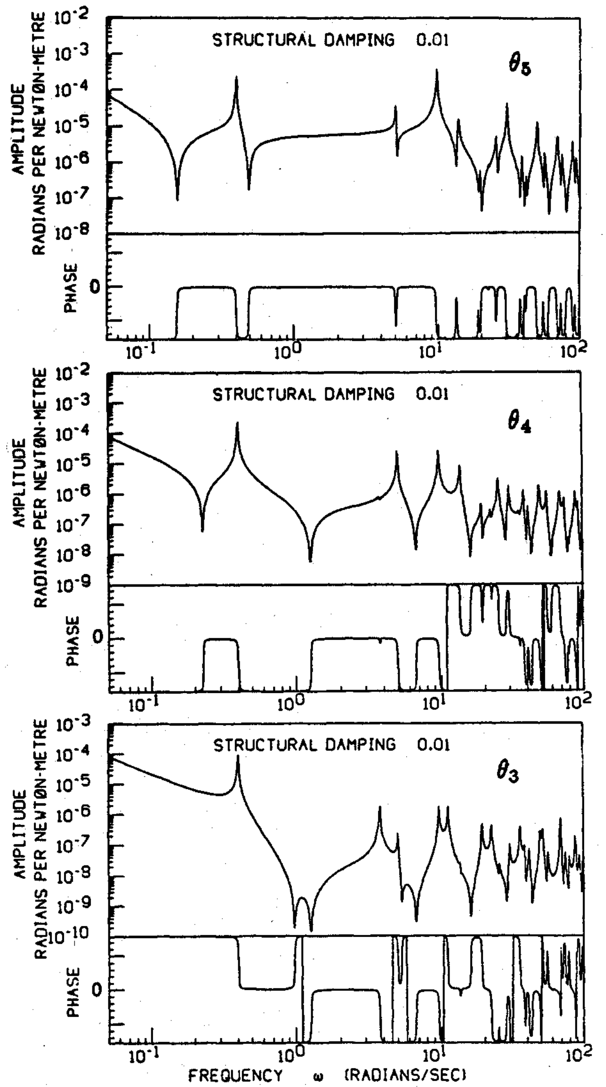

Fig. 8 Open-loop responses of the structure described in Fig. 7, as calculated by an exact solution of the governing partial differential equations. Structural hysteretic damping, with the imaginary part of the stiffness equal to 1% of the real part, is used.

If these eight compensators could be applied to the structure, the structural responses would be modified to those of Fig. 10. These should be compared with the open-loop responses of Fig. 8. Once again, there is a dramatic change in the structural response. Seen from the driven point, the structure behaves as a semi-infinite member; no energy comes back to the driven point. The response of the rotation of junction 4 is also that of a semi-infinite member; the response is lagged by 156 ms, the travel time of torsional disturbances on member A. In contrast, the response of θ_3 has not been damped at all; the magnitude of its response has been increased by approximately a factor of ten. The locations of poles and zeroes in that response have been drastically changed.

Simplified and realizable derivatives of this ideal matching compensator no doubt could be proposed. This development, and the resulting degradation in performance, is left as a topic for future research. It would also have been possible to design a compensator for junction 4 that cancels more of the outgoing wave modes. The boundary conditions for this junction admit one further external force. Thus with external forces alone, two wave modes can be cancelled. The use of two relative rotations as additional controls would permit the cancellation of all outgoing wave modes from this junction. All energy incident upon the junction would be absorbed, and since the junction is centrally located in the structure, all structural transfer functions would be heavily damped. The previously discussed limitations would limit this accomplishment to high frequencies and/or low amplitudes. These extensions are also left as topics for future work.

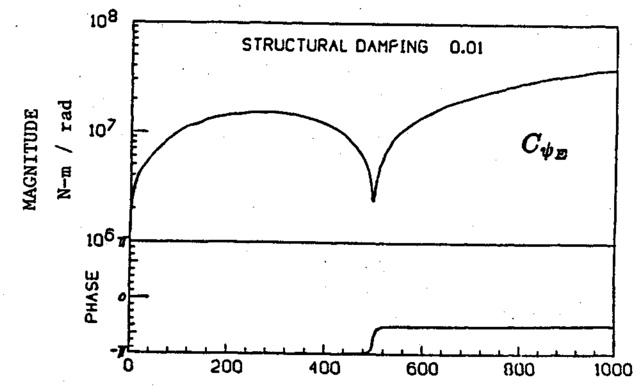

Fig. 9 Frequency dependence of one of the eight compensators of Eq. (7). This compensator feeds local rotation at junction 4 of the structure of Fig. 7 back to a colocated external torque.

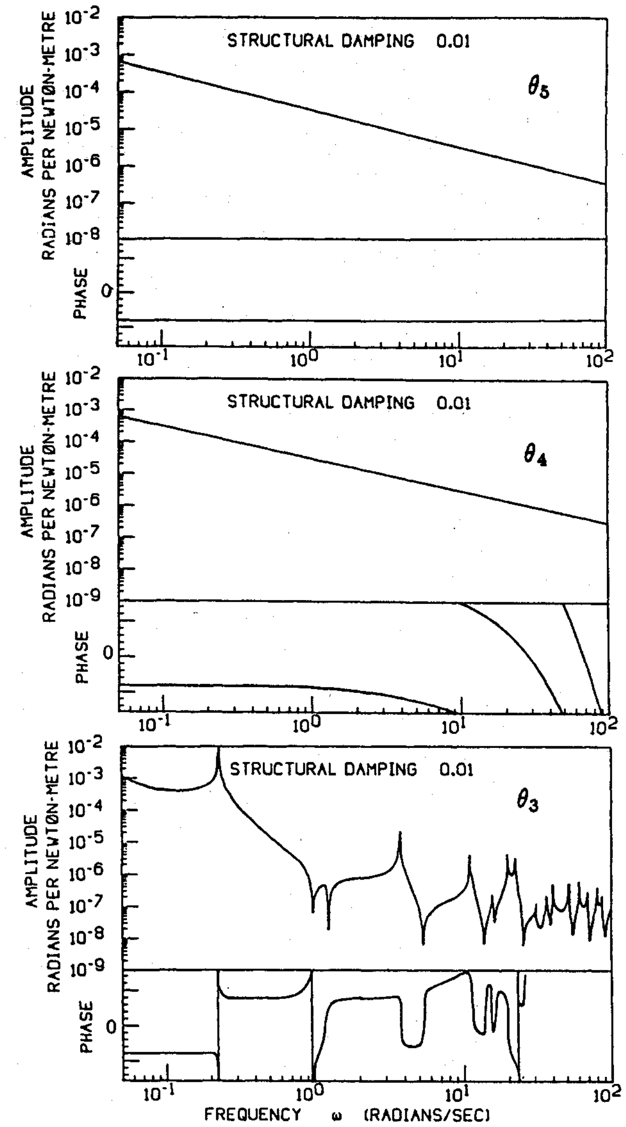

Fig. 10 Closed-loop response of the structure of Fig. 7. The compensator described by Eq. (7) and Fig. 9 is active at junction 4. Note that the response of θ_3 is approximately ten times greater than for the open-loop case, and that it has a new set of poles and zeroes.

Discussion

This paper has introduced the view-point that the elastic response of large spacecraft structures may be aptly viewed in terms of the disturbance propagation characteristics of the structure. This viewpoint (and the associated mathematics) leads to the idea of actively modifying these disturbance propagation characteristics as a first step towards controlling the structure. Several considerations suggest that this interference with the natural disturbance paths will be impractical at very low frequencies and large amplitudes. Thus there still will be a need for a controller operating as an outer loop, and control based upon disturbance propagation concepts might be termed "low-authority control."

Since this paper was written, disturbance propagation control has been applied (in its simplest form) to a laboratory structure.[17] Many of the anticipated difficulties were encountered. One surprising result of the work was the great similarity between a wave-absorbing and an "optimum" modal-based controller.

Acknowledgments

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