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A NEW APPROACH FOR VIBRATION CONTROL
IN LARGE SPACE STRUCTURES

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Abstract

A new approach for augmenting vibration damping characteristics in space structures with large panels is presented. It is based on generation of bending moments rather than forces. The moments are generated using bimetallic strips, suitably mounted at selected stations on both sides of the large panels, under the influence of differential solar heating, giving rise to thermal gradients and stresses. The collocated angular velocity sensors are utilized in conjunction with mini-servos to regulate the control moments by flipping the bimetallic strips. A simple computation of the rate of dissipation of vibrational energy is undertaken to assess the effectiveness of the proposed approach.

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Introduction

Recent trends in space technology have been toward significantly larger and lighter weight structures.[1-3] The successful experimental mission of Communications Technology Satellite (Hermes) which employed a pair of solar panels measuring 11 x 7.3 m each is already behind us. One of the Large Space Structures (LSS) currently under development is the wrap-rib mesh deployable antenna[4,5] with a diameter of 100 m and surface thickness of 0.0254 mm. Another example is the spinning sail[6] which consists of a spacecraft propelled by photon light pressure impinging upon 12 blade-like surfaces, each measuring 8 m x 4000 m x 0.00254 mm and made of extra-light material. With Space Transportation System already in operation, several other space missions demanding varying sizes of LSS are under consideration for technological applications such as power generation and transmission, earth-resources observations and large scale communications related to mass education and electronic mail systems.[7]

Space structures with large dimensions and minimal weight characterized by distributed mass and elasticity properties are extremely flexible and, thus, are highly susceptible to on-board disturbances causing surface deformations and vibrations. Since, achieving adequate performance of LSS generally relies on precise pointing and/or precise structural configuration, it is necessary to minimize their structural response to disturbances through judicious design and by incorporating suitable control devices.

The dynamic analysis, design and control of an LSS poses a formidable technological challenge, calling for a major all-out effort.

The structural flexibility may interact with attitude control systems of spacecraft in a variety of ways. A NASA special report[8] has presented an excellent review of the earlier efforts in this area. It is interesting to note that what began as isolated efforts to understand the anomalous behavior of some specific orbiting spacecrafts by accounting for flexibility has emerged into a new field of astroelasticity.[9] In the early era, there had been considerable emphasis on developing adequate models[10] of the LSS which could account for the flexibility of the appendages. Even concise formulations of the spacecraft structures treating them as "hybrid" assembly of rigid and flexible components seem to represent a significant achievement.[11,12] Simulations of the dynamics of spinning and non-spinning systems with flexible arrays were attempted in several specific cases. The results of some of these analyses have also been verified by ground testing. During the last ten years, the emphasis seems to have shifted to investigations related to the problems involved in controlling the LSS, the stability of controls and the interaction of dynamics and control functions.

The methods most often proposed for controls are active ones although some passive approaches not requiring direct use of energy have also been suggested. For example, stiffening of an LSS can be utilized to improve its dynamic characteristics in a number of situations but only at the expense of severe weight penlaties imposed. Besides, additional stiffness does not automatically add to structural damping. A feasible damping technique considered in the past is to incorporate mechanical dampers such as linear viscous dashpots into the structure.

Even this approach appears to be inadequate and has not gained much acceptance with spacecraft designers. This is perhaps due to only marginal improvements in damping characteristics associated with the rather large weight penalties involved.

Suggested here is a new approach for achieving positive damping action based on the use of moments rather than forces. In order to provide for the development of the controlling bending moments, it is proposed to have a number of moment generating units suitably spread on both sides of the large spacecraft appendages (Fig. 1). A suitable combination of strips of two different metals forms these units with constraints at the ends to obstruct any possible relative elongation or contraction of the strips under differential solar heating. This in turn generates the tensile and compressive stresses in the two members giving rise to a bending moment M_Y at the pin through which the whole unit is mounted on the panel.

For control implementation, the use of an "on-off" controller is proposed here. The sensors, essentially collocated with the bimetallic strips are required to measure the local angular velocities of the appendages. The mini-servo units are utilized to rotate/flip the strips in "on-off" positions depending upon the direction of the local angular velocity as indicated by the sensor. While in "on" position, the strips generate the bending moment through their differential thermal heating under the influence of the solar radiations. When the positive Z face of the strips is exposed to the sun and the local angular velocity of the panel about Y-axis is positive, the corresponding moment about this axis remains negative causing dissipation of vibrational energy. On the

other hand, during the vibration phase of negative local angular velocity, the strips are turned into "off" position by flipping them through 90° about x-axis when the controlling moment virtually disappears. In this sun-orientation, the strips on the other side of the panel remaining unexposed to the sun do not generate any moment and may be kept in "off" position. However, when the sun moves to the other side of the panel, heating the negative Z face of the strips, the moment generated about the Y-axis becomes positive in the "on" position, thus requiring the control law to be reversed. Now, the strips are kept in the "on" position, generating positive moment while the local angular velocity of the panels remains negative. At other times, the strips are flipped into the "off" position.

Rates of Energy Dissipation

The success of the proposed control mechanism depends upon the level of rates of vibrational energy dissipation that can be achieved. The computation of these is the objective of the analysis undertaken here. It may be emphasized that since the objective is to establish the feasibility of the proposed concept, only an approximate analysis is attempted.

The feasibility study is based on a simple cantilever beam model of an LSS. Several moment units/packets are assumed distributed along the length of the beam. Both the metallic pieces in the packets are assumed to have rather small cross-sectional dimensions as compared to the length. Therefore, 1-d analysis permitting temperature variation only along the thickness would be sufficient. In steady thermodynamic state, the differential equation governing the temperature distribution across

the thickness of any of the two pieces can be written as (Fig. 2)

$$\frac{d^2 T}{d\xi^2} = 0 \qquad (1)$$

where T is the temperature at position ξ. In order to maximize the temperature differences, an insulating material is introduced into the gap between the two pieces. Then, the boundary conditions for the "hot" piece can be stated as follows:

$$- K_1 \frac{\partial T}{\partial \xi} = \alpha_a S - \alpha_e \sigma T^4 = 0 \text{ at } \xi = - a_t \qquad (2)$$

where

K_1 = Thermal conductivity of material for the "hot" piece.

$\alpha_a, \alpha_\epsilon$ = Absorptivity and emisivity of the metallic surface, respectively.

S = Rate of solar energy received per unit area of the exposed surface; is same as the solar constant at normal incidence.

σ = Stefan–Boltzmann constant.

From Eq. (2), the temperature of the "hot" piece is given by

$$T = (\frac{\alpha_a S}{\alpha_\epsilon \sigma})^{1/4} \qquad (3)$$

Since the other piece virtually does not receive any heat radiations, it can be assumed to be at 0° absolute. Under the temperature differences thus set up and neglecting the effect of insulating material on the thermal stresses, the resulting tensile and compressive forces in the two members can be obtained using geometric compatibility condition

$$F = \frac{\alpha_1 \Delta T}{(A_1 E_1)^{-1} + (A_2 E_2)^{-1}} \qquad (4)$$

where

$$\Delta T = (\alpha_a S / \alpha_\epsilon \sigma)^{1/4}$$

α_i = coefficient of thermal expansion; i = 1,2

A_i = area of cross section; i = 1,2

E_i = modulus of rigidity; i = 1,2

The subscripts 1 and 2 refer to the "hot" and "cold" members of the assembly. At 0° absolute, the two members in the assembly are assumed to have been kept in "free-force" condition.

The resulting moment in the steady state is given by

$$M_Y = \frac{\alpha_1 \delta (\alpha_a S / \alpha_\epsilon \sigma)^{1/4}}{(A_1 E_1)^{-1} + (A_2 E_2)^{-1}} \tag{5}$$

where δ = mean distance between the two pieces.

The local angular rate of the large panel $(\overset{\circ}{\theta})$ can be expressed as follows:

$$\overset{\circ}{\theta} = - \frac{\partial^2 W}{\partial X \partial t}$$

where W = deflection along Z-axis at the station X.

The instantaneous rate of energy dissipation $(-\overset{\circ}{E})$ is then given by

$$(-\overset{\circ}{E}) = - M_Y \overset{\circ}{\theta} \tag{6}$$

The standard modal analysis of the appendage vibrations leads to the following expression for $\overset{\circ}{\theta}$.

$$\overset{\circ}{\theta} = - \sum_{j=1}^{\infty} \left(\frac{d\phi_j}{dX} \overset{\circ}{\xi}_j \right)$$

where

$$\phi_j = c_1 (\sinh u - \sin u) + c_2 (\cosh u - \cos u)$$

$$u = (i\pi/L)X$$

$$L = \text{length of the appendage}$$

$$\xi_j = \xi_{jmax} \sin (\omega_j t + B)$$

$$\xi_{jmax} = \text{amplitude of the jth mode of panel vibrations}$$

$$\omega_j = (i^2\pi^2/L^2) a$$

$$a = \sqrt{EI/m}$$

$$i = 0.597 \text{ for } j=1; \ 1.49 \text{ for } j=2; \ (n-1/2) \text{ for } j=3,4,\ldots$$

$$(EI), m = \text{section modulus and mass/length of the appendage,}$$
respectively.

On isolating the rate of energy dissipation corresponding to the jth mode and integrating it, one finds that the average loss of kinetic energy for this mode per vibration cycle is given by

$$\Delta E_j = \frac{2\alpha_1 \delta(\alpha_a S/\alpha_\varepsilon \sigma)^{1/4}}{(A_1 E_1)^{-1} + (A_2 E_2)^{-1}} (\frac{d\phi_j}{du})\bigg|_{X=L_i} \xi_{jmax} \sqrt{\omega_j/a} \tag{7}$$

where L_i specifies the position of the ith station where the moment-unit is located. This now must be compared to the maximum kinetic energy of vibrations corresponding to the jth mode (E_j). It is easy to show that

$$E_j = (1/2) \ m\xi_{jmax}^2 \ \omega_j^2 \int_{X=0}^{X=L} \phi_j^2 \ du \ \sqrt{a/\omega_j} \tag{8}$$

On dividing (7) by (8), the loss factor η_j is obtained as

$$\eta_j = \frac{4\alpha_1 \delta(\alpha_a S/\alpha_\varepsilon \sigma)^{1/4}}{(A_1 E_1)^{-1} + (A_2 E_2)^{-1}} \ \frac{1}{\xi_{jmax}} \ \frac{1}{\omega_j \sqrt{EI \ m}} \ \frac{(d\phi_j/du)\big|_{X=L_i}}{\int_{X=0}^{X=L} \phi_j^2 \ du}$$

In case the moment generating units distributed along the length of the appendage are at distances L_1, L_2,... from the clamped end of the appendage, the above results modify to:

$$\eta_j = \eta_{oj} \, f_j \tag{9}$$

where

$$\eta_{oj} = P_m / [\omega_j \, \xi_{jmax} \sqrt{EIm}] \tag{10}$$

$$P_m = \frac{4\alpha_1 \delta (\alpha_a S / \alpha_\epsilon \sigma)^{1/4}}{(A_1 E_1)^{-1} + (A_2 E_2)^{-1}} \text{, the moment parameter} \tag{11}$$

$$f_j = [(d\phi_j/du)_{X=L_1} + (d\phi_j/du)_{X=L_2} + ---]/\int_{X=0}^{X=L} \phi_j^2 \, du \tag{12}$$

Results and Discussion

The approximate analytical results developed here are of considerable significance. Mere observation suggests the effects of various system and design parameters on the loss factor, denoting the fraction of kinetic energy dissipated per cycle in various vibrational modes. The loss coefficient rapidly increases with increase in length of the appendage and hence decreasing frequency. In fact, this factor increases in proportion to L^2. It also suggests that the damping effectiveness progressively declines for higher modes of vibrations, as is usually the case with other damping techniques as well.

It is evident that the vibrational energy dissipation can be maximized by maximizing the design parameters P_m and f_j. The maximization of P_m demands the use of materials which provide the highest values of thermal coefficients, α_1, and modulii of rigidity, E_1 and E_2.

Further increase in p_m can be achieved by moving the two pieces apart to maximize the moment arm and of course by taking the higher areas of cross-section.

The parameters f_j merit special attention. By changing the longitudinal distribution of the moment-strips, it is possible to augment the damping effects in certain chosen/critical modes. However, it will result in compromising with dissipation rates associated with other modes. Rather simple algebraic calculations can be performed to determine the "best" locations. It is interesting to note that this approach enables attaining fairly significant values of loss factor even for higher modes. Since the control effectiveness increases with decreasing frequency, it appears particularly attractive for LSS.

Conceptually, the approach suggested here is a simple one. However, it involves several challenging problems at the design stage. The problems of mounting of the moment-strip units at various stations on a flexible light-weight structure, providing for the measuring sensors and the mini-servos which can control the orientation of the moment-strip unit according to the control policy proposed have to be tackled. The loss factors undergo periodic variation becoming zero during the periods of grazing solar incidence on the strips. Thus, in this phase, the damping mechanism appears to be ineffective. However, this problem can be overcome to a large extent simply by flipping the moment-strips through small angles (~30-45°) so as to ensure a significant differential solar heating even in this situation.

No doubt, the proposed control mechanism involves significant weight penalty associated with the use of the moment-strips and other

accessories; however, it is hoped that a judicious design can keep it to within acceptable limits. Besides, the addition of this mass may also serve the secondary purpose of stiffening the system. Furthermore, the power required to drive the servos is expected to remain at low level. In a nutshell, this investigation brings out several interesting and useful features of the proposed scheme to augment the system damping.

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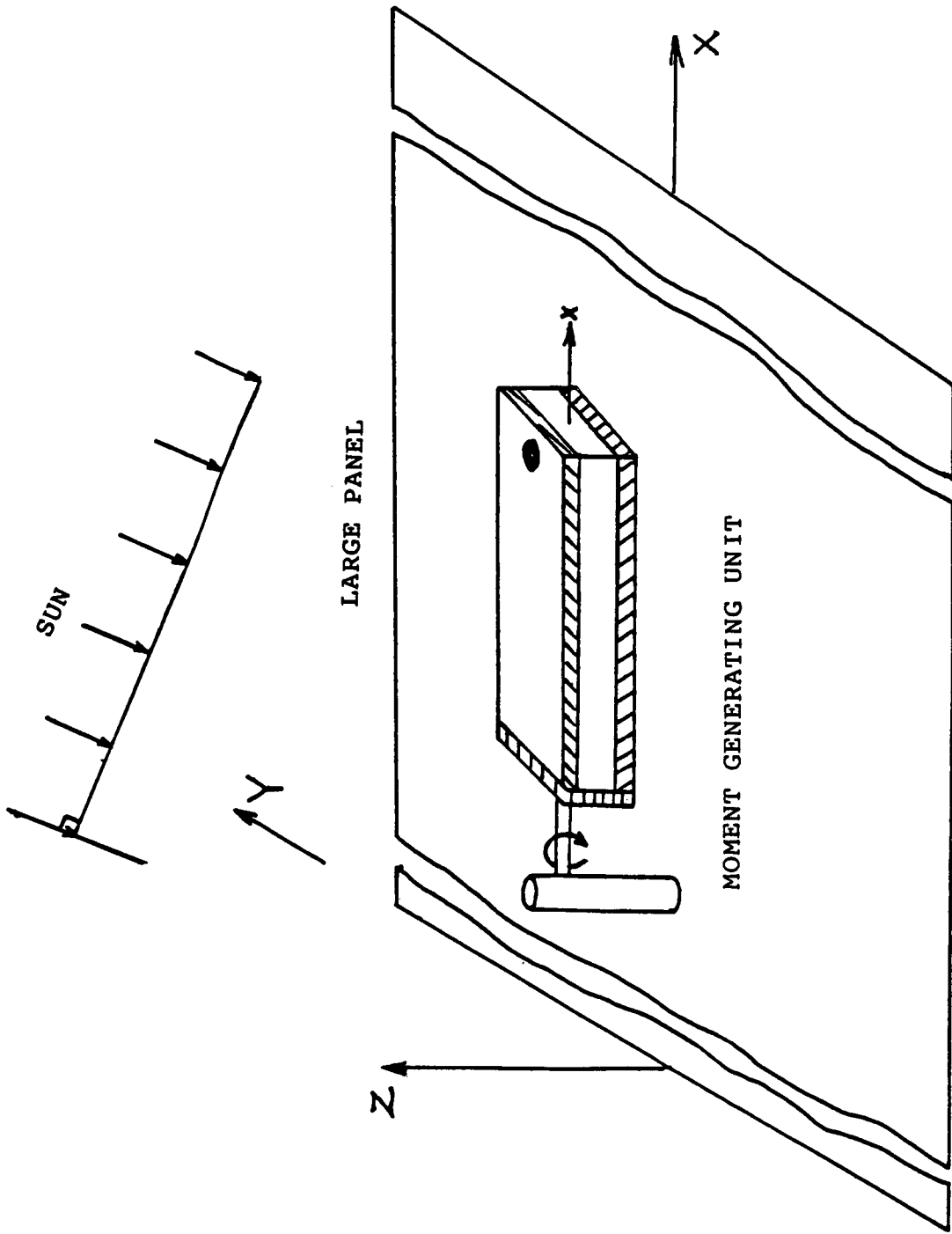
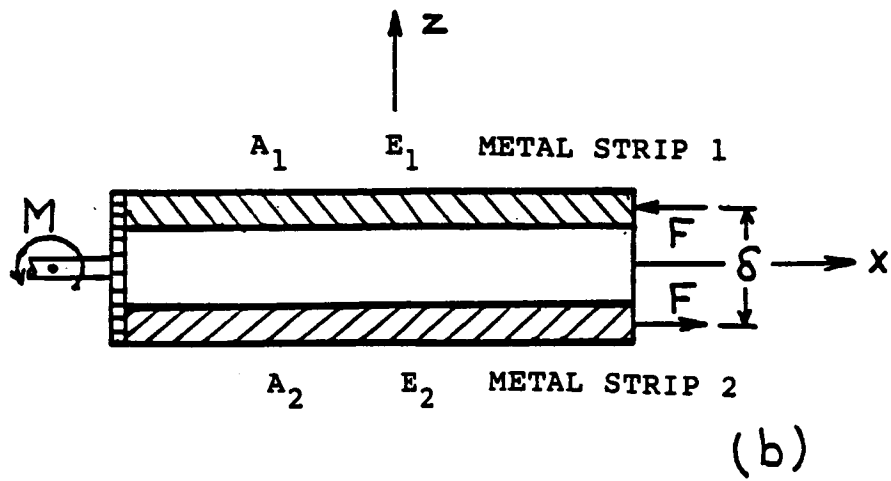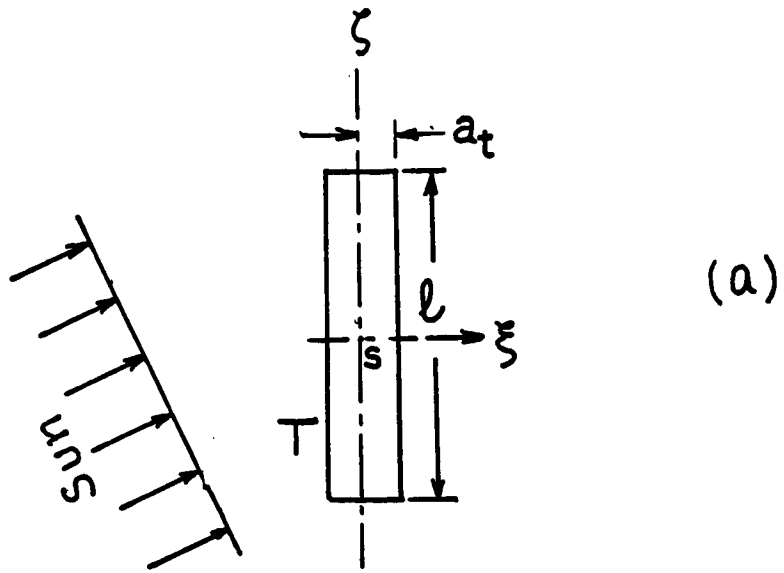Fig 1.   A moment generating unit mounted on the large panel.

Fig. 2.   (a)   Coordinate System for Thermal Analysis

         (b)   Geometry of forces and moments in the moment-packet