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MICROMECHANICAL DYNAMIC ANALYSIS OF AN ADAPTIVE BEAM WITH EMBEDDED DISTRIBUTIONS OF PIEZOELECTRIC ACTUATOR/SENSOR DEVICES

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ABSTRACT

Adaptive materials/structures involve embedding or surface-mounting of actuator/sensor devices and associated control electronics in host structures. The purpose of this paper is to summarize previous work and to provide improved predictions for modeling the interactions between embedded microdevices and the host in adaptive structures using Eshelby's classical equivalent inclusion methods. Eshelby's technique offers a unified method to address both the sensors and the actuators by treating sensors as devices without real induced actuation strains. The application of this method is shown through an example of a "smart" beam with embedded arrays of piezoelectric microdevices. Preliminary analytical results provide important clues regarding the change in the vibrational characteristics of the structure.

1. INTRODUCTION

Piezoelectric materials have attracted significant attention for their potential application as sensors and actuators for controlling the response of structures. Distributions of sensors, actuators, and data processing capability are used to modify, tune, and control the response of adaptive structures to sensed stimuli. Actuators or sensors are either embedded within the structure or bonded to the surface of the structure.

Numerous studies have modeled the interactions between devices and hosts in smart structures. For example, fiber optic sensors, as embedded cylindrical devices, have been analyzed using displacement function methods [Dasgupta and Sirkis, 1992; Pak, 1992; Carman and Reifsnider, 1992; Sirkis, 1993]. The response of adaptive structures with laminated assemblies of piezoelectric wafers/films in composite beams have been analyzed by laminate analysis [Crawley and Lazarus, 1991], simple beam models [Bailey and Hubbard, 1987; Crawley and Lazarus, 1991], pin force models [Crawley and de Luis, 1987; Lin and Rogers, 1992], large deformation beam theory [Im and Atluri, 1989], and one-dimensional eigen-function approximations [Crawley and de Luis, 1987; Lin and Rogers, 1992]. Variational methods have also been developed for solving the coupled boundary value problems in adaptive structures. These include a strain energy method [Wang and Rogers, 1991], finite element methods [Allik and Hughes, 1979; Tzou and Tseng, 1990; Robbins and Reddy, 1991; Ha et al, 1991], and Rayleigh Ritz methods [Hagood et al, 1990]. These models address situations where the size of the device is of the same length scale as the surrounding host structure. The focus of the present paper is on the behavior of a beam with embedded microdevices. The devices are small compared to the characteristic dimensions of the host and the maximum volume fraction of the devices is below 5 %. This corresponds to a spacing greater than 3.5 times the actuator width.

In this paper we present new improvements of recent results from previous work [Alghamdi and Dasgupta, 1993] on modeling the interaction mechanics between embedded microdevices and the host structure, using Eshelby's "equivalent inclusion method" for modeling the perturbation of a uniform applied far field strain, by an ellipsoidal inhomogeneity [Eshelby, 1957]. This method is illustrated here for the dynamic behavior of a simply supported beam made of ALPLEX plastic containing two rows of devices. The system equation of motion is solved and the Rayleigh quotient is used to study the change in the natural frequency of the structure due to harmonic excitation of the actuators.

2. PROBLEM STATEMENT

The equivalent inclusion method is illustrated here through an example of an adaptive beam. As shown schematically in Figure 1, two rows of uniformly spaced microdevices are assumed to be embedded in the beam at a distance d/2 symmetrically about the neutral plane of the beam. The change in natural frequency of the beam is studied in order to investigate the influence of the devices on the vibrational characteristics of the beam. As the beam flexes, every alternate device in each row acts as a sensor and the outputs are used in a closed-loop feedback circuit to actuate the active half (actuators) of the opposite row of devices. This actuation strain is assumed to be opposed in sign to the bending strain seen by the sensors. The result is an apparent stiffening of the structure and an accompanying increase in the natural frequency ω, if all losses in the system are ignored. The aim of this paper is to review and update the electro-mechanical interaction information, necessary for combining the device response with that of the host beam, in an integrated dynamical simulation of the adaptive structure.

The analytical study is simplified significantly by several assumptions. I)- Euler-Bernoulli beam theory is assumed to apply. II)- Each embedded microdevice is assumed to be a piezoelectric micro-cylinder of elliptical cross-section, whose polarization axis is oriented along the length of the beam. III)- The length scale of each device is limited to at least an order of magnitude less than the beam. Hence, IV)- The bending strain is assumed to be approximately uniform over the length scale of the device. This approximation greatly simplifies the algebra of the eigenstrain solution. Further, V)- Each device is assumed to be embedded far enough below the free surface of the beam such that Eshelby's eigenstrain solution for infinite domains is applicable. VI)- The distance between neighboring devices is assumed to be large enough to prevent mutual interactions. Thus, this solution is only valid for beams with low volume fraction of devices. Finally, VII)- The electrical field in each device is assumed to be uniform, as a first order approximation.

As a result of the assumptions presented above, each microdevice is approximated to act like elastic heterogeneities embedded in a large host structure. Perfect bonding is assumed at the interfaces between the devices and the host. The actuator/sensor material is assumed to be PZT-5H. Host and devices materials are approximated to be linear and mechanically isotropic. All material properties are listed in Table 1. The linearizing assumption limits the validity of this approximate analysis to small far field strains and small electrical potentials.

3. ANALYSIS

Eshelby's classical equivalent-inclusion technique is applied to obtain the elastic interaction fields, both in the device and in the host, under external applied loads and under internal actuation loads [Eshelby, 1957]. External loads are handled through equivalent inclusion techniques as a fictitious eigenstrain. The internal actuation loads are treated as real eigenstrains and are obtained from the linearized, isothermal, coupled electro-mechanical constitutive models given below. In the present analytical context, sensors have only a fictitious eigenstrain due to external loads, while actuators have both fictitious and real eigenstrains.

The linearized, isothermal, coupled electro-mechanical constitutive model is [Ikeda, 1990]:

$$\vec{\sigma} = \underline{C}\,\vec{\epsilon} - \underline{h}^T \vec{E} \qquad (1)$$
$$\vec{D} = \underline{h}\,\vec{\epsilon} + \underline{e}\,\vec{E}\ ;$$

where $\vec{\sigma}$, the mechanical stress and $\vec{\epsilon}$, the total strain, are expressed in condensed notation as vectors instead of tensors. As an example, the stress vector $\vec{\sigma}$ is $\lfloor \sigma_{xx}\ \sigma_{yy}\ \sigma_{zz}\ \tau_{yz}\ \tau_{xz}\ \tau_{xy} \rfloor$. The total strain $\vec{\epsilon}$ includes mechanical as well as electro-mechanical contributions. $\vec{E}$ is the electrical field vector, and $\vec{D}$ is the electrical displacement vector. $\underline{C}$ is the mechanical stiffness matrix, $\underline{h}$ is the piezoelectric coupling matrix indicating the stress caused by completely constrained excitation of the PZT material under a unit applied electrical field, and $\underline{e}$ is the fully constrained dielectric matrix for the PZT material. Arrows over a quantity are used to denote vector quantities, while an underscore is used to denote matrix quantities. Clearly, only the mechanical portion of this constitutive model applies to the ALPLEX host material.

Eshelby's method is based upon postulating an equivalent inclusion with a fictitious eigenstrain which has the same stress field as the real heterogeneity, under both external loads and internal actuation strains. Thus, in the heterogeneity:

$$\begin{aligned}\vec{\sigma}^0 + \vec{\sigma}^I &= \underline{C}^D\,(\vec{\epsilon}^0 + \vec{\epsilon}^I - \vec{\epsilon}^r) \\ &= \underline{C}^H\,(\vec{\epsilon}^0 + \vec{\epsilon}^I - \vec{\epsilon}^*)\end{aligned} \qquad (2)$$

where $\vec{\epsilon}^* = \vec{\epsilon}^r + \vec{\epsilon}^f$; superscripts D and H on the stiffness indicate the PZT device and the ALPLEX host, respectively; superscripts 0, I, r, f and * on the stress and strain terms indicate applied far-field value, disturbance due to the presence of the heterogeneity, real actuation eigenstrains, fictitious eigenstrains due to external loading, and total eigenstrains, respectively. The real actuation eigenstrain is obtained from Equation (1) as:

$$\vec{\epsilon}^r = \underline{d}^T \vec{E} \qquad (3)$$
$$\text{where} \quad \underline{d} = \underline{h}\,\underline{S}^D$$

$\underline{d}$ represents the free-expansion of the piezoelectric actuator for a unit applied electric field and $\underline{S}^D$ is the compliance tensor of the device material.

The total eigenstrain is now related to the disturbance strain by Eshelby's strain concentration tensor $\underline{S}^E$:

$$\vec{\epsilon}^I = \underline{S}^E\,\vec{\epsilon}^* = \underline{S}^E\,(\vec{\epsilon}^r + \vec{\epsilon}^f) \qquad (4)$$

Explicit forms for Eshelby's tensor are readily available in the literature for embedded isotropic heterogeneities of ellipsoidal geometries [Eshelby, 1957].

Substituting Equation (4) in Equation (2), we obtain:

$$\underline{C}^D\ [\vec{\epsilon}^0 + \underline{S}^E\ (\vec{\epsilon}^r + \vec{\epsilon}^t) - \vec{\epsilon}^r] = \underline{C}^H\ [\vec{\epsilon}^0 + \underline{S}^E\ (\vec{\epsilon}^r + \vec{\epsilon}^f) - \vec{\epsilon}^f - \vec{\epsilon}^r] \quad (5)$$

Equation (5) can now be solved for the unknown fictitious eigenstrain $\vec{\epsilon}^f$ in terms of the applied external strain $\vec{\epsilon}^o$ and the real actuation eigenstrain $\vec{\epsilon}^r$. If the applied strain is uniform, so is the fictitious eigenstrain.

Now, the mechanical, electro-mechanical, and electrical energy terms are computed to obtain the stiffening of the structure under harmonic excitation, through a suitable variational scheme. The variational principle is a generalized form of Hamilton's principle, and may be written as [Tiersten, 1967];

$$\delta \left[\int_{t_0}^{t_1} (L + W)\, dt \right] = 0 \quad (6)$$

where the Lagrangian functional L is the difference between the kinetic energy T and the electric enthalpy H. The work term, W, includes the potential of all applied mechanical loads and the electrical charges. Thus, using the definition of electric enthalpy [Tiersten, 1967]

$$L+W = \int_v \left[-\left(\frac{1}{2}\right) \vec{\epsilon}^T \underline{C}\, \vec{\epsilon} + \left(\frac{1}{2}\right) \vec{E}^T \underline{e}\, \vec{E} + \vec{\epsilon}^T \underline{h}^T \vec{E} \right.$$
$$\left. \left(\frac{1}{2}\right) \rho \omega^2 \vec{u}^T \vec{u} \right] dV - \int_A \phi\, \vec{n}^T (\underline{h}\, \vec{\epsilon} + \underline{e}\, \vec{E}\,)\, dA \quad (7)$$

where ω is the natural frequency, ρ is the density of host or device, $\vec{u}$ is the displacement field, ϕ is the electric potential specified over the surface area A of the devices, and $\vec{n}$ is the unit outward normal vector on the surface of the devices.

The variation of (L+W) yields stationary solutions for which, the Rayleigh quotient can be presented as [EerNisse, 1967]:

$$\omega^2 = \frac{\int_v (\vec{\epsilon}^T \underline{C}\, \vec{\epsilon})\, dV + \int_\Omega (\vec{E}^T\, \underline{e}\, \vec{E}\,)\, dV}{\int_v \rho\, \vec{u}^T\, \vec{u}\, dV} \quad (8)$$

This equation has been used to predict the natural frequencies of smart structures with embedded actuators [Alghamdi and Dasgupta, 1993]. New updated results are summarized below.

The first integral in the numerator is the sum of electro-mechanical and mechanical energy terms and is denoted as U_{mech}, for convenience. Using Eshelby's analysis, U_{mech} is obtained as

$$2\, U_{mech} = \int_v \vec{\epsilon}^{oT} \underline{C}^H\, \vec{\epsilon}^o\, dV + \int_\Omega \vec{\epsilon}^{*T}\, \underline{C}^H\, \underline{S}^E\, \vec{\epsilon}^*\, dV$$
$$+ \int_\Omega \vec{\epsilon}^{oT}\, \underline{\Delta C}\, \vec{\epsilon}^o\, dV + 2 \int_\Omega \vec{\epsilon}^{oT}\, \underline{\Delta C}\, \underline{S}^E\, \vec{\epsilon}^*\, dV$$
$$+ \int_\Omega \vec{\epsilon}^{*T}\, \underline{S}^{ET}\, \underline{\Delta C}\, \underline{S}^E\, \vec{\epsilon}^*\, dV \quad (9)$$

where Ω is the volume of the devices, and $\vec{\epsilon}^*$ is obtained from Equation (5). $\underline{\Delta C}$ represents the difference in the stiffness of device $\underline{C}^D$ and host $\underline{C}^H$. Note that Equation (9) represents an improvement over previous results published by the authors [Alghamdi and Dasgupta, 1993; Dasgupta and Alghamdi, 1993], since the last three terms were missing in the previous work. After some manipulations, we obtain a more simplified equation for U_{mech} as

$$2\, U_{mech} = \int_v \vec{\epsilon}^{oT}\, \underline{C}^H\, \vec{\epsilon}^o\, dV - \int_\Omega \vec{\epsilon}^{*T}\, \underline{C}^H\, \vec{\epsilon}^o\, dV$$
$$+ \int_\Omega \vec{E}^T\, \underline{h}\, \vec{\epsilon}\, dV \quad (9a)$$

In order to perform the integrations in Equations (8) and (9), all that remains now is to assume explicit representations for the applied flexural strain field, and the actuation eigenstrain. For example, in the Rayleigh scheme for estimating the natural frequency of conservative systems, an approximate displacement field can be assumed. In this example, the approximate bending field is assumed to be harmonic in time and sinusoidal in space. Thus, the transverse deformation of the beam neutral axis is given as;

$$w = \sum_1^n a_n\, Sin(\omega_n t)\ Sin\left(\frac{n\pi y}{L}\right) \quad (10)$$

where the y axis is oriented along the length of the beam, w is the transverse displacement in the z direction, ω_n and a_n are the natural frequency and amplitude, respectively, of the n^{th} mode. L is the length of the beam, and t is time. Only the fundamental mode (n=1) is of interest in this study. In view of the Euler-Bernoulli assumptions, the only non-zero term in the bending strain field $\vec{\epsilon}^o$ is ϵ_2^o, and is given as:

$$\epsilon_2^0 = z\, \frac{\pi^2}{L^2} a_1\, Sin(\omega_1 t)\ \ Sin\left(\frac{\pi y}{L}\right) \quad (11)$$

where, z is the distance of the microdevice from the neutral axis of the beam. The only non-zero component of the actuation voltage vector is now E_2 and is assumed to be proportional to the output of the sensory devices, and hence, to the bending strains. Thus E_2 is written as:

$$E_2 = E_2^* \, Sin(\omega_1 t) \, Sin(\frac{\pi y}{L}) \qquad (12)$$

where the amplitude E_2^* is assumed to be proportional to the amplitude of the bending strain due to the fundamental vibrational mode of the beam, and the non-zero terms of the actuation strain vector are now written as:

$$\epsilon_i^r = d_{2i} \, E_2 \quad , \quad i = 1\text{-}6 \qquad (13)$$

Equations (10-13) are used in Equation (8) to compute the natural frequency of the system.

4. RESULTS AND DISCUSSIONS

Figure (2) shows the increase in the natural frequency (ω) as a function of the actuation strain (ϵ^r) for different numbers of actuators (n). Increasing n increases the volume fraction (V_f) of devices and decreases the spacing between devices. The minimum spacing (c, in Figure 1) is maintained above 3.6 times the actuator width (a, in Figure 1) in order to minimize the interactions between them. For convenience, the natural frequency (ω) is normalized with respect to that for zero electrical excitation (ω_0). The actuation strain amplitude (ϵ^r) is normalized with respect to the far field strain (ϵ^0). The maximum field strength shown here corresponding to $\epsilon^r = 4\,\epsilon^0$, is approximately 54 KV/m (less than 6% of typical fields achievable in PZT-5H). The corresponding voltage is 2800 V for the nominal device size (a = 0.0052 m) assumed in this study. The increase in the natural frequency is a measure of the stiffening of the beam due to the actuation loads. In Figure (3), the volume fraction (V_f) of devices has been increased, not by increasing the number of devices as in Figure (2), but by increasing the size of each device (a and b), relative to the host beam dimensions. The minimum spacing between devices is again maintained above 3.6a. Figure (4) illustrates the dependence of the stiffening effect (natural frequency) on the location of the actuators, relative to the neutral axis. As expected, moving the actuators away from the neutral axis increases the natural frequency, for the same excitation. Figure (5) illustrates the dependence of this stiffening effect (natural frequency) on the Young's modulus of the host material. As expected the stiffening effect decreases with increasing host stiffness. Figure (6) illustrates the relative contributions of the mechanical interaction energy (U_{mech}), and the electrical energy (U_{elect}), towards stiffening of the structure. The mechanical term is larger than the electrical term and its relative contribution increases as the actuation is increased and decreases as the host stiffness is increased. The relatively large contribution from U_{mech} illustrates the importance of accurate modeling of the electromechanical interactions in the adaptive beam.

5. CONCLUSIONS

This paper has summarized recent research improvements on the use of Eshelby's techniques for modeling the interaction between microdevices and the surrounding host. This method presents a unified approach for addressing the interaction mechanics of microdevices embedded in an adaptive structure. Preliminary results suggest that accurate models for electromechanical interaction are extremely important for predicting the behavior of adaptive structures.

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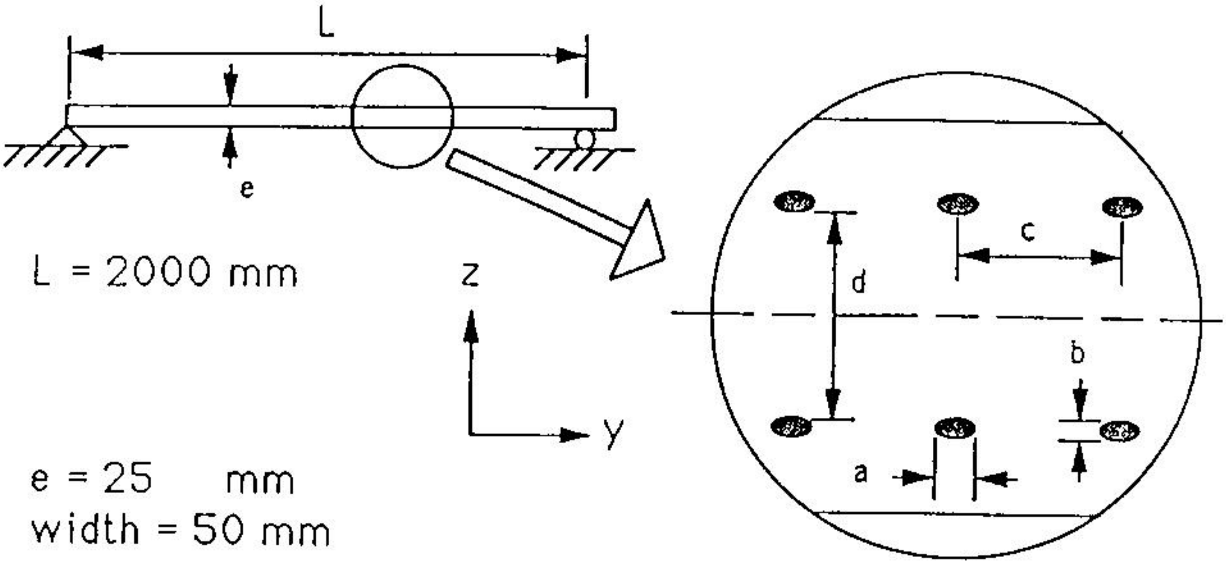

FIGURE 1. ADAPTIVE BEAM WITH EMBEDDED ROWS OF MICRODEVICS.

TABLE 1. ELECTROMECHANICAL PROPERTIES OF DEVICE AND HOST MATERIALS.

	E (GPa)	υ	d_{21} (C/N) * 1e-12	d_{22}	d_{16}	ε_{11} (C/mV) * 1e-10	ε_{22}
PZT-5H	64	0.39	-274	593	741	150	66
ALPLEX	3.5	0.35	—	—	—	—	—

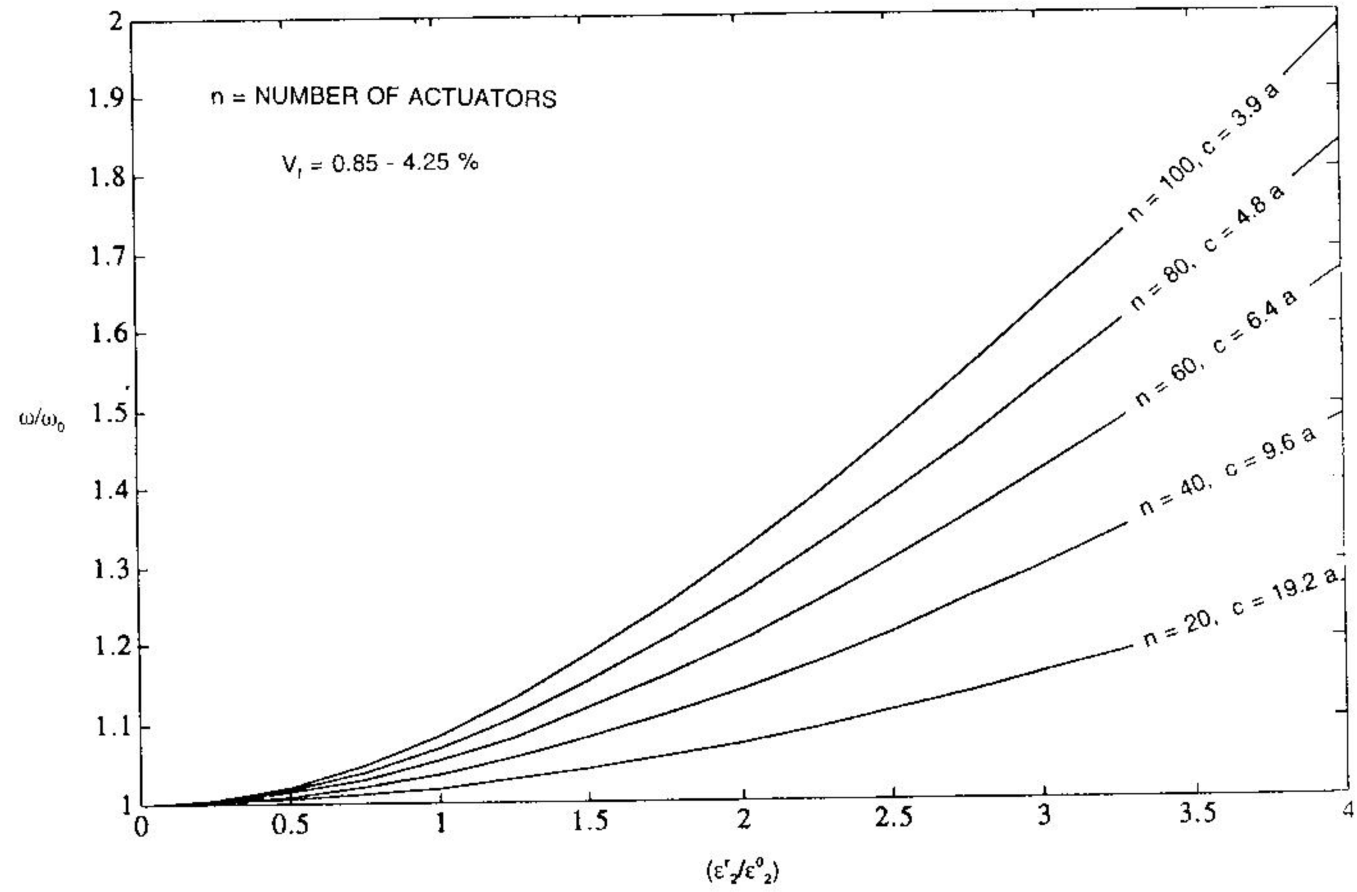

FIGURE 2. NORMALIZED NATURAL FREQUENCY AS A FUNCTION OF THE NORMALIZED ACTUATION STRAIN $(\varepsilon^r_2/\varepsilon^0_2)$, FOR DIFFERENT DEVICE DENSITIES (DIFFERENT NUMBERS OF ACTUATORS, n).

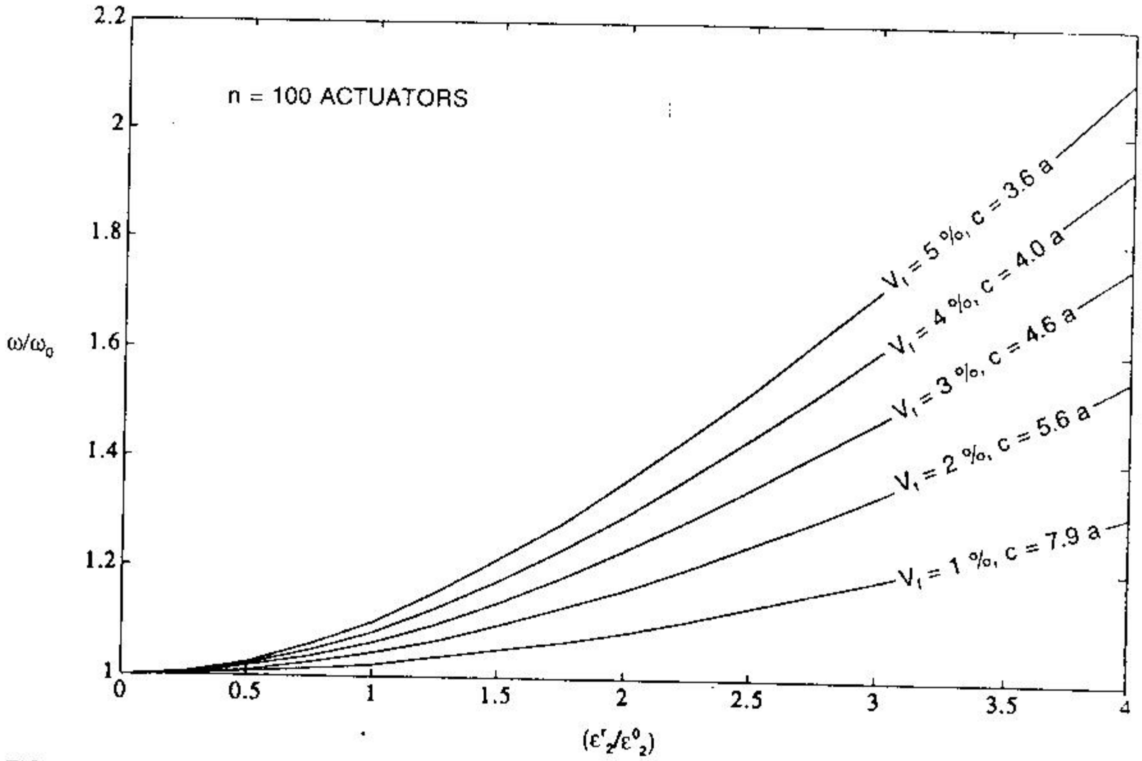

FIGURE 3. NORMALIZED NATURAL FREQUENCY AS A FUNCTION OF NORMALIZED ACTUATION STRAIN $(\varepsilon^r_2/\varepsilon^0_2)$ FOR DIFFERENT DEVICE DENSITIES (DIFFERENT DEVICE SIZES a AND b).

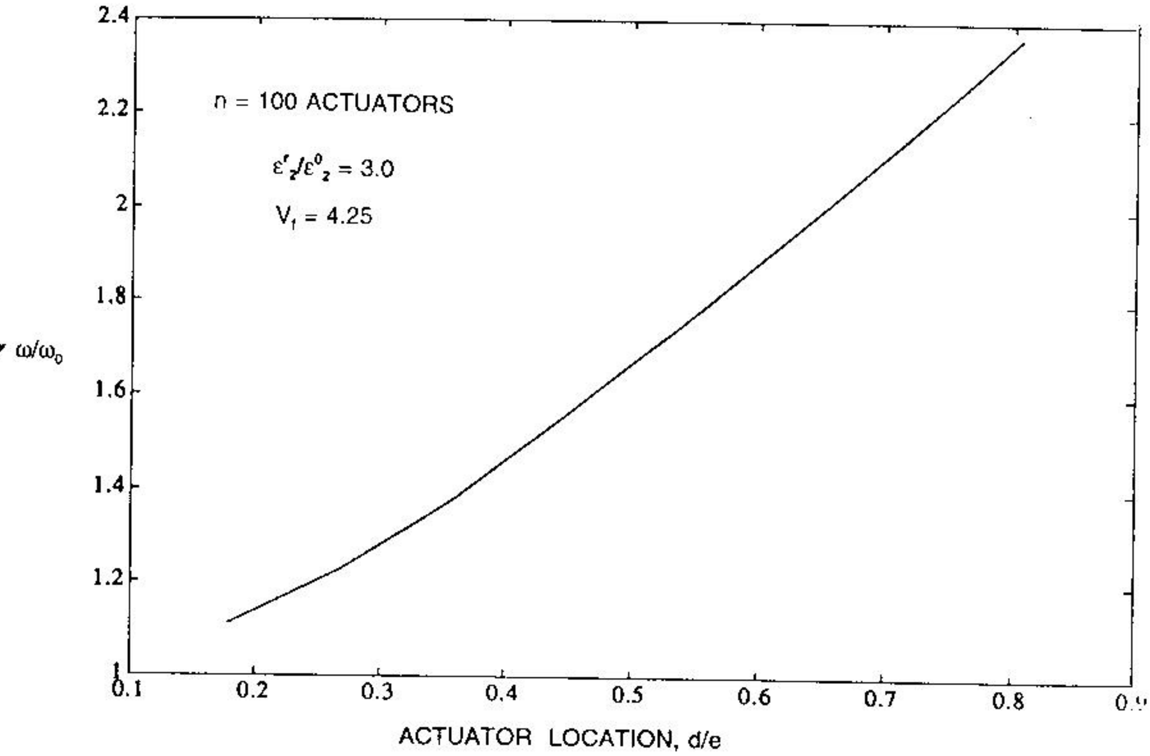

FIGURE 4. INFLUENCE OF ACTUATOR POSITION ON STIFFENING OF THE HOST BEAM.

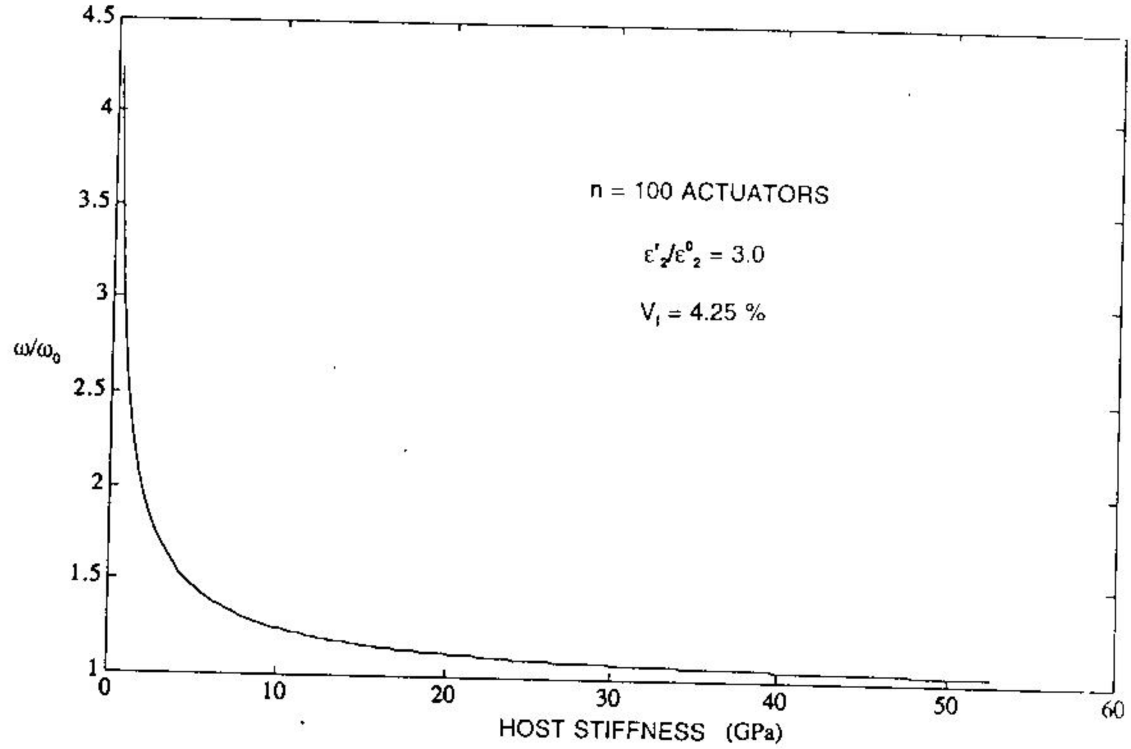

FIGURE 5. PLOT OF NORMALIZED NATURAL FREQUENCY VS. HOST STIFFNESS.

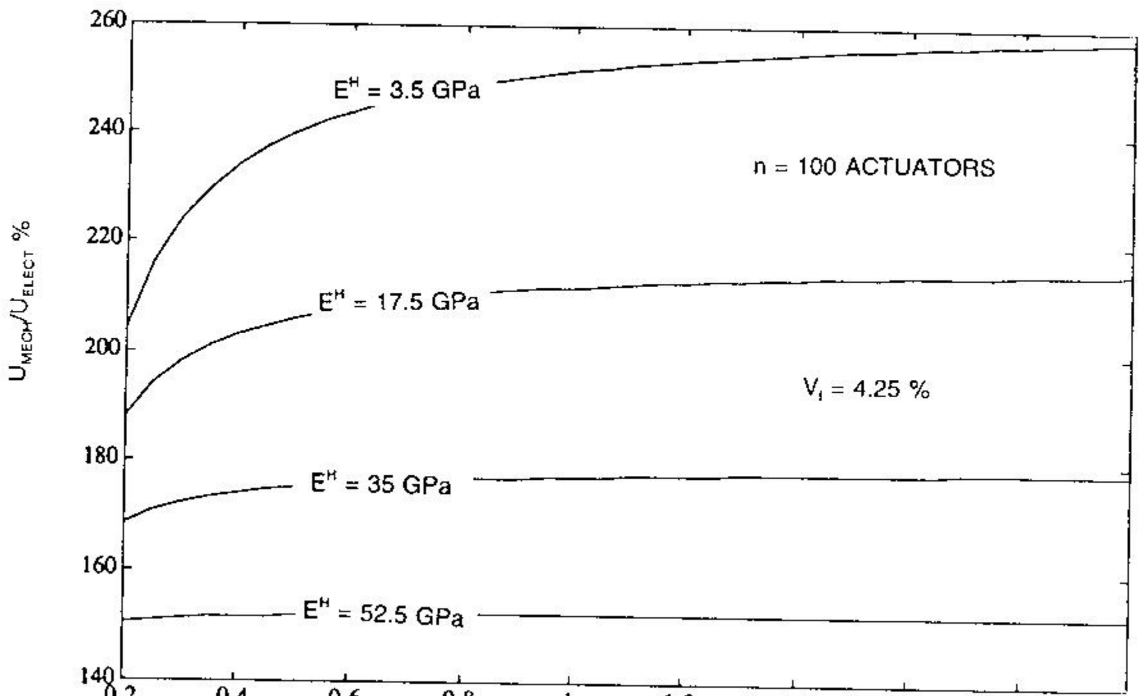

FIGURE 6. THE PERCENTAGE CHANGE IN THE MECHANICAL AND ELECTROMECHANICAL ENERGY TERMS, RELATIVE TO ELECTRICAL ENERGY TERM AS A FUNCTION OF NORMALIZED ACTUATION STRAIN ($\varepsilon'_2/\varepsilon^0_2$) FOR DIFFERENT HOST STIFFNESS.

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