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STATIC DEFLECTION CONTROL OF FLEXIBLE BEAMS
BY PIEZO-ELECTRIC ACTUATORS

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NOMENCLATURE

b width of beam and piezo-actuator, m

d electric charge constant of piezo-actuator, m/v

D distance to neutral axis of composite beam measured from its lower edge, m

$E_{1,2}$ Young's modulus of elasticity of piezo-actuator and beam respectively, N/m^2

E_i Young's modulus of elsticity of element i of the beam, N/m^2

F vector of external forces and moments

$I_{1,2}$ area moment of inertia of actuator and beam about the neutral axis of the composite beam respectively, m^4

I_i area moment of inertia of the element i, m^4

K_i stiffness matrix of the element i

K overall stiffness matrix of beam-actuator system

l_i length of element i

M_{ei} external moment acting on i^{th} node of beam, Nm

M_f piezo-electric moment generated by piezo-film, Nm

N number of elements of the beam

$t_{1,2}$ thickness of piezo-actuator and beam respectively, m

U deflection criterion

v voltage applied across the piezo-electric film, volts

V_i external force acting on the ith element of the beam, N

y_i the linear translation of node i, m

Greek Letters

δ_i deflection of node i

δ deflection vector of all nodes of the beam, m or rad.

ϵ_f piezo-electric strain in piezo-actuator, m/m

$\sigma_{1,2}$ bending stresses in piezo-actuator and beam respectively, N/m^2

σ_f piezo-electric stress in actuator, N/m^2

ABSTRACT

This study deals with the utilization of piezo-electric actuators in controlling the static deformation of flexible beams.

An optimum design procedure is presented to enable the selection of the optimal location, thickness and excitation voltage of the piezo-electric actuators in a way that would minimize the deflection of the beam to which these actuators are bonded.

Numerical examples are presented to illustrate the application of the developed optimization procedure in minimizing the structural deformation of beams of different materials when subjected to different loading and end conditions using ceramic or polymeric piezo-electric actuators.

The obtained results emphasize the importance of the devised rational procedure in designing beam-actuator systems with minimal elastic distortions.

INTRODUCTION

The construction and operation of large structures which are extremely flexible have posed new and challenging problems particularly because these structures are intended to provide stable bases for observations and communications such as in space structures. With strict constraints imposed on the structural deformations, it became essential to suppress such deformations to a minimum through the use of active control systems of one form or another.

Distinct among the presently available active control systems are those that rely in their operation on piezo-electric actuators. Such systems have proven to be experimentally effective in controlling the vibrations of simple structural elements such as rectangular beams [1-2]* and hollow cylindrical masts [3]. The effectiveness of these systems is coupled also with the light weight, high force and low power consumption capabilities of the piezo-electric actuators [4-8]. These features rendered this class of actuators to be an attractive candidate for controlling structural deformation.

The present state-of-the-art of this type of actuators has been limited to the analysis of their characteristics [9-11] as influenced by their geometrical or operational conditions. But, no attempt has been made towards selecting their optimal geometrical parameters or location which are best suited for a

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* Numbers between brackets designate references

particular structure subjected to known loading conditions. Such a synthesis procedure is essential to the successful integration of the actuators into the structure in order to minimize the structural deformation.

It is therefore, the purpose of this study to develop an optimum design procedure that would enable the selection, on rational basis, of the optimum design parameters and location of piezo-electric actuators to satisfy certain structural deformation requirements. The procedure will take into account the effect that the actuators have on changing the elastic and inertial properties of the structure to which they are bonded to.

THE PIEZO-ELECTRIC ACTUATOR BEAM SYSTEM

A. General Layout

Figure (1) shows a general layout of a flexible beam (A) whose deflection is to be controlled by a piezo-electric actuator (B). The beam, under consideration, can generally be made of several steps which are not necessarily of the same thickness or the same material. The interfacial nodes between the different steps can be subjected to external forces, moments or both. Further, the degrees of freedom of any node can be limited to linear translation, angular rotations or restrained completely depending on the nature of support at the node under consideration.

In this study, the beam is assumed to have rectangular cross

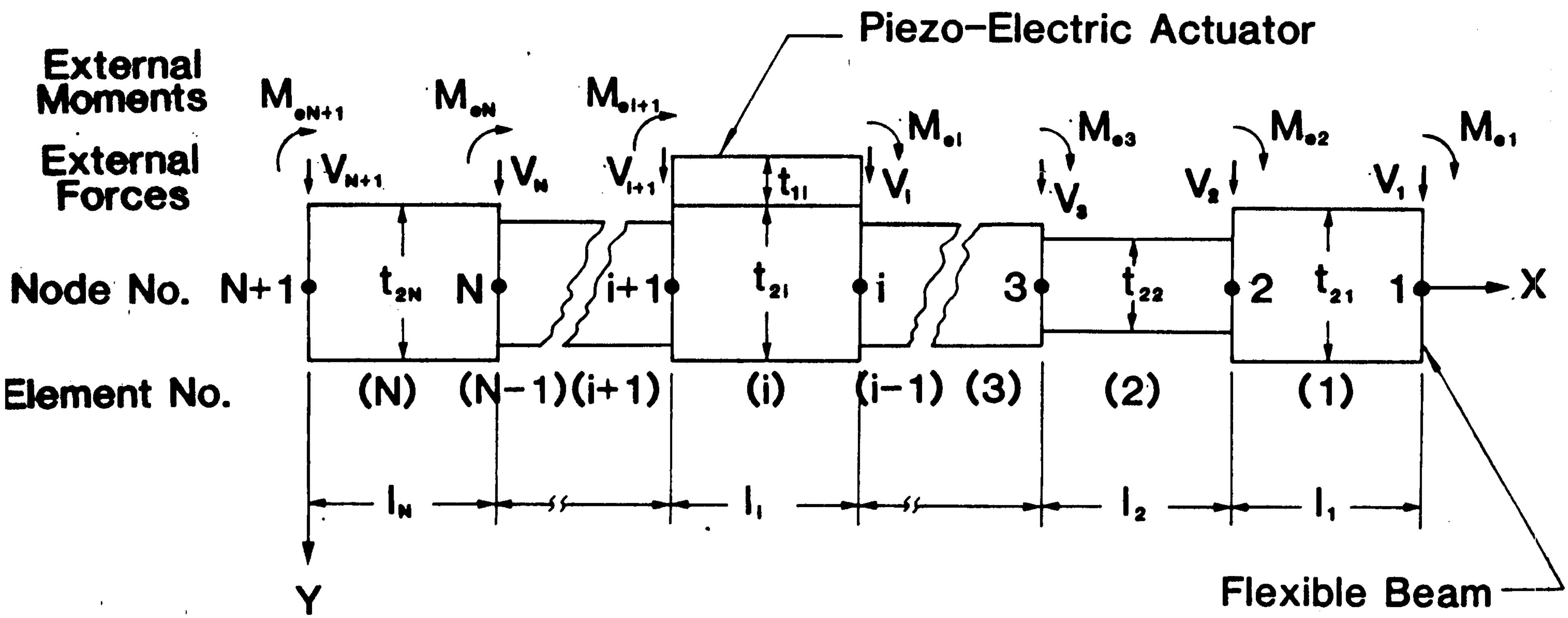

Figure (1) – General layout of the piezo-electric actuator–beam system section of constant width (b). The beam is considered to deflect in the transverse direction due to the flextural action of the external forces and moments.

In Figure (1), the piezo-electric actuator B is shown bonded to the element i of the flexible beam to form a composite beam. When an electric field is applied across the film, then it will expand if the field is, for example, along the polarization axis of the film and will contract if the two were out of phase. The expansion or contraction of the film relative to the beam, by virtue of the piezo-electric effect, creates longitudinal bending stresses in the composite beam which tend to bend the beam in a manner very similar to a bimetallic thermostat.

With proper selection, placement and control of the actuator, it would be possible to generate enough piezo-electric bending stresses to counter balance the effect of the external forces and moments acting on the beam in a way that minimizes its structural deformation.

B. Model Of An Actuator-Beam Element

Figure (2) shows a schematic drawing of a piezo-film A bonded to an element B of the flexible beam.

If a voltage v is applied across the film, a piezo-electric strain ϵ_f is introduced in the film and can be computed from :

$$\epsilon_f = (d/t_1)*v \tag{1}$$

where d is the electric charge constant of the film, m/v

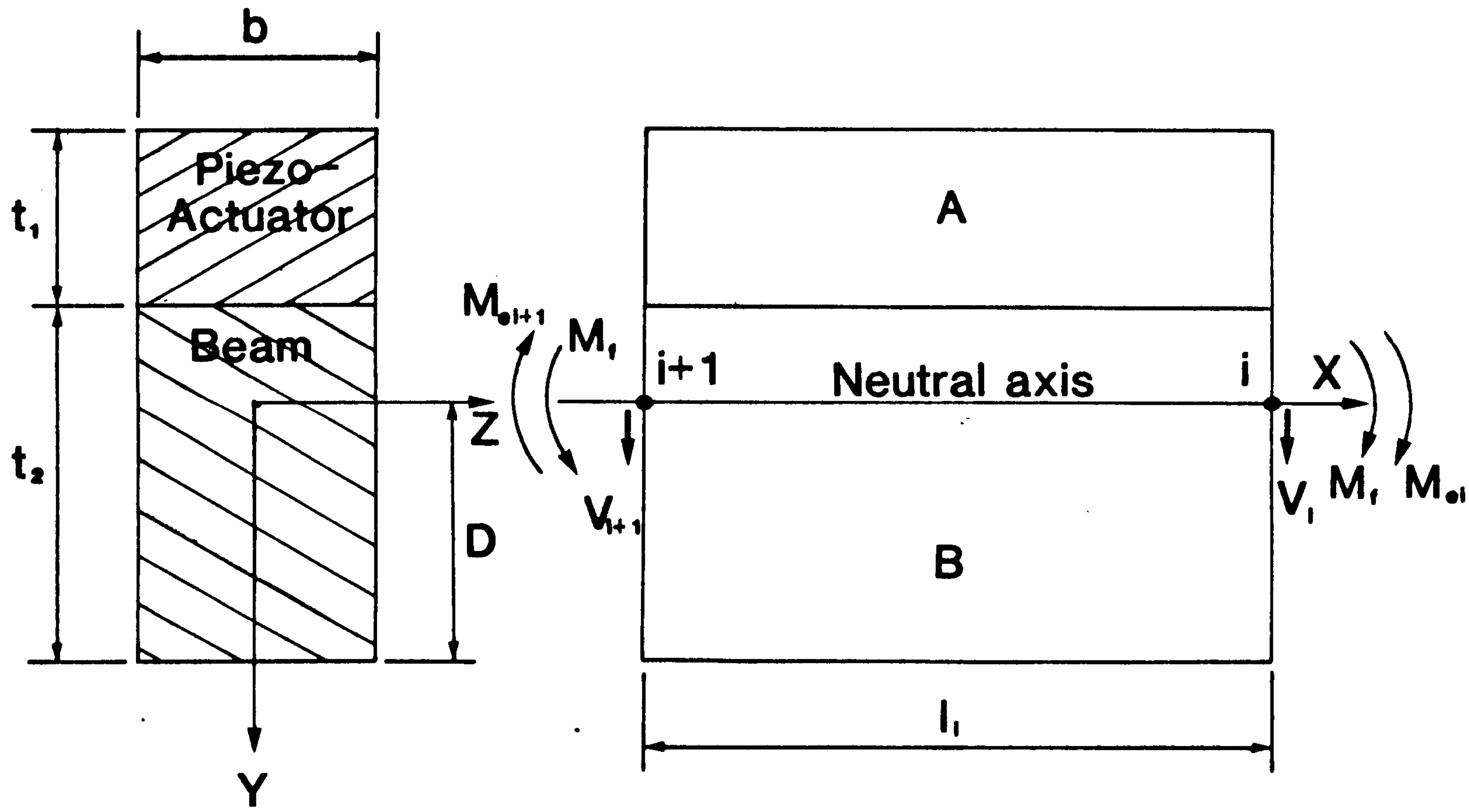

Figure (2) – Schematic drawing of an actuator bonded to a flexible beam element

t_1 is the thickness of the piezo-actuator, m

This strain results in a longitudinal stress σ_f given by :

$$\sigma_f = E_1(d/t_1) \tag{2}$$

where E_1 is the Young's modulus of elasticity of the film, N/m^2

This, in turn generates a bending moment M_f, around the neutral axis of the composite beam, given by :

$$M_f = \int_{-(t_2-D)}^{-(t_1+t_2-D)} \sigma_f(bY)dy \tag{3}$$

where t_2 is the thickness of the beam, m

b is the width of the beam and the piezo-film, m

In equation (3), D is the distance of the neutral axis from the lower edge of the beam which can be determined by considering the force balance in the longitudinal direction X of the beam; or :

$$\int_{film} \sigma_1 dA + \int_{beam} \sigma_2 dA = 0 \tag{4}$$

or

$$E_1 b \int_{-(t_1+t_2-D)}^{-(t_2-D)} ydy + E_2 b \int_{-(h_2-D)}^{D} ydy = 0 \tag{5}$$

where E_2 is Young's modulus of elasticity of the beam

Equation (5) yields the following expression for D :

$$D = \frac{E_1 t_1^2 + 2E_1 t_1 t_2 + E_2 t_2^2}{2(E_1 t_1 + E_2 t_2)} \tag{6}$$

Equation (2), (3) and (6) can be combined to determine the bending moment M_f generated by the piezo-film on the composite beam as follows :

$$M_f = db*(t_1+t_2)/2*(E_1E_2t_2)/(E_1t_1+E_2t_2)*v \tag{7}$$

For this composite beam, it can be easily shown [12] that it has a flextural rigidity (E_iI_i) given by :

$$E_iI_i = E_1I_1 + E_2I_2 \tag{8}$$

where I_1 and I_2 are the area moments of inertia of the film and the beam about the neutral axis respectively.

Let us now assume that the composite beam, shown in Figure (2), extends a length l_i between two nodes (i) and (i+1). Further, it is assumed that the external forces V_i and V_{i+1} as well as the external moments M_{ei} and M_{ei+1} are acting on the beam at nodes i and i+1 respectively. Then, the resulting linear and angular deformations of the beam y_i and θ_i as well as y_{i+1} and θ_{i+1} at the nodes i and i+1, respectively, can be related to the loads acting on the element as follows [13] :

$$\begin{bmatrix} V_i \\ M_{ei}+M_f \\ V_{i+1} \\ M_{ei+1}-M_f \end{bmatrix} = \frac{E_iI_i}{l_i^3} \begin{bmatrix} 12 & 6l_i & -12 & 6l_i \\ 6l_i & 4l_i^2 & -6l_i & 2l_i^2 \\ -12 & -6l_i & 12 & -6l_i \\ 6l_i & 2l_i^2 & -6l_i & 4l_i^2 \end{bmatrix} \begin{bmatrix} y_i \\ \theta_i \\ y_{i+1} \\ \theta_{i+1} \end{bmatrix} \tag{9}$$

Equation (8) can be rewritten as :

$$F_i = K_i\delta_i \tag{10}$$

where F_i is the resultant forces and moments vector acting on the beam element i

K_i is the stiffness matrix of the composite beam element i

δ_i is the deflection vector of the nodes bounding the beam element

Equation (9) constitutes the basic finite element model that relates the external loads (V and M_e) and piezo-electric moments (M_f) to the deflections (y and θ) of the element as a function of its elastic and inertial parameters.

The equation can be equally used for any element of the beam whether it has a piezo-film bonded to it or not. In the latter case, M_f is set to zero and flextural rigidity E_iI_i is set equal to that of the flexible beam element under consideration.

C. Model Of The Overall Actuator-Beam System

The force-displacement characteristics of the individual elements of the beam-actuator system, as given for element i by equation (9), are combined to determine the behavior of the overall structure.

The equilibrium conditions of the overall structure will be expressed as :

$$\begin{array}{l}\text{External forces and moments} \\ \text{acting on the nodes} \\ \text{of the overall system}\end{array} = \sum \begin{array}{l}\text{Forces and moments} \\ \text{acting on the elements} \\ \text{at these nodes}\end{array}$$

or

$$F = \sum_{i=1}^{N+1} F_i = \sum_{i=1}^{N+1} K_i \delta_i = K\delta \tag{11}$$

where K is the overall stiffness matrix of the system (2n*2n)

Bathe and Wilson [14], Yang [15] and Fenner [16], for example, show how to generate the overall matrix K from the stiffness matrices K_i of the individual elements.

In equation (11), the external force vector F and the displacement vector δ are given by :

$$F^T = [\ V_1\ M_1\ V_2\ M_2\ \ldots\ \ldots\ V_{N+1}\ M_{N+1}\ ] \tag{12}$$

and

$$\delta^T = [\ Y_1\ \theta_1\ Y_2\ \theta_2\ \ldots\ \ldots\ Y_{N+1}\ \theta_{N+1}\ ] \tag{13}$$

where superscript T denotes the transpose of the vector

D. Solution Algorithm

For a particular flexible beam configuration, external loading conditions and end conditions the effect of applying certain voltage V on a piezo-electric actuator of a specific thickness t_1 placed at a particular location i on modifying the elastic deflection δ of the beam can be determined by solving equation (11).

In equation (11), the overall stiffness matrix K is modified to account for the end conditions and type of restraints imposed on the different nodes. If, for example, a node(j) is simply-supported and is therefore restrained in the Y direction, i.e. y_j=0, then the off diagonal elements of 2*j row of the overall stiffness matrix K are set equal to zero together with the corresponding externally applied force. If the j^{th} node is also restrained in its angular motion then $\theta_j=0$ and the off diagonal elements of the (2*j+1) row of the K matrix are eliminated along with the corresponding externally applied moment.

After such a modification process, the elastic deflection δ of the beam can be computed from :

$$\delta = K^{-1} * F \tag{14}$$

where K^{-1} is the inverse of the overall stiffness matrix

Figure (3) shows a flowchart of the solution algorithm.

OPTIMIZATION OF THE ACTUATOR-BEAM SYSTEM

The presented analysis algorithm of the actuator-beam system is utilized as a basis for the development of the optimization procedure for selecting the thickness t_1, excitation voltage v and location i of the actuator in order to minimize the deflection of the beam.

The optimum design problem is formulated as follows :

$$\left\langle \begin{array}{lll} \text{Find thickness t1, voltage v and location i} & & \\ \text{to Minimize} \quad U = \sum_{i=1}^{N+1} (Y_i^2 + \theta_i^2) & & \\ \text{Such that} \qquad V_i = V^*_i & & i=1,2,\ldots,N+1 \\ \qquad\qquad\qquad M_{ei} = M^*_{ei} & & i=1,2,\ldots,N+1 \\ \qquad\qquad\qquad v/t_1 \leqslant v^* & & \end{array} \right\rangle$$

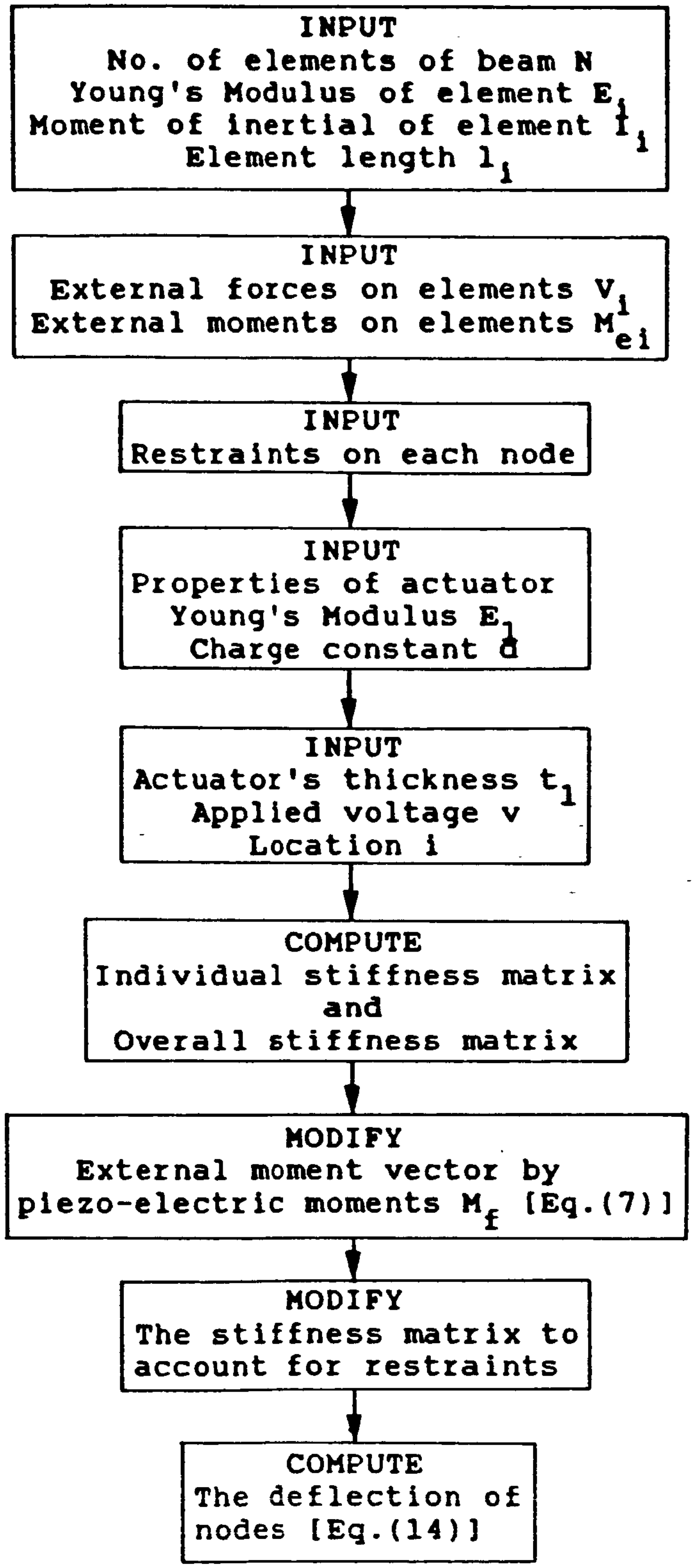

Figure (3) - Flowchart of the analysis algorithm

$$\left\backslash \begin{array}{c} E_1 dv/t_1 \leq \sigma^* \\ t_1 \leq t_1^* \\ v_{min} \leq v \leq v_{max} \end{array} \right/$$

Such a formulation results in a nonlinear optimization problem in 3 design variables t_1, v and i which is subjected to five inequality constraints. The first inequality constraint guards against the application of a high voltage across the actuator that may result in its depolirization. The second inequality constraint limits the stresses in the piezo-film within the acceptable safe limits defined by the allowable stress σ^*.

The fact that the actuator location is integer makes the optimization problem of the Mixed-Integer Programming (MIP) type.

A linearization algorithm is used to reduce the problem to a series of successive Linear Programs [17] that are then optimized using the Branch and Bound method [18-19] to search for the optimal parameters.

Figure (4) shows a block diagram of the optimization algorithm.

NUMERICAL EXAMPLES

The developed optimization procedure is used to optimally select, place and control the action of a single piezo-electric actuator bonded to a three-element straight beam that is 0.0125m wide, 0.0021m thick and 0.15m long. The effect of varying the

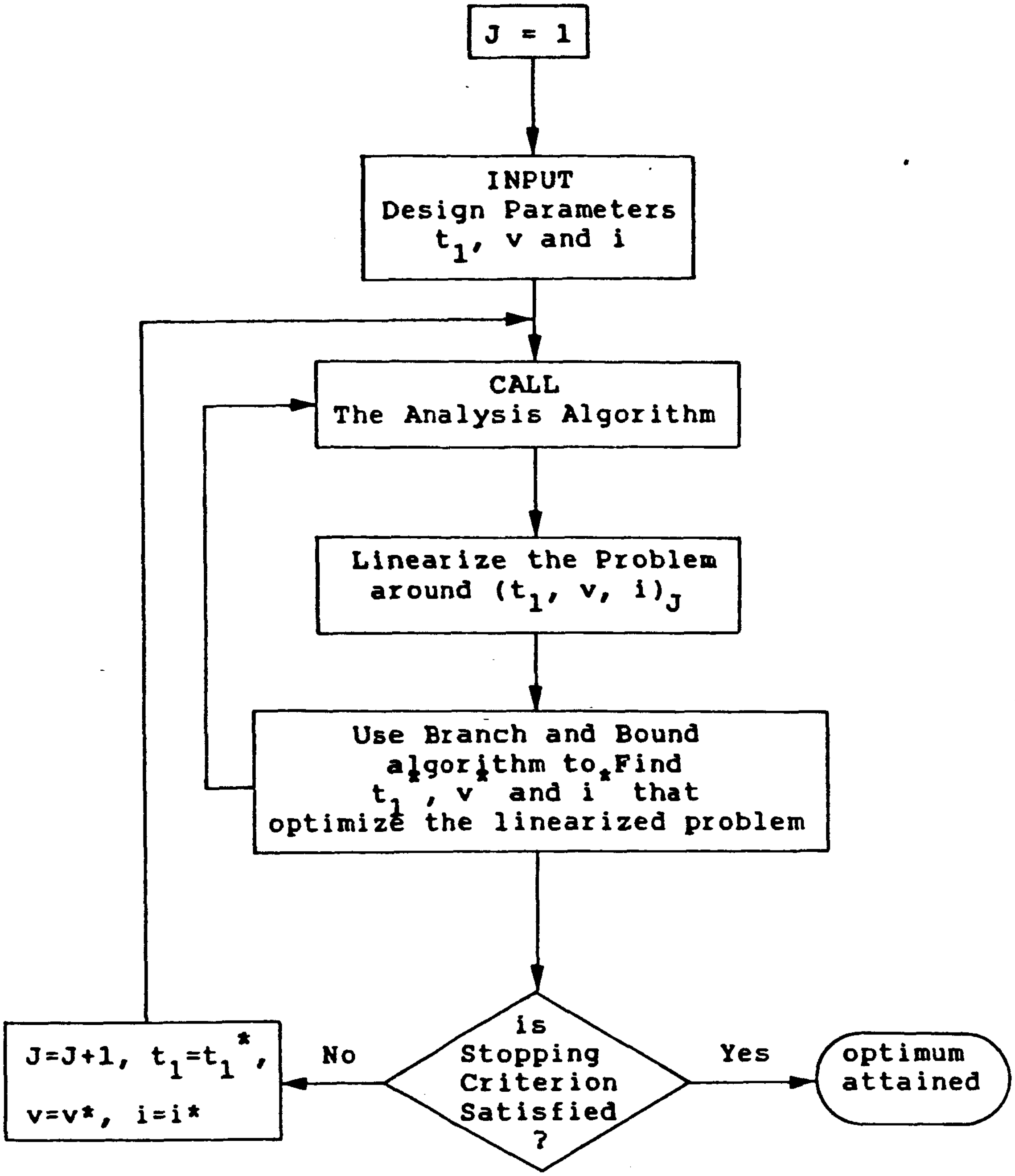

Figure (4) - Flowchart of the optimization algorithm beam loading and end conditions as well as beam and actuator materials on the optimal configuration of the system is considered in great details.

A. Effect Of Loading Conditions

(i) Transverse Load

A steel cantilever beam is considered by fixing node 4 of the three element beam shown in Figure (5). A load of 0.1N is applied to the beam at node 1 perpendicular to its longitudinal axis.

Table (1) - Properties of piezo-electric actuators

Actuator Type	Ceramic(Ref.8 and 10)		Polymeric(Ref.20)
Material	PZT	G1278	KYNAR (PVF2)
charge coefficient d 10^{-12} (m/v)	123	270	23
Young's modulus (G N/m^2)	139	60	2
max. voltage v^* (Mv/m)	1	0.7	30
max. tensile strength σ^* (M N/m^2)	45	47.5	33 - 55
density (kg/m^3)	7500	7400	1780

If one PZT ceramic actuator, whose properties are given in Table (1), is to be used to control the beam deflection, then this actuator can be placed at element 1, 2 or 3. The effect of

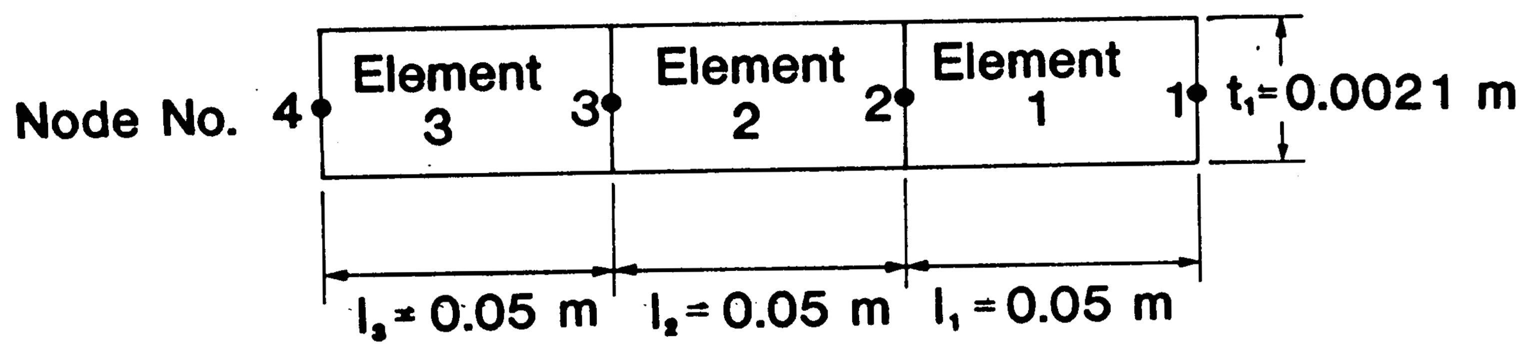

Figure (5) – Layout of a 3-element straight beam these three placement strategies on the sum of the deflection squared criterion of the beam is shown in Figures (6-a), (6-b) and (6-c) respectively. The figures display the iso-contour maps of the deflection criterion drawn in the applied voltage v, actuator thickness t_1 plane with the actuator thickness normalized with respect to the beam thickness, t_2.

Figure (6-a) indicates that if the actuator is placed at element 1, the deflection criterion attains a minimum value of 0.346×10^{-6} when the applied voltage is 120v and the actuator thickness t_1 is 0.00021m or $t_1/t_2=0.1$. This value of the deflection criterion can be reduced further to 0.0995×10^{-6} when an optimal actuator, that has a thickness of 0.00425m ($t_1/t_2=0.4$), is placed at element 2 and the applied voltage is controlled at 120 volts as can be seen from Figure (6-b).

A global minimum of the deflection criterion is obtained when the actuator is placed at element 3. The optimum point is constrained as it lies on the maximum applied voltage/unit thickness constraint. The optimum thickness of the actuator is 0.000105m, or $t_1/t_2=0.05$, and the applied voltage is adjusted at 80 volts as shown in Figure (6-c).

A summary of the optimal parameters of the actuator is given in Table (2) as a function of the actuator location. The table includes also the ratio between the deflection criterion of the optimally controlled beam to that of the uncontrolled beam which is subjected to the same loading and end conditions. This ratio serves as a measure of the effectiveness of the piezo-actuator in

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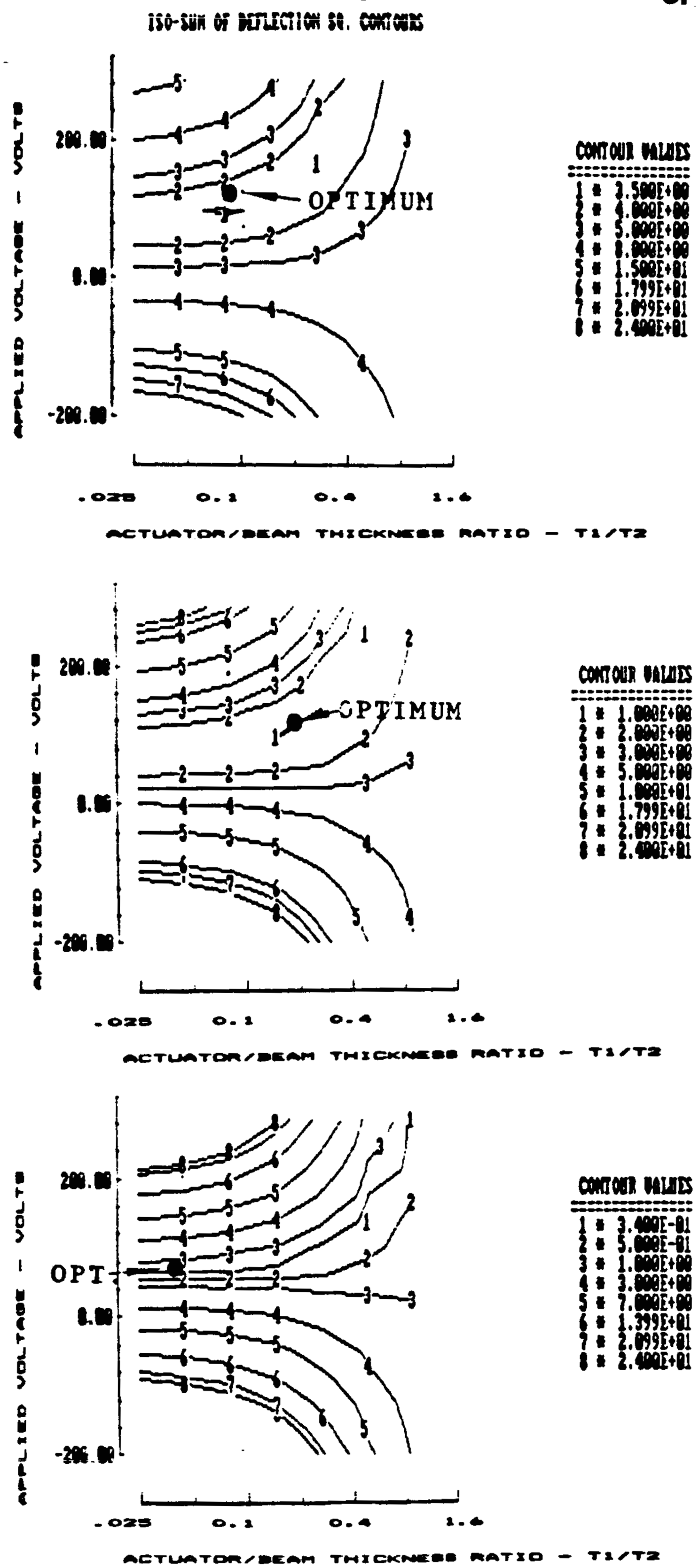


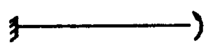
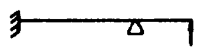
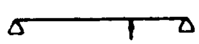
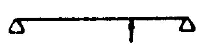
a) - Actuator at 1

b) - Actuator at 2

c) - Actuator at 3

Figure (6) - Iso-contours of the deflection criterion of a steel cantilever beam controlled with a PZT actuator when subjected to an external end load.

Table (2) - Optimum parameters of piezo-actuators for different loads, end conditions, beam materials and actuator types.

Case No.	Beam Type	Beam and Support Configuration	Actuator Location	Beam Material	Actuator Type	Load	Pt. of Appl.	Opt. Volt	Opt. t_1 (mm)	$U_{controlled}$ $(x10^7)$	$U_{uncontrolled}$ $(x10^7)$	$U_{uncontr.}$ / $U_{contr.}$
1	Cantilever		1 2 3	Steel	PZT	0.1N	Node 1	120 120 80	0.21 0.42 0.105	3.480 0.995 0.344	6.000	1.72 6.03 17.44
2	Cantilever		1 2 3	Steel	PZT	0.015Nm	Node 1	320 200 160	0.84 0.42 0.42	7.050 2.110 2.800	19.600	2.78 9.29 7.00
3	Overhung		1 2 3	Steel	PZT	0.1N	Node 1	120 120 -120	3.36 3.36 3.36	0.042 0.024 0.042	0.197	4.71 8.17 4.72
4	Simply-Supported		1 2 3	Steel	PZT	0.1N	Node 2	-120 - 80 - 40	3.36 3.36 1.68	0.039 0.011 0.057	0.098	2.54 8.91 1.73
5	Simply-Supported		1 2 3	Aluminum	PZT	0.1N	Node 2	-120 - 80 -120	1.68 1.68 1.68	0.338 0.095 0.501	0.868	2.57 9.10 1.73
6	Simply-Supported		1 2 3	Aluminum	Kynar	0.1N	Node 2	-200 -200 -200	3.36 3.36 3.36	0.616 0.400 0.741	0.868	1.41 2.17 1.17
7	Simply-Supported		1 2 3	Aluminum	G1278	0.1N	Node 3	-120 - 40 -160	3.36 1.68 3.36	0.338 0.096 0.500	0.868	3.47 9.09 1.74

controlling the elastic deformation of the beam.

Table (2) indicates that the ratio of the deflection criteria is 1.72, 6.03 and 17.44 when the actuator is bonded to elements 1, 2 and 3 respectively.

Accordingly, the proper placement of the piezo-actuator along the beam is very critical in controlling the elastic deflection of the beam. With the actuator placed at element 3, it would be possible to have an optimally controlled beam that is 17.44 times less distorted than the uncontrolled beam.

In order to gain an insight into the effect that the actuator location has on the deflection characteristics of the beam, let us consider the profiles of the different elastic lines obtained by the optimized actuators. The linear and angular elastic deformation are shown in Figures (7.a) and (7-b) respectively for both the uncontrolled and optimally controlled beams. It is evident from the figures that placing an optimized actuator at element 1 produces the least compensation for the external loads and its effects are limited to the first element only. However, placing the actuator at the third element, near the fixed end, results in considerable improvement that is not confined to element 3 but is manifested clearly all over the three elements.

(ii) An End Bending Moment

If the considered cantilever beam is subjected to an external bending moment of 0.015Nm applied to its free end, then

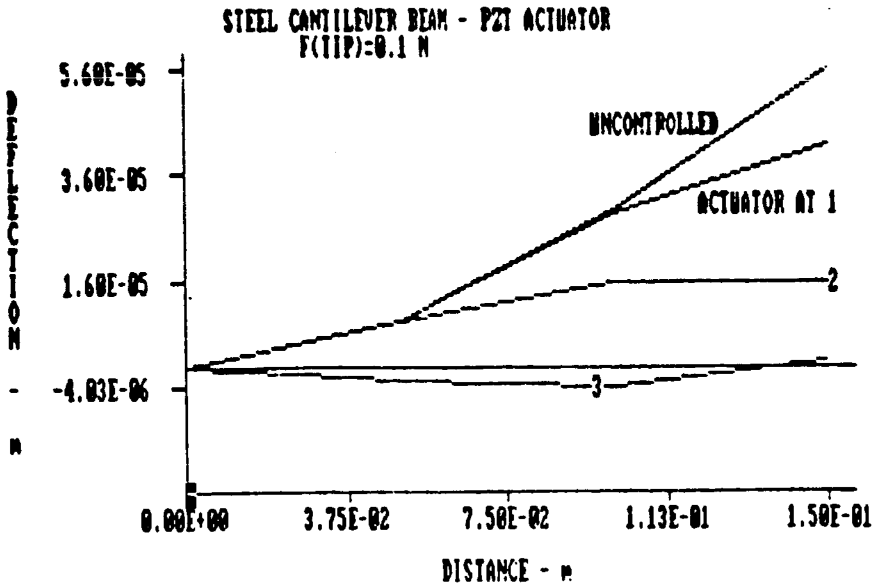


Figure (7-a) - Effect of the placement strategy of a PZT actuator on the linear deflection characteristics of a steel cantilever beam subjected to an external load.

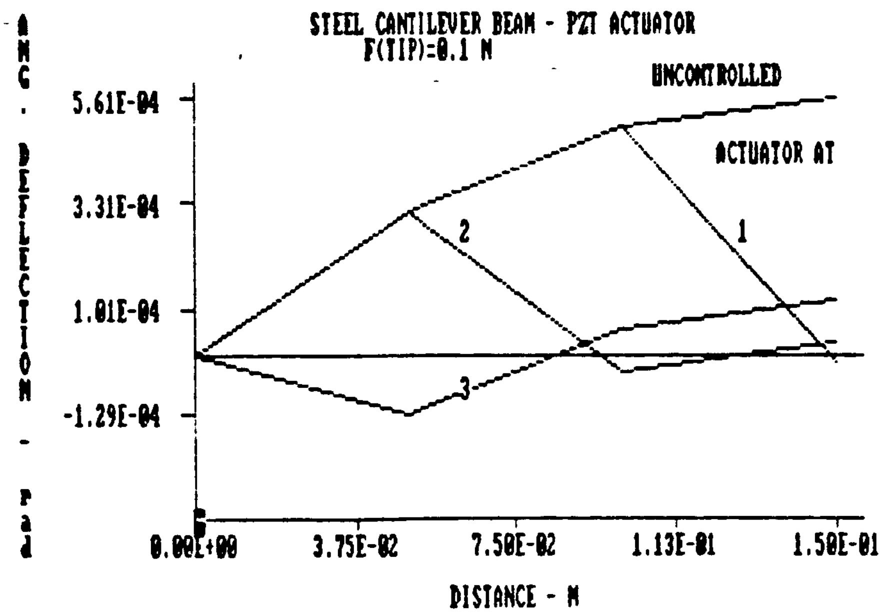


Figure (7-b) - Effect of the placement strategy of a PZT actuator on the angular deflection characteristics of a steel cantilever beam subjected to an external load.

the effect of placement of the actuator at element 1, 2 or 3 on the deflection criterion is as shown in the iso-contour maps of Figure (8-a), (8-b) and (8-c) respectively.

Figure (8-a) indicates that placing the actuator at element 1 makes the deflection criterion attains a minimum of 7.05×10^{-7} when the applied voltage is 320v and the actuator thickness is 0.00084m. When the actuator is placed at 2 the deflection criterion assumes a much lower minimum value of 2.11×10^{-7}. This minimum value is obtained by an actuator that is only half as thick which is excited by a much lower voltage of only 200 volts as compared to the 320 volts required if the optimal actuator is placed at element 1.

Accordingly, the proper placement of the actuator along the structure results not only on minimizing the elastic distortion of the beam but also in reducing the size and power consumption of the actuator.

If the actuator is placed, however, at 3, the deflection criterion starts to increase and even with an optimized actuator the best sum of the deflection squared that could be obtained is 2.8×10^{-7}.

A summary of the optimum parameters of the actuator along with the resulting improvement as measured with reference to the uncontrolled beam is given in Table (2).

The obtained results indicate that in the case of a moment applied to the end of the cantilever beam it is found beneficial

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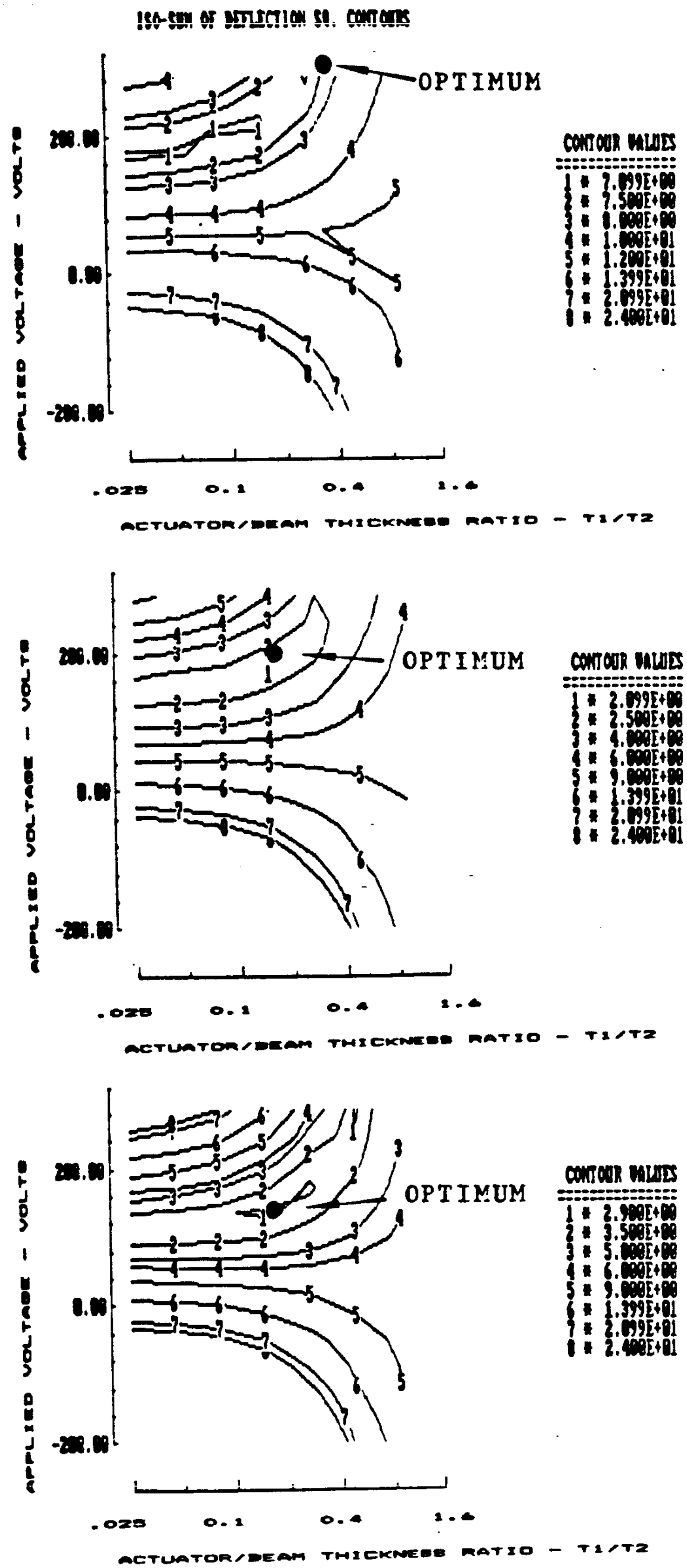


a) - Actuator at 1

b) - Actuator at 2

c) - Actuator at 3

Figure (8) - Iso-contours of the deflection criterion of a steel cantilever beam controlled with a PZT actuator when subjected to an external bending moment.

to place the piezo-actuator at element 2 to achieve the minimum distortion. This can also be emphasized by considering the linear and angular profiles of the elastic lines shown in Figures (9-a) and (9-b) respectively for the controlled and uncontrolled beam.

It should be pointed out here that the obtained optimal location of the actuator in case of an end moment is different from that obtained when the load is a transverse load applied to the beam end. This is inspite of the fact the load produces the same moment at the fixed end of the cantilever beam.

B. Effect Of End Conditions

Two end conditions, other than the fixed-free condition of the cantilever beam, are considered for the three element steel beam of Figure (5). Under these conditions, the beam is subjected to same transverse load of 0.1N.

(i) Overhung Beam

The beam is fixed at node 4 and simply-supported at node 2. The load is applied to the beam at its free end. In this case, the minimum deflection criterion is obtained when the actuator is bonded to element 2. The optimal actuator should have a thickness of 0.000336m and be subjected to 120 volts. The improvement in the deflection criterion with respect to the uncontrolled case is 8.17 times.

A summary of the optimal parameters of the actuator is given in Table (2) as function of the actuator location.

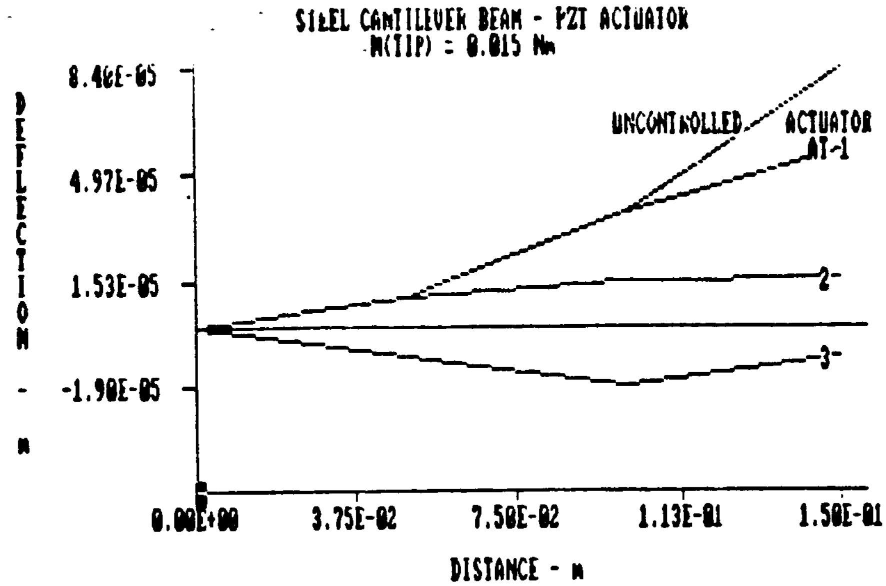


Figure (9-a) - Effect of the placement strategy of a PZT actuator on the linear deflection characteristics of a steel cantilever beam subjected to an external bending moment.

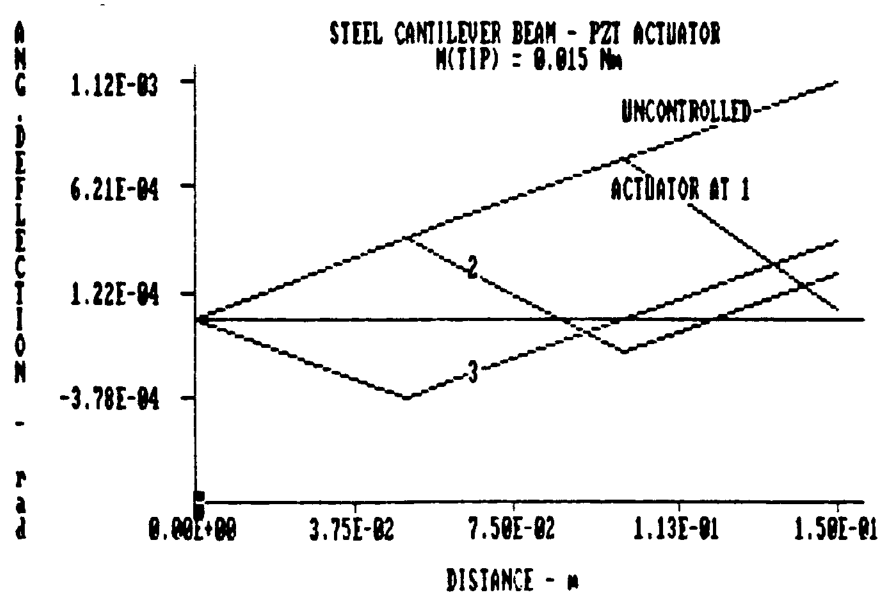


Figure (9-b) - Effect of the placement strategy of a PZT actuator on the angular deflection characteristics of a steel cantilever beam subjected to an external bending moment.

Figures (10-a) and (10-b) display the linear and angular deformations respectively of the overhung beam with and without the actuators. It is evident, from these two figures, that placement of the actuator at element 2 results in the minimal deflection profiles.

(ii) Simply-Supported Beam

The three element beam is considered now with its two end nodes 1 and 4 are simply-supported. The transverse load is applied to the beam at node 2.

The results of the optimization procedure, summarized in Table (2), indicates that the optimal deflection control can be achieved when the actuator is placed at element 2. The resulting improvement over the uncontrolled beam is 8.91 times. This requires the actuator to be 0.00336m thick and excited by 80 volts.

Figures (11-a) and (11-b) show the corresponding linear and angular deformations respectively of the controlled and uncontrolled beam. The figures illustrate clearly the effectiveness of the optimal actuator 2 in compensating for the externally applied load.

C. Effect Of Beam Material

The effect of changing the beam material from steel to aluminum is considered for the case of a simply-supported beam with a transverse load of 0.1N applied at node 2.

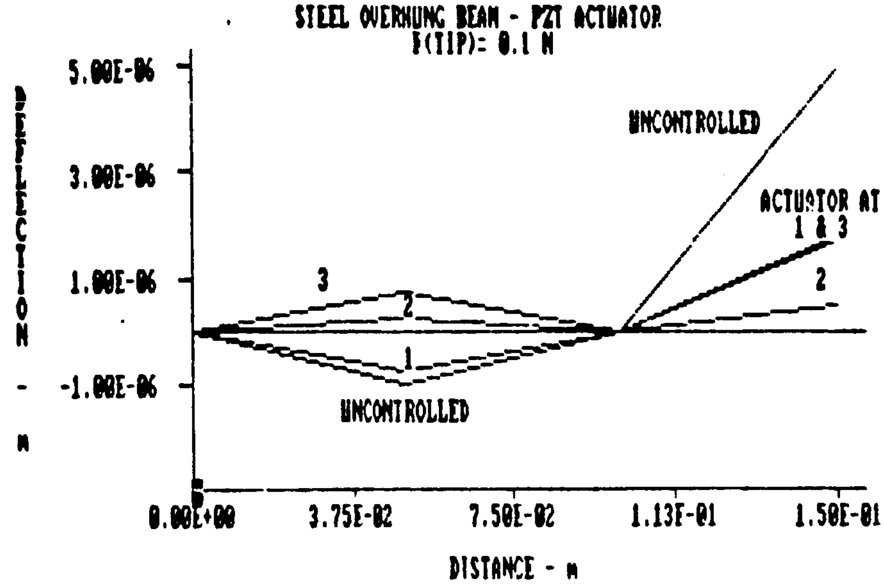


Figure (10-a) - Effect of the placement strategy of a PZT actuator on the linear deflection characteristics of a steel overhung beam.

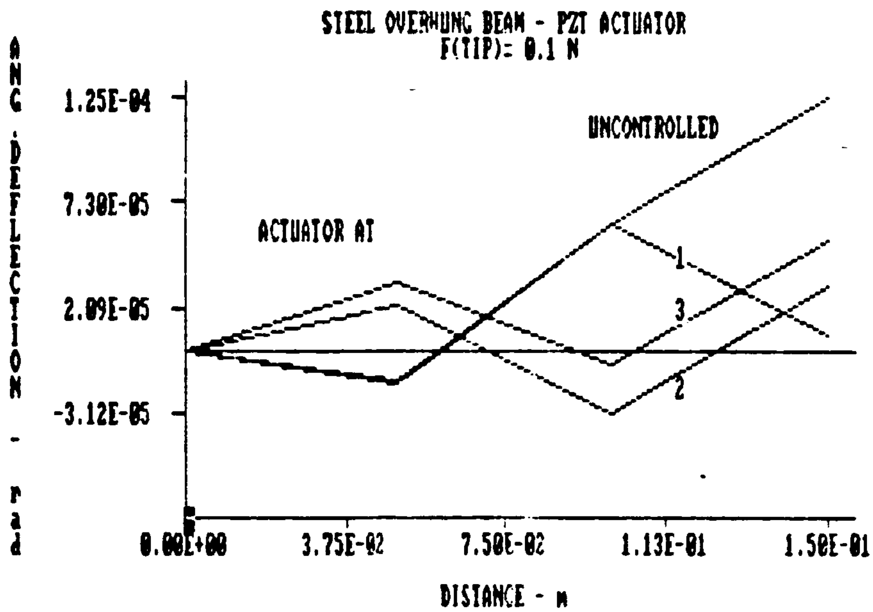


Figure (10-b) - Effect of the placement strategy of a PZT actuator on the angular deflection characteristics of a steel overhung beam.

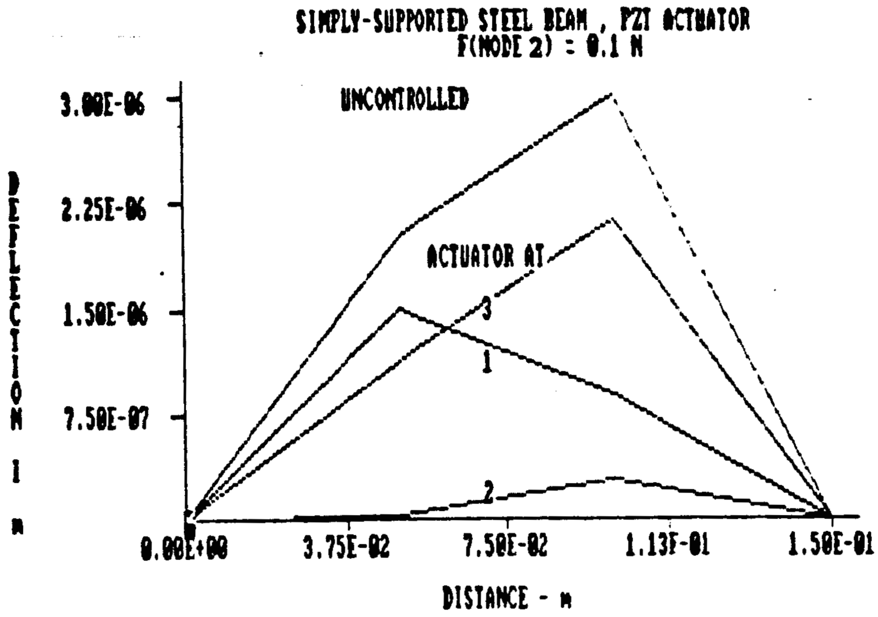


Figure (11-a) - Effect of the placement strategy of a PZT actuator on the linear deflection characteristics of a simply-supported steel beam.

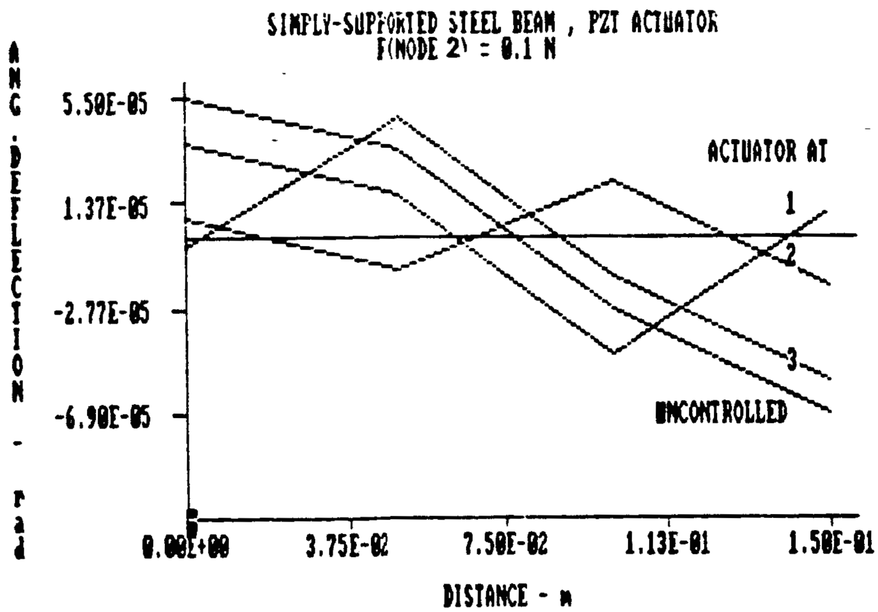


Figure (11-b) - Effect of the placement strategy of a PZT actuator on the angular deflection characteristics of a simply-supported steel beam.

The optimization algorithm yields results that are similar to those obtained for the steel beam. The optimum location of the actuator is on element 2 and the improvement over the uncontrolled beam is 9.10 which is slightly higher than the steel beam case. Important here to note that the thickness of the optimal actuator is only half that required to control the steel beam whereas the applied voltage is of the same magnitude.

The optimal linear and angular deflection profiles of the aluminum beam, which are shown in Figures (12-a) and (12-b), are very similar to those of the steel beam. However, the magnitude of these deflections are about 3 times higher than those of the steel beam. This difference is attributed mainly to the difference in Young's modulus of elasticity of the two materials.

D. Effect Of Actuator Material

Two actuator materials are considered other than the PZT-ceramic actuators. These two materials are :

(i) G1278-Ceramic

This type of actuator, as indicated in Table (1), has about half the Young's modulus of elasticity of the PZT actuators but at the same time it has almost twice as much the charge constant. These two factor combined result in nearly the same piezo-electric stress and bending moment.

The output of the optimization algorithm indicates, as expected, that the optimal parameters of the G1278 and PZT

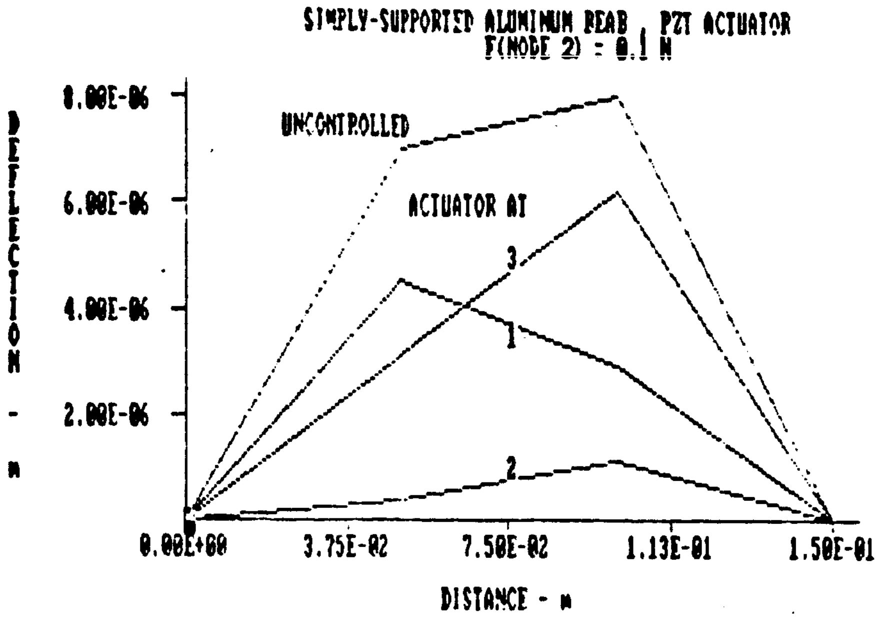


Figure (12-a) - Effect of the placement strategy of a PZT actuator on the linear deflection characteristics of a simply-supported aluminum beam.

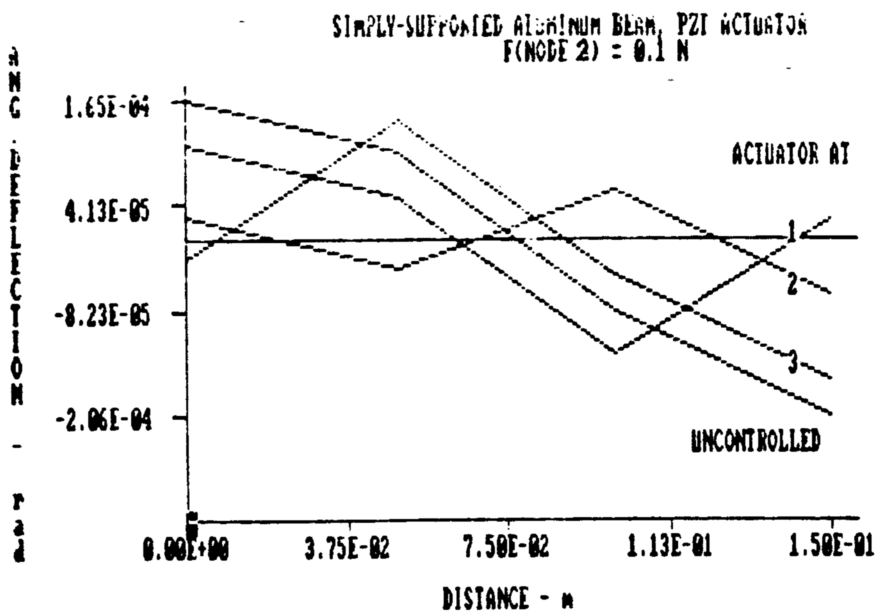


Figure (12-b) - Effect of the placement strategy of a PZT actuator on the angular deflection characteristics of a simply-supported aluminum beam.

actuators are nearly the same as shown in Table (2), This is clearly emphasized in the resulting linear and angular deformations profiles of Figure (13-a) and (13-b) respectively.

(ii) Kynar-Polymeric

A kynar actuator is considered as a typical representative of the polymeric piezo-electric actuators. This type of actuators has very low Young's modulus of elasticity and also very small charge constant. Accordingly, it would only be capable of producing small piezo-electric moments unless the voltage applied across the actuator is increased considerably to counter balance the low E and d parameters.

Considering this actuator on the aluminum beam, the optimization routine indicates that the best results are obtained with an actuator placed at location 2. This actuator should be twice as thick as the optimal PZT actuator and should be excited by a voltage which is 2.5 times as high. With these optimal parameters, the obtained improvement is only 2.17 times compared to 9.10 times for the PZT actuators.

Figures (14-a), (14-b) and (14-c) display the iso-contour maps of the deflection criterion when the kynar actuator is placed at element 1, 2 or 3 respectively. These figures emphasize the results of the optimization routine as it can be seen that the optimal actuator parameters tend to move towards the large thickness and large voltage constraint boundaries.

Figures (15-a) and (15-b) show the linear and angular

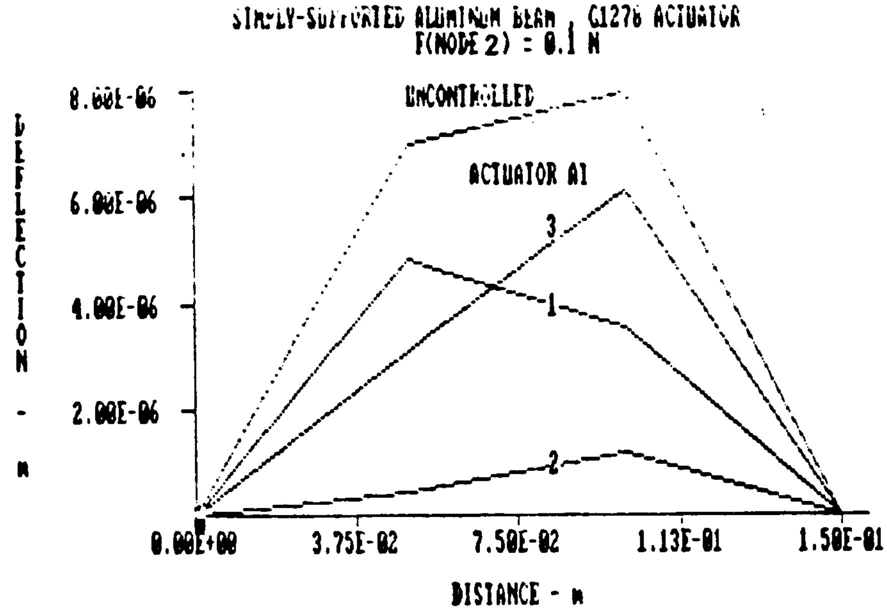


Figure (13-a) - Effect of the placement strategy of a G1278 actuator on the linear deflection characteristics of a simply-supported aluminum beam.

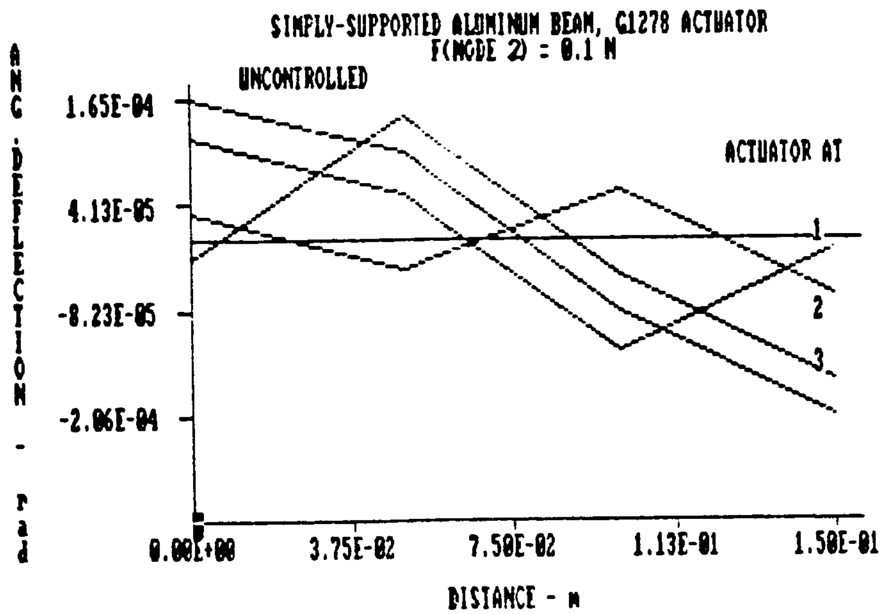
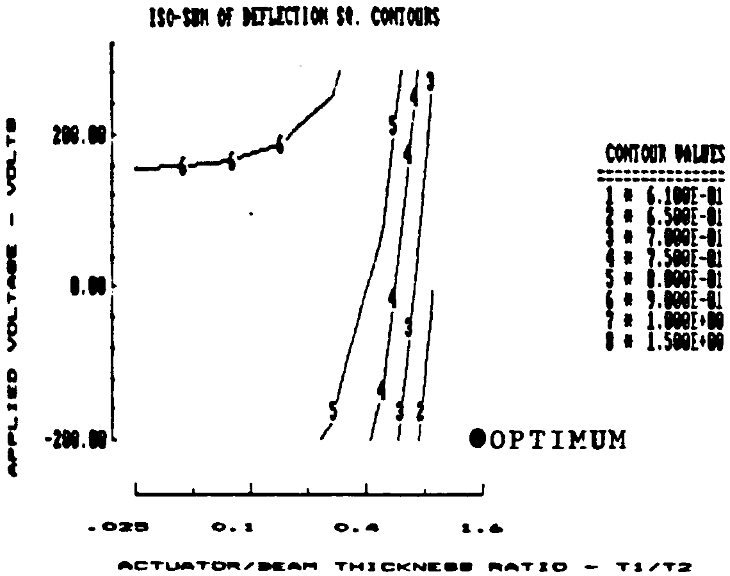
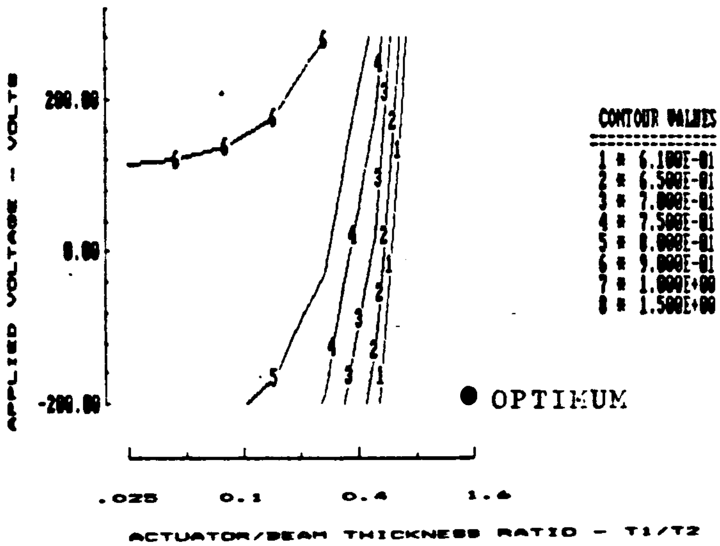


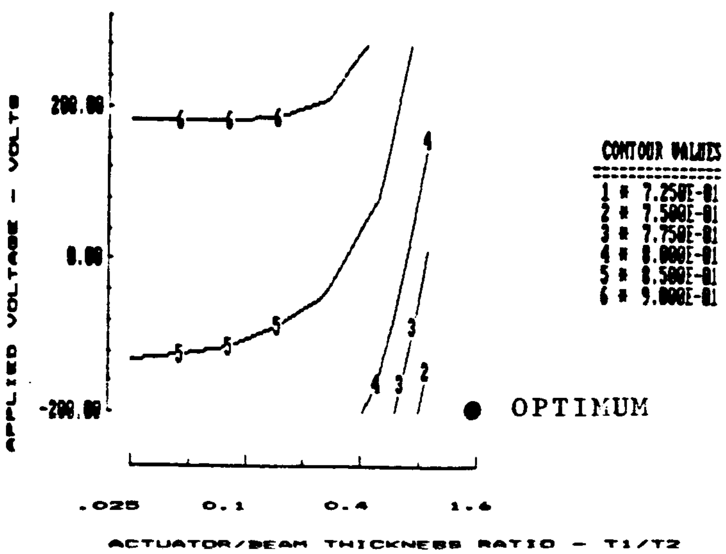
Figure (13-b) - Effect of the placement strategy of a G1278 actuator on the angular deflection characteristics of a simply-supported aluminum beam.



a) - Actuator at 1



b) - Actuator at 2



c) - Actuator at 3

Figure (14) - Iso-contours of the deflection criterion of a simply-supported aluminum beam controlled with a kynar actuator.

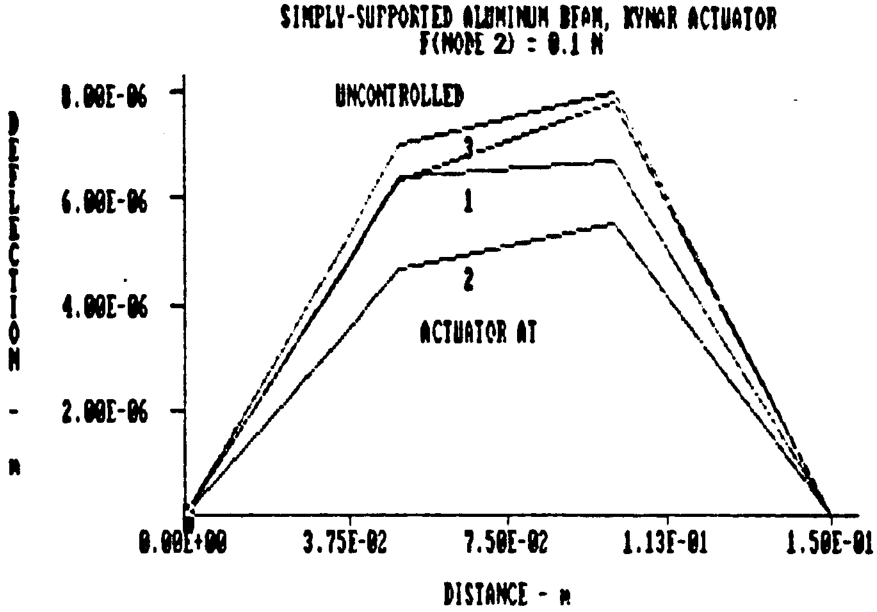


Figure (15-a) - Effect of the placement strategy of a kynar actuator on the linear deflection characteristics of a simply-supported aluminum beam.

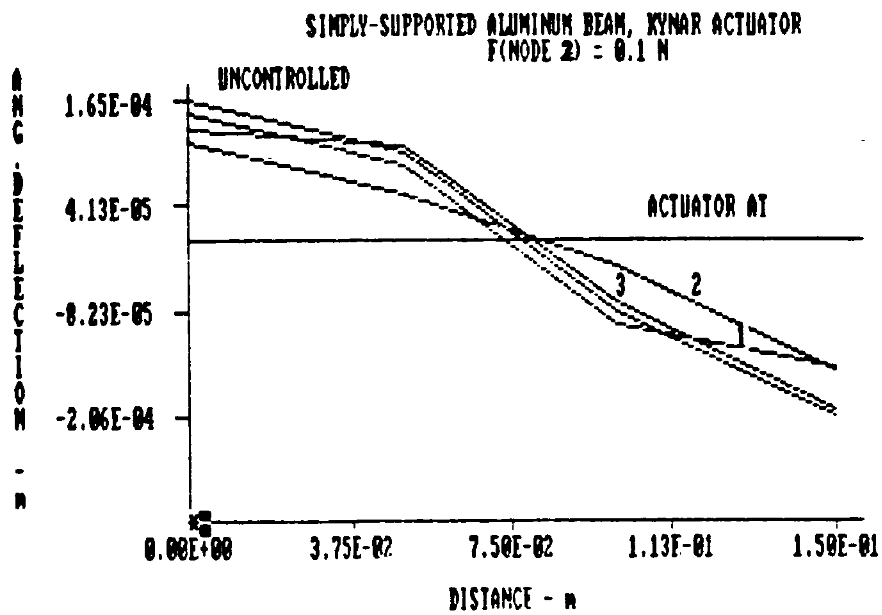


Figure (15-b) - Effect of the placement strategy of a kynar actuator on the angular deflection characteristics of a simply-supported aluminum beam.

deformations respectively of the aluminum beam. The obtained results indicate that the optimal deflections, with the actuator at 2, are about 4 times higher than those obtained with the G1278 or PZT actuators.

It should however be pointed out here that the more effective deflection control could be obtained by the kynar actuator if the applied voltage is increased beyond the set limits of the constraints.

CONCLUSIONS

This study has addressed the important problems of optimal selection, placement and control of piezo-electric actuators in order to minimize the static deflection of elastic beams under different loading conditions. Emphasis has been placed in this study on the effect that the actuators has on changing the elastic and inertial properties of the structural element to which they are bonded to.

The developed optimum design procedure is applied, with a large degree of flexibility, to elastic beam of various configurations and subjected to wide variety of end conditions.

The obtained results suggest the importance of the developed strategies in controlling the structural deformations of elastic systems.

Further work is in progress to integrate the presented procedure into the optimal control of the dynamics of more complex structures.

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