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Current Trends in PZT Control of 2D Cylindrical Panels Made of Advanced Materials – Dynamic Instability and Wave Propagation

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Current Trends in PZT Control of 2D Cylindrical Panels Made of Advanced Materials—Dynamic Instability and Wave Propagation

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Abstract:

The paper presents a methodology for the passive and active displacement and vibration damping of beams an d multilayered composite cylindrical shells. The displacement suppression systems consist of piezoelectric extension actuators bonded to the external surface. The second solution applies piezoelectric shear actuators/sensors embedded in the structure, An analysis on active vibration control of smart laminated cylindrical shells based on Hamilton's principle and the higher order deformation theory is discussed. Then, the investigated cylindrical structure is modelled using two different approaches: solutions with the aid of series with the use of numerical symbolic package Mathematica and finite element NISA II models. Active vibration suppression is implemented with positive position feedback (PPF) and velocity feedback. The non-local formulation shows the influence of governing relations on eigenfrequencies. The effects of the position of the piezoelectric actuators on shell structures is demonstrated .

Keywords: laminated beams; laminated cylindrical panels; active vibration control; passive displacement control; extension actuators, shear actuators; optimization problems

1. Introduction

Intelligent materials with multifield coupling properties become an important aspect of modern science and technology with applications in many industrial fields such as biomedical, electronic and mechanical engineering. The concept involves combined sensing, actuation and control capabilities embedded within materials and structures at different scales [1,2].

The subject of the analysis is connected with the variety of problems that due to the lack of space are given in the very shortened form: 1) the type of the material – multilayered composites, piezoelectrics, functionally graded materials, nanostructures, 2) the failure mode considered – static (buckling) or dynamic instabilities (divergence), free vibrations, 3) the form of PZT – extension actuators bonded to the surface or shear actuators/sensors embedded in the structure, 4) the form of the PZT control – passive or active, 5) the accuracy of the kinematical hypothesis used in the description – classical, first or higher order shear deformation theory.

As it may be seen from the above list of factors that may affect on the electromechanical behaviour of 2D smart structures the complex studies/ investigations do not exist in the open literature. Different researchers conduct analysis of the particular problems presented in the above list not going deeper insight the problems and not even trying to generalize various results.

Therefore, the fundamental research problems that we intend to solve in the paper can be formulated in the following way:

the determination of the most significant/important elements arising in the PZT vibration suppression for the considered 2D smart structures made of advanced materials.

the definition of the most suitable design variables that can be implemented in the optimal design.

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The novelties introduced in the present work cannot be formulated only as the proposal of generalization of some results but they are also connected with the nonlinear description of unidirectional fibrous composites and nonlocal approach to problems dealing with nanostructures.

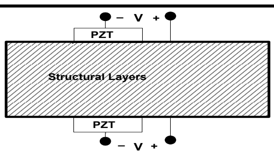
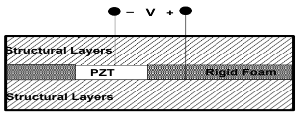
The present state of knowledge is illustrated by a series of works:

the PZT control and buckling analysis of plated/shell structures – [2–9]:
dynamic instability (non-conservative forces; pipelines and aeroelastic bahaviour) – [10–12];
free vibrations – [13–16];
functionally graded materials – [17–19];
nanostructures – [20–23];
material nonlinearities – [24,25];
nonlocal theories – [26–30];
sandwich structures – [31–37];
wave propagation and detection of subsurface defects – [38–43];
experimental studies – [44,45].

In the above list of publications more than 2000 references is presented and discussed so that it is impossible to demonstrate herein the review of the broader literature dealing with various aspects of the paper. The above list consists of four monographs and two review papers, not saying about research paper. The review of the cited literature illustrate evidently the necessity of carrying out the research and the lack of concise approach to design of 2D smart structures.

The present work intends to discuss and demonstrate a comprehensive methodology for the structural analysis and vibration suppression of 2D PZT- plated/shell constructions made of advanced materials. The subject of the analysis is connected with the form of PZT – extension actuators bonded to the surface or shear actuators/sensors embedded in the structure – Table 1.

Table 1. Cross-sections of structures with different actuators.

	
(a) extension actuator	(b) shear actuator

2. Formulation of the Local Coupled Electro-Mechanical Problem

2.1. Kinematic Relations

The displacements $\tilde{U}_i(x,y,z)$ (i=1,2,3) at any point of the shell in the *x*, *y* and *z* directions, respectively, are expressed in the following form [46]:

$$\begin{aligned}
&\tilde{U}_1(x,y,z)=u(x,y)+z\varphi_1+z^2\psi_1+z^3\gamma_1+z^4\theta_1\\
&\tilde{U}_2(x,y,z)=v(x,y)(1+z/R)+z\varphi_2+z^2\psi_2+z^3\gamma_2+z^4\theta_2 \qquad (1)\\
&\tilde{U}_3(x,y,z)=w(x,y)+z\chi_1+z^2\chi_2+z^3\chi_3
\end{aligned}$$

where *u*, *v*, *w* are the displacements of a generic point on the reference mid-surface, R denotes the radius of the cylindrical panel, ϕ_1, ϕ_2 are the rotations of normal to the mid-surface about the *y*- and *x*-axes, respectively and χ_1, χ_2 and χ_3 are three parameters characterizing the thickness stretching. Assuming that the transverse shear strains are equal to zero at the top and bottom of the shell surface the variables $u(x,y)$, φ_1, ψ_1, γ_1, θ_1, $v(x,y)$, φ_2, ψ_2, γ_2, θ_2 can be expressed as the linear functions of *u*, *v*, *w*, ϕ_1, ϕ_2, χ_1, χ_2, χ_3. The in-plane displacement have been expanded to the 4-th order and to the 3-rd order in z. Further simplifications of the number of the kinematic variables are described in Table 2.

Table 2. Variants of shell theories (TSE – transverse shear effects, SN – stretching of the normal, L-K Love Kirchhoff).

	Number of variables	Number of indepen-dent variables

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TSE/ SN – [47,48]	14 - $u, v, w, \phi_1, \phi_2, \psi_1, \psi_2, \gamma_1, \gamma_2, \theta_1, \theta_2, \chi_1, \chi_2, \chi_3$	**8** - $u, v, w, \phi_1, \phi_2, \chi_1, \chi_2, \chi_3$	4-th order TSE/ SN
TSE/ SN — [49]	13 - $u, v, w, \phi_1, \phi_2, \psi_1, \psi_2, \gamma_1, \gamma_2, \theta_1, \theta_2, \chi_1, \chi_2$	*7* - $u, v, w, \phi_1, \phi_2, \chi_1, \chi_2$	3-rd order TSE/ SN
TSE/ SN – [50],	12 - $u, v, w, \phi_1, \phi_2, \psi_1, \psi_2, \gamma_1, \gamma_2, \theta_1, \theta_2, \chi_1$	**6** - $u, v, w, \phi_1, \phi_2, \chi_1$	3-rd order TSE/ SN
TSE [51]	5 - u, v, w, ϕ_1, ϕ_2	5 - u, v, w, ϕ_1, ϕ_2	3-rd order TSE
TSE	5 - u, v, w, ϕ_1, ϕ_2	5 - u, v, w, ϕ_1, ϕ_2	1-rd order TSE
L-K	3 - u, v, w	3 - u, v, w	L-K

Based on the above expressions, the total, linear 3-D strain tensor can be written as follows:

$\varepsilon_{ij}(x,y,z)=0.5(\delta\tilde{U}_i/\delta x_j+\delta\tilde{U}_j/\delta x_j), x_1=x, x_2=y, x_3=z$ (2)

To the linear membrane terms the non-linear components are added in the following form:

$\bar{e}_{xx}=0.5(\delta w/\delta x)^2, \bar{e}_{yy}=0.5(\delta w/\delta y)^2, \bar{e}_{xy}=\delta^2 w/(\delta x\delta y)$ (3)

Taking into account Eqs (1-3) the strain-displacement relations can be written in the following way – the explicit form is written in [44]:

$[\varepsilon_{ij}]=[\varepsilon_{ij}]^0+z[\kappa_{ij}]^{(0)}+z^2[\kappa_{ij}]^{(1)}+z^3[\kappa_{ij}]^{(2)}$

i, j=1 or 2 or 3

dimension of matrices [6x1]

2.2. Constitutive Equations and Variational Formulation

Modelling of composite structures having smart piezoelectric sensors or actuators is very similar to that for conventional composite layered structures, however, there is one difference reflected in the constitutive laws in the form of the electromechanical coupling. It affects also the additional complexities in the FE formulation. The constitutive model for laminated panels with embedded piezoelectric sensors/actuators (S/A(s) (Figure 1) is established in the global cylindrical coordinate *x-y-z* system (*x* – a longitudinal direction, *y* – a circumferential direction, *z* – a normal direction to a shell mid-surface). It is given by:

$[\sigma]=[C][\varepsilon]$ (4)

Dimension of matrices [9x1] =[9x9][1x9]

The symbols having the bar over them have the standard mechanical interpretation, i.e. stresses *[σ]* and strains *[ε]* – see Eq. (4). The coefficients in *[Q]* matrix can be calculated from the elastic moduli, Poisson's ratios, piezoelectric moduli and dielectric constants of the lamina. *[D]* is the vector of electric displacements (three components), *[e]* is the matrix of piezoelectric coefficients of size 6x3, *[μ]* is the permittivity matrix of size 3x3 and *[E]* is the applied electric field. Its field is defined as the gradient of the electric potential Φ_{el}, i.e.: $[E]=-grad\ \Phi_{el}$.

$\Phi_{el}(x,y,z)=\varphi_0+\varphi_1 z/t,\ \varphi_0=0.5(V^{+}+V^{-})$ (5)

Through integrating along the plate/shell thickness *z* the membrane, bending stress and transverse shear stress resultants can be determined in the classical way.

Introducing the total potential energy of a typical element of the cylindrical panel given by the potential energy:

$U_L=0.5\int[[\varepsilon]^{Tr}[C][\varepsilon]]Rdxdydz$ (6)

and the kinetic energy of the laminate:

$T=0.5\int\varrho\{(\partial u/\partial\tau)^2+(\partial v/\partial\tau)^2+(\partial w/\partial\tau)^2+$

$z^2(\partial\psi_1/\partial\tau)^2+z^2(\partial\psi_2/\partial\tau)^2\}Rdxdydz$ (7)

where Ω and ϱ are the volume (the Cartesian product $[0,L_x]x[0,L_y]x[-t/2,t/2]$) and density of the laminated structures. Besides, the work done by the external thermo-mechanical loads on the panel (see Figure 1) is as follows [47]:

$W^{ext}=0.5\int\{N_x^0(\partial w/\partial)^2+N_y^0(\partial w/\partial y)^2\}Rdxdydz$ (8)

and the work of an electric force:

$W_{el}=\int\{q\Phi_{el}+q(\Phi_{el}-\Phi^{*}_{el})\}Rdxdydz$ (9)

where the symbol q denotes the electric charge distributed over the surfaces (boundary electric conditions) and Φ_{el} denotes the electric potential.

Using Hamilton's Principle:

$\delta\{T-U_L+W^{ext}+W_{el}\}=0$ (10)

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one can derive the fundamental system of equations describing the deformations of the structures. Let us notice that the above operations can be carried out in a symbolic way searching for the possible variations of unknown functions - u, v, w, ϕ_1, ϕ_2, χ_1, χ_2, χ_3, Φ_{el} with the aid of the operation "*Variational Calculus*" that can be found in the package Mathematica. Finally, the variations of the Hamilton functional (16) leads to nine differential equations. The first two characterize the in-plane effects and are functions of in-plane displacements u and v. The next three describe bending and stretching effects and are directly connected with the description of the failure effects, and the last corresponds to the electric potential.

2.3. Non-Local Formulation

The above relations (1)-(10) corresponds to the classical (so-called local formulation). The non-local formulations can be included easily by adding differential operators.

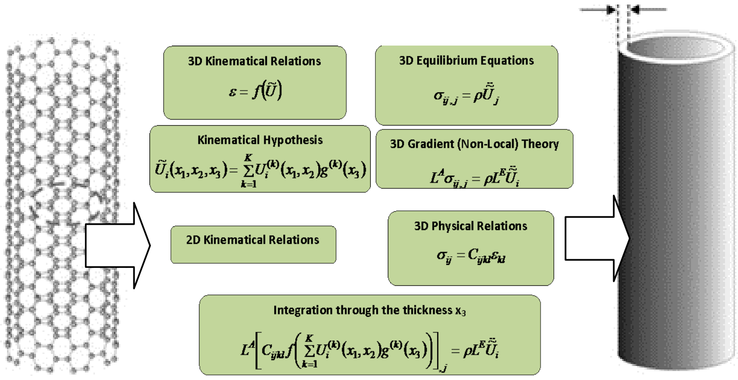

Figure 1. Continuum model of 2D structures in the non-local theories (L^A – differential operators).

The detailed explanation those problems is derived in [30]. The flowchart of such analysis is presented in Figure 1. Let us note that the non-local formulation deals with the classical stresses and strains [ε] and strains [σ] , and not to the electric fields.

3. Numerical Results

3.1. Active Vibration Control–Shear Actuator

Vibration suppression is a concept that is well suited for an intelligent structure because of its ability to sense and respond to external stimulus. The active vibration suppression uses sensors and actuators to control the response of the system Figure 2.

Strain Sensor **Charge amplifier & signal conditioner** **Data Acquisition Real Time Control**

System Signal Amplifier &Conditioner

Figure 2. Feedback vibration suppression control scheme using piezoelectric elements.

The system shown in Figure 2 is a closed-loop feedback control system since the signal sent to the actuator is a direct result of the incoming sensor signal.

The structure/compensator interaction is as follows (PPF positive position feedback – see Ref. [52]):

the structure: $\xi'' + 2\zeta\omega_n\xi' + \omega_n^2\xi = -G\,\omega_c^2\eta$

the compensator: $\eta'' + 2\,\zeta_c\omega_c\eta' + \omega_c^2\eta = \omega_c^2\xi'$ (11)

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where ξ is the modal coordinate, $\eta$is the compensator coordinate, ζ is the damping ratio of the structure, ω_n is the natural frequency of the structure, ζ_c is the compensator damping ratio, ω_c is the compensator frequency and G is a scalar gain.

There are really three quantities that must be appropriately chosen to implement this feedback control: the scalar gain, the compensator natural frequency and the compensator damping ratio. The rest of the quantities are a direct result of the physical system.

To verify the correctness of the present FE computations the vibration modes and the damping properties were compared with the example shown in Ref. [53]. An infinite cylindrical shell was analyzed loaded by a harmonic distributed load (along the shell circumference γ) of the following form:

$$q(\gamma,\tau)=q_0\sin(2(\gamma+\gamma_0))e^{i\omega\tau} \qquad (12)$$

The shell is composed of four composite layers (FRP) oriented at 0^0 along the circumference and two PZT layers – see Figure 4. The extent of the shell is 90^0 (i.e. in Eq.(4) $\gamma_0=45^0$ and $\gamma ě[-45^0,45^0]$). The shell radius R=250 mm. The thicknesses of individual composite layers are equal to 0.2h, whereas PZT layers 0.15h and 0.05h. The mechanical properties of the used materials are following: PZT - E_1=111 [GPa], E_2=E_3=121 [GPa], ϱ=7750 [kg/m^3] and composites - E_1=183.4 [GPa], E_2=E_3=11.7 [GPa], ϱ=1590 [kg/m^3]. Others material constants, used herein, can be found in [53].

In Ref. [53] the authors computed the vibration modes using 3D FE discretization available in the ABAQUS code. In our approach the calculations were carried out with the use of the NISA II package and the structure (Figure 3) was modelled by plane and 3D FE. For the first nine vibration modes the results are demonstrated in Table 3. The agreement is very good, both for different FE codes and plane and 3D formulations. The modes of vibrations are different: bending along the circumference (the number of waves is different), out of plane modes and the thickness shear mode (the fifth).

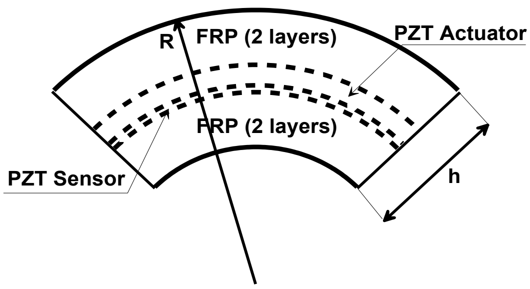

Figure 3. Cross-section of a cylindrical panel with PZT layers.

Table 3. Comparison of the natural frequencies (R/h=5).

Mode	Natural frequency [Hz]		
	[51] – FE 3D Model- ABAQUS	Present analysis – NISA II	
1.Bending	683.3	690.7	689.7
2. Out of plane	2393.7	2454.1	2505.2
3. Bending	2858.2	2907.4	2791.9
4. Out of plane	4780.8	4783.4	4693.2
5. Bending	5254.7	5292.1	5387.2
6. Out of plane	7156.8	7342.6	7245.3
7.Bending	7636.6	7600.4	7795.6
8. Bending	9984.2	9645.2	9342.3
9. Shear	10223.9	10443.9	10396.1

The two electrical boundary conditions are prescribed at $r = R - h$ and R in the sense that the top and bottom surfaces are electrically closed, i.e. Φ_{el}=0 – see Eq. (6).

A controller is employed for active vibration suppression. The second order PPF controller is forced by the electric potential of the sensor at γ = 0. The feedback voltage (evaluated from the sensor and

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compensator relations (12)) is applied to the PZT-5A actuator layer. For a given choice of control parameters ω_c, ζ_c and G – Eq. (11), the steady-state response of the system is computed for a given forcing frequency ω. The magnitude and phase of the radial deflection is plotted as function of the forcing frequency ω for different controller parameters to obtain frequency response curves – Figure 4.

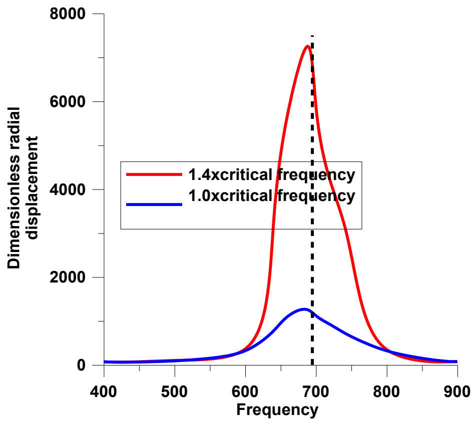

Figure 4. Damping of radial displacements (NISAII).

3.2. Smart Beam Structure – Extension Actuator

The use of passive actuators affects on the structural behaviour. In Ref. [55] the authors studied the behaviour of aluminium beam subjected to the action of actuators and sensors – Figure 5. The results given in Table 4 demonstrates a very good agreement between the ANSYS and NISA II analysis. The reduction of fundamental frequency reaches 97%.

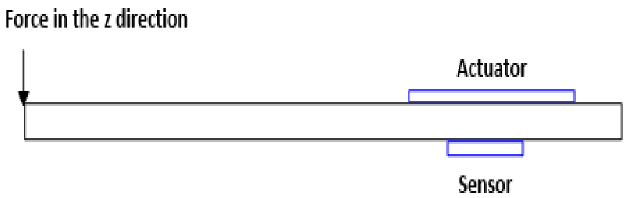

Figure 5. The smart beam structure.

Table 4. Comparison of the natural frequencies.

Mode	Natural Frequency dB		
	Ansys Karagulle FE 3D Model	Present analysis – NISA II	
		FE – 2D	FE- 3D
	29.7	24.5	24.4

The plotted results were evaluated with the use of the NISA II FE numerical package for the first critical frequency presented in Table 2. They are similar to those derived in Ref. [53]. However the author of the cited work calculated the results in an analytical, symbolic way (the Mathematica pack).

3.3. Location of Passive Actuators

Their influence of the position of sensors/actuators can be determined with the use of the optimal design procedures. For the passive reduction of normal displacements the positions of the PZT layers significant – see Figure 6, although the values of the resultant displacements are changed. Figure 6 represents the influence of the passive actuator located at the middle of the cylindrical panel. The

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normal displacements are reduced as the PZT actuators are bounded to the ends of the panel – Figure 6. Those problems can be solved with the use of the evolutionary algorithms [55].

The plots drawn in Figure 6 were obtained with the use of NISA II FE package and 3D finite elements.

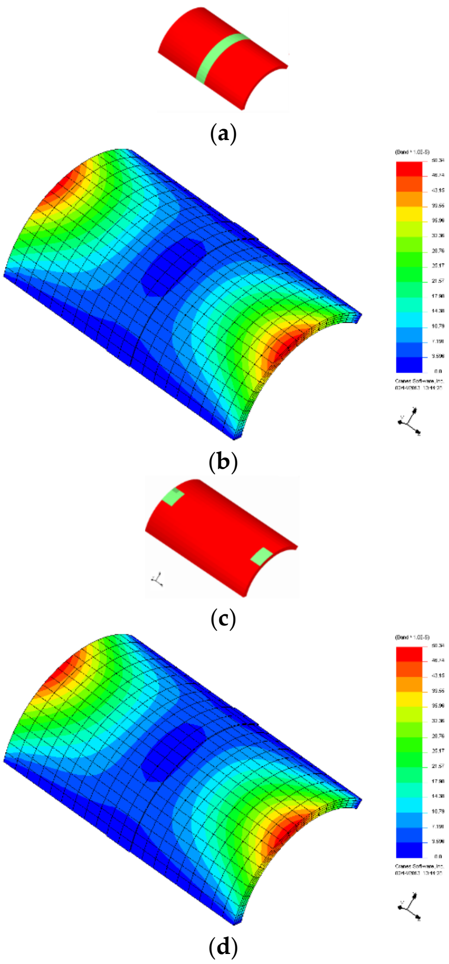

Figure 6. a. The form of passive actuator (green colour). b. Resultant displacements of compressed panels – R/h=100, V=100 [V]. c. The form of passive actuator (green colour). d. Resultant displacements of compressed panels – R/h=100, V=100 [V].

4. Non-Local Formulation

The influence of employing the non-local formulations is shown in Figure 7. Similarly as for free vibrations the use of the stress gradient theory [56] decreases values of frequencies ω (in the comparison to the classical theories), whereas the strain gradient theory [57] have opposite effects. The magnitude of the growth/decrease is a function of the length scale parameters The use of the Mindlin theory [58] does not change values of frequencies since. In general, the Mindlin model, unifies the gradient elasticity theories of Eringen and Aifantis. Let us note that different branches (modes) of the solution can be plotted in plots ω-s for different values of n.

The formulation of the different non-local theories is presented and discussed in Ref. [30].

The plotted results were evaluated with the use of the NISA II FE numerical package for the first critical frequency presented in Table 2. They are similar to those derived in Ref. [53]. However the author of the cited work calculated the results in an analytical, symbolic way (the Mathematica pack

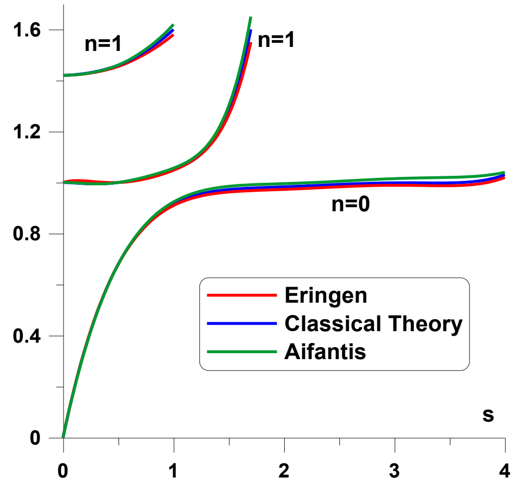

Figure 7. The comparison between classical and nonlocal theories.

5. Concluding Remarks

The present paper presents a variety of the numerical results dealing with the influence of the formulation of governing relations on the vibration/displacements

suppression of cylindrical panels having PZT layers. The general formulation with the use the Hamilton's principle for higher order 2D theories is discussed.

Various results are presented and solved:

- Local formulation and shear vibration control.
- Local formulation and extension displacement control.
- Non-local vibration control.

The conducted with the use of the Mathematica package and the NISA II commercial package demonstrate a very good agreement with the results shown in the literature.

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