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Research Article

Three-Dimensional Exact Free Vibration Analysis of Spherical, Cylindrical, and Flat One-Layered Panels

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The paper proposes a three-dimensional elastic analysis of the free vibration problem of one-layered spherical, cylindrical, and flat panels. The exact solution is developed for the differential equations of equilibrium written in orthogonal curvilinear coordinates for the free vibrations of simply supported structures. These equations consider an exact geometry for shells without simplifications. The main novelty is the possibility of a general formulation for different geometries. The equations written in general orthogonal curvilinear coordinates allow the analysis of spherical shell panels and they automatically degenerate into cylindrical shell panel, cylindrical closed shell, and plate cases. Results are proposed for isotropic and orthotropic structures. An exhaustive overview is given of the vibration modes for a number of thickness ratios, imposed wave numbers, geometries, embedded materials, and angles of orthotropy. These results can also be used as reference solutions to validate two-dimensional models for plates and shells in both analytical and numerical form (e.g., closed solutions, finite element method, differential quadrature method, and global collocation method).

1. Introduction

Exact solutions for the three-dimensional analysis of plates and shells have been developed by several researchers. In particular, three-dimensional static and dynamic analysis of cylinders and cylindrical/spherical shells has received considerable attention in the past few years. The development of plate and shell elements highlights the importance of research on the design of these geometries because they are fundamental for the analysis of general structures in the engineering field. For this reason, plate and shell elements need accurate validation. Three-dimensional exact solutions allow such validations and checks to be made, and they also give further details about three-dimensional effects and their importance. In the literature, works about exact three-dimensional solutions analyze separately the various geometries and they do not give a general overview of plate and shell elements. The present paper aims to fill this gap by proposing a general formulation for the equations of motion in orthogonal curvilinear coordinates that is valid for square and rectangular plates, cylindrical shell panels, spherical shell

panels, and cylinders. A general overview is given for those readers interested in both plate and shell analysis. This paper exactly solves the equations of motion in general curvilinear orthogonal coordinates including an exact geometry for shell structures without simplifications. To the best of the present author's knowledge, this is the first time that this solution is proposed by means of the exponential matrix method for the three-dimensional elastic free vibration analysis of plates and shells.

The next paragraph discusses the most relevant works about three-dimensional shell analysis. Chen et al. [1] exactly studied the coupled free vibrations of a transversely isotropic cylindrical shell embedded in an elastic medium. The behavior of the cylindrical shell was analyzed based on three-dimensional elasticity and three displacement functions were chosen to represent three displacement components to decouple the three-dimensional equations of motion of a transversely isotropic body. Fan and Zhang [2] presented the exact solutions for statics, dynamics, and buckling of thick open laminated cylindrical shells by means of the Cayley-Hamilton theorem. The state equation for orthotropy

was established in a cylindrical coordinate system without any assumption about displacement models and stress distribution. Gasemzadeh et al. [3] studied free vibrations of cylindrical shells on the basis of three-dimensional exact theory. Simply supported boundary conditions were studied and extensive frequency parameters were obtained by solving frequency equations. The free vibrations of simply supported cross-ply cylindrical and doubly curved laminates were investigated in [4]. The three-dimensional equations of motion, expressed in terms of displacements, were reduced to a system of coupled ordinary differential equations, that were solved using the power series method. Sharma et al. [5] studied the three-dimensional free vibrations of a homogeneous isotropic, viscothermoelastic hollow sphere whose surfaces were subjected to stress-free, thermally insulated or isothermal boundary conditions. The governing partial differential equations were solved into a coupled system of ordinary differential equations. Fröbenius matrix method was employed to obtain the solution. The same method was also used for the exact three-dimensional vibration analysis of a transversally isotropic, thermoelastic solid sphere [6]. Soldatos and Ye [7] proposed exact, three-dimensional, free vibration analysis of angle-ply laminated cylinders. Results were presented and discussed for relatively thick cylinders having a regular symmetric or a regular antisymmetric angle-ply lay-up. Armenakas et al. [8] proposed a self-contained treatment of the problem of plane harmonic waves propagation along a hollow circular cylinder in the framework of the three-dimensional theory of elasticity. The results presented frequencies of free vibrations of such cylinders for a wide range of axial and circumferential wave-lengths and of shell dimensions. An interesting comparison between frequencies obtained from a refined two-dimensional analysis, a shear deformation theory, the Flügge theory, and an exact elasticity analysis were proposed in [9]. Further details about the Flügge classical thin shell theory concerning the free vibrations of cylindrical shells with elastic boundary conditions can be found in [10]. Other comparisons between two-dimensional closed form solutions and available exact 3D elastic and analytical solutions for the free vibration analysis of simply supported and clamped homogeneous isotropic circular cylindrical shells were also proposed in [11]. Exact elasticity solutions were extended to functionally graded cylindrical shells in [12]. The three-dimensional linear elastodynamics equations, simplified to the case of generalized plane strain deformation in the axial direction, were solved using suitable displacement functions that identically satisfy the boundary conditions. Loy and Lam [13] proposed a layerwise approach to study the vibration of thick circular cylindrical shells on the basis of three-dimensional theory of elasticity. The governing equation was obtained using an energy minimization principle. Another interesting extension was proposed by Wang et al. [14], who investigated the free vibrations of magnetoelastic cylindrical panels based on the three-dimensional theory. Further results about three-dimensional analysis of shells that are not given in closed form were shown by Efraim and Eisenberger [15] for the dynamic stiffness matrix method and by Kang and

Leissa [16] and Liew et al. [17] for the three-dimensional Ritz method for vibration of spherical shells.

Three-dimensional analysis of plates is usually performed by using simpler equations that do not allow the analysis of shell geometries. Demasi [18] proposed a Navier-type method for finding the exact three-dimensional solution for isotropic thick and thin rectangular plates. The method used the mixed form of Hooke Law to write the boundary conditions on the top and bottom surfaces of the plate directly in terms of transverse stresses. Aimmanee and Batra [19] proposed an analytical solution for free vibrations of a simply supported rectangular plate made of an incompressible homogeneous linear elastic isotropic material. Some frequencies missing in previous analytical solutions for plates made of compressible materials were identified. Some of the in-plane modes of free vibration of a simply supported rectangular plate that were missed in previous exact solutions were also proposed in [20]. A three-dimensional linear elastic, small deformation theory obtained by the direct method was developed for the free vibration of simply supported, homogeneous, isotropic, thick rectangular plates in [21]. The same method was used in [22] for the flexure of simply supported homogeneous, isotropic, thick rectangular plates under arbitrary loading. This solution, in terms of infinite series, was formally exact and yielded accurate numerical results without undue effort. Useful comparisons between two-dimensional models and an exact three-dimensional solution for the free vibrations of a simply supported rectangular orthotropic thick plate are shown in [23]. On the basis of three-dimensional elasticity, Ye [24] presented a free vibration analysis of cross-ply laminated rectangular plates with clamped boundaries. The analysis was based on a recursive solution suitable for three-dimensional vibration analysis of simply supported plates. Comparisons between 2D-displacement-based models and exact results of the linear three-dimensional elasticity were also proposed in [25] for natural frequencies, displacement, and stress quantities in multilayered plates. Exact three-dimensional elasticity theory was investigated in [26] for isosceles triangular plates by means of a global three-dimensional Ritz formulation. The three-dimensional elasticity free vibration analysis of a circular plate was analyzed in [27] by means of the Ritz method with a set of orthogonal polynomial series to approximate the spatial displacements. Theoretical analysis of high frequency vibrations is indispensable in a variety of engineering designs. Zhao et al. [28] introduced a novel computational approach, the discrete singular convolution (DSC) algorithm, for high frequency vibration analysis of plate structures. A completely independent approach, the Levy method, was employed to provide exact solutions to validate the proposed method. The importance of high frequency analysis of multilayered composite plates was also confirmed in [29]. Further works about the analysis of high frequency vibrations of structures are given in [30], where the discrete singular convolution (DSC) algorithm was used and then compared with a completely independent approach (the Levy method). In [31] the first nine frequency parameters of circular and annular plates with variable thickness and combined boundary conditions were computed for different thickness to radius ratios. Three-dimensional elasticity theory was used in [31] and the

Ritz method was applied to derive the eigenvalue equation. Xing and Liu [32] proposed the separation of variables to solve the Hamiltonian dual form of eigenvalue problem for transverse free vibrations of thin plates, and formulation of the natural modes in closed form was performed. The closed form natural mode exactly satisfied the governing equation of the eigenvalue problem of thin plate and was applicable for any types of boundary conditions. The extension of 3D exact analysis to functionally graded plates was performed in [33–35]. Exact closed form solutions (based on three-dimensional elasticity theory) were carried out in [33] for both in-plane and out-of-plane free vibration of thick homogeneous simply supported rectangular plates coated by a functionally graded (FG) layer. Vel and Batra [34] presented a three-dimensional exact solution for free and forced vibrations of simply supported functionally graded rectangular plates and displacement functions that identically satisfy boundary conditions. Xu and Zhou [35] used the three-dimensional elasticity theory to derive the general expressions for the displacements and stresses of the plate under static loads, which exactly satisfy the governing differential equations and the simply supported boundary conditions at four edges of the plate. The extension of three-dimensional solutions to plates embedding piezoelectric layers was given in [36–41]. The three-dimensional exact solution by Haojiang et al. [36] allowed the free vibration analysis of a circular plate made of piezoelectric material. Baillargeon and Vel [37] obtained an exact three-dimensional solution for the cylindrical bending vibration of simply supported laminated composite plates with an embedded piezoelectric shear actuator. The bending and free vibration of an imperfectly bonded orthotropic piezoelectric rectangular laminate were investigated in [38] using a three-dimensional state-space approach. Cheng et al. [39] presented an exact three-dimensional method for the distribution of mechanical and electric quantities in interior points of a symmetrically inhomogeneous and laminated piezoelectric plate in the framework of linear theory of piezoelectricity. An exact three-dimensional (3D) piezothermoelasticity solution was presented in [40] for static, free vibration and steady state harmonic response of simply supported cross-ply piezoelectric (hybrid) laminated rectangular plates with interlaminar bonding imperfections. Zhong and Shang [41] presented an exact three-dimensional analysis for a functionally graded piezoelectric rectangular plate that was simply supported and grounded along its four edges. The state equations of the functionally graded piezoelectric plate were developed based on the state space approach.

The revision of the literature about three-dimensional analysis of shells and plates demonstrates that there are a variety of interesting works concerning plate or shell geometry, and they include static and dynamic analysis, functionally graded, composite and piezoelectric materials, displacement, and mixed models. The present work gives a general formulation for all the geometries (square and rectangular plates, cylindrical and spherical shell panels, and cylindrical closed shells). The equations of motion for the dynamic case are written in general orthogonal curvilinear coordinates by using an exact geometry for shells. The system of second order differential equations is reduced to a system

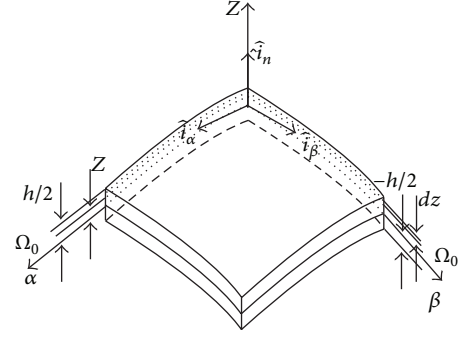


FIGURE 1: Notation and reference system for shells.

of first order differential equations, and afterwards it is exactly solved by using the exponential matrix method. Similar approaches have been used in [25] for the three-dimensional analysis of plates in rectilinear orthogonal coordinates and in [7] for an exact, three-dimensional, free vibration analysis of angle-ply laminated cylinders in cylindrical coordinates. The equations of motion written in orthogonal curvilinear coordinates allow general exact solutions for simply supported plate and shell geometries with constant radii of curvature. These equations are exactly solved, for the first time in this paper, by means of the exponential matrix method.

2. Geometrical and Constitutive Equations

A shell is a three-dimensional body bounded by two closely spaced curved surfaces where one dimension (the distance between the two surfaces) is small in comparison with the other two dimensions in the plane directions. The middle surface of the shell is the locus of points which lie midway between these surfaces. The distance between the surfaces measured along the normal to the middle surface is the *thickness* of the shell at that point [42, 43]. Geometry and the reference system are indicated in Figure 1 where curvilinear orthogonal coordinates (α, β, z) are shown. The three-dimensional space (the space in which the three independent variables α , β , and z vary) is an Euclidean space. R_α and R_β are the principal radii of curvature along the coordinates α and β , respectively. z is the thickness coordinate that varies from $-h/2$ to $+h/2$ (where h is the thickness of the structure as shown in Figure 1).

In this work, we will focus on shells with constant radii of curvature (e.g., cylindrical and spherical geometries). The geometrical relations written for shells with constant radii of curvature are a particular case of the strain-displacement equations of three-dimensional theory of elasticity in orthogonal curvilinear coordinates (see also [44–46]):

$$\epsilon_{\alpha\alpha} = \frac{1}{H_\alpha} \frac{\partial u}{\partial \alpha} + \frac{w}{H_\alpha R_\alpha},$$

$$\epsilon_{\beta\beta} = \frac{1}{H_\beta} \frac{\partial v}{\partial \beta} + \frac{w}{H_\beta R_\beta},$$

$$\begin{aligned}
\epsilon_{zz} &= \frac{\partial w}{\partial z}, \\
\gamma_{\alpha\beta} &= \frac{1}{H_\alpha} \frac{\partial v}{\partial \alpha} + \frac{1}{H_\beta} \frac{\partial u}{\partial \beta}, \\
\gamma_{\alpha z} &= \frac{1}{H_\alpha} \frac{\partial w}{\partial \alpha} + \frac{\partial u}{\partial z} - \frac{u}{H_\alpha R_\alpha}, \\
\gamma_{\beta z} &= \frac{1}{H_\beta} \frac{\partial w}{\partial \beta} + \frac{\partial v}{\partial z} - \frac{v}{H_\beta R_\beta},
\end{aligned} \tag{1}$$

where u , v , and w are the displacement components in α , β , and z direction, respectively. Symbol ∂ indicates the partial derivatives; the parametric coefficients for shells, with constant radii of curvature, that use the orthogonal curvilinear coordinate system are

$$H_\alpha = \left(1 + \frac{z}{R_\alpha}\right), \quad H_\beta = \left(1 + \frac{z}{R_\beta}\right), \quad H_z = 1. \tag{2}$$

General geometrical relations for spherical shells degenerate into geometrical relations for cylindrical shells when R_α or R_β is infinite (with H_α or H_β equals one), and they degenerate into geometrical relations for plates when both R_α and R_β are infinite (with $H_\alpha = H_\beta = 1$) [46].

Geometrical relations (see (2)) are substituted in three-dimensional linear elastic constitutive equations in orthogonal curvilinear coordinates (α , β , z) for orthotropic material in the structural reference system (see [47, 48]). Partial derivatives $\partial/\partial\alpha$, $\partial/\partial\beta$, and $\partial/\partial z$ are indicated with subscripts $_{,\alpha}$, $_{,\beta}$, and $_{,z}$. A closed form solution of differential equations of equilibrium for shells is obtained when coefficients C_{16} , C_{26} , C_{36} , and C_{45} are set to zero (this means orthotropic angle θ equals 0° or 90°):

$$\begin{aligned}
\sigma_{\alpha\alpha} &= \frac{C_{11}}{H_\alpha} u_{,\alpha} + \frac{C_{11}}{H_\alpha R_\alpha} w + \frac{C_{12}}{H_\beta} v_{,\beta} + \frac{C_{12}}{H_\beta R_\beta} w + C_{13} w_{,z}, \\
\sigma_{\beta\beta} &= \frac{C_{12}}{H_\alpha} u_{,\alpha} + \frac{C_{12}}{H_\alpha R_\alpha} w + \frac{C_{22}}{H_\beta} v_{,\beta} + \frac{C_{22}}{H_\beta R_\beta} w + C_{23} w_{,z}, \\
\sigma_{zz} &= \frac{C_{13}}{H_\alpha} u_{,\alpha} + \frac{C_{13}}{H_\alpha R_\alpha} w + \frac{C_{23}}{H_\beta} v_{,\beta} + \frac{C_{23}}{H_\beta R_\beta} w + C_{33} w_{,z}, \\
\sigma_{\beta z} &= \frac{C_{44}}{H_\beta} w_{,\beta} + C_{44} v_{,z} - \frac{C_{44}}{H_\beta R_\beta} v, \\
\sigma_{\alpha z} &= \frac{C_{55}}{H_\alpha} w_{,\alpha} + C_{55} u_{,z} - \frac{C_{55}}{H_\alpha R_\alpha} u, \\
\sigma_{\alpha\beta} &= \frac{C_{66}}{H_\alpha} v_{,\alpha} + \frac{C_{66}}{H_\beta} u_{,\beta}.
\end{aligned} \tag{3}$$

3. Equilibrium Equations

The three differential equations of equilibrium written for the case of free vibration analysis of spherical shells with constant

radii of curvature R_α and R_β are here given (the most general form for variable radii of curvature can be found in [45, 49]):

$$\begin{aligned}
H_\beta \frac{\partial \sigma_{\alpha\alpha}}{\partial \alpha} + H_\alpha \frac{\partial \sigma_{\alpha\beta}}{\partial \beta} + H_\alpha H_\beta \frac{\partial \sigma_{\alpha z}}{\partial z} \\
+ \left(\frac{2H_\beta}{R_\alpha} + \frac{H_\alpha}{R_\beta} \right) \sigma_{\alpha z} &= \rho H_\alpha H_\beta \ddot{u}, \\
H_\beta \frac{\partial \sigma_{\alpha\beta}}{\partial \alpha} + H_\alpha \frac{\partial \sigma_{\beta\beta}}{\partial \beta} + H_\alpha H_\beta \frac{\partial \sigma_{\beta z}}{\partial z} \\
+ \left(\frac{2H_\alpha}{R_\beta} + \frac{H_\beta}{R_\alpha} \right) \sigma_{\beta z} &= \rho H_\alpha H_\beta \ddot{v}, \\
H_\beta \frac{\partial \sigma_{\alpha z}}{\partial \alpha} + H_\alpha \frac{\partial \sigma_{\beta z}}{\partial \beta} + H_\alpha H_\beta \frac{\partial \sigma_{zz}}{\partial z} - \frac{H_\beta}{R_\alpha} \sigma_{\alpha\alpha} - \frac{H_\alpha}{R_\beta} \sigma_{\beta\beta} \\
+ \left(\frac{H_\beta}{R_\alpha} + \frac{H_\alpha}{R_\beta} \right) \sigma_{zz} &= \rho H_\alpha H_\beta \ddot{w},
\end{aligned} \tag{4}$$

where ρ is the mass density and \ddot{u} , \ddot{v} , and \ddot{w} indicate the second temporal derivative of the three displacement components.

Equation (3) is introduced in (4), and the closed form is obtained for simply supported shells and plates made of isotropic material or orthotropic material with 0° or 90° orthotropic angle (in both cases $C_{16} = C_{26} = C_{36} = C_{45} = 0$). The three displacement components have the following harmonic form:

$$\begin{aligned}
u(\alpha, \beta, z, t) &= U(z) e^{i\omega t} \cos(\bar{\alpha}\alpha) \sin(\bar{\beta}\beta), \\
v(\alpha, \beta, z, t) &= V(z) e^{i\omega t} \sin(\bar{\alpha}\alpha) \cos(\bar{\beta}\beta), \\
w(\alpha, \beta, z, t) &= W(z) e^{i\omega t} \sin(\bar{\alpha}\alpha) \sin(\bar{\beta}\beta),
\end{aligned} \tag{5}$$

where U , V , and W are the displacement amplitudes in α , β , and z directions, respectively. i is the coefficient of the imaginary unit, $\omega = 2\pi f$ is the circular frequency where f is the frequency value, and t is the time. In coefficients $\bar{\alpha} = m\pi/a$ and $\bar{\beta} = n\pi/b$, m and n are the half-wave numbers and a and b are the shell dimensions in α and β directions, respectively. Equilibrium equations are for spherical shell panels; they automatically degenerate into equilibrium equations for cylindrical closed/open shell panels when R_α or R_β is infinite (with H_α or H_β equals one) and into equilibrium equations for plates when R_α and R_β are infinite (with H_α and H_β equal one). In this way a unique and general closed formulation is possible for any geometry:

$$\begin{aligned}
\left(-\frac{C_{55}H_\beta}{H_\alpha R_\alpha^2} - \frac{C_{55}}{R_\alpha R_\beta} - \bar{\alpha}^2 \frac{C_{11}H_\beta}{H_\alpha} \right. \\
\left. - \bar{\beta}^2 \frac{C_{66}H_\alpha}{H_\beta} + \rho H_\alpha H_\beta \omega^2 \right) U + (-\bar{\alpha}\bar{\beta}C_{12} - \bar{\alpha}\bar{\beta}C_{66}) V
\end{aligned}$$

TABLE 1: Assessment 1. Simply supported aluminum square plate with Poisson ratio $\nu = 0.3$. First six exact natural circular frequencies in no-dimensional form $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for imposed half-wave numbers $m = n = 1$. Comparison between present three-dimensional analysis and three-dimensional analyses by Vel and Batra [34] and Srinivas et al. [22]. M is the number of fictitious layers and N is the order of expansion for the exponential matrix.

	Mode					
	I	II	III	IV	V	VI
$a/h = 10$						
Present 3D ($M = 1, N = 3$)	5.3370	27.554	46.493	154.39	168.09	268.36
Present 3D ($M = 1, N = 7$)	5.7769	27.554	46.502	192.91	197.98	329.35
Present 3D ($M = 1, N = 10$)	5.7769	27.554	46.502	196.21	201.87	308.34
Present 3D ($M = 1, N = 15$)	5.7769	27.554	46.502	196.77	201.34	358.23
Present 3D ($M = 1, N = 18$)	5.7769	27.554	46.502	196.77	201.34	357.42
Present 3D ($M = 10, N = 3$)	5.7769	27.554	46.502	196.71	201.27	355.40
Present 3D ($M = 50, N = 3$)	5.7769	27.554	46.502	196.77	201.34	351.90
Present 3D ($M = 100, N = 3$)	5.7769	27.554	46.502	196.77	201.34	357.42
3D [34]	5.7769	27.554	46.503	196.77	201.34	357.42
3D [22]	5.7769	27.554	46.502	196.77	201.34	357.42
$a/h = \sqrt{10}$						
Present 3D ($M = 1, N = 3$)	/	8.7132	14.182	17.512	/	/
Present 3D ($M = 1, N = 7$)	4.6598	8.7132	14.463	20.987	23.999	29.925
Present 3D ($M = 1, N = 10$)	4.6582	8.7132	14.463	21.383	25.009	31.399
Present 3D ($M = 1, N = 15$)	4.6581	8.7132	14.463	21.343	24.830	34.049
Present 3D ($M = 1, N = 18$)	4.6581	8.7132	14.463	21.343	24.830	33.982
Present 3D ($M = 10, N = 3$)	4.6580	8.7132	14.463	21.337	24.821	33.939
Present 3D ($M = 50, N = 3$)	4.6581	8.7132	14.463	21.343	24.830	33.923
Present 3D ($M = 100, N = 3$)	4.6581	8.7132	14.463	21.343	24.830	33.982
3D [34]	4.6582	8.7132	14.463	21.343	24.830	33.982
3D [22]	4.6582	8.7134	14.462	21.343	24.830	33.982

$$\begin{aligned}
& + \left(\bar{\alpha} \frac{C_{11}H_\beta}{H_\alpha R_\alpha} + \bar{\alpha} \frac{C_{12}}{R_\beta} + \bar{\alpha} \frac{C_{55}H_\beta}{H_\alpha R_\alpha} + \bar{\alpha} \frac{C_{55}}{R_\beta} \right) W \\
& + \left(\frac{C_{55}H_\beta}{R_\alpha} + \frac{C_{55}H_\alpha}{R_\beta} \right) U_{,z} + (\bar{\alpha} C_{13}H_\beta + \bar{\alpha} C_{55}H_\beta) W_{,z} \\
& + (C_{55}H_\alpha H_\beta) U_{,zz} = 0, \\
& (-\bar{\alpha}\bar{\beta}C_{66} - \bar{\alpha}\bar{\beta}C_{12}) U \\
& + \left(-\frac{C_{44}H_\alpha}{H_\beta R_\beta^2} - \frac{C_{44}}{R_\alpha R_\beta} \right. \\
& \quad \left. - \bar{\alpha}^2 \frac{C_{66}H_\beta}{H_\alpha} - \bar{\beta}^2 \frac{C_{22}H_\alpha}{H_\beta} + \rho H_\alpha H_\beta \omega^2 \right) V \\
& + \left(\bar{\beta} \frac{C_{44}H_\alpha}{H_\beta R_\beta} + \bar{\beta} \frac{C_{44}}{R_\alpha} + \bar{\beta} \frac{C_{22}H_\alpha}{H_\beta R_\beta} + \bar{\beta} \frac{C_{12}}{R_\alpha} \right) W \\
& + \left(\frac{C_{44}H_\alpha}{R_\beta} + \frac{C_{44}H_\beta}{R_\alpha} \right) V_{,z} + (\bar{\beta} C_{44}H_\alpha + \bar{\beta} C_{23}H_\alpha) W_{,z} \\
& + (C_{44}H_\alpha H_\beta) V_{,zz} = 0, \\
& + (C_{44}H_\alpha H_\beta) V_{,zz} = 0, \\
& \left(\bar{\alpha} \frac{C_{55}H_\beta}{H_\alpha R_\alpha} - \bar{\alpha} \frac{C_{13}}{R_\beta} + \bar{\alpha} \frac{C_{11}H_\beta}{H_\alpha R_\alpha} + \bar{\alpha} \frac{C_{12}}{R_\beta} \right) U \\
& + \left(\bar{\beta} \frac{C_{44}H_\alpha}{H_\beta R_\beta} - \bar{\beta} \frac{C_{23}}{R_\alpha} + \bar{\beta} \frac{C_{22}H_\alpha}{H_\beta R_\beta} + \bar{\beta} \frac{C_{12}}{R_\alpha} \right) V \\
& + \left(\frac{C_{13}}{R_\alpha R_\beta} + \frac{C_{23}}{R_\alpha R_\beta} - \frac{C_{11}H_\beta}{H_\alpha R_\alpha^2} - \frac{2C_{12}}{R_\alpha R_\beta} - \frac{C_{22}H_\alpha}{H_\beta R_\beta^2} \right. \\
& \quad \left. - \bar{\alpha}^2 \frac{C_{55}H_\beta}{H_\alpha} - \bar{\beta}^2 \frac{C_{44}H_\alpha}{H_\beta} + \rho H_\alpha H_\beta \omega^2 \right) W \\
& + (-\bar{\alpha} C_{55}H_\beta - \bar{\alpha} C_{13}H_\beta) U_{,z} + (-\bar{\beta} C_{44}H_\alpha - \bar{\beta} C_{23}H_\alpha) V_{,z} \\
& + \left(\frac{C_{33}H_\beta}{R_\alpha} + \frac{C_{33}H_\alpha}{R_\beta} \right) W_{,z} + (C_{33}H_\alpha H_\beta) W_{,zz} = 0.
\end{aligned} \tag{6}$$

TABLE 2: Assessment 2 (part I). Simply supported aluminum cylinder with Poisson ratio $\nu = 0.3$ and thickness ratio $h/R_\alpha = 0.12$. Lowest exact natural circular frequencies in no-dimensional form $\bar{\omega} = \omega(h/\pi)\sqrt{\rho/G}$ for imposed axial half-wave number $n = 1$ and several half-wave numbers m in circumferential direction. Comparison between present three-dimensional analysis and three-dimensional analysis by Armenakas et al. [8] as adapted in Bhimaraddi [9]. M is the number of fictitious layers and N is the order of expansion for the exponential matrix.

	$h/R_\alpha = 0.12$			
	$m = 2$	$m = 4$	$m = 6$	$m = 8$
$b/R_\alpha = 2$				
Present 3D ($M = 1, N = 3$)	0.03724	0.02323	0.02175	0.02846
Present 3D ($M = 10, N = 3$)	0.03730	0.02360	0.02464	0.03687
Present 3D ($M = 50, N = 3$)	0.03730	0.02359	0.02463	0.03686
Present 3D ($M = 100, N = 3$)	0.03730	0.02359	0.02462	0.03686
3D [8, 9]	0.03730	0.02359	0.02462	0.03686
$b/R_\alpha = 1$				
Present 3D ($M = 1, N = 3$)	0.05769	0.04772	0.04229	0.04272
Present 3D ($M = 10, N = 3$)	0.05853	0.04979	0.04790	0.05546
Present 3D ($M = 50, N = 3$)	0.05853	0.04978	0.04789	0.05545
Present 3D ($M = 100, N = 3$)	0.05853	0.04978	0.04789	0.05545
3D [8, 9]	0.05853	0.04978	0.04789	0.05545
$b/R_\alpha = 0.5$				
Present 3D ($M = 1, N = 3$)	0.08017	0.07732	0.07308	0.06667
Present 3D ($M = 10, N = 3$)	0.10057	0.10234	0.10688	0.11528
Present 3D ($M = 50, N = 3$)	0.10057	0.10234	0.10688	0.11528
Present 3D ($M = 100, N = 3$)	0.10057	0.10234	0.10688	0.11528
3D [8, 9]	0.10057	0.10234	0.10688	0.11528
$b/R_\alpha = 0.25$				
Present 3D ($M = 1, N = 3$)	0.48156	0.48619	0.49378	0.50419
Present 3D ($M = 10, N = 3$)	0.27489	0.27848	0.28446	0.29286
Present 3D ($M = 50, N = 3$)	0.27491	0.27849	0.28447	0.29287
Present 3D ($M = 100, N = 3$)	0.27491	0.27849	0.28447	0.29287
3D [8, 9]	0.27491	0.27849	0.28447	0.29287

The system of equation (6) can be written in a compact form by introducing coefficients A_r for each block ():

$$\begin{aligned}
 A_1 U + A_2 V + A_3 W + A_4 U_{,z} + A_5 W_{,z} + A_6 U_{,zz} &= 0, \\
 A_7 U + A_8 V + A_9 W + A_{10} V_{,z} + A_{11} W_{,z} + A_{12} V_{,zz} &= 0, \\
 A_{13} U + A_{14} V + A_{15} W + A_{16} U_{,z} \\
 + A_{17} V_{,z} + A_{18} W_{,z} + A_{19} W_{,zz} &= 0.
 \end{aligned} \tag{7}$$

Equation (7) is a system of three second order differential equations. They are written for spherical shell panels with constant radii of curvature but they automatically degenerate into equations for cylindrical shells and plates (see also [7, 25]).

3.1. Solution for the System of Differential Equations. The system of second order differential equations can be reduced to a system of first order differential equations by using the

method described in [50, 51]. This methodology allows the following system to be obtained:

$$\begin{bmatrix} A_6 & 0 & 0 & 0 & 0 & 0 \\ 0 & A_{12} & 0 & 0 & 0 & 0 \\ 0 & 0 & A_{19} & 0 & 0 & 0 \\ 0 & 0 & 0 & A_6 & 0 & 0 \\ 0 & 0 & 0 & 0 & A_{12} & 0 \\ 0 & 0 & 0 & 0 & 0 & A_{19} \end{bmatrix} \begin{bmatrix} U \\ V \\ W \\ U' \\ V' \\ W' \end{bmatrix}' = \begin{bmatrix} 0 & 0 & 0 & A_6 & 0 & 0 \\ 0 & 0 & 0 & 0 & A_{12} & 0 \\ 0 & 0 & 0 & 0 & 0 & A_{19} \\ -A_1 & -A_2 & -A_3 & -A_4 & 0 & -A_5 \\ -A_7 & -A_8 & -A_9 & 0 & -A_{10} & -A_{11} \\ -A_{13} & -A_{14} & -A_{15} & -A_{16} & -A_{17} & -A_{18} \end{bmatrix} \begin{bmatrix} U \\ V \\ W \\ U' \\ V' \\ W' \end{bmatrix}. \tag{8}$$

Equation (8) can be written in a compact form as

$$\mathbf{D} \frac{\partial \mathbf{U}}{\partial \bar{z}} = \mathbf{A} \mathbf{U}, \tag{9}$$

TABLE 3: Assessment 2 (part II). Simply supported aluminum cylinder with Poisson ratio $\nu = 0.3$ and thickness ratio $h/R_\alpha = 0.18$. Lowest exact natural circular frequencies in no-dimensional form $\bar{\omega} = \omega(h/\pi) \sqrt{\rho/G}$ for imposed axial half-wave number $n = 1$ and several half-wave numbers m in circumferential direction. Comparison between present three-dimensional analysis and three-dimensional analysis by Armenakas et al. [8] as adapted in Bhimaraddi [9]. M is the number of fictitious layers and N is the order of expansion for the exponential matrix.

$h/R_\alpha = 0.18$				
	$m = 2$	$m = 4$	$m = 6$	$m = 8$
$b/R_\alpha = 2$				
Present 3D ($M = 1, N = 3$)	0.05613	0.03607	0.03241	0.02570
Present 3D ($M = 10, N = 3$)	0.05652	0.03930	0.04998	0.07822
Present 3D ($M = 50, N = 3$)	0.05652	0.03929	0.04996	0.07821
Present 3D ($M = 100, N = 3$)	0.05652	0.03929	0.04996	0.07821
3D [8, 9]	0.05652	0.03929	0.04996	0.07821
$b/R_\alpha = 1$				
Present 3D ($M = 1, N = 3$)	0.08833	0.07237	0.05717	0.01965
Present 3D ($M = 10, N = 3$)	0.09403	0.08546	0.09094	0.11205
Present 3D ($M = 50, N = 3$)	0.09402	0.08545	0.09093	0.11205
Present 3D ($M = 100, N = 3$)	0.09402	0.08545	0.09093	0.11205
3D [8, 9]	0.09402	0.08545	0.09093	0.11205
$b/R_\alpha = 0.5$				
Present 3D ($M = 1, N = 3$)	0.36506	0.37946	0.40165	0.43001
Present 3D ($M = 10, N = 3$)	0.18895	0.19467	0.20616	0.22449
Present 3D ($M = 50, N = 3$)	0.18894	0.19467	0.20616	0.22450
Present 3D ($M = 100, N = 3$)	0.18894	0.19467	0.20616	0.22450
3D [8, 9]	0.18894	0.19467	0.20616	0.22450
$b/R_\alpha = 0.25$				
Present 3D ($M = 1, N = 3$)	0.72230	0.72911	0.74032	0.75570
Present 3D ($M = 10, N = 3$)	0.50335	0.50934	0.51931	0.53321
Present 3D ($M = 50, N = 3$)	0.50338	0.50937	0.51934	0.53325
Present 3D ($M = 100, N = 3$)	0.50338	0.50937	0.51934	0.53325
3D [8, 9]	0.50338	0.50937	0.51934	0.53325

where $\partial \mathbf{U} / \partial \bar{z} = \mathbf{U}'$ and $\mathbf{U} = [U \ V \ W \ U' \ V' \ W']$. Equation (9) can be written as

$$\mathbf{D}\mathbf{U}' = \mathbf{A}\mathbf{U}, \quad (10)$$

$$\mathbf{U}' = \mathbf{D}^{-1}\mathbf{A}\mathbf{U}, \quad (11)$$

$$\mathbf{U}' = \mathbf{A}^*\mathbf{U}, \quad (12)$$

with $\mathbf{A}^* = \mathbf{D}^{-1}\mathbf{A}$.

In the case of plate geometry coefficients $A_3, A_4, A_9, A_{10}, A_{13}, A_{14}$, and A_{18} are zero because the radii of curvature R_α and R_β are infinite. The other coefficients $A_1, A_2, A_5, A_6, A_7, A_8, A_{11}, A_{12}, A_{15}, A_{16}, A_{17}$, and A_{19} are constant because parametric coefficients $H_\alpha = H_\beta = 1$ and they do not depend on the thickness coordinate \bar{z} ; therefore matrices \mathbf{D} , \mathbf{A} , and \mathbf{A}^* are constant in the plate case. The solution of (12) can be written as [51, 52]

$$\mathbf{U}(\bar{z}) = \exp(\mathbf{A}^*\bar{z})\mathbf{U}(0) \quad \text{with } \bar{z} \in [0, h], \quad (13)$$

where \bar{z} is the thickness coordinate referred to the bottom of the structure that has values from 0 at the bottom to h at the

top (see Figure 2). The exponential matrix is calculated with $\bar{z} = h$ as

$$\begin{aligned} \mathbf{A}^{**} = \exp(\mathbf{A}^*h) &= \mathbf{I} + \mathbf{A}^*h + \frac{\mathbf{A}^{*2}}{2!}h^2 \\ &+ \frac{\mathbf{A}^{*3}}{3!}h^3 + \dots + \frac{\mathbf{A}^{*N}}{N!}h^N, \end{aligned} \quad (14)$$

where \mathbf{I} is the 6×6 identity matrix. This expansion has a fast convergence as indicated in [53] and it is not time consuming from the computational point of view. In the case of one-layered plate there is not a necessity to impose the continuity conditions at the interfaces for displacements and transverse shear/normal stresses; for this reason the transfer matrix \mathbf{T} is equal to the identity matrix \mathbf{I} :

$$\mathbf{T} = \mathbf{I}, \quad (15)$$

Equation (15) allows the calculation of the matrix \mathbf{H} as

$$\mathbf{H} = \mathbf{A}^{**}\mathbf{I} = \mathbf{A}^{**}; \quad (16)$$

TABLE 4: Assessment 3. Simply supported orthotropic cylinder with thickness ratio $h/R_\alpha = 0.3$ and orthotropic angle θ equal to 0° or 90° . First three exact natural circular frequencies in no-dimensional form $\bar{\omega} = \omega(R/\pi)\sqrt{\rho/G_{12}}$ for imposed half-wave numbers $m = 0$ and $n = 1$. Comparison between present three-dimensional analysis and three-dimensional analysis by Soldatos and Ye [7]. M is the number of fictitious layers and N is the order of expansion for the exponential matrix.

	Mode					
	I	II	III	I	II	III
	$(m = 0, n = 1), \theta = 0^\circ$			$(m = 0, n = 1), \theta = 90^\circ$		
Present 3D ($M = 50, N = 3$)	1.02407	2.55671	4.66395	0.20406	0.49393	0.91564
Present 3D ($M = 60, N = 3$)	0.32274	0.41827	3.29830	0.30493	0.81346	7.51166
Present 3D ($M = 80, N = 3$)	0.99999	1.29840	1.85418	0.95308	1.00000	3.04558
Present 3D ($M = 90, N = 3$)	1.00000	1.29714	2.48260	0.95308	1.00000	3.25249
Present 3D ($M = 100, N = 3$)	1.00000	1.29714	2.48260	0.95308	1.00000	3.25249
3D [7]	1.00000	1.29730	2.48260	0.95307	1.00000	3.25250

TABLE 5: Benchmark 1. Simply supported isotropic square plate. First three vibration modes in term of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 1$ fictitious layers and $N = 18$ order of expansion for the exponential matrix.

a/h	100	50	10	5
	$(m = 0, n = 1)$			
I mode	2.9861	2.9846	2.9360	2.8012
II mode	194.83	97.417	19.483	9.7417
III mode	329.33	164.66	32.908	16.415
	$(m = 1, n = 0)$			
I mode	2.9861	2.9846	2.9360	2.8012
II mode	194.83	97.417	19.483	9.7417
III mode	329.33	164.66	32.908	16.415
	$(m = 1, n = 1)$			
I mode	5.9713	5.9650	5.7769	5.3036
II mode	275.54	137.77	27.554	13.777
III mode	465.73	232.86	46.502	23.136
	$(m = 1, n = 2)$			
I mode	14.920	14.881	13.805	11.645
II mode	435.66	217.83	43.566	21.783
III mode	736.37	368.14	73.350	36.157
	$(m = 2, n = 1)$			
I mode	14.920	14.881	13.805	11.645
II mode	435.66	217.83	43.566	21.783
III mode	736.37	368.14	73.350	36.157
	$(m = 2, n = 2)$			
I mode	23.860	23.761	21.214	16.882
II mode	551.07	275.54	55.107	27.554
III mode	931.42	465.63	92.545	45.071

this last matrix permits linking vector \mathbf{U} calculated at the top of the plate ($\bar{z} = h$) with vector \mathbf{U} calculated at the bottom of the plate ($\bar{z} = 0$):

$$\mathbf{U}(h) = \mathbf{H}\mathbf{U}(0). \quad (17)$$

In the case of shell geometry matrices \mathbf{D} , \mathbf{A} , and \mathbf{A}^* are not constant because of parametric coefficients $H_\alpha = (1 +$

$z/R_\alpha) = (1 + (\bar{z} - h/2)/R_\alpha)$ and $H_\beta = (1 + z/R_\beta) = (1 + (\bar{z} - h/2)/R_\beta)$ that depend on z or \bar{z} coordinate (see Figure 2). A first method could be the use of hypothesis $z/R_\alpha = z/R_\beta = 0$ (it is valid only for very thin shells) that means $H_\alpha = H_\beta = 1$. In this case the solution is the same already seen for the plate because matrices \mathbf{D} , \mathbf{A} , and \mathbf{A}^* are constant. This method is not used in the benchmarks proposed in this paper because it is an approximation that is valid only for very thin shells, and it does not consider the exact geometry of the structure. However, some comparisons between the method using exact geometry and the method using $H_\alpha = H_\beta = 1$ will be given at the end of the section for results. The second method (used in this paper) is the introduction of several j fictitious layers where H_α and H_β can exactly be calculated with the \bar{z} thickness coordinate of the middle of the fictitious layers; in this way (12), (13), and (14) are valid for each j layer because \mathbf{A}_j^* is here constant. The exponential matrix \mathbf{A}_j^{**} is calculated in each fictitious j layer with the thickness h_j of the layer. The results proposed in this paper for the benchmarks are given with a number of fictitious layers $j = M = 100$; this value guarantees the exact convergence to the correct geometry of shells for each case investigated as demonstrated in the section for the preliminary results. In the case of M layers for shell geometry, it is necessary to calculate $M - 1$ transfer matrices $\mathbf{T}_{j-1,j}$ by using for each interface the following conditions:

$$\begin{aligned} u_j^b &= u_{j-1}^t, & v_j^b &= v_{j-1}^t, & w_j^b &= w_{j-1}^t, \\ \sigma_{zzj}^b &= \sigma_{zzj-1}^t, & \sigma_{\alpha zj}^b &= \sigma_{\alpha zj-1}^t, & \sigma_{\beta zj}^b &= \sigma_{\beta zj-1}^t; \end{aligned} \quad (18)$$

that means each displacement and transverse stress component at the top (t) of the $j-1$ layer is equal to displacement and transverse stress component at the bottom (b) of the j layer. The calculated $\mathbf{T}_{j-1,j}$ matrices allow linking \mathbf{U} at the bottom of the j layer with \mathbf{U} at the top of the $j-1$ layer:

$$\mathbf{U}_j(0) = \mathbf{T}_{j-1,j} \mathbf{U}_{j-1}(h_{j-1}). \quad (19)$$

\mathbf{U} at the top of the j layer is linked with \mathbf{U} at the bottom of the same layer j by means of the exponential matrix \mathbf{A}_j^{**} :

$$\mathbf{U}_j(h_j) = \mathbf{A}_j^{**} \mathbf{U}_j(0). \quad (20)$$

TABLE 6: Benchmark 2. Simply supported orthotropic square plate. First three vibration modes in term of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 1$ fictitious layers and $N = 18$ order of expansion for the exponential matrix.

a/h	100	50	10	5	100	50	10	5
		$(m = 0, n = 1), \theta = 0^\circ$				$(m = 0, n = 1), \theta = 90^\circ$		
I mode	2.8553	2.8537	2.8058	2.6731	10.007	9.9712	9.0075	7.2059
II mode	227.77	113.88	22.777	11.388	227.77	113.88	22.777	11.388
III mode	314.89	157.44	31.457	15.679	1104.7	552.35	110.41	46.861
		$(m = 1, n = 0), \theta = 0^\circ$				$(m = 1, n = 0), \theta = 90^\circ$		
I mode	10.007	9.9712	9.0075	7.2059	2.8553	2.8537	2.8058	2.6731
II mode	227.77	113.88	22.777	11.388	227.77	113.88	22.777	11.388
III mode	1104.7	552.35	110.41	46.861	314.89	157.44	31.457	15.679
		$(m = 1, n = 1), \theta = 0^\circ$				$(m = 1, n = 1), \theta = 90^\circ$		
I mode	11.367	11.325	10.226	8.2183	11.367	11.325	10.226	8.2183
II mode	382.03	191.01	38.169	19.032	382.03	191.01	38.169	19.032
III mode	1130.2	565.09	112.94	49.636	1130.2	565.09	112.94	49.636
		$(m = 1, n = 2), \theta = 0^\circ$				$(m = 1, n = 2), \theta = 90^\circ$		
I mode	17.719	17.648	15.817	12.656	41.011	40.436	29.614	19.220
II mode	652.40	326.17	65.004	32.100	549.29	274.64	54.882	27.380
III mode	1204.5	602.22	120.31	56.667	2222.2	1111.0	190.33	53.344
		$(m = 2, n = 1), \theta = 0^\circ$				$(m = 2, n = 1), \theta = 90^\circ$		
I mode	41.011	40.436	29.614	19.220	17.719	17.648	15.817	12.656
II mode	549.29	274.64	54.882	27.380	652.40	326.17	65.004	32.100
III mode	2222.2	1111.0	190.33	53.344	1204.5	602.22	120.31	56.667
		$(m = 2, n = 2), \theta = 0^\circ$				$(m = 2, n = 2), \theta = 90^\circ$		
I mode	45.302	44.664	32.873	21.768	45.302	44.664	32.873	21.768
II mode	764.05	381.98	76.128	37.625	764.05	381.98	76.128	37.625
III mode	2260.4	1130.1	198.54	59.836	2260.4	1130.1	198.54	59.836

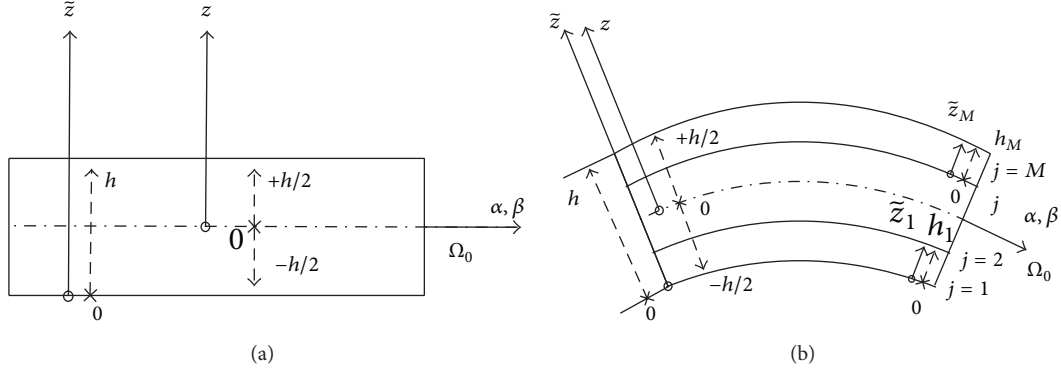


FIGURE 2: Thickness coordinates for plates and shells.

Equation (19) can recursively be introduced in (20) for the $M - 1$ interfaces to obtain

$$\mathbf{U}_M(h_M) = \mathbf{A}_M^{**} \mathbf{T}_{M-1,M} \mathbf{A}_{M-1}^{**} \mathbf{T}_{M-2,M-1} \cdots \mathbf{A}_2^{**} \mathbf{T}_{1,2} \mathbf{A}_1^{**} \mathbf{U}_1(0), \quad (21)$$

by defining the matrix \mathbf{H}_m for the multi-fictitious-layer shells, it is possible to write (21) in analogy with (17):

$$\mathbf{U}_M(h_M) = \mathbf{H}_m \mathbf{U}_1(0). \quad (22)$$

It links \mathbf{U} calculated at the top of the last M fictitious layer with \mathbf{U} calculated at the bottom of the first layer. Matrices \mathbf{A}_j^{**} are constant because they are evaluated with R_α , R_β , $\bar{\alpha}$, and $\bar{\beta}$ calculated in the midsurface Ω_0 of the shell and with H_α and H_β calculated in the middle of each j layer. Matrices $\mathbf{T}_{j-1,j}$ are constant because they are evaluated with R_α , R_β , $\bar{\alpha}$, and $\bar{\beta}$ calculated in the midsurface Ω_0 of the shell and with H_α and H_β calculated at each fictitious interface. Equation (17) is valid for one-layered plates and (22) is valid for one-layered

TABLE 7: Benchmark 3. Simply supported isotropic rectangular plate. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 1$ fictitious layers and $N = 18$ order of expansion for the exponential matrix.

a/h	100	50	10	5
$(m = 0, n = 1)$				
I mode	0.3318	0.3318	0.3312	0.3293
II mode	64.944	32.472	6.4944	3.2472
III mode	109.78	54.888	10.977	5.4870
$(m = 1, n = 0)$				
I mode	2.9861	2.9846	2.9360	2.8012
II mode	194.83	97.417	19.483	9.7417
III mode	329.33	164.66	32.908	16.415
$(m = 1, n = 1)$				
I mode	3.3179	3.3159	3.2562	3.0923
II mode	205.37	102.69	20.537	10.269
III mode	347.14	173.56	34.685	17.297
$(m = 1, n = 2)$				
I mode	4.3130	4.3097	4.2099	3.9450
II mode	234.16	117.08	23.416	11.708
III mode	395.80	197.89	39.537	19.699
$(m = 2, n = 1)$				
I mode	12.270	12.243	11.497	9.9094
II mode	395.04	197.52	39.504	19.752
III mode	667.72	333.83	66.559	32.908
$(m = 2, n = 2)$				
I mode	13.264	13.233	12.369	10.572
II mode	410.74	205.37	41.074	20.537
III mode	694.26	347.10	69.186	34.170

shells divided in j fictitious layers to correctly consider the geometry.

The structures are simply supported and free stresses at the top and at the bottom; this feature means:

$$\sigma_{zz} = \sigma_{\alpha z} = \sigma_{\beta z} = 0, \quad \text{for } z = \frac{-h}{2}, \frac{+h}{2} \text{ or } \tilde{z} = 0, h, \quad (23)$$

$$w = v = 0, \quad \sigma_{\alpha\alpha} = 0, \quad \text{for } \alpha = 0, a, \quad (24)$$

$$w = u = 0, \quad \sigma_{\beta\beta} = 0, \quad \text{for } \beta = 0, b. \quad (25)$$

Equation (23) imposed at the top of the structure has R_α , R_β , $\bar{\alpha}$, and $\bar{\beta}$ calculated in the midsurface Ω_0 of the shell, and H_α^t and H_β^t are calculated at the top of the shell for $\tilde{z} = h$:

$$\begin{aligned} \sigma_{zz}^t = & -\bar{\alpha} \frac{C_{13}}{H_\alpha^t} U^t + \frac{C_{13}}{H_\alpha^t R_\alpha} W^t - \bar{\beta} \frac{C_{23}}{H_\beta^t} V^t \\ & + \frac{C_{23}}{H_\beta^t R_\beta} W^t + C_{33} W_{,z}^t = 0, \end{aligned}$$

$$\sigma_{\beta z}^t = \bar{\beta} \frac{C_{44}}{H_\beta^t} W^t + C_{44} V_{,z}^t - \frac{C_{44}}{H_\beta^t R_\beta} V^t = 0, \quad (26)$$

$$\sigma_{\alpha z}^t = \bar{\alpha} \frac{C_{55}}{H_\alpha^t} W^t + C_{55} U_{,z}^t - \frac{C_{55}}{H_\alpha^t R_\alpha} U^t = 0.$$

Equation (23) imposed at the bottom of the structure has R_α , R_β , $\bar{\alpha}$, and $\bar{\beta}$ calculated in the midsurface Ω_0 of the shell, and H_α^b and H_β^b are calculated at the bottom of the shell for $\tilde{z} = 0$:

$$\begin{aligned} \sigma_{zz}^b = & -\bar{\alpha} \frac{C_{13}}{H_\alpha^b} U^b + \frac{C_{13}}{H_\alpha^b R_\alpha} W^b - \bar{\beta} \frac{C_{23}}{H_\beta^b} V^b \\ & + \frac{C_{23}}{H_\beta^b R_\beta} W^b + C_{33} W_{,z}^b = 0, \end{aligned} \quad (27)$$

$$\sigma_{\beta z}^b = \bar{\beta} \frac{C_{44}}{H_\beta^b} W^b + C_{44} V_{,z}^b - \frac{C_{44}}{H_\beta^b R_\beta} V^b = 0,$$

$$\sigma_{\alpha z}^b = \bar{\alpha} \frac{C_{55}}{H_\alpha^b} W^b + C_{55} U_{,z}^b - \frac{C_{55}}{H_\alpha^b R_\alpha} U^b = 0.$$

Equations (26) in matrix form are $(U_M(h_M))$ means U calculated at the top of the shell)

$$\begin{bmatrix} -\bar{\alpha} \frac{C_{13}}{H_\alpha^t} & -\bar{\beta} \frac{C_{23}}{H_\beta^t} & \left(\frac{C_{13}}{H_\alpha^t R_\alpha} + \frac{C_{23}}{H_\beta^t R_\beta} \right) & 0 & 0 & C_{33} \\ 0 & -\frac{C_{44}}{H_\beta^t R_\beta} & \bar{\beta} \frac{C_{44}}{H_\beta^t} & 0 & C_{44} & 0 \\ -\frac{C_{55}}{H_\alpha^t R_\alpha} & 0 & \bar{\alpha} \frac{C_{55}}{H_\alpha^t} & C_{55} & 0 & 0 \end{bmatrix}$$

$$\times \begin{bmatrix} U_M(h_M) \\ V_M(h_M) \\ W_M(h_M) \\ U'_M(h_M) \\ V'_M(h_M) \\ W'_M(h_M) \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}.$$

(28)

TABLE 8: Benchmark 4. Simply supported orthotropic rectangular plate. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 1$ fictitious layers and $N = 18$ order of expansion for the exponential matrix.

a/h	100	50	10	5	100	50	10	5
		$(m = 0, n = 1), \theta = 0^\circ$				$(m = 0, n = 1), \theta = 90^\circ$		
I mode	0.3173	0.3173	0.3167	0.3148	1.1130	1.1126	1.0987	1.0586
II mode	75.922	37.961	7.5922	3.7961	75.922	37.961	7.5922	3.7961
III mode	104.98	52.483	10.495	5.2459	368.24	184.12	36.822	18.408
		$(m = 1, n = 0), \theta = 0^\circ$				$(m = 1, n = 0), \theta = 90^\circ$		
I mode	10.007	9.9712	9.0075	7.2059	2.8553	2.8537	2.8058	2.6731
II mode	227.77	113.88	22.777	11.388	227.77	113.68	22.777	11.388
III mode	1104.7	552.35	110.41	46.861	314.89	157.44	31.457	15.679
		$(m = 1, n = 1), \theta = 0^\circ$				$(m = 1, n = 1), \theta = 90^\circ$		
I mode	10.127	10.091	9.1126	7.2879	3.4236	3.4213	3.3511	3.1611
II mode	249.70	124.85	24.967	12.480	312.77	156.38	31.251	15.586
III mode	1107.6	553.78	110.69	47.184	441.11	220.55	44.099	22.031
		$(m = 1, n = 2), \theta = 0^\circ$				$(m = 1, n = 2), \theta = 90^\circ$		
I mode	10.538	10.500	9.4774	7.5865	6.1047	6.0947	5.8032	5.1319
II mode	306.16	153.08	30.604	15.284	341.88	170.94	34.157	17.031
III mode	1116.1	558.04	111.54	48.130	774.35	387.17	77.407	38.652
		$(m = 2, n = 1), \theta = 0^\circ$				$(m = 2, n = 1), \theta = 90^\circ$		
I mode	40.001	39.445	28.902	18.694	11.868	11.841	11.093	9.5179
II mode	466.89	233.44	46.684	23.337	560.57	280.28	56.021	27.943
III mode	2210.8	1105.3	187.77	51.133	656.59	328.31	65.435	32.318
		$(m = 2, n = 2), \theta = 0^\circ$				$(m = 2, n = 2), \theta = 90^\circ$		
I mode	40.364	39.801	29.152	18.872	13.685	13.649	12.644	10.634
II mode	499.39	249.69	49.921	24.938	625.52	312.73	62.343	30.811
III mode	2215.1	1107.5	188.74	51.982	882.21	441.09	88.126	43.879

Equation (27) in matrix form is ($\mathbf{U}_1(0)$ means \mathbf{U} calculated at the bottom of the shell)

$$\begin{bmatrix} -\bar{\alpha} \frac{C_{13}}{H_\alpha^b} & -\bar{\beta} \frac{C_{23}}{H_\beta^b} \left(\frac{C_{13}}{H_\alpha^b R_\alpha} + \frac{C_{23}}{H_\beta^b R_\beta} \right) & 0 & 0 & C_{33} \\ 0 & -\frac{C_{44}}{H_\beta^b R_\beta} & \bar{\beta} \frac{C_{44}}{H_\beta^b} & 0 & C_{44} & 0 \\ -\frac{C_{55}}{H_\alpha^b R_\alpha} & 0 & \bar{\alpha} \frac{C_{55}}{H_\alpha^b} & C_{55} & 0 & 0 \end{bmatrix} \times \begin{bmatrix} U_1(0) \\ V_1(0) \\ W_1(0) \\ U_1'(0) \\ V_1'(0) \\ W_1'(0) \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}. \quad (29)$$

Equations (28)-(29) in compact form to express the free stress state at the top and bottom of the shell are

$$\mathbf{B}_M(h_M) \mathbf{U}_M(h_M) = 0, \quad (30)$$

$$\mathbf{B}_1(0) \mathbf{U}_1(0) = 0; \quad (31)$$

in the case of plate, there are no j fictitious layers:

$$\begin{aligned} \mathbf{B}(h) \mathbf{U}(h) &= 0, \\ \mathbf{B}(0) \mathbf{U}(0) &= 0. \end{aligned} \quad (32)$$

Equation (22) can be substituted in (30):

$$\begin{aligned} \mathbf{B}_M(h_M) \mathbf{H}_m \mathbf{U}_1(0) &= 0; \\ \mathbf{B}_1(0) \mathbf{U}_1(0) &= 0, \end{aligned} \quad (33)$$

in this way (33) are grouped in the following system:

$$\begin{bmatrix} \mathbf{B}_M(h_M) \mathbf{H}_m \\ \mathbf{B}_1(0) \end{bmatrix} [\mathbf{U}_1(0)] = [0], \quad (34)$$

and introducing the (6×6) \mathbf{E} matrix, (34) becomes:

$$[\mathbf{E}] [\mathbf{U}_1(0)] = [0]. \quad (35)$$

TABLE 9: Benchmark 5. Simply supported isotropic cylindrical shell panel. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5
$(m = 0, n = 1)$				
I mode	1067.1	106.71	10.683	5.3415
II mode	1068.3	106.83	10.690	5.3727
III mode	1855.7	185.57	18.557	9.2797
$(m = 1, n = 0)$				
I mode	2.5478	2.5184	2.4853	2.3948
II mode	2040.3	204.03	20.411	10.216
III mode	3635.3	363.51	36.229	17.917
$(m = 1, n = 1)$				
I mode	222.72	22.503	3.8758	3.2156
II mode	2364.5	236.45	23.656	11.842
III mode	4017.4	401.73	40.060	19.846
$(m = 1, n = 2)$				
I mode	552.89	55.574	7.7851	5.7590
II mode	3031.5	303.16	30.328	15.181
III mode	5048.7	504.86	50.367	24.983
$(m = 2, n = 1)$				
I mode	70.382	14.052	11.533	10.021
II mode	4230.8	423.08	42.325	21.174
III mode	7214.2	721.38	71.724	35.120
$(m = 2, n = 2)$				
I mode	232.90	27.451	13.793	11.643
II mode	4639.2	463.92	46.414	23.226
III mode	7847.2	784.68	78.020	38.182

Equation (35) is also valid for plate case where the fictitious layers are not introduced and $\mathbf{U}(0) = \mathbf{U}_1(0)$, $\mathbf{H} = \mathbf{H}_m$, and $\mathbf{B} = \mathbf{B}_M(h_M) = \mathbf{B}_1(0)$. The solution is implemented in a Matlab code where only the spherical shell method is considered; it automatically degenerates into cylindrical open/closed shell and plate methods.

The free vibration problem means finding the nontrivial solution of $\mathbf{U}_1(0)$ in (35); this means imposing the determinant of matrix \mathbf{E} equal to zero:

$$\det[\mathbf{E}] = 0. \quad (36)$$

Equation (36) needs to find the roots of a higher order polynomial in $\lambda = \omega^2$. For each pair of half-wave numbers (m, n) a certain number of circular frequencies are obtained depending on the order N chosen for the exponential matrices \mathbf{A}_j^{**} and the M number of fictitious layers.

3.2. Vibration Modes. A certain number of circular frequencies ω_l are found when half-wave numbers m and n are imposed in the structures. For each frequency ω_l , it is possible to find the vibration mode through the thickness in terms of the three displacement components. If the frequency ω_l is substituted in (6×6) matrix \mathbf{E} , this last matrix has six

eigenvalues. We are interested to the null space of matrix \mathbf{E} that means finding the (6×1) eigenvector related to the minimum of the six eigenvalues proposed. This null space is, for the chosen frequency ω_l , the vector \mathbf{U} calculated at the bottom:

$$\mathbf{U}_{1\omega_l}(0) = [U_1(0) \ V_1(0) \ W_1(0) \ U'_1(0) \ V'_1(0) \ W'_1(0)]_{\omega_l}^T. \quad (37)$$

T means the transpose of the vector and the subscript ω_l means that the null space is calculated for the circular frequency ω_l .

It is possible to find $\mathbf{U}_{j\omega_l}(\bar{z}_j)$ (with the three displacement components $U_{j\omega_l}(\bar{z}_j)$, $V_{j\omega_l}(\bar{z}_j)$, and $W_{j\omega_l}(\bar{z}_j)$ through the thickness) for each j layer by using (19)–(22); the thickness coordinate \bar{z} can assume all the values from the bottom to the top of the structure. For the plate case the procedure is simpler because there are no j fictitious layers and the M number equals 1.

4. Results

The three-dimensional exact solution proposed has been validated by means of a comparison with three assessments given in the literature for a one-layered isotropic plate [22, 34], one-layered isotropic cylinder [8, 9], and one-layered orthotropic cylinder [7]. After this preliminary validation and the appropriate convergence study, the method has been used to investigate the free vibration analysis of one-layered isotropic and orthotropic square and rectangular plates, cylindrical shell panels, cylinders, and spherical shell panels (see Figure 3). The effects of the curvature terms in H_α and H_β parametric coefficients have also been investigated.

4.1. Preliminary Assessments. The assessment proposed in Vel and Batra [34] compares their 3D solution with the 3D solution by Srinivas et al. [22]. The square plate is simply supported embedding one aluminium alloy layer with Young modulus $E = 70$ GPa, Poisson ratio $\nu = 0.3$, and mass density $\rho = 2702$ kg/m³. The thickness value is $h = 1$. Two different thickness ratios are considered, that is, $a/h = 10$ and $a/h = \sqrt{10}$. The first six circular frequencies in no-dimensional form $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ are given in Table 1 for imposed half-wave numbers $m = n = 1$. The 3D exact solution proposed in this paper coincides with those proposed by Vel and Batra [34] and by Srinivas et al. [22] for both thickness ratios and for each vibration mode (from the first to the sixth one). A convergence study has been proposed where the M number of fictitious layers and the N order of expansion for the exponential matrix have conveniently been chosen. For the plate case one fictitious layer is used because there is no effect of parametric coefficients H_α and H_β . For $M = 1$, an order of expansion $N = 18$ for the exponential matrix always gives the correct solution for each vibration mode and thickness ratio a/h . The calculation of the exponential matrix is not time consuming. An alternative method could be the use of M fictitious layers and a lower order of expansion $N = 3$ for the exponential matrix. In this last case, $M = 100$ fictitious layers always give the correct solution for each

TABLE 10: Benchmark 6. Simply supported orthotropic cylindrical shell panel. First three vibration modes in term of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5	1000	100	10	5
		$(m = 0, n = 1), \theta = 0^\circ$				$(m = 0, n = 1), \theta = 90^\circ$		
I mode	1248.9	124.89	12.489	6.2444	1096.5	109.67	11.281	6.0108
II mode	1721.6	172.16	17.230	8.6360	1248.9	124.89	12.489	6.2444
III mode	3858.5	385.85	38.590	19.293	6057.8	605.78	60.570	30.272
		$(m = 1, n = 0), \theta = 0^\circ$				$(m = 1, n = 0), \theta = 90^\circ$		
I mode	8.4450	8.4392	7.6484	6.1872	2.3272	2.4079	2.3754	2.2863
II mode	2385.2	238.52	23.861	11.941	2385.2	238.52	23.861	11.943
III mode	12194	1219.4	121.48	51.358	3476.0	347.58	34.633	17.114
		$(m = 1, n = 1), \theta = 0^\circ$				$(m = 1, n = 1), \theta = 90^\circ$		
I mode	230.37	24.568	8.0953	6.3709	352.48	35.490	5.3604	4.1227
II mode	2939.9	294.00	29.454	14.782	3588.6	358.85	35.788	17.735
III mode	12257	1225.7	122.10	52.106	6559.0	655.90	65.574	32.756
		$(m = 1, n = 2), \theta = 0^\circ$				$(m = 1, n = 2), \theta = 90^\circ$		
I mode	641.70	64.887	10.877	6.8670	644.27	65.496	12.428	9.0851
II mode	4155.7	415.58	41.604	20.824	4164.3	416.43	41.609	20.709
III mode	12445	1244.4	123.94	54.240	12370	1237.0	123.62	54.652
		$(m = 2, n = 1), \theta = 0^\circ$				$(m = 2, n = 1), \theta = 90^\circ$		
I mode	79.744	39.112	28.352	18.587	160.90	20.150	11.545	9.9019
II mode	5062.8	506.29	50.681	25.371	6476.9	647.66	64.461	31.673
III mode	23492	2349.1	205.91	56.149	7983.5	798.33	79.684	39.571
		$(m = 2, n = 2), \theta = 0^\circ$				$(m = 2, n = 2), \theta = 90^\circ$		
I mode	243.62	46.084	29.034	19.077	364.43	40.611	16.527	13.080
II mode	5849.5	584.96	58.566	29.293	6951.1	695.08	69.132	33.884
III mode	23598	2359.6	208.24	58.169	13115	1311.5	131.01	62.003

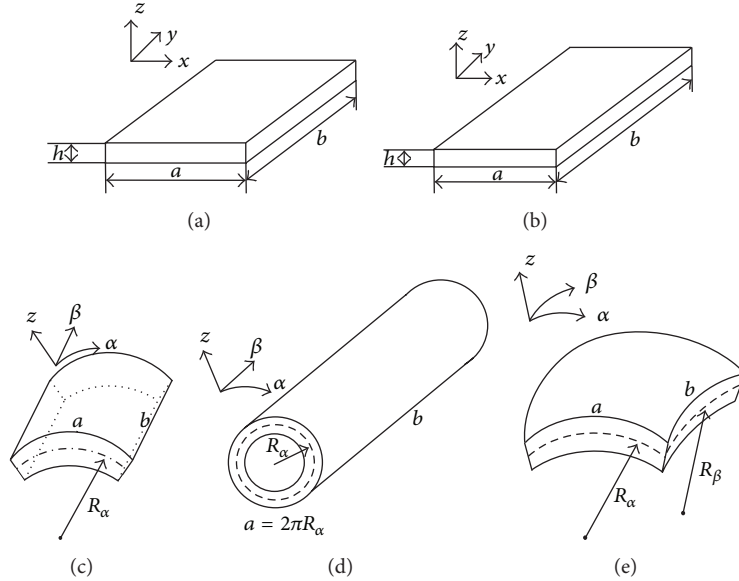


FIGURE 3: Geometries considered for the assessments and benchmarks: (a) square plate, (b) rectangular plate, (c) cylindrical shell panel, (d) cylinder, and (e) spherical shell panel.

TABLE 11: Benchmark 7. Simply supported isotropic cylinder. First three vibration modes in term of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5
$(m = 0, n = 1)$				
I mode	7691.7	769.17	76.917	38.459
II mode	12342	1234.2	123.42	61.710
III mode	41586	4158.6	416.30	208.81
$(m = 2, n = 1)$				
I mode	2445.9	244.59	24.492	12.295
II mode	27270	2727.0	272.81	136.56
III mode	59189	5918.8	591.08	294.28
$(m = 2, n = 2)$				
I mode	7712.1	771.23	77.252	38.820
II mode	33375	3337.5	333.85	167.09
III mode	61353	6135.3	612.74	305.15
$(m = 2, n = 3)$				
I mode	13536	1353.7	135.69	68.323
II mode	40081	4008.1	400.89	200.56
III mode	65490	6548.9	654.17	325.96

vibration mode and thickness ratio. The use of $M = 100$ layers is not time consuming because the E matrix has always 6×6 dimension for any value of M layers, even if the proposed method uses a layerwise approach. Solutions with $M = 1$ and $N = 18$ or with $M = 100$ and $N = 3$ are coincident for the plate case and their time solution needs a few seconds. In conclusion, the three-dimensional exact solution presented here is correct for the free vibration analysis of one-layered plates even when the structures are thick and higher order vibration modes are considered.

The assessment proposed by Khalili et al. [11] compared their 2D closed form solutions for isotropic circular cylindrical shells with the three-dimensional analysis by Armenakas et al. [8] adapted from Bhimaraddi [9]. The cylinder is isotropic and one-layered with Poisson ratio $\nu = 0.3$. The structure is simply supported and it has radius of curvature $R_\alpha = 1$ and infinite radius of curvature R_β in β direction. The circular dimension is $a = 2\pi R_\alpha$; the thickness h is 0.12 and 0.18; that means thickness ratios are $h/R_\alpha = 0.12$ and $h/R_\alpha = 0.18$, respectively. Several ratios cylinder length/radius of curvature (b/R_α) are investigated in the case of $b = 2, 1, 0.5, 0.25$. The fundamental circular frequency in no-dimensional form $\bar{\omega} = \omega(h/\pi)\sqrt{\rho/G}$ is given in Tables 2 and 3 (thickness ratio h/R_α equals 0.12 and 0.18, resp.) for axial half-wave number $n = 1$ and several circular half-wave numbers $m = 2, 4, 6, 8$. The three-dimensional exact solution proposed here is coincident with the 3D analysis by Armenakas et al. [8] for each half-wave number, thickness ratio, and cylinder length proposed. For the case of shell, the use of fictitious layers is necessary to approximate the parametric coefficients H_α and H_β . The convergence study is shown for a value $N = 3$ for the expansion of the exponential matrix. $M = 100$ fictitious layers always give the correct

solution for each thickness ratio h/R_α (see Tables 2 and 3), length b of the cylinder, and half-wave numbers m in circumferential direction. The present method is not time consuming; $M = 100$ layers solution gives the correct results in a few seconds. In conclusion, the solution proposed can be considered as validated for one-layered isotropic shells.

The assessment proposed by Soldatos and Ye [7] gives three-dimensional results for multilayered orthotropic cylinders $(-\theta/+ \theta)_j$. A comparison with the present method is made for the cases of orthotropic angle θ equal to 0° or 90° (they allow the closed form solution and it also means one-layered structure). The cylinder is simply supported with a great degree of orthotropy ($E_1/E_2 = E_1/E_3 = 40$, $G_{12}/E_2 = G_{13}/E_2 = 0.6$, $G_{23}/E_2 = 0.5$, $\nu_{12} = \nu_{13} = \nu_{23} = 0.25$, and homogeneous mass density ρ). The length b is 10 and the radius of curvature R_α equals 10. The circular dimension is $a = 2\pi R_\alpha$. The cylinder is very thick ($h = 3$); that means thickness ratio is $h/R_\alpha = 0.3$. In Table 4 the first three circular frequencies in no-dimensional form $\bar{\omega} = \omega(R/\pi)\sqrt{\rho/G_{12}}$ are given for half-wave numbers $m = 0$ and $n = 1$, and orthotropic angle θ equals 0° or 90° . The 3D solution by Soldatos and Ye [7] is correctly obtained for each mode and angle θ by using $M = 100$ fictitious layers and $N = 3$ order of expansion. ($M = 100$, $N = 3$) solution is not heavy from the computational point of view because the E matrix has always 6×6 dimension even if the method uses a layerwise approach. In conclusion, the solution is also validated for orthotropic single-layered thick shells.

All the new benchmarks proposed in Section 4.2 will use ($M = 1$, $N = 18$) solution for plate cases and ($M = 100$, $N = 3$) solution for shell cases. Both solutions are very convenient from the computational point of view (they need only a few seconds) and they allow the correct analysis for each thick and thin isotropic/orthotropic plate and shell.

4.2. Benchmarks. Ten different benchmarks are proposed to show a complete overview of the free vibration analysis of one-layered plates and shells. Five different geometries are taken into account in Figure 3. All these structures are simply supported. The square plate has dimensions $a = b = 1$ and thickness ratios $a/h = 100, 50, 10, 5$. The rectangular plate has dimensions $a = 1$ and $b = 3a$ and the thickness ratios are $a/h = 100, 50, 10, 5$. The cylindrical shell panel has a radius of curvature $R_\alpha = 10$ and an infinite radius of curvature R_β in β direction. The dimensions are $a = (\pi/3)R_\alpha$ and $b = 20$, and the thickness ratios are $R_\alpha/h = 1000, 100, 10, 5$. The cylinder has the same radii of curvature of the cylindrical shell panel, but it is closed in circumferential direction that means $a = 2\pi R_\alpha$. The other dimension is $b = 100$ and the thickness ratios are $R_\alpha/h = 1000, 100, 10, 5$. The last geometry is the spherical shell panel with radii of curvature $R_\alpha = R_\beta = 10$, dimensions $a = b = (\pi/3)R_\alpha$, and thickness ratios $R_\alpha/h = 1000, 100, 10, 5$. Each geometry includes two materials: one isotropic layer in aluminium alloy Al2024 with Young modulus $E = 73$ GPa, Poisson ratio $\nu = 0.3$, and mass density $\rho = 2800$ kg/m³ and one orthotropic layer in Gr/Ep with Young modulus components $E_1 = 132.38$ GPa and $E_2 = E_3 = 10.756$ GPa, shear modulus components

TABLE 12: Benchmark 8. Simply supported orthotropic cylinder. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5	1000	100	10	5
		$(m = 0, n = 1), \theta = 0^\circ$				$(m = 0, n = 1), \theta = 90^\circ$		
I mode	8991.9	899.19	89.919	44.960	8991.9	899.19	89.919	44.959
II mode	12402	1240.2	124.16	62.281	39097	3909.7	391.39	196.33
III mode	138826	13883	1389.6	696.66	44039	4403.9	440.45	220.32
		$(m = 2, n = 1), \theta = 0^\circ$				$(m = 2, n = 1), \theta = 90^\circ$		
I mode	2486.4	248.65	24.921	12.541	5295.1	529.53	53.084	26.738
II mode	31697	3169.8	317.14	158.81	50390	5039.0	503.83	251.80
III mode	196438	19643	1961.3	974.72	58023	5802.3	579.62	288.86
		$(m = 2, n = 2), \theta = 0^\circ$				$(m = 2, n = 2), \theta = 90^\circ$		
I mode	8071.9	807.21	80.903	40.715	11839	1184.0	119.10	60.562
II mode	38989	3898.9	390.16	195.44	56607	5660.7	565.47	281.81
III mode	196773	19677	1964.6	976.14	92402	9240.2	924.00	461.97
		$(m = 2, n = 3), \theta = 0^\circ$				$(m = 2, n = 3), \theta = 90^\circ$		
I mode	14640	1464.1	146.76	73.903	17656	1765.9	178.90	92.578
II mode	48229	4823.0	482.66	241.82	58769	5876.8	587.41	293.24
III mode	197336	19733	1970.0	978.55	134283	13428	1342.8	671.29

$G_{12} = G_{13} = 5.6537$ GPa and $G_{23} = 3.603$ GPa, Poisson ratio components $\nu_{12} = \nu_{13} = 0.24$ and $\nu_{23} = 0.49$, and mass density $\rho = 1600$ kg/m³. The benchmarks are one-layered isotropic square plate (see Table 5), one-layered orthotropic square plate (see Table 6), one-layered isotropic rectangular plate (see Table 7), one-layered orthotropic rectangular plate (see Table 8), one-layered isotropic cylindrical shell panel (see Table 9), one-layered orthotropic cylindrical shell panel (see Table 10), one-layered isotropic cylinder (see Table 11), one-layered orthotropic cylinder (see Table 12), one-layered isotropic spherical shell panel (see Table 13), and one-layered orthotropic spherical shell panel (see Table 14). The first three circular frequencies in no-dimensional form $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ are calculated in Tables 5–14 for various pairs of half-wave numbers and several thickness ratios. The vibration modes plotted in Figures 4–12 are given in terms of no-dimensional values such as $u^* = u/|u_{\max}|$, $v^* = v/|v_{\max}|$, $w^* = w/|w_{\max}|$, and $z^* = \bar{z}/h$.

Table 5 presents a square isotropic plate (benchmark 1). It shows the first three vibration modes for each possible combination of half-wave numbers (m, n) from $(0, 1)$ until to $(2, 2)$. Both thick and thin plates are analyzed. The plate is square and isotropic; therefore results for half-wave numbers $(0, 1)$ are equal to those for $(1, 0)$, and the same considerations are made for the half-wave numbers $(1, 2)$ and $(2, 1)$. The first three vibration modes through the thickness of a thick plate ($a/h = 10$) in terms of the three displacement components u^* , v^* , and w^* are given in Figure 4 for half-wave numbers $(1, 0)$, $(1, 1)$, $(2, 1)$, and $(2, 2)$. The second vibration mode is an in-plane mode for each pair of half-wave numbers because transverse displacement w^* through the thickness is zero. The first mode has a constant transverse displacement w^* through the thickness and linear in-plane displacements u^* and v^* . The transverse displacement is not perfectly constant

for higher values of half-wave number (effect of higher order modes). The in-plane displacement v^* is zero for the first case because the half-wave number $n = 0$. The third vibration mode always has linear transverse displacement w^* through the thickness for each pair of half-wave numbers. In-plane displacements u^* and v^* are constant (they are not perfectly constant for higher values of half-wave numbers; it is a typical effect of higher order modes). The in-plane displacement v^* is zero for the first case because of zero half-wave number n .

Table 6 shows a square orthotropic plate (benchmark 2) and lists the first three vibration modes for each possible combination of half-wave numbers (m, n) from $(0, 1)$ until to $(2, 2)$ and for two different fibre orientation angles θ (0° and 90°). For the angle $\theta = 0^\circ$ results for half-wave numbers $(0, 1)$ are not equal to those for $(1, 0)$, and the same considerations are made for the half-wave numbers $(1, 2)$ and $(2, 1)$. This is due to the in-plane orthotropy. The same considerations are valid for the angle $\theta = 90^\circ$. It is important to notice that frequencies for $(0, 1)$ and $\theta = 0^\circ$ are equal to those for $(1, 0)$ and $\theta = 90^\circ$ and frequencies for $(1, 0)$ and $\theta = 0^\circ$ are equal to those for $(0, 1)$ and $\theta = 90^\circ$. The same considerations remain valid for $(1, 2)$ and $(2, 1)$. These properties are valid for square plates and they are not true for rectangular plates. Figures 5 and 6 show the first three vibration modes through the thickness in terms of the three displacement components u^* , v^* , and w^* for half-wave numbers $(1, 0)$, $(1, 1)$, $(2, 1)$, and $(2, 2)$ and for a thick plate ($a/h = 10$) with fiber orientation $\theta = 0^\circ$ or $\theta = 90^\circ$. The considerations already made for the first benchmark in Figure 4 are also valid in this case. The orthotropy also generates a coupling between in-plane and out-of-plane behavior that gives displacements that are not constant or linear through the thickness even when low half-wave numbers are considered. Moreover, thickness modes are more complicated for higher half-wave numbers. For these

TABLE 13: Benchmark 9. Simply supported isotropic spherical shell panel. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5
$(m = 0, n = 1)$				
I mode	999.01	99.924	10.213	5.3986
II mode	2040.3	204.02	20.360	10.115
III mode	3785.7	378.55	37.677	18.556
$(m = 1, n = 0)$				
I mode	999.01	99.924	10.213	5.3986
II mode	2040.3	204.02	20.360	10.116
III mode	3785.7	378.55	37.677	18.556
$(m = 1, n = 1)$				
I mode	1046.3	104.76	11.564	6.9554
II mode	2885.4	288.53	28.793	14.302
III mode	5111.6	511.13	50.821	24.937
$(m = 1, n = 2)$				
I mode	1076.3	108.52	16.906	12.400
II mode	4562.2	456.21	45.524	22.598
III mode	7857.9	785.72	77.933	37.847
$(m = 2, n = 1)$				
I mode	1076.3	108.52	16.906	12.400
II mode	4562.2	456.21	45.524	22.598
III mode	7857.9	785.72	77.933	37.847
$(m = 2, n = 2)$				
I mode	1084.1	110.77	23.273	17.513
II mode	5770.8	577.07	57.581	28.565
III mode	9869.7	986.87	97.645	46.869

reasons a 2D model will have more difficulties to obtain the 3D behavior of the benchmark 2 with respect to the 3D behavior of the isotropic benchmark 1. The change of orthotropic angle θ has an important effect on the frequency values and a negligible effect on the vibration modes through the thickness.

Tables 7 and 8 show the same cases already seen in Tables 5 and 6, but a rectangular plate is analyzed in place of a square plate. This geometry gives a difference in Table 7 (benchmark 3) between modes with (0, 1) and those with (1, 0) and between modes with (1, 2) and those with (2, 1); in fact, the two in-plane dimensions a and b are not the same ($b = 3a$). This difference due to the in-plane dimensions is also important in Table 8 (benchmark 4) where frequencies for (0, 1) and $\theta = 0^\circ$ are not equal to the frequencies for (1, 0) and $\theta = 90^\circ$ and frequencies for (1, 0) and $\theta = 0^\circ$ are not equal to the frequencies for (0, 1) and $\theta = 90^\circ$. The same considerations can be made for higher half-wave numbers (1, 2) and (2, 1).

Table 9 gives the first three vibration modes of the isotropic cylindrical shell panel (benchmark 5) for several combinations of half-wave numbers (m, n) . The shell is isotropic but the frequencies are always different for each couple of (m, n) because of the geometry with different a

and b dimensions. The same considerations are also valid for benchmark 6 in Table 10 (the orthotropic cylindrical shell panel). The geometry with different a and b dimensions does not give any similarity when angle θ changes in combination with the half-wave numbers. The orthotropic effects give a coupling between in-plane and out-of-plane behavior that will require higher order 2D models to obtain the 3D analysis in terms of frequencies and vibration modes.

Tables 11 and 12 show results for isotropic and orthotropic one-layered cylinders, respectively. Results do not add further considerations with respect to those already given for the open cylindrical shell. The main difference is in terms of vibration modes through the thickness as shown in Figure 7 for isotropic cylinder (benchmark 7) and in Figures 8 and 9 for orthotropic cylinder (benchmark 8 with orthotropic angle θ equals 0° or 90°). In Figure 7 the first mode for each pair of (m, n) always has constant displacements u^* and w^* , and a quasi-constant v^* displacement, but in the case $m = 0$ and $n = 1$ the only component different from zero is u^* . The second mode always has constant displacements v^* and w^* and a linear u^* displacement, but in the case $m = 0$ and $n = 1$ u^* is zero and w^* and v^* are constant. The third mode has quasi-constant u^* , v^* , and w^* displacements, but in the case $m = 0$ and $n = 1$ the component u^* is zero. In Figure 8 the first three modes are given for the orthotropic case with $\theta = 0^\circ$. The considerations about the first vibration mode for each pair of (m, n) are the same as those already given for the isotropic benchmark. The second and third modes are different with respect to the isotropic case because of the coupling due to the in-plane orthotropy (see in particular w^* displacement for the second mode and v^* displacement for the third mode). The change of angle of orthotropy θ gives large differences in terms of frequency values and small differences in terms of vibration modes (see the comparison between Figure 8 for $\theta = 0^\circ$ and Figure 9 for $\theta = 90^\circ$).

Tables 13 and 14 investigate isotropic spherical shell panel (benchmark 9) and orthotropic spherical shell panel (benchmark 10), respectively. The shell geometry considered here has the same radii of curvature in directions α and β and the same dimensions a and b . For this reason the considerations made for the square plate (Tables 5 and 6) are also valid in these two last cases: results for half-wave numbers (0, 1) are equal to those for (1, 0), and the same similarity is valid for the half-wave numbers (1, 2) and (2, 1) (isotropic shell). For the angle $\theta = 0^\circ$, results for half-wave numbers (0, 1) are not equal to those for (1, 0), and the same considerations are made for the half-wave numbers (1, 2) and (2, 1). This is due to the in-plane orthotropy. Similar considerations are also valid for the angle $\theta = 90^\circ$. It is important to notice that frequencies for (0, 1) and $\theta = 0^\circ$ are equal to the frequencies for (1, 0) and $\theta = 90^\circ$ and frequencies for (1, 0) and $\theta = 0^\circ$ are equal to those for (0, 1) and $\theta = 90^\circ$. The same feature is confirmed for (1, 2) and (2, 1). Figure 10 shows the vibration modes for isotropic case, while Figures 11 and 12 show the vibration modes for $\theta = 0^\circ$ and $\theta = 90^\circ$ orthotropic cases, respectively. The curvature effect is very important and gives rather complicated modes through the thickness (in particular for the orthotropic spherical shells in Figures 11 and 12). These demonstrate that the

TABLE 14: Benchmark 10. Simply supported orthotropic spherical shell panel. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5	1000	100	10	5
		$(m = 0, n = 1), \theta = 0^\circ$				$(m = 0, n = 1), \theta = 90^\circ$		
I mode	2385.2	238.51	23.802	11.825	1037.9	104.08	12.585	7.7554
II mode	2902.0	290.20	29.026	14.513	2385.2	238.51	23.802	11.824
III mode	4371.6	437.15	43.559	21.518	12223	1222.3	121.65	51.888
		$(m = 1, n = 0), \theta = 0^\circ$				$(m = 1, n = 0), \theta = 90^\circ$		
I mode	1037.9	104.08	12.583	7.7554	2385.2	238.51	23.802	11.825
II mode	2385.2	238.51	23.802	11.824	2902.0	290.20	29.026	14.513
III mode	12223	1222.3	121.65	51.888	4371.6	437.15	43.559	21.518
		$(m = 1, n = 1), \theta = 0^\circ$				$(m = 1, n = 1), \theta = 90^\circ$		
I mode	1209.7	121.32	14.754	9.1755	1209.7	121.32	14.754	9.1755
II mode	4016.5	401.63	40.041	19.819	4016.5	401.63	40.041	19.819
III mode	12456	1245.5	123.88	54.506	12456	1245.5	123.88	54.506
		$(m = 1, n = 2), \theta = 0^\circ$				$(m = 1, n = 2), \theta = 90^\circ$		
I mode	1682.9	169.01	21.986	14.305	1114.0	117.90	30.685	19.645
II mode	6838.7	683.84	68.035	33.373	5758.3	575.82	57.401	28.398
III mode	13147	1314.6	130.57	60.847	23599	2359.8	208.50	58.572
		$(m = 2, n = 1), \theta = 0^\circ$				$(m = 2, n = 1), \theta = 90^\circ$		
I mode	1114.0	117.90	30.685	19.645	1682.9	169.01	21.986	14.305
II mode	5758.3	575.82	57.401	28.398	6838.7	683.84	68.035	33.373
III mode	23599	2359.8	208.50	58.572	13147	1314.6	130.57	60.847
		$(m = 2, n = 2), \theta = 0^\circ$				$(m = 2, n = 2), \theta = 90^\circ$		
I mode	1260.9	133.19	34.372	22.411	1260.9	133.19	34.372	22.411
II mode	8009.4	800.89	79.625	39.004	8009.4	800.89	79.625	39.004
III mode	23985	2398.3	216.62	65.349	23985	2398.3	216.62	65.349

use of 3D analysis or refined 2D models is fundamental to calculate these frequencies and modes. The change of angle of orthotropy has a large influence on the frequency values, but this influence is smaller for vibration modes in Figures 11 and 12.

The parametric coefficient effects, in particular the H_α effect in cylindrical shell panels and both H_α and H_β effects in spherical shell panels, are investigated in Tables 15 and 16. Table 15 shows the comparison between the present 3D solution using the exact geometry and the same solution using $H_\alpha = 1$ (which means $z/R_\alpha \approx 0$) for the benchmark 5 concerning the isotropic cylindrical shell panel. The approximation $H_\alpha = 1$ is valid for very thin shells (R_α/h equals 1000 or 100) even if it could have some problems for particular vibration modes and higher values of half-wave numbers (m, n). For thick shells the use of the exact geometry without the approximation $H_\alpha = 1$ is mandatory. Table 16 shows the comparison between the present 3D solution using the exact geometry and the same solution using the approximation $H_\alpha = H_\beta = 1$ (which means $z/R_\alpha = z/R_\beta \approx 0$) for the benchmark 10 concerning the orthotropic spherical shell panel. The approximation $H_\alpha = H_\beta = 1$ is valid for thin shells (R_α/h equals 1000 or 100), for low values of

half-wave numbers, and for the lower vibration modes. The approximation gives significative errors for thick shells where the hypothesis $z/R_\alpha = z/R_\beta \approx 0$ is not valid.

5. Conclusions

The differential equations of equilibrium in orthogonal curvilinear coordinates for the free vibrations of simply supported structures have been exactly solved in three-dimensional form by using the exponential matrix method. The proposed general 3D formulation uses an exact geometry for shells, and it allows results for spherical, open cylindrical, closed cylindrical and flat panels to be obtained. The first three vibration modes have been investigated for several geometries, both isotropic and orthotropic layers, various thickness ratios, and imposed half-wave numbers. The modes plotted through the thickness make it possible to recognize the most complicated cases and these results will be useful benchmarks to validate future refined 2D models. The method is simple and intuitive and it will be extended to multilayered structures (also embedding functionally graded layers). This extension will give a global three-dimensional overview of the free vibration problem of one-layered and multilayered plates and shells.

TABLE 15: Effects of $H_\alpha = (1 + z/R_\alpha)$ terms in benchmark 5 for simply supported isotropic cylindrical shell panel. First three vibration modes in terms of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5
$(m = 0, n = 1)$				
I mode	1067.1	106.71	10.683	5.3415
II mode	1068.3	106.83	10.690	5.3727
III mode	1855.7	185.57	18.557	9.2797
I mode ($H_\alpha = 1$)	1067.1	106.71	10.686	5.3486
II mode ($H_\alpha = 1$)	1068.3	106.83	10.688	5.3684
III mode ($H_\alpha = 1$)	1855.7	185.56	18.550	9.2651
$(m = 1, n = 0)$				
I mode	2.5478	2.5184	2.4853	2.3948
II mode	2040.3	204.03	20.411	10.216
III mode	3635.3	363.51	36.229	17.917
I mode ($H_\alpha = 1$)	2.7747	2.7473	2.7029	2.5828
II mode ($H_\alpha = 1$)	2040.3	204.03	20.403	10.201
III mode ($H_\alpha = 1$)	3635.3	363.52	36.303	18.076
$(m = 1, n = 1)$				
I mode	222.72	22.503	3.8758	3.2156
II mode	2364.5	236.45	23.656	11.842
III mode	4017.4	401.73	40.060	19.846
I mode ($H_\alpha = 1$)	222.73	22.535	4.0440	3.3738
II mode ($H_\alpha = 1$)	2364.5	236.45	23.668	11.868
III mode ($H_\alpha = 1$)	4017.4	401.73	40.115	19.965

TABLE 16: Effects of $H_\alpha = (1 + z/R_\alpha)$ and $H_\beta = (1 + z/R_\beta)$ terms in benchmark 10 for simply supported orthotropic spherical shell panel. First three vibration modes in term of no-dimensional circular frequency $\bar{\omega} = \omega(a^2/h)\sqrt{\rho/E_2}$ for several combinations of half-wave numbers m and n . $M = 100$ fictitious layers and $N = 3$ order of expansion for the exponential matrix.

R_α/h	1000	100	10	5	1000	100	10	5
$(m = 0, n = 1), \theta = 0^\circ$					$(m = 0, n = 1), \theta = 90^\circ$			
I mode	2385.2	238.51	23.802	11.825	1037.9	104.08	12.585	7.7554
II mode	2902.0	290.20	29.026	14.513	2385.2	238.51	23.802	11.824
III mode	4371.6	437.15	43.559	21.518	12223	1222.3	121.65	51.888
I mode ($H_\alpha = H_\beta = 1$)	2385.2	238.52	23.858	11.939	1037.9	104.09	12.602	7.7410
II mode ($H_\alpha = H_\beta = 1$)	2902.0	290.20	29.030	14.529	2385.2	238.52	23.856	11.934
III mode ($H_\alpha = H_\beta = 1$)	4371.6	437.15	43.623	21.667	12223	1222.3	122.04	51.743
$(m = 1, n = 0), \theta = 0^\circ$					$(m = 1, n = 0), \theta = 90^\circ$			
I mode	1037.9	104.08	12.583	7.7554	2385.2	238.51	23.802	11.825
II mode	2385.2	238.51	23.802	11.824	2902.0	290.20	29.026	14.513
III mode	12223	1222.3	121.65	51.888	4371.6	437.15	43.559	21.518
I mode ($H_\alpha = H_\beta = 1$)	1037.9	104.09	12.602	7.7410	2385.2	238.52	23.858	11.939
II mode ($H_\alpha = H_\beta = 1$)	2385.2	238.52	23.856	11.934	2902.0	290.20	29.030	14.529
III mode ($H_\alpha = H_\beta = 1$)	12223	1222.3	122.04	51.743	4371.6	437.15	43.623	21.667
$(m = 1, n = 1), \theta = 0^\circ$					$(m = 1, n = 1), \theta = 90^\circ$			
I mode	1209.7	121.32	14.754	9.1755	1209.7	121.32	14.754	9.1755
II mode	4016.5	401.63	40.041	19.819	4016.5	401.63	40.041	19.819
III mode	12456	1245.5	123.88	54.506	12456	1245.5	123.88	54.506
I mode ($H_\alpha = H_\beta = 1$)	1209.7	121.33	14.810	9.2169	1209.7	121.33	14.810	9.2169
II mode ($H_\alpha = H_\beta = 1$)	4016.5	401.64	40.137	20.022	4016.5	401.64	40.137	20.022
III mode ($H_\alpha = H_\beta = 1$)	12456	1245.6	124.29	54.312	12456	1245.6	124.29	54.312

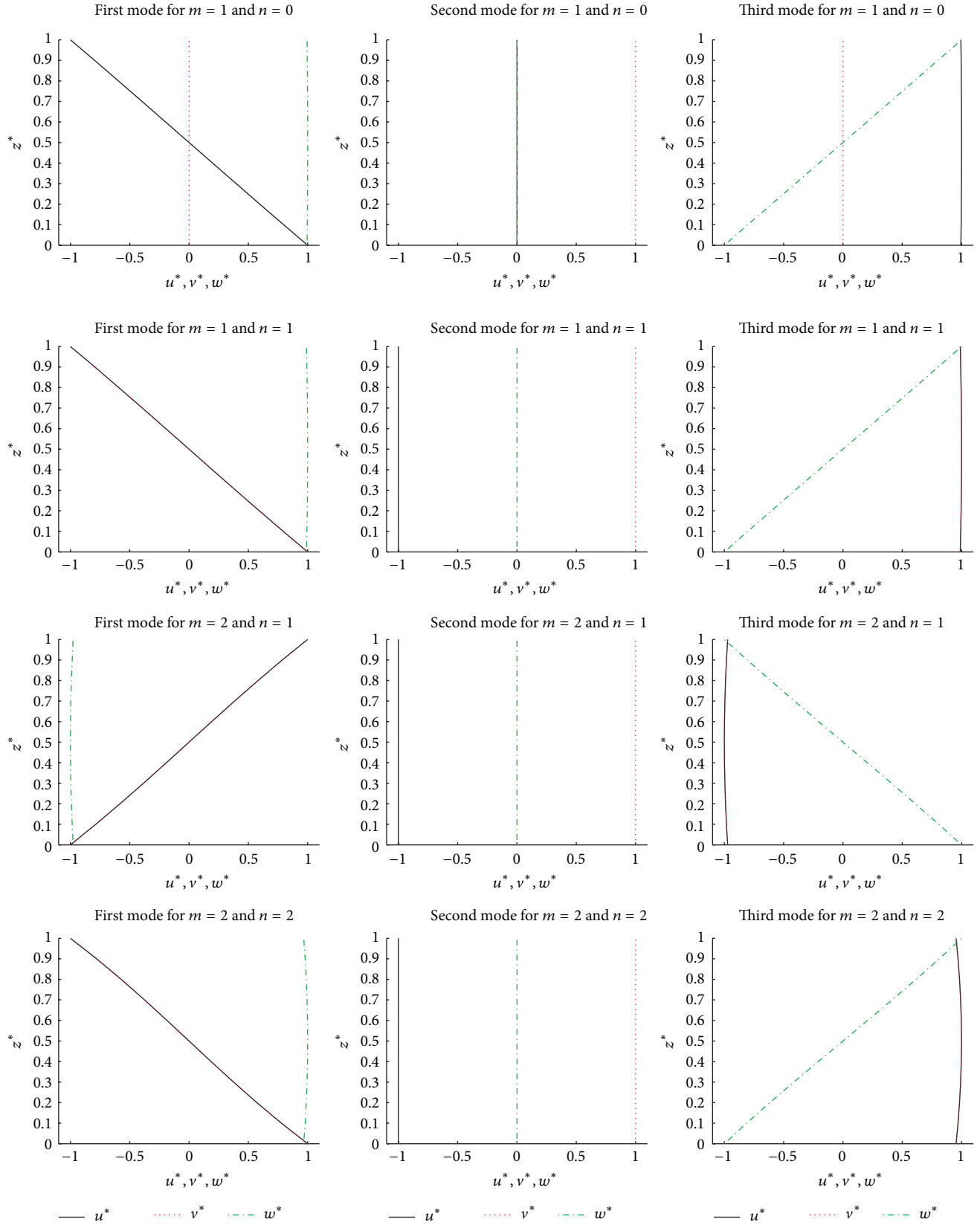


FIGURE 4: Benchmark 1, simply supported isotropic square plate. First three vibration modes in terms of displacement components through the thickness for thickness ratio $a/h = 10$ and several combinations of half-wave numbers.

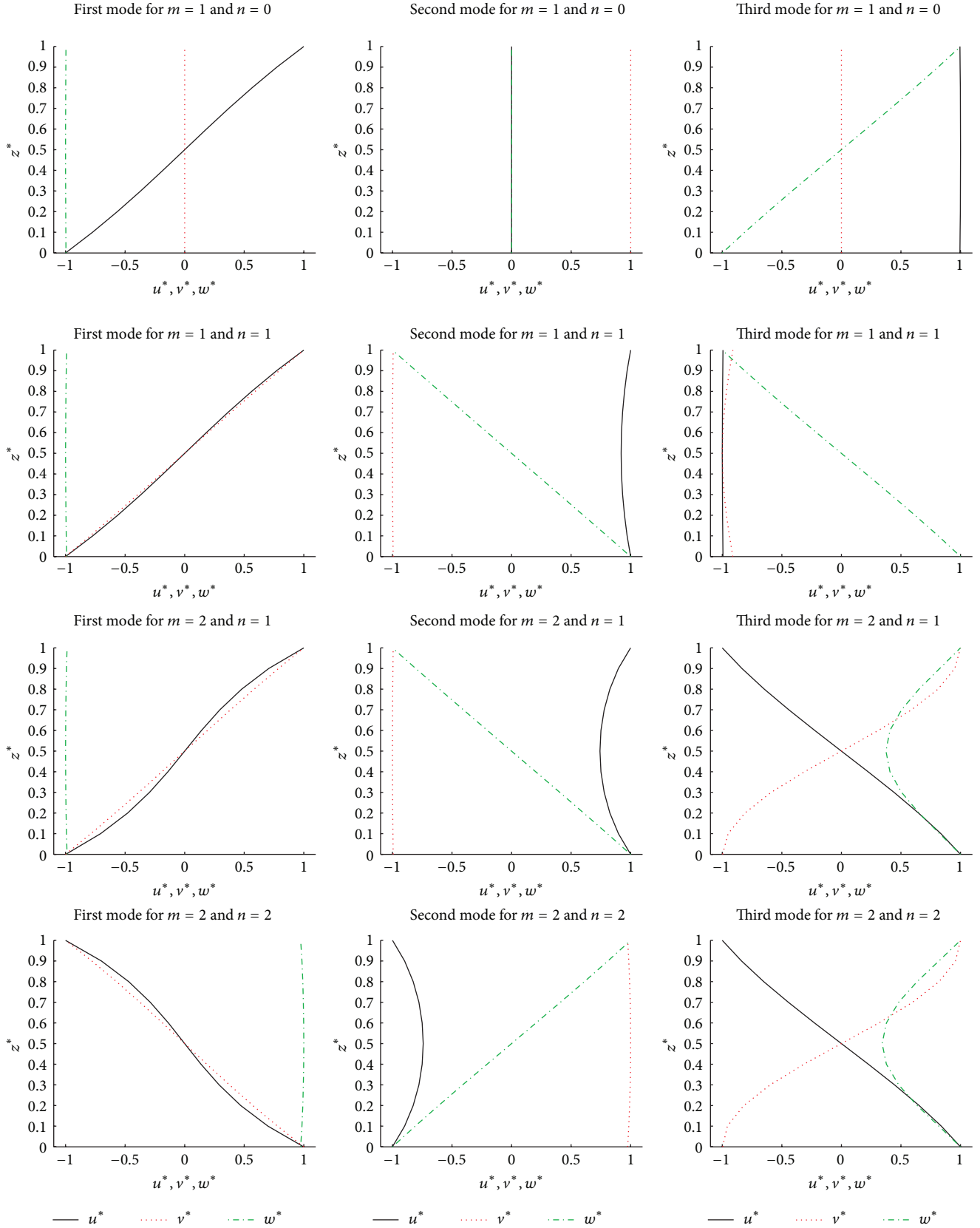


FIGURE 5: Benchmark 2, simply supported orthotropic square plate ($\theta = 0^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $a/h = 10$ and several combinations of half-wave numbers.

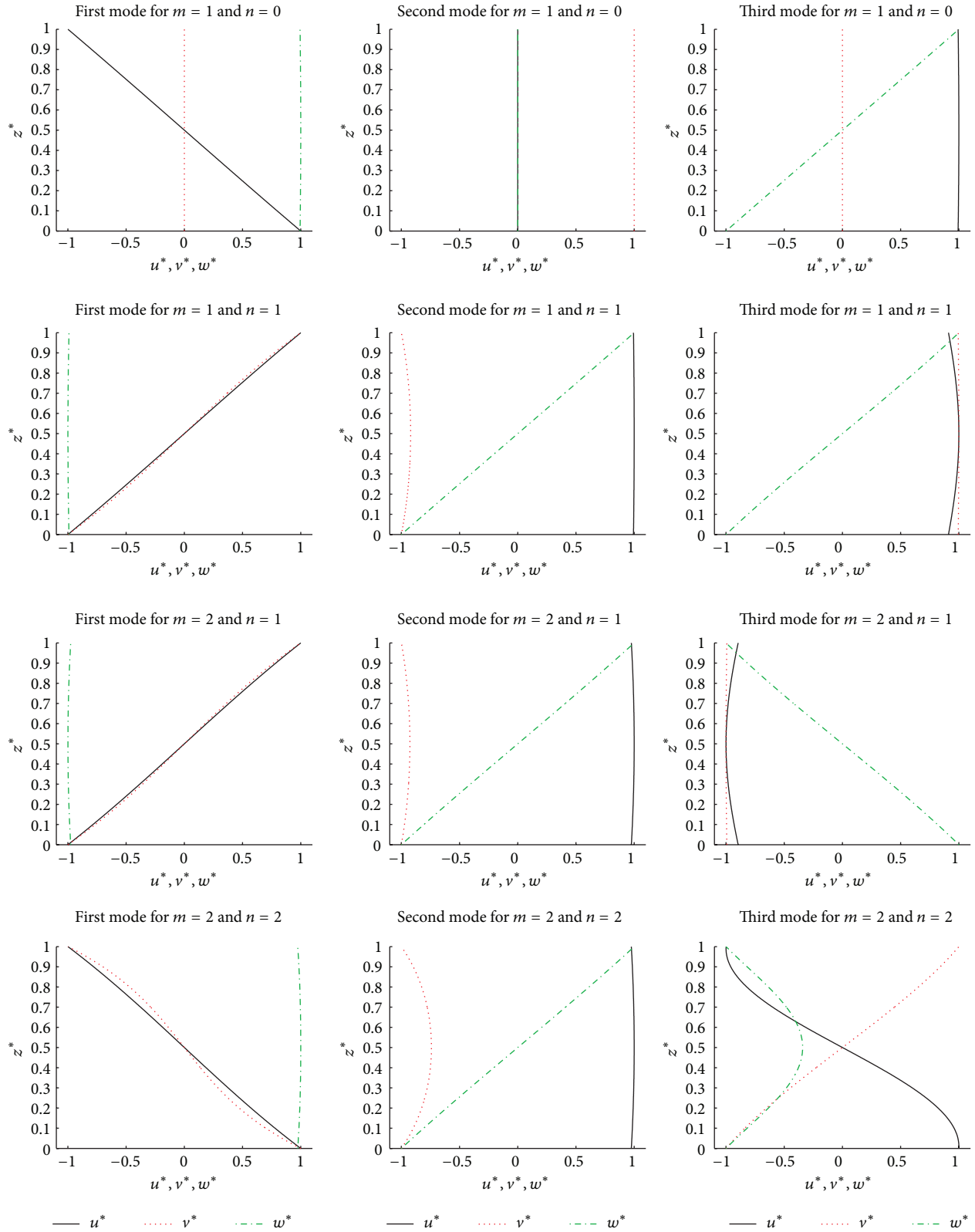


FIGURE 6: Benchmark 2, simply supported orthotropic square plate ($\theta = 90^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $a/h = 10$ and several combinations of half-wave numbers.

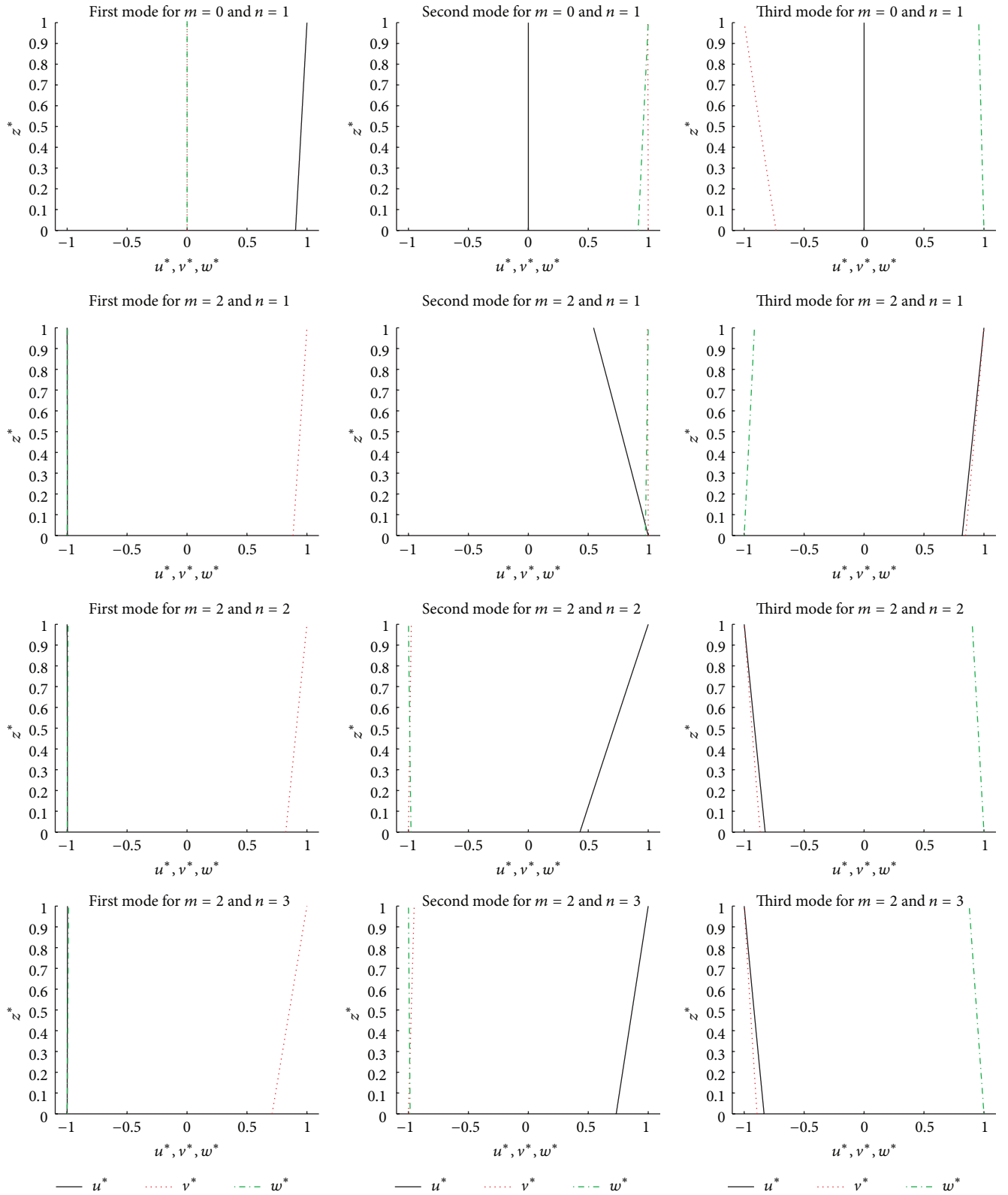


FIGURE 7: Benchmark 7, simply supported isotropic cylinder. First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 10$ and several combinations of half-wave numbers.

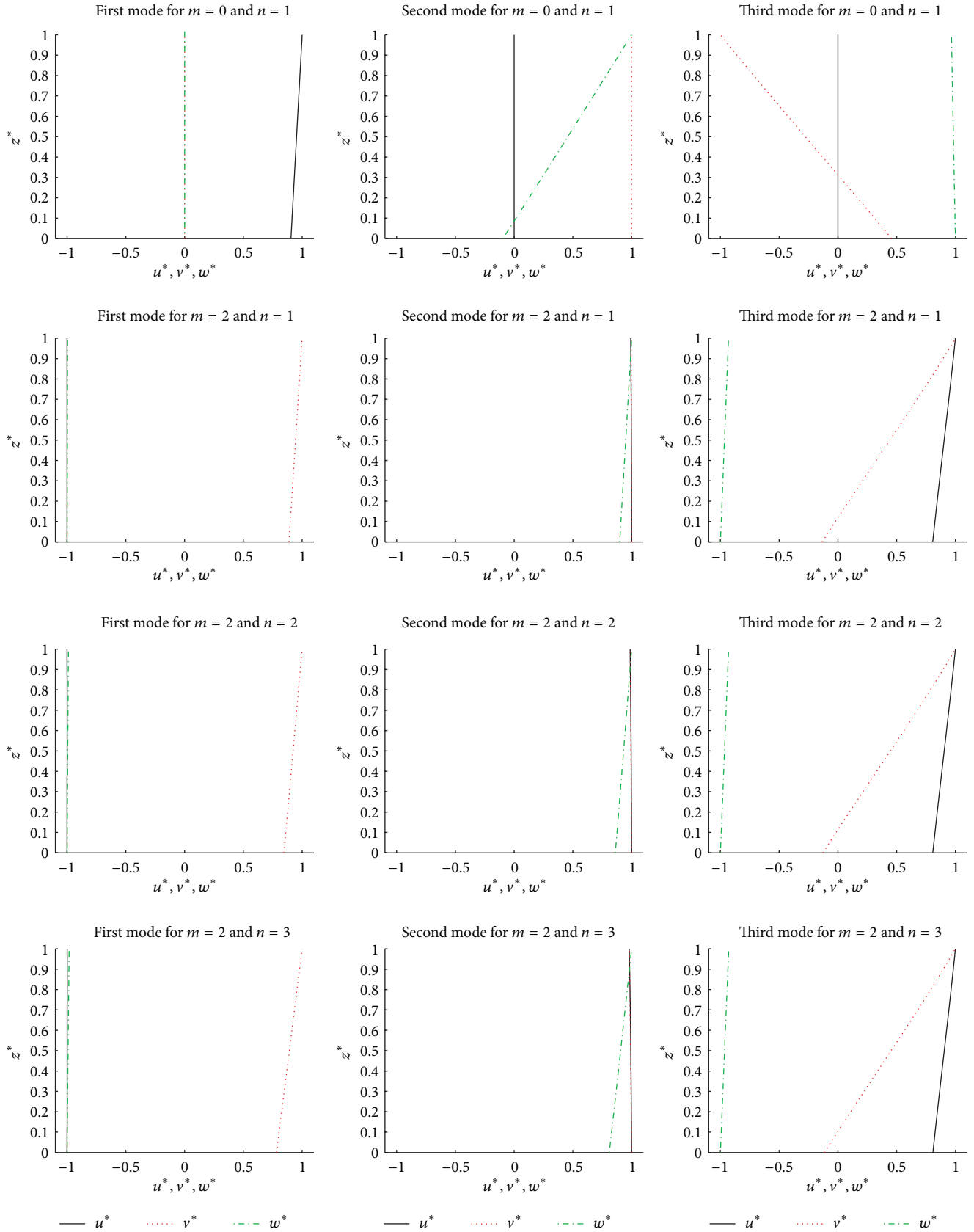


FIGURE 8: Benchmark 8, simply supported orthotropic cylinder ($\theta = 0^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 10$ and several combinations of half-wave numbers.

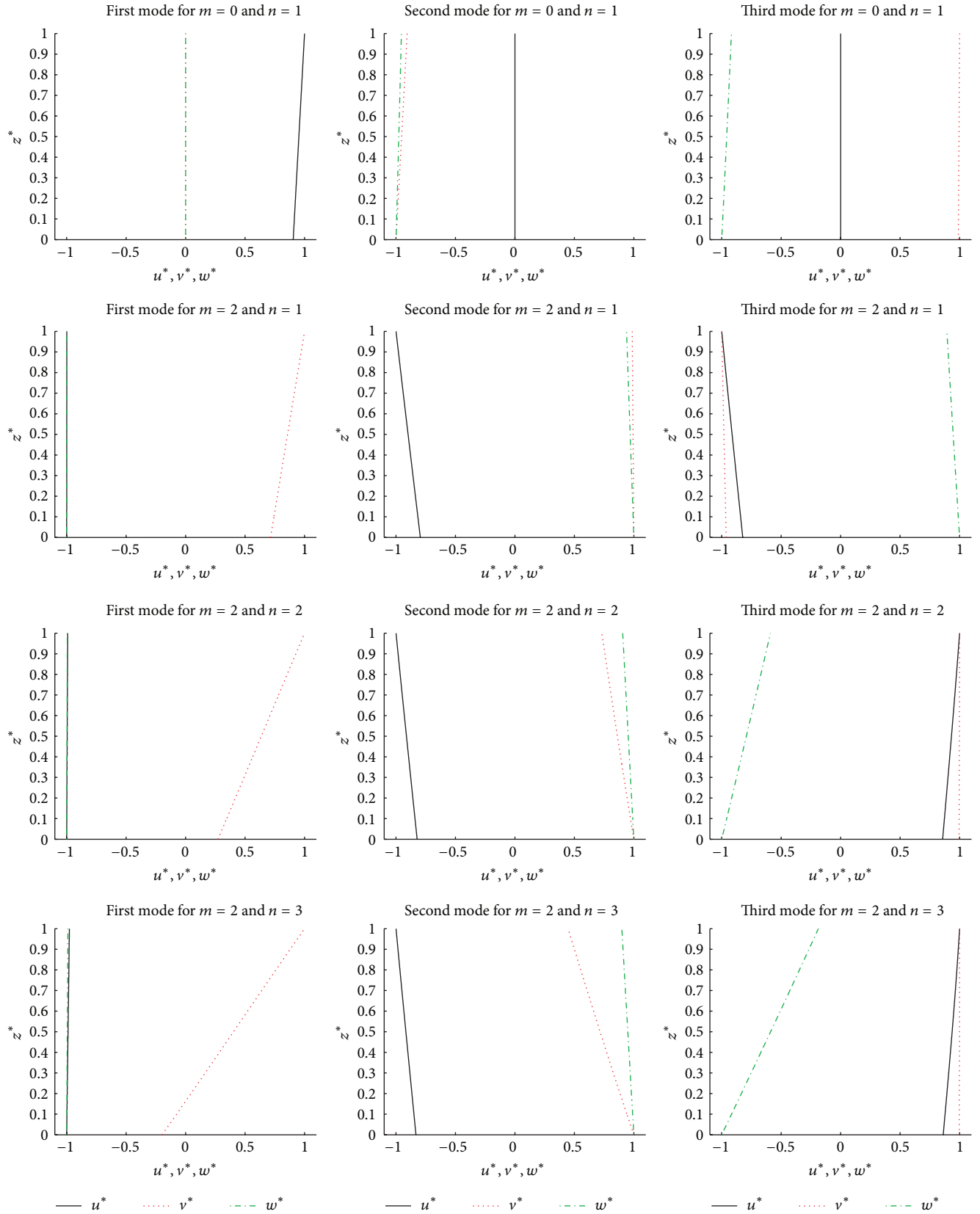


FIGURE 9: Benchmark 8, simply supported orthotropic cylinder ($\theta = 90^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 10$ and several combinations of half-wave numbers.

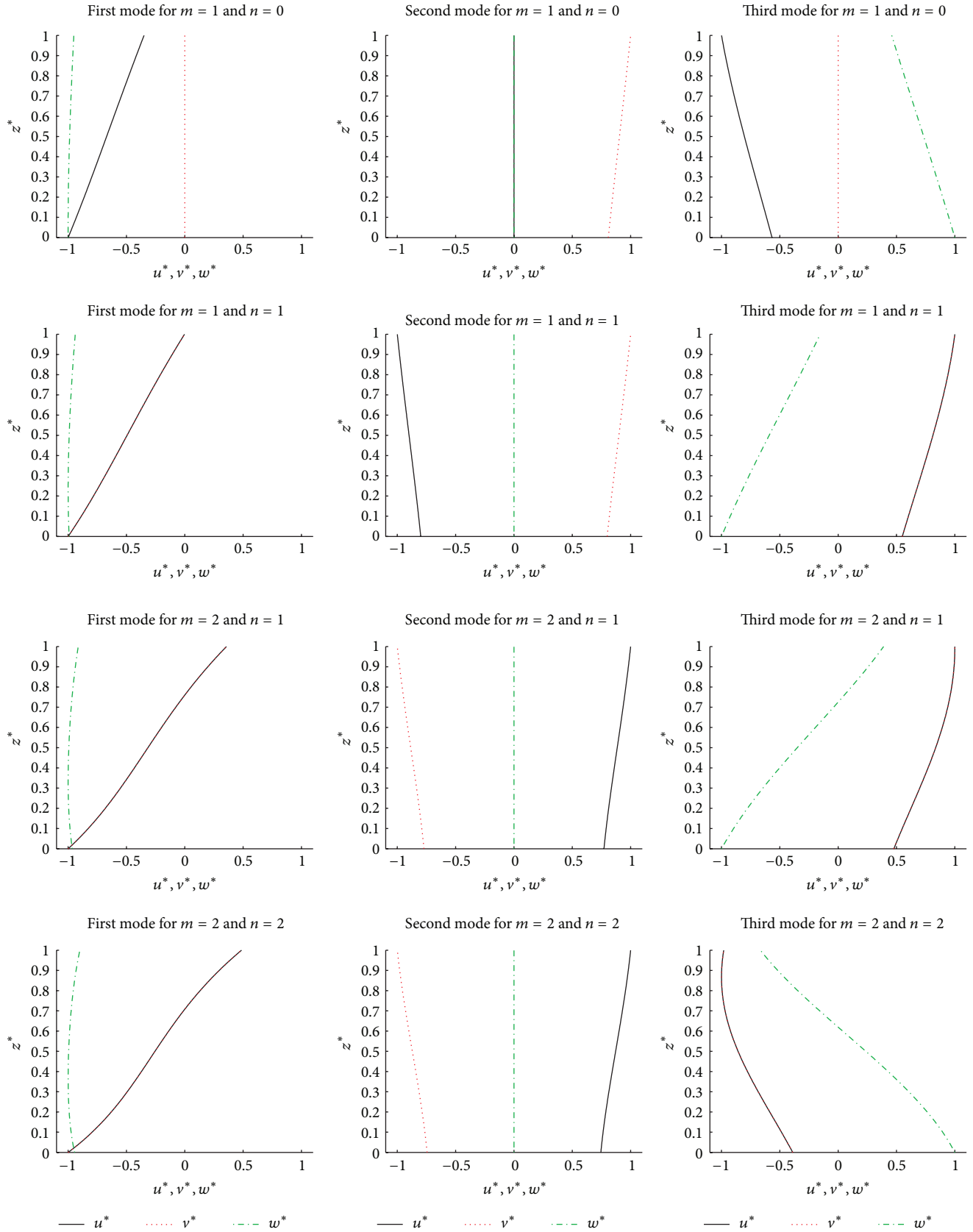


FIGURE 10: Benchmark 9, simply supported isotropic spherical shell panel. First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 5$ and several combinations of half-wave numbers.

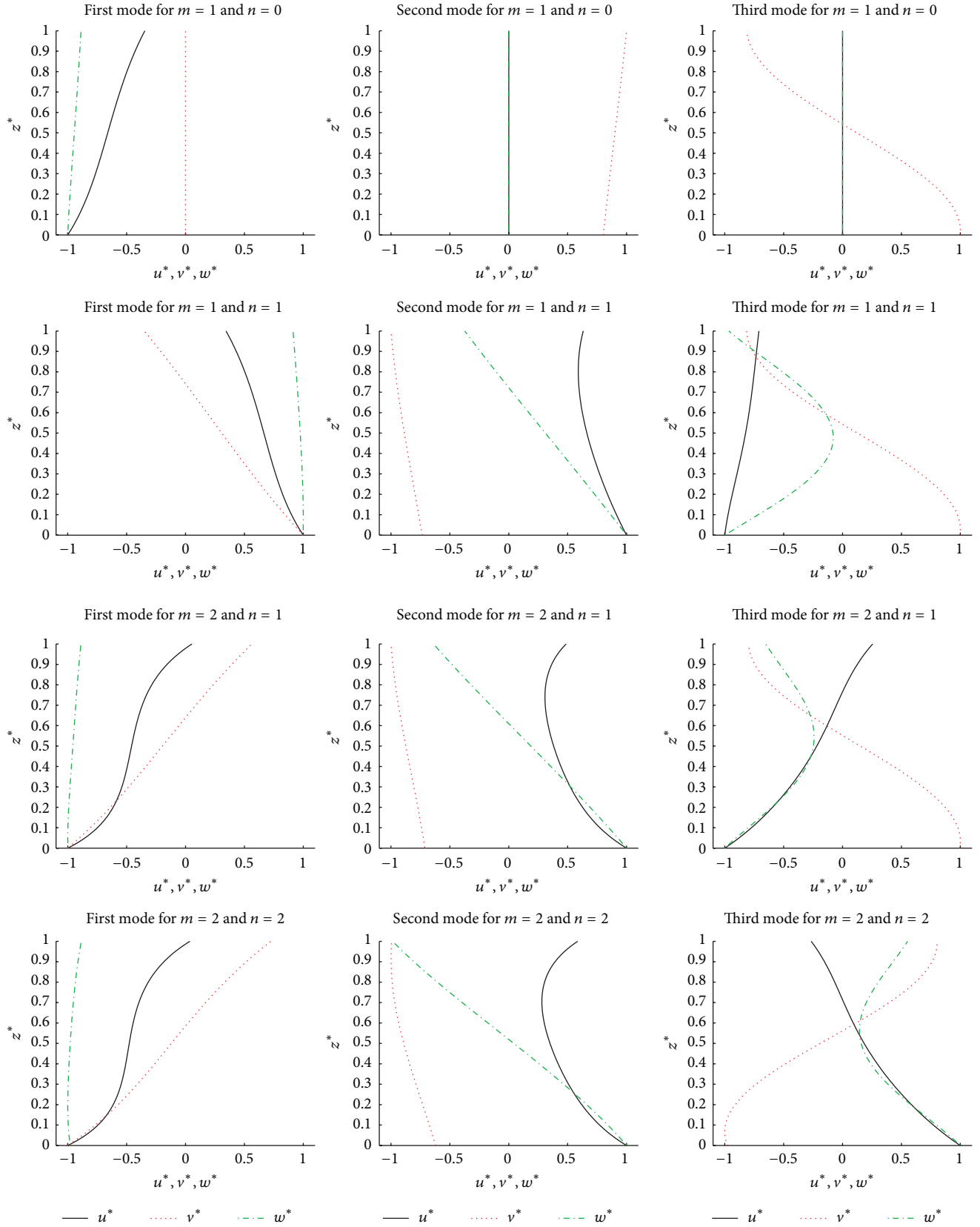


FIGURE 11: Benchmark 10, simply supported orthotropic spherical shell panel ($\theta = 0^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 5$ and several combinations of half-wave numbers.

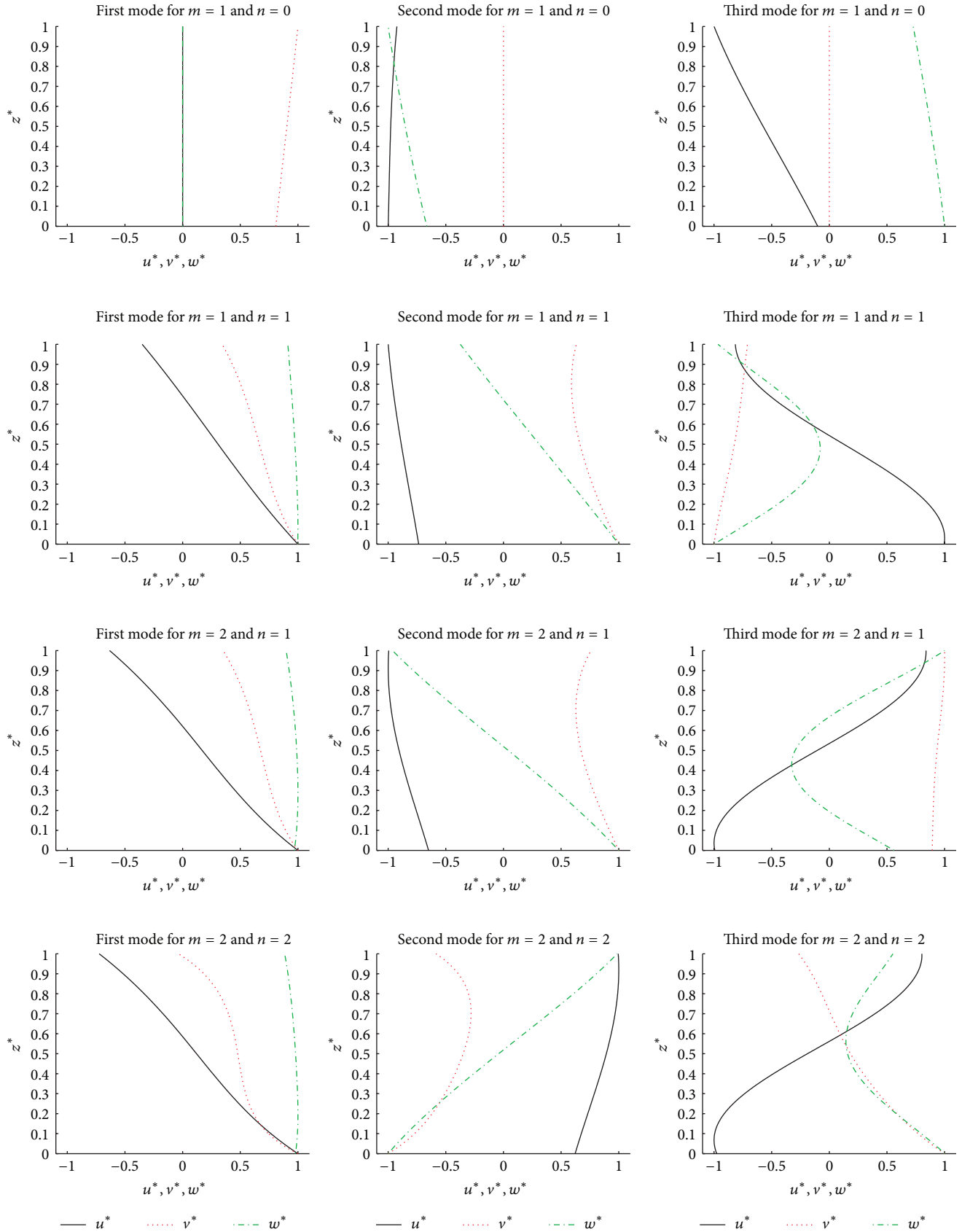


FIGURE 12: Benchmark 10, simply supported orthotropic spherical shell panel ($\theta = 90^\circ$). First three vibration modes in terms of displacement components through the thickness for thickness ratio $R_\alpha/h = 5$ and several combinations of half-wave numbers.

The present method has a fast convergence and it is not heavy from the computational point of view for each geometry (plates and shells). This feature is an important advantage with respect to other methods proposed in the literature that are valid only for a chosen geometry.

Conflict of Interests

The author declares that there is no conflict of interests regarding the publication of this paper.

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