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ARAŞTIRMA MAKALESİ

FREE VIBRATION ANALYSIS OF ELASTICALLY POINT SUPPORTED ORTHOTROPIC PLATES

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ELASTİK NOKTA MESNETLİ ORTOTROP PLAKLARIN SERBEST TİTREŞİMLERİNİN İNCELENMESİ

ÖZET

Bu çalışmada elastik nokta mesnetli ortotrop dikdörtgen plakların serbest titreşimleri incelenmiştir. Problemin çözümü için klasik Ritz metodu kullanılmıştır. Çalışmada, Ritz metodunun uygulanması için plağın yerdeğiştirmelerini ifade eden çözüm fonksiyonunun oluşturulmasında polinomlar kullanılmıştır. Ritz metodu kullanılarak problem cebrik denklem sisteminin çözümüne indirgenmiştir. Elastik nokta mesnetli ortotrop dikdörtgen plağın mod şekilleri üzerinde mekanik özelliklerin etkisi çeşitli boyut oranları, nokta mesnetlerin rijitlikleri ve plak malzemesinin anizotropisini karakterize eden özellikler için sayısal olarak incelenmiştir. Sayısal değerler çok sayıda ortotropi parametresi, nokta mesnetlerin rijitlik parametreleri ve iki adet boyut oranı durumunda, dört mod ailesi için tablolaştırılmıştır. Her bir mod ailesi için, ilk üç titreşim moduna ait özdeğerler verilmiştir. Tablolaştırılan sonuçların tasarımcılar için faydalı olacağı ve diğer araştırmacıların sonuçlarını karşılaştırmada referans olacağına inanılmaktadır. Göz önüne alınan problemler Kirchhoff-Love hipotezi çerçevesinde çözülmüştür.

ABSTRACT

Free vibration of orthotropic rectangular plates having elastically point supports at the corners is analyzed. The classical Ritz method is used for solving the problem. In the study, for applying the Ritz method, the trial function denoting the deflection of the plate is expressed in the polynomial form. By using the Ritz method, the problem is reduced to the solution of a system of algebraic equations. The influence of the mechanical properties on the mode shapes of the elastically point supported rectangular orthotropic plates is investigated numerically for various values of aspect ratios, stiffness parameters of point supports and mechanical properties characterizing the anisotropy of the plate material. The eigenvalues are tabulated for a wide range of orthotrophy parameters, stiffness parameters of point supports, and two aspect ratios for four mode families. For each mode family, the eigenvalues are provided for the first three vibration modes. It is believed that the tabulated results will prove useful to designers and provide a reference against which other researchers can compare their results. The problems considered are solved within the framework of the Kirchhoff-Love hypothesis.

1. INTRODUCTION

This problem is of considerable interest to the engineers designing panels at isolated points. The free vibration analysis of rectangular isotropic plates supported at various points and based on Kirchhoff-Love plate theory or Mindlin theory is investigated and is well known. However, it appears that there are only a limited number of studies on the vibrations of elastically point supported plates.

A considerable number of publications have been concerned with the free vibration analysis of rectangular isotropic and orthotropic plates supported at various points and based on Kirchhoff-Love or Mindlin plate theories. Kerstens[1] analyzed vibration of a rectangular plate supported at an arbitrary number of points by using the eigenfunctions of a free vibrating beam for expressing the plate displacement functions and by taking into account the Lagrangian generalized forces of constraints. The free vibration of thin rectangular orthotropic plates resting on point supports symmetrically located about the plate central axis were examined by Gorman [2]. Accurate free vibration analysis of the completely free orthotropic rectangular plate by the superposition method was studied by Gorman[3]. Free vibration analysis of point supported Mindlin plates by the superposition method was analyzed by Gorman[4]. Free vibration analysis of rectangular plates with symmetrically distributed point supports along the edges was investigated by Gorman[5]. Vibrations of point supported rectangular plates is solved by Narita[6] by using a classical Ritz method, with a trial function expressed in terms of double power series, wherein the constraint conditions of the supports are taken into account by Lagrange multipliers. By using a finite element method, the first few frequencies of point supported plates were obtained by Venkateswara Rao et al. [7]. Although there are lots of studies on the free vibration analysis of rectangular plates supported at various points, there are only a limited number of studies on the vibration of elastically/viscoelastically point supported rectangular plates: An elastically point supported plate was analyzed by Leuner [8] and Laura and Gutierrez [9]. Free vibration analysis of rectangular Mindlin plates with elastic restraints uniformly distributed along the edges was studied by Saha et. al.[10]. A general solution for the free vibration of rectangular plates resting on uniform elastic edge supports was analyzed by Gorman[11]. The steady state response to a sinusoidally varying force is determined for a viscoelastically point-supported square or rectangular plate by Yamada et. al.[12] by using the generalized Galerkin method, a generalization of this study to orthotropic rectangular plates was investigated by Kocatürk [13]. In many branches of modern industry, the structural elements, such as plates, are fabricated from composite materials. For this reason, the present investigation may be considered to be a problem of the mechanics of elements fabricated from composite materials. The eigenvalues are tabulated for a wide range of orthotrophy parameters, stiffness parameters of point supports, and two aspect ratios for four mode families. For each mode family, the eigenvalues are provided for the first three vibration modes.
The accuracy of the results is established by comparison with previously published accurate results for the corner point supported and completely free plates based on the thin plate theory and Mindlin plate theory. The considered problems are solved within the framework of the Kirchhoff-Love hypothesis.

2. ANALYSIS

Consider an elastically point-supported rectangular orthotropic plate of side lengths a, b and thickness h as shown in Figure 1, where k_i is the spring constant (siffness parameter of the i th support), $P_i\left(X_{1i}, X_{2i}\right)$ is the support force of a point support at the i th support. The axes of the elastic symmetry of the plate material coincides with the OX_1 and OX_2 axes. Also, the coordinate axes OX_1 and OX_2 are oriented along the edges of the plate with the origin at o. Although it is possible to take lots of point supports at arbitrary points, in the numerical investigations here, it will be considered that the plate is

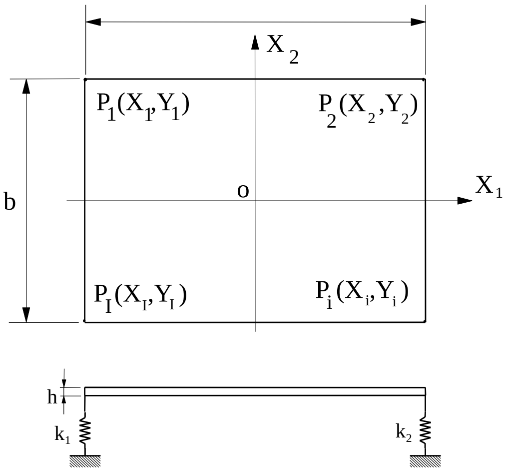

Figure 1. Elastically point-supported orthotropic plate

supported symmetrically at the four corner points, where the parameter k_i is taken to have the same value at all the supports denoted by $k_i = k$.

For an orthotropic plate that is vibrating harmonically with amplitude W and radian frequency ω, the maximum potential energy of bending in Cartesian coordinates is given as

$$U = \frac{1}{2}\int_{-\frac{a}{2}}^{\frac{a}{2}}\int_{-\frac{b}{2}}^{\frac{b}{2}}\left[D_{11}\left(\frac{\partial^2 W}{\partial X_1^2}\right)^2 + 2D_{11}\nu_{21}\frac{\partial^2 W}{\partial X_1^2}\frac{\partial^2 W}{\partial X_2^2} + D_{22}\left(\frac{\partial^2 W}{\partial X_2^2}\right)^2 + 4D_{66}\left(\frac{\partial^2 W}{\partial X_1 \partial X_2}\right)^2\right]dX_1 dX_2 \quad (1)$$

In Equation (1), D_{11}, D_{22} and D_{66} are expressed as follows,

$$D_{11} = \frac{E_1 \ h^3}{12\left(1 - \frac{\nu_{21}^2}{e}\right)} \ , \ D_{22} = \frac{E_2 \ h^3}{12\left(1 - \frac{\nu_{21}^2}{e}\right)} \ , \ D_{66} = \frac{G_{12} \ h^3}{12} \quad (2)$$

where G_{12} is shear modulus. In deriving the above expressions, the following expressions are used:

$$\frac{\nu_{12}}{E_1} = \frac{\nu_{21}}{E_2} \ , \ e = \frac{E_2}{E_1} \quad (3)$$

Here E_1, E_2 are Young's moduli in the OX_1 and OX_2 directions, respectively, and ν_{21} is the Poisson's ratio for the strain response in the X_1 direction due to an applied stress in the X_2 direction. The maximum kinetic energy of vibration is

$$T = \frac{\rho h \omega^2}{2} \int_{-\frac{a}{2}}^{\frac{a}{2}} \int_{-\frac{b}{2}}^{\frac{b}{2}} W^2 dX_1 dX_2 \tag{4}$$

and the additive strain energy of elastic supports is

$$F_s = \frac{1}{2} \sum_{i=1}^{4} k_i W_{Si}^2 \tag{5}$$

where W_s is the amplitude of the displacement of the plate at the *i* th corner and therefore is the amplitude of the displacement of the i th support. Introducing the following non-dimensional parameters

$$x_1 = \frac{X_1}{a}, \quad x_2 = \frac{X_2}{b}, \quad W = aw(x_1, x_2), \quad \alpha = b/a \tag{6}$$

the above energy expressions can be written as

$$U = \frac{D_{11}}{2} \int_{-\frac{1}{2}}^{\frac{1}{2}} \int_{-\frac{1}{2}}^{\frac{1}{2}} \left[\alpha \left(\frac{\partial^2 w}{\partial x_1^2} \right)^2 + \frac{2\nu_{21}}{\alpha} \frac{\partial^2 w}{\partial x_1^2} \frac{\partial^2 w}{\partial x_2^2} + \frac{e}{\alpha^3} \left(\frac{\partial^2 w}{\partial x_2^2} \right)^2 + \frac{4D_{66}}{\alpha D_{11}} \left(\frac{\partial^2 w}{\partial x_1^2 \partial x_2^2} \right)^2 \right] dx_1\, dx_2 \tag{7a}$$

$$T = \frac{a^3 b \rho h \omega^2}{2} \int_{-\frac{1}{2}}^{\frac{1}{2}} \int_{-\frac{1}{2}}^{\frac{1}{2}} w^2 dx_1\, dx_2 \tag{7b}$$

$$F_s = \frac{a^2}{2} \sum_{i=1}^{4} k_i\, w_i^2 \tag{7c}$$

It is known that some expressions satisfying geometrical boundary conditions are chosen for W and by minimizing the energy functional $(U + F_s) - T$, the natural boundary conditions are also satisfied. The direct application of the calculus of variations to minimize the energy functional for the orthotropic rectangular plate supported elastically at the corners leads to the partial differential equation for a vibrating plate. Therefore, instead of performing this procedure, using the classical Ritz method by assuming the displacement $W(X_1, X_2)$ to be representable by a linear series of admissible functions and adjusting the coefficients in the series to minimize the energy functional, an approximate solution is found for the displacement function. For applying the Ritz method, the trial function denoting the deflection of the plate is expressed in the following polynomial form which is used by Narita [6] for analyzing vibrations of point supported rectangular plates:

$$w(x_1, x_2) = \sum_{m=0}^{M} A_{mn} x_1^m x_2^n \tag{8}$$

Each term, x_1^m and x_2^n must satisfy the geometrical boundary conditions. However, in the considered problem, there is no geometrical boundary condition to be satisfied. As it is known, there is no need for these functions to satisfy natural boundary conditions. However, if the natural boundary conditions are also satisfied when selecting the functions, then the rate of convergence will be high.

When the function $w(x_1, x_2)$, which is given by equation (8), is substituted in equations (7a, b, c), the right hand side becomes functions of the coefficients A_{mn} . Minimization of the energy functional leads to the following expression:

$$\frac{\partial(U + F_s)}{\partial A_{kl}} - \frac{a^3 b \rho h \omega^2}{2} \frac{\partial}{\partial A_{kl}} \left(\int_{-\frac{1}{2}}^{\frac{1}{2}} \int_{-\frac{1}{2}}^{\frac{1}{2}} w^2 dx_1 \, dx_2 \right) = 0 \tag{9}$$

Introducing the following non-dimensional parameters

$$\kappa_i = \frac{k_i \ a^3}{b \ D_{11}} \ , \ \lambda^2 = \frac{\rho \ h \ \omega^2 a^4}{D_{11}} \tag{10}$$

and by using Equation (8) and (9), the set of equations (9) can be reduced to the following form

$$\sum_{m=0}^{M} \sum_{n=0}^{N} C_{mn}^{kl} A_{mn} = 0 \tag{11}$$

where

$$\begin{aligned} C_{mn}^{ij} = {} & E_{km} g_{nl} + e F_{nl} g_{mk} / \alpha^4 + \nu_{21} \left(E_{mk} F_{ln} + E_{km} F_{nl} \right) / \alpha^2 + 4 D_{66} / D_{11} H_{km} K_{ln} / \alpha^2 - \\ & \lambda^2 g_{mk} g_{nl} + \kappa (0.5)^m (0.5)^k (0.5)^n (0.5)^l + \kappa (0.5)^m (0.5)^k (-0.5)^n (-0.5)^l + \\ & \kappa (-0.5)^m (-0.5)^k (0.5)^n (0.5)^l + \kappa (-0.5)^m (-0.5)^k (-0.5)^n (-0.5)^l \end{aligned} \tag{12}$$

In writing equation (11), the following notation is used:

$$E_{km} = \int_{-0.5}^{0.5} x_1^k \frac{d^2 x_1^m}{dx_1^2} \quad E_{mk} = \int_{-0.5}^{0.5} x_1^m \frac{d^2 x_1^k}{dx_1^2} \quad F_{ln} = \int_{-0.5}^{0.5} x_2^l \frac{d^2 x_2^n}{dx_2^2} \quad F_{nl} = \int_{-0.5}^{0.5} x_2^n \frac{d^2 x_2^l}{dx_2^2}$$

$$g_{km} = \int_{-0.5}^{0.5} x_1^m x_1^k \quad g_{nl} = \int_{-0.5}^{0.5} x_2^n x_2^l \ H_{km} = \int_{-0.5}^{0.5} \frac{d^2 x_1^k}{dx_1^2} \frac{d^2 x_1^m}{dx_1^2} \quad K_{ln} = \int_{-0.5}^{0.5} \frac{d^2 x_2^l}{dx_2^2} \frac{d^2 x_2^n}{dx_2^2} \tag{13}$$

The eigenvalues (characteristic values) λ are found from the condition that the determinant of the system of equations given by equation (11) must vanish.

3. NUMERICAL RESULTS

The parameter κ_i is taken as having the same value at all the supports denoted by $\kappa_i = \kappa$. For a rectangular plate elastically point supported at the corners, four types of vibration mode exist: i.e., SS, SA, AS, AA modes. Here S means symmetric, A means antisymmetric. In the frequency equation, m and n are odd or even integers depending on the mode. For example, the SA mode is symmetric about the x_2 axis and antisymmetric about the x_1 axis. Therefore, for the SA mode, $m = 0, 2, 4, ...$ and $n = 1, 3, 5, ...$. In the numerical calculations, G_{12} is assumed as follows [14]:

$$G_{12} \approx \frac{E_1 \sqrt{e}}{2\left(1 + \nu_{21}\sqrt{1/e}\right)} \tag{14}$$

It is possible to simulate infinite lateral support stiffness by setting the translational stiffness coefficient equal to $1x10^8$ at all the supports for comparing the obtained results with the existing results of the point supported plates. Also, by setting the translational stiffness coefficient equal to zero at all the supports, a completely free plate situation is obtained. In Table 1, the calculated frequency parameters λ are compared with those of the other researchers for the SS-1, SA(AS)-1, SS-2, AA-1, SS-3 natural frequencies of an isotropic square plate supported at the corners for $\nu_{21} = 0.3$. Also, the convergence is tested in the table by taking the number of terms $M \times N = 3\times3, 4\times4, 5\times5, 6\times6$. The corresponding determinant size becomes $9\times9, 16\times16, 25\times25, 36\times36$ respectively. It is seen that the present converged values show excellent agreement with those of references [6, 7].

Table 1a. Comparison of the obtained results with the existing results and convergence study of frequency parameters λ. $\kappa = \infty, \nu_{21} = 0.3$, $\alpha = 1$, $e = 1$.

	Determinant size	SS-1	SA(AS)-1	SS-2	AA-1	SS-3
Present study $\kappa = 1\times10^8$	9x9	7.11180	15.77153	19.72570	38.45861	45.55991
	16x16	7.11093	15.77028	19.59627	38.43224	44.37619
	25x25	7.11089	15.77022	19.59614	38.43163	44.36968
	36x36	7.11088	15.77021	19.59614	38.43155	44.36961
Narita[6]		7.11089	15.7702	19.5961	38.4316	44.3696
Venkateswara Rao et. al. [7]		7.11088	15.7702	19.5961	------	------
Kerstens[1]		7.15	15.64	19.49	38.62	43.89

Table 1b. Comparison of the obtained results with the existing results and convergence study of frequency parameters λ. $\kappa = \infty, \nu_{21} = 0.333, \alpha = 1, e = 1$.

	Determinant size	SS-1	SA(AS)-1	SS-2	AA-1	SS-3
Present study $\kappa = 1\times10^8$	9x9	7.10274	15.54378	19.34726	38.20547	45.19786
	16x16	7.10196	15.54264	19.22399	38.17963	44.04902
	25x25	7.10192	15.54258	19.22386	38.17904	44.04289
	36x36	7.10192	15.54257	19.22386	38.17898	44.04282
Narita[6]		7.10192	15.5426	19.2239	38.1790	44.0428
Gorman[4]		7.112	15.55	19.22	38.18	44.080
Gorman[5]		7.18	15.564	19.22	38.36	44.28

In Table 1b, the Poison's ratio is taken as $\nu_{21} = 0.333$ and the obtained results are compared with those of Gorman[4, 5] and Narita[6]. Gorman's values [4, 5] are somewhat higher than the present ones and those of Narita[6]. It is shown that the convergence with respect to the number of the polynomial terms is excellent in the considered cases. As it is observed from Table 1, the frequency parameter decreases as the number of the polynomial terms increases: It means that the convergence is from above. By increasing the number of the polynomial terms, the exact value can be approached from above. It should be remembered that energy methods always overestimate the fundamental frequency, so with more refined analyses the exact value can be approached from above. Since the Ritz method always yields upper bounds and the convergence study indicates that the calculated values are converged to within five significant figures, it seems that the present results, as well as those in reference [7] are closer to the exact ones than the results of Gorman[4, 5]. As it was pointed out earlier, while the constraint conditions of the point supports are taken into account by Lagrange multipliers in the study of Narita[6], they are taken into account by taking the stiffness parameters $\kappa = 1\times10^8$ for infinitely rigidly point supported plates in the present study. Therefore, while the determinant size is 36x36 in the present study, it is 37x37 in that of Narita[6] for the same accuracy level.
In Table 2, the obtained numerical results are compared with those of obtained by Gorman [3] for a completely free orthotropic plate and it is observed from that table that the results are in good agreement.

Table 2. Comparison of the obtained results with the existing results and convergence study of frequency parameters λ. $\kappa = 0, \nu_{21} = 0.25, \alpha = 1, e = 1$.

	Determinant size	SS-1	SA(AS)-1	SS-2	AA-1	SS-3
Present study $\kappa = 0$	9x9	0.00	36.10585	20.27049	13.90559	24.27027
	16x16	0.00	35.60235	20.13153	13.90505	24.01574
	25x25	0.00	35.60191	20.13139	13.90501	24.01533
	36x36	0.00	35.60186	20.13139	13.90500	24.01533
Gorman[3]		0.00	35.6	20.132	13.904	24.016

Table 3. Comparison of the obtained frequency parameters λ with the existing results for the AA modes. $\sqrt{\nu_{21}\nu_{12}} = 0.25, \kappa = 0, \alpha = 1$.

		Present study		Gorman[3]	
Mode		$e=1.5$	$e=2.0$	$e=1.5$	$e=2.0$
AA-1	$\alpha=1$	15.38317	16.51963	15.384	16.52
	$\alpha=2$	7.61808	8.20249	7.62	8.204
AA-2	$\alpha=1$	75.73373	78.26101	75.72	78.28
	$\alpha=2$	29.79398	32.96248	29.792	32.964
AA-3	$\alpha=1$	89.11444	100.44466	89.12	100.44
	$\alpha=2$	64.97077	65.90557	64.96	65.92

In the calculation of the results of the present study for Tables 4, 5, 6, 7, 6x6 terms of the polynomial series were used for each type of vibration, namely the size of the determinant is 36x36.

In Tables, the values $\kappa = 0$ and $\kappa = 10^8 \approx \infty$, respectively, represent the frequency parameters of an unconstrained free plate and a simply point supported plate. Although the SS-1 mode existing in the case of point supports does not occur in the completely free plate, it occurs for every value of stiffness parameter, which is different from zero. Therefore, the frequency value of the first SS mode of the completely free plate is taken as zero and the frequency value after the zero value is assumed as the frequency value of the second mode for convenience while tabulating the results for stiffness parameters varying from zero to 10^8. However, the first frequency value, which is different from zero, is taken as the frequency value of the first SS mode in the studies on the completely free plates. When increasing the stiffness parameter κ, the frequency parameters increase monotonically and ultimately become the values of a simply point-supported plate. In the isotropic case, in the $SS-2$ vibration mode, for $E_2/E_1 = 1$, the frequency parameter remains constant without being affected by the variation of κ [12, 13], as can be seen from Table 4. However, in the orthotropic case, in the $SS-2$ vibration mode, frequency parameters change with the variation in κ [13]. Although Yamada[12] obtained some of the values of frequency parameters for isotropic plates and Kocatürk[13] for orthotropic plates in the case of elastic point supports, the values were shown graphically. Therefore there is no exact value to compare the present obtained results.

4. CONCLUSIONS

By using the classical Ritz method, the natural frequencies of elastically point supported orthotropic rectangular plates has been studied and compared with the existing results. By the application of the above mentioned solution technique, for each vibration mode family, the first three value of SS, SA, AS, AA vibration mode natural frequencies are determined, the converge characteristics of the frequency parameters are investigated numerically for orthotropic square plates elastically supported at four points at the corners. It is seen that the rate of convergence is very high. The effect of the orthotropy, stiffness of the supports and the aspect ratios on the frequency parameters is investigated and shown in the tables. These results are the unique results for the elastically point supported orthotropic plates in the case of stiffness parameters different from zero and ∞. It is expected that the tabulated data will be very useful to the researches to compare their results. It should be noted that the obtained results may easily be extended to multiple support conditions and may be useful for designing mechanical and structural systems.

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Hazırlamış olduğumuz Free Vibration Analysis Of Elastically Point Supported Orthotropic Plates isimli makale hakemlerin görüşleri doğrultusunda düzeltilmiştir. İlgili makalenin Yıldız Teknik Üniversitesi Dergisi'nde basılması için gereğini arz ederiz.

Saygılarımızla.

Prof. Dr. Turgut KOCATÜRK

Ar. Gör. Nihat İLHAN

Eki : 1) Makalenin düzeltilmiş çıktısı.
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Table 4. SS frequency parameters λ calculated for various support stiffness values, orthotrophy values

Table 5. SA frequency parameters λ calculated for various support stiffness values, orthotrophy values