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Vibration and Buckling Analysis of Skew Sandwich Plate using Radial Basis Collocation Method

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KEYWORDS

Skew plate;
HSDT;
Sandwich;
Meshfree;
Vibration;
Buckling.

ABSTRACT

This paper presents the free vibration and buckling responses of a skew sandwich plate using higher-order shear deformation theory (HSDT). The governing differential equations (GDEs) for the skew sandwich plate are obtained using Hamilton's principle, which states that the actual motion of a system minimizes the total potential energy of the system. The GDEs obtained are discretized using radial basis function (RBF), which is a meshfree based numerical method. The vibration and buckling results for skew sandwich plates using meshfree methods and the effect of node distribution are not available in the open literature to the best of the author's knowledge. Numerous results are presented showing the non-dimensional frequency and buckling parameters of the skew sandwich plates for different values of the plate geometry, material properties, and boundary conditions. These results provide insights into the vibration and buckling behavior of skew sandwich plates and can be used to optimize the design and performance of these plates for various applications, such as aerospace structures, marine structures, and civil engineering structures. Convergence studies of present results are checked, and the results obtained are also validated with the results available in the open literature. The effect of span-to-thickness ratio, core-to-face thickness ratio, aspect ratio, boundary conditions, boundary node distribution, and skew angle is examined. The results presented in this paper can be useful for engineers and researchers working in the field of structural mechanics and can contribute to the development of safer and more efficient structures.

1. Introduction

Because of their advantageous features like high strength and stiffness-to-weight ratios, thermal properties, and a variety of other multi-physical aspects, laminated composites and sandwich configurations are used in many lightweight structures in aviation, structural, maritime, and civil engineering applications. By giving designers of mechanical parts the flexibility to customize the distribution of materials of various qualities in accordance with the loading routes, these designs also enable custom optimization. Laminated composites are constructed by layering piles of composites with specific fiber orientations in each layer to create the structures. Shi et al. [1] applied a semi-analytical approach to the buckling response of sandwich plates. Karakoti et al. [2] investigated skew-edge sandwich plates via the finite element method. Katariya et al. [3] studied statics and natural frequency of skew sandwich composite plates using the HSDT model.

Radial basis function-based mesh-free methods have been used for the analysis of plates by Singh et al.[4] for free vibration of laminated composite plates, and Solanki et al. [5] for the flexure behavior of laminated plates. Singh et al. [6] for buckling of square sandwich plates, Shukla and Singh [7] for flexure analysis of angle-ply rectangular plates, Solanki et al. [8] for nonlinear vibrations of square laminated composite and sandwich plates, Singh et al. [9] for flexure of laminated composite and sandwich plates. RBFs have been also used to analyze the buckling behavior of functionally graded materials rectangular plates by Kumar et al. [10, 11]. Civalek [12] obtained the frequencies and buckling loads of skew laminated composite plates using the discrete singular convolution method. Ashour [13] studied the free vibration

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response of symmetric laminated angle-ply thin skew plates using the finite strip transition matrix method. Malekzadeh and Alibeygi Beni [14] investigated the frequency response of skew FGM plates via DQM.

The buckling response of a CFRP plate has been studied by Yidris et al. [15] Civalek and Jalaei [16] analyzed the shear buckling of an FG skew plate. Vibrational analysis of FG skew sandwich plates of geometric distortions has been carried out by Khanke and Tande [17]. Kiani and Żur.[18] studied the effect of vibrations on graphene platelet-reinforced composite skew plates resting on point supports. Investigating Responses due to the nonlinear moving load of FG-GPLRC skew plates has been carried out by Noroozi and Malekzadeh [19].

Sayyad and Ghugal [20] used sinusoidal beam theory for analysis of functionally graded sandwich curved beams. Flexure response of laminated plates [21] stress analysis of orthotropic laminated doubly-curved shells on rectangular planform under concentrated Force have been analysed by Sayyad and Ghugal [22].

Bending, buckling, vibration 0f rectangular laminated composite and sandwich plates has been studied by Sayyad and Ghugal [23–27].

Ghugal and Sayyad [28] has studied stress analysis of thick laminated plates using trigonometric shear deformation theory. Laminated composite, sandwich and FGM beams have been analyzed by Shinde and Sayyad [29].

Thermoelastic analysis of laminated plates has been done by Sayyad et al. [30] using four variable plate theory.

The literature review indicates that there has been limited research on the buckling and vibration analysis of skew-laminated sandwich plates. Therefore, this paper aims to address these gaps by using theoretical approaches.

The paper derives the GDEs for buckling and vibration characteristics and reveals the influence of skew angles. The GDEs are derived via Hamilton's principle and discretized via the RBF approach.

Overall, this study provides valuable insights into the behavior of skew laminated plates, which can be applied in various engineering applications, such as aviation and civil engineering.

2. Mathematical Formulation

Figure 1 shows the geometry of a skew plate with a skew angle (ψ) where thickness 'h' along the z-axis has been considered.

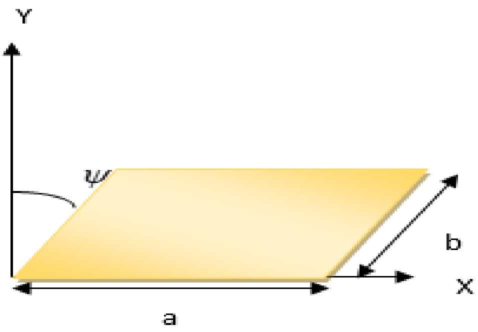

Figure.1. Geometry of skew plate

The displacement variables are expressed as Srivastava and Singh [31], [32]:

$$\begin{aligned} u &= u_0 - z\frac{\partial w_0}{\partial x} + f(z)\phi_x, \\ v &= v_0 - z\frac{\partial w_0}{\partial y} + f(z)\phi_y, \\ w &= w_0 \end{aligned} \qquad (1)$$

For the present analysis, the transverse shear stress has been considered as proposed by Touratier [33].

The GDEs of the skew plate along with boundary conditions are derived using Hamilton's principle, which is expressed as [34]:

$$\int_{t_1}^{t_2} \delta L\, dt = 0 \qquad (2)$$

where, L is Lagrangian and defined as

$$L = KE - (UE + VE) \qquad (3)$$

where,

KE = Kinetic energy, UE = Strain energy,

VE = Potential energy due to external loads

The expressions for kinetic energy and strain energy of the plate can be written as[35]:

$$KE = \frac{1}{2}\int_{-\frac{h}{2}}^{\frac{h}{2}} \int_A \rho\left\{\left(\frac{\partial u}{\partial \tau}\right)^2 + \left(\frac{\partial v}{\partial \tau}\right)^2 + \left(\frac{\partial w}{\partial \tau}\right)^2\right\} dz\, dA \qquad (4)$$

$$UE = \frac{1}{2}\int_{-\frac{h}{2}}^{\frac{h}{2}} \int_A (\sigma_{xx}\varepsilon_{xx} + \sigma_{yy}\varepsilon_{yy} + \sigma_{xy}\gamma_{xy} + \sigma_{yz}\gamma_{yz} + \sigma_{xz}\gamma_{xz})\, dz\, dA \qquad (5)$$

The potential energy due to in-plane mechanical loading can be expressed as [36],[37];

$$VE = \frac{1}{2}\int_A \left\{ N_x^b\left(\frac{\partial w}{\partial x}\right)^2 + N_y^b\left(\frac{\partial w}{\partial y}\right)^2 + 2N_{xy}^b\left(\frac{\partial w}{\partial x}\right)\left(\frac{\partial w}{\partial y}\right)\right\} dA \qquad (6)$$

In Equation (6) N_x^b, N_y^b and N_{xy}^b are the applied in-plane compressive loadings in x and y directions and shear loading, respectively. The governing differential equations of the plate are obtained by collecting the coefficients of $\partial\mu_o$, ∂v_o, $\partial\omega_o$, $\partial\psi_x$ and $\partial\psi_y$ can be expressed as:

$$\delta u_0: \frac{\partial N_{xx}}{\partial x}+\frac{\partial N_{xy}}{\partial y}=\left(I_0\frac{\partial^2 u_0}{\partial\tau^2}-I_1\frac{\partial^3 w_0}{\partial x\partial\tau^2}+I_3\frac{\partial^2\phi_x}{\partial\tau^2}\right) \tag{7}$$

$$\delta v_0: \frac{\partial N_{xy}}{\partial x}+\frac{\partial N_{yy}}{\partial y}=\left(I_0\frac{\partial^2 v_0}{\partial\tau^2}-I_1\frac{\partial^3 w_0}{\partial y\partial\tau^2}+I_3\frac{\partial^2\phi_y}{\partial\tau^2}\right) \tag{8}$$

$$\begin{aligned}\delta w_0: &\frac{\partial^2 M_{xx}}{\partial x^2}+\frac{\partial^2 M_{yy}}{\partial y^2}+2\frac{\partial^2 M_{xy}}{\partial x\partial y}+N_x^b\frac{\partial^2 w_0}{\partial x^2}+N_y^b\frac{\partial^2 w_0}{\partial y^2}+2N_{xy}^b\frac{\partial^2 w_0}{\partial x\partial y}\\ &=I_0\frac{\partial^2 w_0}{\partial t^2}+I_1\left(\frac{\partial^3 u_0}{\partial x\partial\tau^2}+\frac{\partial^3 v_0}{\partial y\partial\tau^2}\right)-I_2\left(\frac{\partial^4 w_0}{\partial x^2\partial\tau^2}+\frac{\partial^4 w_0}{\partial y^2\partial\tau^2}\right)\\ &+I_4\left(\frac{\partial^3\phi_x}{\partial x\partial\tau^2}+\frac{\partial^3\phi_y}{\partial y\partial\tau^2}\right)\end{aligned} \tag{9}$$

$$\delta\phi_x: \frac{\partial M_{xx}^f}{\partial x}+\frac{\partial M_{xy}^f}{\partial y}-Q_x^f=\left(I_3\frac{\partial^2 u_0}{\partial\tau^2}-I_4\frac{\partial^3 w_0}{\partial x\partial\tau^2}+I_5\frac{\partial^2\phi_x}{\partial\tau^2}\right) \tag{10}$$

$$\delta\phi_y: \frac{\partial M_{xy}^f}{\partial x}+\frac{\partial M_{yy}^f}{\partial y}-Q_y^f=\left(I_3\frac{\partial^2 v_0}{\partial\tau^2}-I_4\frac{\partial^3 w_0}{\partial y\partial\tau^2}+I_5\frac{\partial^2\phi_y}{\partial\tau^2}\right) \tag{11}$$

where,

$$N_{ij},\ M_{ij},\ M_{ij}^f = \int_{-h/2}^{+h/2}(\sigma_{ij},\ z\sigma_{ij},\ f(z)\sigma_{ij})\,dz \tag{12}$$

$$Q_x^f,\ Q_y^f = \int_{-h/2}^{+h/2}\left(\sigma_{xz},\ \sigma_{yz}\right)\left(\frac{\partial f(z)}{\partial z}\right)dz \tag{13}$$

The simply supported boundary conditions are taken as:

$$\begin{aligned}&u_s,\phi_s,w_0,N_{nn},M_{nn}=0 \quad \textit{Simply supported}\,(S)\\ &u_s,u_n,w,\phi_s,\phi_n=0 \quad \textit{Clamped}\,(C)\end{aligned} \tag{14}$$

where,

$$\begin{aligned}&u_s=-n_y.u_o+n_x.v_0\\ &\phi_s=-n_y.\phi_x+n_x.\phi_y\\ &N_{nn}=n_x^2N_{xx}+2n_xn_yN_{xy}+n_y^2N_{yy}\\ &M_{nn}=n_x^2M_{xx}+2n_xn_yM_{xy}+n_y^2M_{yy}\\ &n_x=\cos(\psi),\quad n_y=\sin(\psi)\end{aligned} \tag{15}$$

3. Solution Methodology

A skew domain with NB boundary nodes and NI interior nodes is obtained and shown in Figure 2.

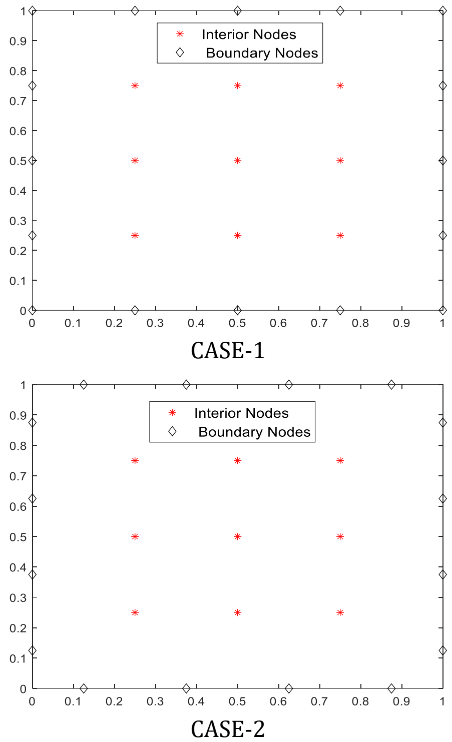

Fig. 2. Geometry of skew plate with nodes

The field variables u_0, v_0, w_0, ϕ_x and ϕ_y of Eq (1) are assumed in terms of radial basis function as [38], [39]:

$$\begin{aligned}&u_0=\sum_{j=1}^{N}\alpha_j^{u_0}g\left(\|X-X_j\|,c\right),\ v_0=\sum_{j=1}^{N}\alpha_j^{v_0}g\left(\|X-X_j\|,c\right),\\ &w_0=\sum_{j=1}^{N}\alpha_j^{w_0}g\left(\|X-X_j\|,c\right)\\ &\phi_x=\sum_{j=1}^{N}\alpha_j^{\phi_x}g\left(\|X-X_j\|,c\right),\ \phi_y=\sum_{j=1}^{N}\alpha_j^{\phi_y}g\left(\|X-X_j\|,c\right)\end{aligned} \tag{16}$$

where, $[\alpha^{u_0}\ \alpha^{v_0}\ \alpha^{w_0}\ \alpha^{\phi_x}\ \alpha^{\phi_y}]$ are unknown coefficients, $g\left(\|X-X_j\|,c\right)$ is RBF, is the radial distance between the nodes, and c is the shape parameter. For the present analysis, RBF taken is a thin plate spline $g=r^{2c}\log r$ with c=3.

$$r=\|X-X_j\|=\sqrt{\left((x-x_j)/a\right)^2+\left((y-y_j)/b\right)^2} \tag{17}$$

The GDEs along with boundary conditions are discretized and expressed in compact matrix form as:

$$\left[[K]+\lambda[K]_G\right]\{\delta\}=\Re\,[M]\{\delta\} \tag{18}$$

For Buckling analysis $\Re$ =0 and λ =1

For Vibration analysis $\Re$ =1 and λ =0

here, [K] is the stiffness matrix obtained using equations [7-11] for interior nodes and using equation [14] for boundary nodes. [M] is the mass matrix obtained using equation [7-14]. The final expressions for vibration analysis can be expressed as [40]

$$\left[\begin{bmatrix}[K]_L \\ [K]_B\end{bmatrix}_{5N\times 5N} + \omega^2 \begin{bmatrix}[M] \\ 0\end{bmatrix}_{5N\times 5N}\right]\{\delta\}_{5N\times 1} = 0 \tag{19}$$

$$\{\delta\} = [\alpha^{u_0}\ \alpha^{v_0}\ \alpha^{w_0}\ \alpha^{\phi_x}\ \alpha^{\phi_y}]^T$$

The eigenvectors (V) and eigenvalues (D) are calculated as:

$$[V, D] = \text{eig}\left[\begin{bmatrix}[K]_L \\ [K]_B\end{bmatrix}_{5N\times 5N}, \begin{bmatrix}[M] \\ 0\end{bmatrix}_{5N\times 5N}\right] \tag{20}$$

Frequency (ω) = $\sqrt{D}$

For buckling analysis, the final equation can be expressed as [41]:

$$\left[\begin{bmatrix}[K]_L \\ [K]_B\end{bmatrix}_{N\times N} + \lambda \begin{bmatrix}[K]_G \\ 0\end{bmatrix}_{N\times N}\right]\{\delta\}_{N\times 1} = 0$$

where *[K]G* is the geometric matric obtained from equation [9]. *[K]L* denotes the stiffness matrix at domain nodes and *[K]B* for boundary nodes.

The eigenvectors (V) and eigenvalues *(D)* are calculated as:

$$[V, D] = \text{eig}\left[\begin{bmatrix}[K]_L \\ [K]_B\end{bmatrix}_{5N\times 5N}, \begin{bmatrix}[K]_G \\ 0\end{bmatrix}_{5N\times 5N}\right] \tag{21}$$

Finally, the buckling load is calculated as (λ) = (D).

4. Result and Discussion

Present section deals with numerical experimentations and validation of obtained results. A square sandwich plate (a/h=10) consisting of two orthotropic face layers of 0.1 times thickness and a core layer of 0.8 times thickness is considered. The material properties taken for the core layer are as Pandit et al. [42].

E22/E11=0.543, E1=1, ν_{12} =0.3,

G12/E11=0.2629, G13/E11=0.1599,

G23/E11=0.2668, ρ =1.

The elastic modulus of face sheets has been varied with a factor Rf.

The frequency and buckling responses are normalized as $\Omega = 100\omega\left(\frac{\rho h^2}{E_1}\right)^{(1/2)}$ and $\overline{\lambda} = \lambda b^2/(E_{2f}h^3)$ respectively.

Two cases of node generation have been considered here to see the effect of change in the position of boundary nodes while applying the boundary conditions. CASE-1 represents the position when boundary nodes are at the corner while CASE-2 represents when boundary nodes are not at the corner.

The convergence and validation study for the fundamental frequency parameter and buckling parameter of a simply supported square sandwich plate are obtained and presented in Table 1 and Table 2 and the same has been depicted in Fig. 3 and Fig. 4 respectively. Results obtained are compared with results due by Srinivas and Rao [43] for free vibration analysis. It can be seen that the present results converged for both cases within 1% and became closer to the results of Srinivas and Rao [43] for vibration analysis at 15X 15 nodes. From Fig. 3 and Fig. 4, it is observed that convergence is better when a plate is rectangular as compared to that of for skew plate with a skew angle of 45.

Table 1. Frequency parameter Ω for different skew angles (ψ)[Rf=10]

No. of Nodes	CASE=1		CASE-2	
	ψ =0	ψ =45	ψ =0	ψ =45
5x5	16.2389	11.5657	18.0937	10.6235
7x7	11.7651	20.0860	5.3638	18.1613
9x9	10.3759	15.3905	10.3735	16.1812
11x11	10.0651	14.4978	10.0258	15.2517
13x13	9.9713	14.0399	9.9311	14.6074
15x15	9.9339	13.7013	9.8979	14.1129
Srinivas and Rao [43]	9.8104	--------	9.8104	---------

Table 2. Buckling parameter λ for different skew angles (ψ) [Rf=10]

No. of Nodes	CASE=1		CASE-2	
	ψ =0	ψ =45	ψ =0	ψ =45
5x5	1.5920	1.3842	3.1811	3.4106
7x7	1.6854	5.0083	4.3924	7.8329
9x9	2.5082	6.7541	2.7162	8.4169
11x11	2.5336	7.4413	2.5470	8.4519
13x13	2.5355	7.3135	2.4984	8.0468
15x15	2.5723	7.0777	2.4791	7.7763

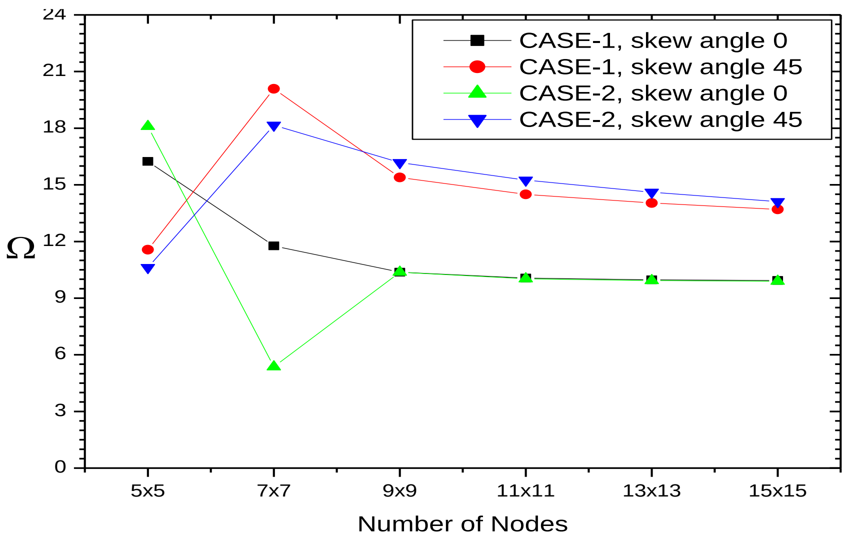

Figure 3. Convergence of frequency parameter Ω

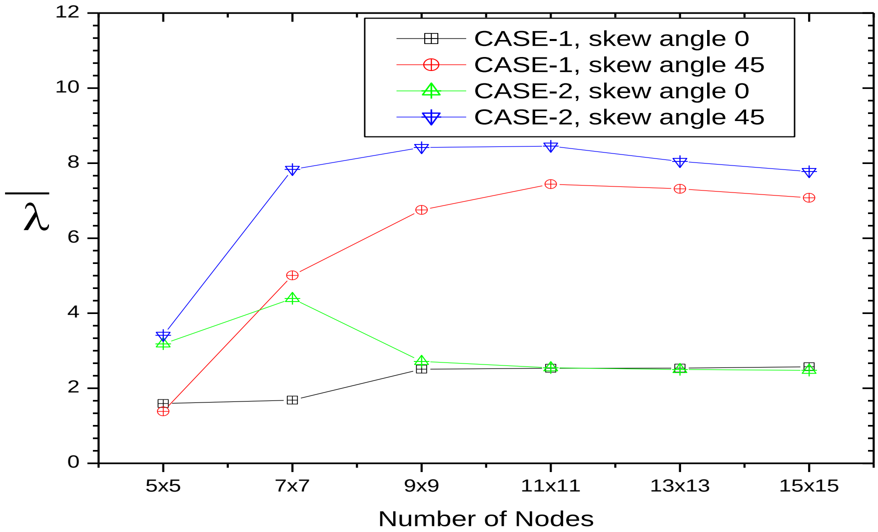

Figure 4. Convergence of Buckling parameter $\overline{\lambda}$

Table 3 presents the comparison of present results with Pandit et al.[42] and Srinivas and Rao [43].

Present results are in good agreement with published results. Further, the effect of the span-to-thickness ratio with the variation of Rf is studied.

The results obtained for the fundamental frequency parameter and buckling parameter are obtained and shown in Fig. 5 and Fig. 6 respectively. It is observed that with increase in span to thickness ratio fundamental frequency parameter decreases while fundamental buckling parameter increases, and its effect is negligible after b/h=40. It can be also seen that with increase in Rf fundamental frequency parameter increases while fundamental buckling parameter decreases.

Table 3. Frequency parameter of a square sandwich plate of orthotropic layers

References	Rf				
	1	2	5	10	15
Pandit et al.[42]	4.7283	5.6871	7.6933	9.7870	11.1816
Srinivas and Rao [43]	4.7419	5.7041	7.7148	9.8104	11.2034
Present	4.7544	5.7134	7.7232	9.8979	11.2123

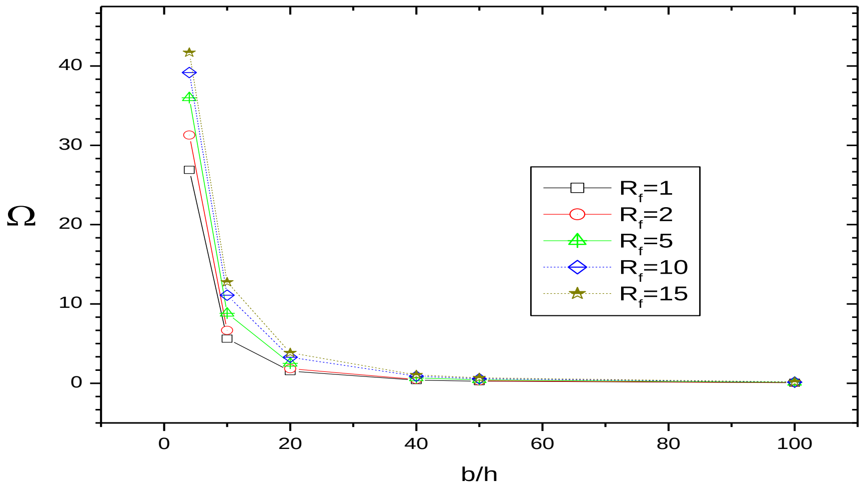

Figure5. Effect of span-to-thickness ratio on frequency parameter Ω for different Rf [ψ =30]

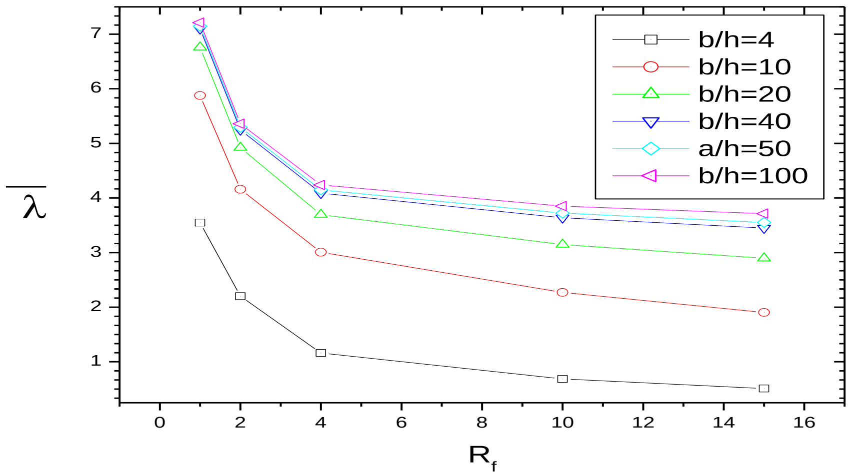

Figure6. Effect of span-to-thickness ratio on buckling parameter λ for different Rf [ψ =30]

The effect of core to face thickness ratio with skew angles are also considered. Results obtained are presented in Table 4 and Table 5 for vibration and buckling analysis respectively. With increase in skew angle both the parameters increases while with increase in core to face thickness ratio it decreases.

Table 4. Frequency parameter Ω for different skew angles (ψ)[Rf =10, b/h==20]

Core-to-face thickness ratio	ψ				
	0	15	30	45	60
2	3.5614	3.6574	4.1697	5.5115	9.2500
4	3.2055	3.2555	3.6937	4.8455	8.1457
6	2.9469	2.9916	3.3934	4.4501	7.4817
8	2.7547	2.8052	3.1844	4.1827	7.0308
10	2.6046	2.6615	3.0253	3.9822	6.6909
12	2.4834	2.5452	2.8974	3.8224	6.4186

Table 5. Buckling parameter $\overline{\lambda}$ for different skew angles (ψ)[Rf =10, b/h==20]

Core-to-face thickness ratio	ψ				
	0	15	30	45	60
2	3.7971	4.0713	5.3611	8.4687	15.1909
4	3.0755	3.1932	4.1918	6.4315	11.7088
6	2.5993	2.6951	3.5371	5.4216	9.8650
8	2.2714	2.3775	3.1181	4.8068	8.7795
10	2.0308	2.1472	2.8174	4.3786	7.9987
12	1.8462	1.9685	2.5867	4.0555	7.3618

Further, the effect of boundary conditions with aspect ratio are also considered presented in Fig. 7. It is observed that with increase in aspect ratio, the buckling parameter decreases for both the cases.

The effect of skew angle for different modes are presented in Table 6 and Table 7 for vibration and buckling analysis respectively.

Table 8 shows the results for buckling parameter with different skew angles and in-plane loadings of all edges simply supported.

Fig. 8 and Fig. 9 shows the contours of different mode shapes. Present methodology shows the capability to obtain the contours for different mode shapes.

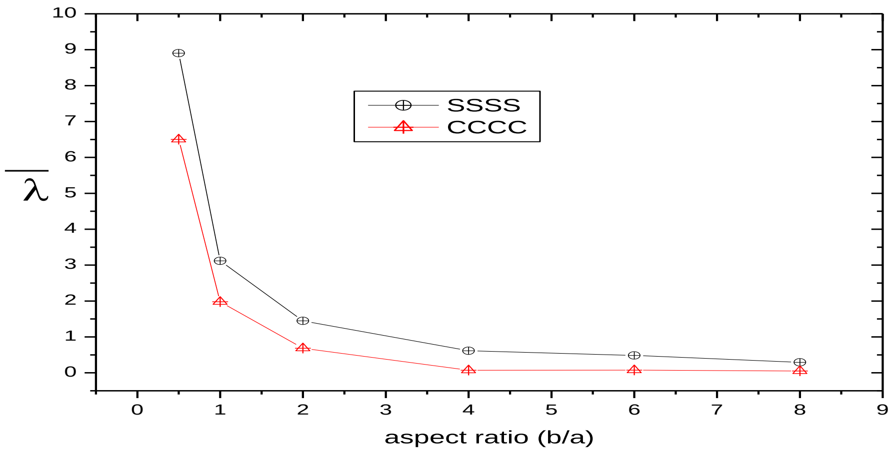

Figure7. Effect of boundary conditions of buckling parameter λ

Table 6. Frequency parameter Ω for different skew angle (ψ)[Rf =10, b/h==20]

Modes	Skew angle			
	0	30	60	75
1	1.9115	2.0409	3.7674	12.2436
2	2.7678	3.0178	5.4304	14.3424
3	4.1911	4.7656	7.8702	17.0028
4	6.1185	5.5724	9.4601	19.8703
5	6.2410	6.3289	11.0003	23.4864
6	7.0057	6.5226	12.0317	25.7191
7	8.2896	8.2046	12.0317	30.6092
8	8.5213	8.7674	13.8923	30.6092

Table 7. Buckling parameter λ for different skew angle (ψ)[Rf =10, b/h==20]

Modes	Skew angle			
	0	30	60	75
1	1.0920	1.4481	4.8325	14.5690
2	2.2901	2.3145	5.9576	14.6059
3	2.9371	2.8061	6.7724	15.8205
4	3.6965	2.9614	7.3035	15.8245
5	4.9850	3.9249	8.1120	17.1792
6	5.1586	4.4415	8.8965	18.3184
7	5.2388	5.4227	9.5923	20.1938
8	5.5972	5.6897	10.5197	20.4136

Table 8. Buckling parameter λ for different square skew angle (ψ)[Rf =5, b/h==10, SSSS]

CASE		Skew angle				
		0	15	30	45	75
	Uniaxial	2.2580	1.5514	2.7289	3.3884	12.0979
CASE-1	Biaxial	1.1293	0.8432	1.0850	1.2930	4.6472
	Pure Shear	3.8800	3.8396	2.5300	2.2732	11.8956
	Uniaxial	2.2680	2.3005	3.0076	3.3884	12.2728
CASE-2	Biaxial	1.1341	1.1118	1.1405	1.2439	4.9167
	Pure Shear	3.8412	3.4927	2.5288	1.9828	12.5058

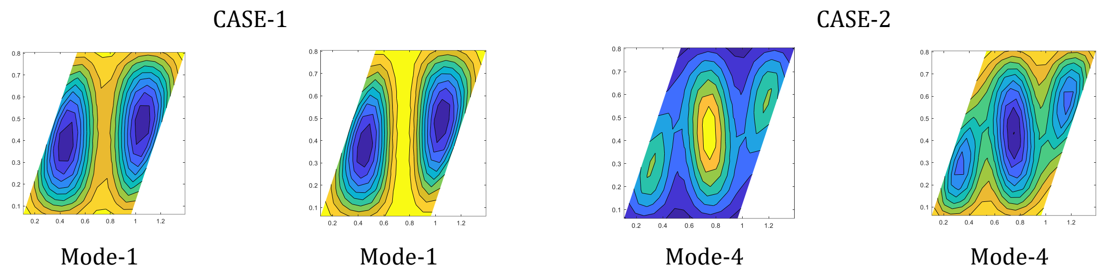

Figure 8. Different mode frequency of sandwich plate [Rf=5, hc/hf=8, ψ =30]

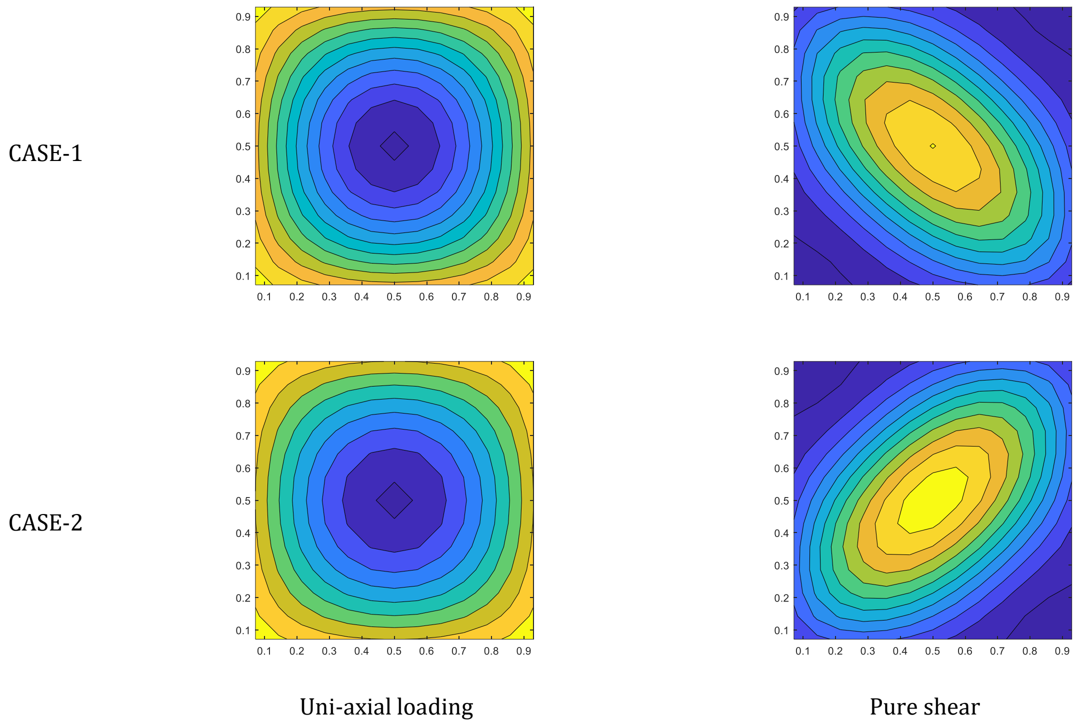

Figure 9. Different modes λ of the sandwich plate under in-plane loading conditions [Rf=5, hc/hf=8, ψ =30]

5. Conclusions

In this study, the buckling and free vibration characteristics of laminated sandwich plates are investigated using the HSDT.

- The present solution methodology is easy to implement for acceptable results.
- With an increase of span to thickness ratio, the frequency parameter decreases and after a/h =40, its effect is negligible.
- With an increase in core-to-face thickness ratio, frequency and buckling parameter decreases.
- With an increase in skew angle, the frequency parameter as well as the buckling parameter increases.
- There is a considerable effect on results with node distribution on boundaries.
- New results for vibration and buckling analysis of skew sandwich plates have been produced, which can be used for validation by other research groups.

The results of this study can be useful for optimizing the design and performance of laminated sandwich plates for various applications, such as aerospace structures, marine structures, and civil engineering structures.

By understanding the buckling and free vibration behavior of these plates, engineers can design structures that can withstand expected loads and vibrations, thereby increasing the safety and reliability of the structures.

Conflicts of Interest

The author declares that there is no conflict of interest regarding the publication of this manuscript. In addition, the authors have entirely observed the ethical issues, including plagiarism, informed consent, misconduct, data fabrication and/or falsification, double publication and/or submission, and redundancy.

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