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Effect of bi-linear thickness variation on vibration of the non-homogeneous orthotropic (S-C-S-C) triangular plate with the bi-linear temperature variations

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ABSTRACT-

In present paper a simple model is presented to study the effect of non-homogeneity on vibration of triangular plate with bilinear thickness variation. Thermal Induced vibration of these plates has been taken as two dimensional temperature distributions. Poisson ratio is assumed to vary bilinearily for non-homogeneity of the plate material. Using separation of variables method, the governing differential equation has been solved. The time period corresponding to the first two modes of vibration has been calculated for a simply supported-clamped triangular plate for various values of aspect ratio, thermal constant, taper constant and skew angle.

Keywords- Vibration, non-homogeneous, triangular plate, bilinear thickness, thermal gradient.

Introduction

Plates with varying shapes, sizes, and thicknesses are increasingly valuable in mechanical and engineering applications. By strategically varying their thickness, these plates can accommodate a broader range of performance requirements compared to plates of uniform thickness. In advanced fields such as aerospace, marine engineering, and optical systems, combining non-homogeneous materials with non-uniform thickness in structural components not only reduces weight and size but also meets the need for high strength and performance.

Leissa published a comprehensive monograph on the vibration behavior of various plate types, providing extensive numerical data on recurrence parameters under different boundary conditions. **Narinder Kaur** explored the effect of thermal gradients on the vibration of triangular plates, assuming bilinear thickness tapering and a linear temperature gradient in one direction. **Gorman** contributed an analytical approach for studying the vibration of simply supported right triangular plates, specifically to validate eigen values for the first four vibration modes. His method is applicable to right triangular plates with different boundary conditions. Using a finite element method, **Mirza and Bijlani** examined the vibration of cantilevered triangular plates with variable thickness. They also analyzed natural frequencies and mode shapes for various combinations of four non-dimensional geometric parameters, particularly aspect ratios and thickness ratios. The frequencies were categorized across different cases, with representative mode shapes illustrated for select configurations. **Gorman** further studied the vibration of right triangular plates under all possible clamped boundary conditions, offering detailed results for mode shapes and frequencies. Using a modified superposition method, he also evaluated the natural frequencies and mode shapes for the first four vibration modes of plates with a wide range of aspect ratios. **Leissa and Jaber** employed the Ritz method to perform a comprehensive study of free vibration in triangular plates. They modeled displacement functions using algebraic polynomials and presented the first six natural frequencies and nodal patterns for 17 different triangular plate configurations, obtained by varying side length ratios. **Liew and Chiam** explored free vibration of isotropic and symmetrically laminated composite plates from the generic triangular plate using the Rayleigh – Ritz approach. The scientists looked at how the plate's natural frequencies were affected by geometry, material qualities, and lamination. **Singh and Hassan** utilised the Rayleigh – Ritz approach to develop numerical solutions to the vibration issue of triangular plates with uniform and arbitrarily changing thicknesses for varied boundary conditions. **Sakiyama and Haung** demonstrated a free vibration analysis of a right triangular plate with various thicknesses and boundary conditions. The authors discovered that the triangular plate was a non-uniformly thick rectangle plate. Based on an exact theory of three-dimensional elasticity, **Cheung and Zhou** investigated free vibration of triangular plates with cantilevered and entirely free isosceles. From the strain energy and the kinetic energy of the plate, the Ritz approach is utilized to get the equation for one's own frequency. **Chakarvarty** offered a wealth of data on the vibration qualities of several plate types under varied boundary circumstances. The author presented accurate data for linear vibration of elastic plates under various boundary conditions. **Zhang and Li** proposed a method for analyzing the vibration of randomly produced triangular plates with elastically bonded edges. The displacement function is written as a two-dimensional Fourier cosine series, with additional one-dimensional series added to increase the convergence and accuracy of the displacement solution. **Chaudhary and Falak** published a paper on various materials on a laminated triangular plate with free-clamped boundary conditions. The study was carried out on the isotropic right triangular plate and the symmetrically laminated / composite triangular plate. The impact of different physical and geometric characteristics on the natural frequencies of the triangular equilateral plate exposed to classical boundary conditions were studied by **Pradhan and Chakraverty**. To acquire the problem of own value in question, the numerical modelling is done using the Rayleigh–Ritz approach. In both tabular and visual representations, **Khanna and Kaur** explored the first two modes of natural frequency parameters for the vibration of a tapered rectangular visco-elastic isotropic plate under various heat circumstances. The Rayleigh–Ritz approach was used to investigate the influence of non-homogeneity on free vibrations of a rectangular plate. **Sharma and Bensal** studied the Effect of vibration on tapered triangular plate with simply supported boundary condition under thermal condition

A non-homogeneous orthotropic triangular plate of various thicknesses in two dimensions is the focus of this investigation into the effects of a bi-linear fluctuating temperature on vibration. In some cases, it's taken as read that the plate is simply supported-clamped (S-C-S-C). The frequencies of the first and second modes of vibration are calculated for different values of the tapering constant and the temperature gradient.

Geometry of Triangular Plate

Let us consider about the u, v, and w depicted in Figure-1, which together define a visco-elastics triangular plate OPQ :

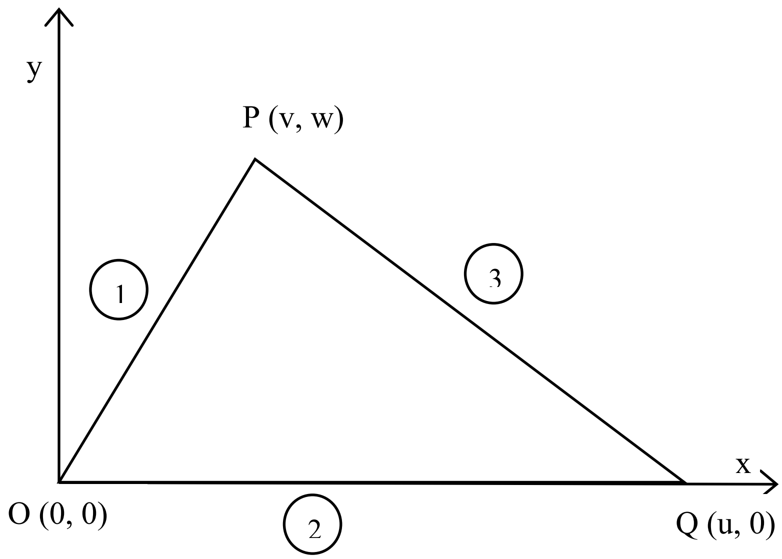

Figure-1. A general triangular domain

Then, as shown in Figure-2, the generic triangle T_G is transformed into a right-angle triangular plate T_S by using the following transformation [13].

$$\begin{cases} x = u\xi + v\eta \\ y = w\eta \end{cases} \tag{1.1}$$

$$\begin{cases} \xi = \dfrac{1}{u}\left(x - \left(\dfrac{b}{c} \right) y \right) \\ \eta = \dfrac{y}{w} \end{cases} \tag{1.2}$$

From above transformation, three points of the triangular plate in x-y plane i.e. O. P and Q map on three points in ξ-η plane i.e. O, P_1 and Q_1 as follows :

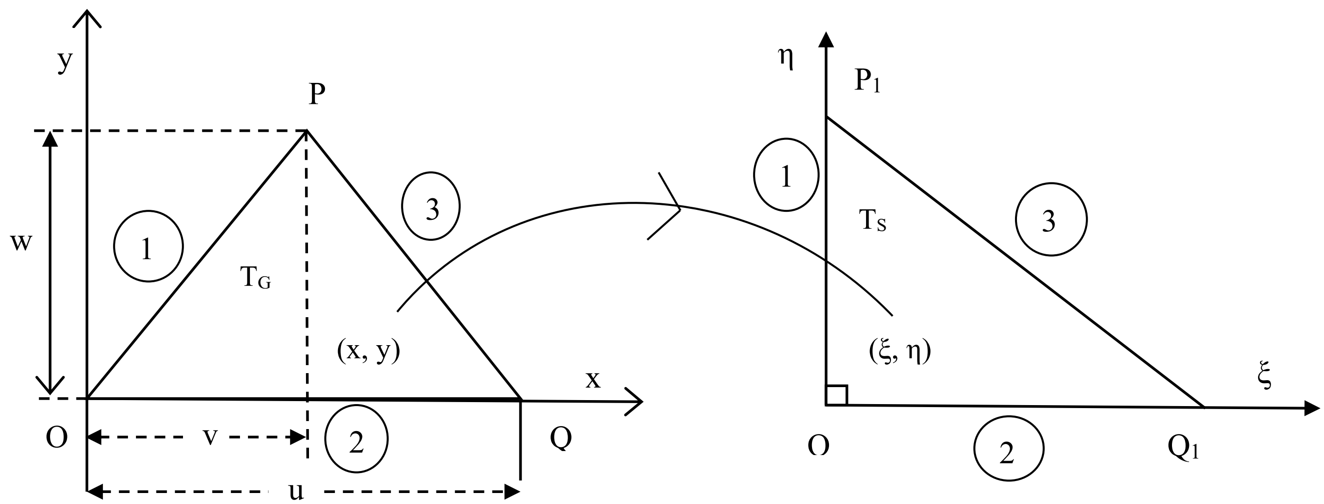

Figure-2

Analysis of Equation of motion

The differential equation of motion for an isotropic triangular plate (OPQ) in rectangular dimensions is :

$$\tilde{D}\begin{bmatrix} D_1\left(w_{xxxx}+2w_{xxyy}+w_{yyyy}\right)+2D_{1_x}\left(w_{xxx}+w_{xyy}\right)+2D_{1_y}\left(w_{yyy}+w_{yxx}\right) \\ +D_{1_{xx}}\left(w_{xx}+\nu w_{yy}\right)+D_{1_{yy}}\left(w_{yy}+\nu w_{xx}\right)+2(1-\nu)D_{1_{xy}}w_{xy} \end{bmatrix}+\rho h w_{tt}=0 \qquad (1.3)$$

As the product of the deflection function and the time function, the deflection of the triangular plate may be expressed as :

$$w(x,y,t)=W(x,y)\times T(t) \qquad (1.4)$$

Deflection function in *x* and *y* *W(x, y)* and *T (t)* is a time function.

Using equation (1.4) into equation (1.3), we obtain

$$\left[\begin{array}{l} D_1\left(W_{xxxx}+2W_{xxyy}+W_{yyyy}\right)+2D_{1_x}\left(W_{xxx}+W_{xyy}\right)+2D_{1_y}\left(W_{yyy}+W_{yxx}\right) \\ +D_{1_{xx}}\left(W_{xx}+\nu W_{yy}\right)+D_{1_{yy}}\left(W_{yy}+\nu W_{xx}\right)+2(1-\nu)D_{1_{xy}}W_{xy} \end{array}\right] \Bigg/ \rho hW = -T_{tt} \Big/ \tilde{D}T \qquad (1.5)$$

Assuming some constant, denoted by p^2, on both sides of (1.5), we get a solution,

$$\left[\begin{array}{l} D_1\left(W_{xxxx}+2W_{xxyy}+W_{yyyy}\right)+2D_{1_x}\left(W_{xxx}+W_{xyy}\right)+2D_{1_y}\left(W_{yyy}+W_{yxx}\right) \\ +D_{1_{xx}}\left(W_{xx}+\nu W_{yy}\right)+D_{1_{yy}}\left(W_{yy}+\nu W_{xx}\right)+2(1-\nu)D_{1_{xy}}W_{xy} \end{array}\right] - \rho p^2 hW = 0 \qquad (1.6)$$

and

$$T_{tt} + p^2 \tilde{D} T = 0 \qquad (1.7)$$

For a triangular plate, the time function and differential equation of motion are given in Equations (1.6) and (1.7), respectively.

As used here, D_1 denotes the flexural stiffness of a triangular plate, as defined by ,

$$D_1 = \frac{Eh^3}{12(1-\nu^2)} \qquad (1.8)$$

where E is elasticity modulus, ν is the Poisson's ratio.

The frequency parameter is then calculated using the differential equations in the Rayleigh-Ritz technique.

Rayleigh-Ritz Method in Triangular Plates

In attempt to address the frequency equation, the Rayleigh-Ritz method is used. In this strategy, which is based exclusively on the law of conservation of energy, the maximum strain energy (P_E) must be equal to the maximum kinetic energy (K_E) :

Mathematically,

$$\delta(P_E - K_E) = 0 \qquad (1.9)$$

where

$$K_E = \frac{1}{2}\rho\omega^2 \int_{x_1}^{x_2}\int_{y_1}^{y_2} hW^2 acdydx \tag{1.10}$$

and

$$P_E = \frac{1}{2}\int_{x_1}^{x_2}\int_{y_1}^{y_2} D_1\left[\left(W_{xx}\right)^2 + \left(W_{yy}\right)^2 + 2\nu W_{xx}W_{yy} + 2(1-\nu)(W_{xy})^2\right]dydx \tag{1.11}$$

On using equation (1.1) in equations (1.10) and (1.11), new equation becomes:

$$K_E{}^* = \frac{1}{2}\rho\omega^2 \int_0^1\int_0^{1-\xi} hW^2 acd\eta d\xi \tag{1.12}$$

and

$$P_E{}^* = \frac{1}{2}\int_0^1\int_0^{1-\zeta} D_1\left[\begin{matrix}\left(\frac{W_{\xi\xi}}{a^4}\right)^2 + \left(\frac{b^2}{a^2c^2}W_{\xi\xi} + \frac{1}{c^2}W_{\eta\eta} - \frac{2b}{ac^2}W_{\xi\eta}\right)^2 + 2(1-\nu)\left(-\frac{b}{a^2c}W_{\xi\xi} + \frac{1}{ac}W_{\xi\eta}\right)^2 \\ +2\nu\left(\frac{W_{\xi\xi}}{a^2}\right)\left(\frac{b^2}{a^2c^2}W_{\xi\xi} + \frac{1}{c^2}W\eta\eta - \frac{2b}{ac^2}W_{\xi\eta}\right)\end{matrix}\right](ac)d\psi d\zeta \tag{1.13}$$

Limitations

Due to the extensive nature of the subject of plate vibration, the author assumed minimal restrictions in order to evaluate the vibrational behaviour of triangular plates under the impact of different plate parameters.

(i) Limitation for thickness variation

As part of this study, the author included a hypothetical change to the triangular plate's thickness. The thickness of the triangular plate is considered to change bi-linear

in two dimensions, i.e.

$$h = h_o(1-\beta_1\xi)(1-\beta_2\eta) \tag{1.14}$$

Where h_0 is the thickness of the triangular plate at $x=0$, and β_1, β_2 are tapering parameters.

(ii) Limitation for temperature variation

In the triangular plate, vibrational qualities are temperature-dependent. Any changes in the structure's features due to thermal effect may be noticed through change in the frequency of vibration

$$\tau = \tau_0 (1-\xi)(1-\eta) \tag{1.15}$$

where τ represents the temperature difference from the reference temperature at any point on the triangular plate, and τ_0 denotes the temperature difference from the reference temperature at ξ=0.

Specifically, the formula for determining E, the elasticity modulus, is as follows :

$$E = E_0\left(1-\gamma\tau\right) \tag{1.16}$$

where E_0 is the Young's modulus at $\tau = 0$ and γ is called slope of variation.

New equation obtained by combining (1.15), (1.16),

$$E = E_0\left(1-\alpha\{1-\xi\}\{1-\eta\}\right) \tag{1.17}$$

where $\alpha = \gamma\tau_0, \left(0 \le \alpha < 1\right)$ is called thermal gradient.

From the value of *h* and *E*, the expression of the flexural rigidity (D_1) becomes:

$$D_1 = \frac{E_0\left(1-\alpha\{1-\xi\}\{1-\eta\}\right)(h_o)^3(1-\beta_1\xi)^3(1-\beta_2\eta)^3}{12\left(1-\nu^2\right)} \tag{1.18}$$

Here,

$$W(\xi,\eta) = \left[\xi^{p}\eta^{q}\left(1-\xi-\eta\right)^{r}\right]\left[C_1 + C_2\left(\xi\right)\left(\eta\right)\left(1-\xi-\eta\right)\right] \quad (1.19)$$

is deflection function and provide a full set of necessary boundary conditions for the triangular plate's edges.

Superscripted p, q, and r in equation establish the boundary conditions along the plate's sides (1.19). Depending on whether the border at ξ=0 is free, merely supported, or clamped, these values may be either 0 or 2. Similar to how q and r govern the top and bottom boundaries, they provide the criteria for the sides :

The author made the assumption that the triangular plate's border is S-C-S-C along all three sides.

Deflection function $W(\xi, \eta)$ is taken as:

$$W(\xi, \eta) = [(\xi)(\eta)^2(1-\xi-\eta^2)][C_1 + C_2(\xi)(\eta)(1-\xi-\eta)] \quad (1.20)$$

where C_1 and C_2 are arbitrary constants.

Solution of frequency parameters

A new equation is obtained by substituting (1.14), (1.17), (1.18), and (1.20) into (1.12), and (1.11), respectively:

$$K_E^{**} = \frac{1}{2}\rho h_0 w^2 \int_0^1 \int_0^{1-\xi} (1-\beta_1\xi)(1-\beta_2\eta) W^2 ac \, d\eta d\xi \quad (1.21)$$

$$P_E^{**} = \frac{E_0 {h_0}^3 1}{24(1-v^2)a^4} \int_0^1 \int_0^{1-\xi} (1-\alpha\{1-\xi\}\{1-\eta\})[(1-\beta_1\xi)(1-\beta_2\eta)]^3 [A_1 W_{\xi\xi}^2 + W_{\eta\eta}^2 + A_2 W_{\xi\eta}^2 + A_3 W_{\xi\xi}W_{\eta\eta} + A_4 W_{\xi\xi}W_{\xi\eta} + A_5 W_{\eta\eta}W_{\xi\eta}](ac)d\eta d\xi \quad (1.22)$$

Here,

$A_1 = (1+\theta^2)^2$
$A_2 = 2\mu^{-2}(2\theta^2 + 1 - \nu)$
$A_3 = 2\mu^{-2}(\nu + \theta^2)$
$A_4 = -4\theta\mu^{-1}(1+\theta^2)$
$A_5 = -4\theta\mu^{-3}$

On using equations (1.21) and (1.22) in equation (1.9), we get:

$$\delta(P'_E - \lambda^2 K'_E) = 0 \tag{1.23}$$

where

$$K'_E = \int_0^1 \int_0^{1-\xi} (1-\beta_1\xi)(1-\beta_2\eta) W^2 ac \, d\eta d\xi \tag{1.24}$$

And

$$P'_E = \int_0^1 \int_0^{1-\xi} (1-\alpha\{1-\xi\}\{1-\eta\})[(1-\beta_1\xi)(1-\beta_2\eta)]^3 \quad [A_1 \quad W_{\xi\xi}^2 + W_{\eta\eta}^2 + A_2 W_{\xi\eta}^2 + A_3 W_{\xi\xi} W_{\eta\eta} + A_4 W_{\xi\xi} W_{\xi\eta} + A_5 W_{\eta\eta} W_{\xi\eta}](ac) d\eta d\xi \tag{1.25}$$

where $\lambda^2 = \dfrac{\rho\omega^2 h_0 a^2}{D_0}$ is the required expression of frequency parameter.

After W is substituted, the preceding formula has two variables, the constants C_1 and C_2, which are determined by the following:

$$\frac{\partial(P'_E - \lambda^2 K'_E)}{\partial C_1} = 0, \frac{\partial(P'_E - \lambda^2 K'_E)}{\partial C_2} = 0 \tag{1.26}$$

Solving Eq. (1.26), we have frequency equation

$$b_{q1}C_1 + b_{q1}C_2 = 0 \text{, q = 1, 2} \tag{1.27}$$

where the temperature gradient and the taper parameter of the triangular plate make up b_{q1} and b_{q2}, respectively. The determinant of the coefficients in equation (1.27) for a non-zero solution, must be zero. The resulting frequency equation is:

$$\begin{vmatrix} b_{11} & b_{12} \\ b_{21} & b_{22} \end{vmatrix} = 0 \qquad (1.28)$$

Results and Discussion

By putting in values for the thermal gradient and taper constant, we can calculate the frequency parameter λ for each combination. For different values of the thermal constant (λ), the taper constants (β_1, β_2), and the angle θ, the frequency parameter corresponding to the first two modes of vibration of a S-C-S-C triangular plate has been determined. The computation uses the following parameters for the plates:

$$E_0 = 7.08 \times 10^{10} \frac{N}{M^2},$$

$$\rho = 2.80 \times 10^3 \frac{kg}{M^3},$$

$$v = 0.345, \; h_0 = 0.01M.$$

These results are summarized in tables (1.1) to (1.6) and plotted in figures (1.1) to (1.6)

Tables 1.1-1.3 includes the frequency parameter of S-C-S-C triangular plate for a range of heat gradient (α), taper constant (β_1, β_2), and fixed (θ=0.0, μ=1.0) values of the first two modes of vibration. From the data presented, it is clear that the frequency parameter (λ) becomes larger when the values of the thermal gradient and taper constants decrease.

Tables 1.4-1.6 gives the value of frequency parameter of S-C-S-C plate at the standard value of $\theta=1/\sqrt{3}$, μ = 1.5 and different combination of α, β_1, and β_2. From the tables and figures, it has been seen that the frequency parameter (λ) decreases when the value of thermal gradient and taper constants increases.

Table 1.1-1.3

Frequency parameter (λ) for First two Mode of S-C-S-C triangular plate with varying values of taper constants (β_1, β_2), thermal gradient (α) = 0.0, θ = 0.0 and μ = 1.0

Table 1.1 includes the frequency parameter of S-C-S-C triangular plate for various values of temperature gradient (α) and the fixed values of θ=0.0, μ=1.0 for the first two modes of vibration. From the data presented, it is clear that when the values of the thermal gradient are raised, the frequency parameter (λ) falls.

Table 1.1

α	$\beta_1 = \beta_2 = 0.2$		$\beta_1 = \beta_2 = 0.4$		$\beta_1 = \beta_2 = 0.6$	
	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	40.13	179.73	32.56	151.94	25.70	126.65
0.2	39.05	173.38	31.59	146.34	24.82	121.71
0.4	37.94	166.80	30.60	140.52	23.91	116.56
0.6	36.79	159.95	29.57	134.45	22.97	111.16
0.8	35.61	152.79	28.50	128.09	21.98	105.50

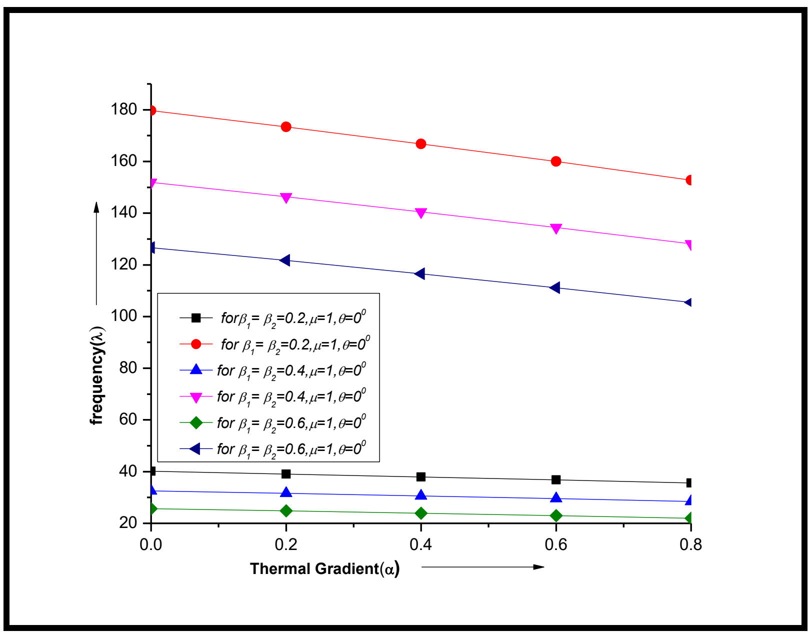

Figure 1.1 Variation of frequency with thermal gradient for triangular plate

Table 1.2 includes the frequency parameter of S-C-S-C triangular plate for various values of taper constant (β_1) and the fixed values of θ=0.0, μ=1.0 for the first two modes of vibration. From the data presented, it is clear that when the values of the thermal gradient are raised, the frequency parameter (λ) falls.

Table 1.2

β_1	$\alpha = \beta_2 = 0.2$		$\alpha = \beta_2 = 0.4$		$\alpha = \beta_2 = 0.6$	
	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	37.46	182.26	31.26	153.55	25.61	127.15
0.2	36.12	179.22	30.03	150.17	24.48	123.50
0.4	34.83	177.25	28.76	129.31	23.38	120.81
0.6	33.58	176.60	27.71	146.75	22.33	119.21
0.8	32.41	174.43	26.64	145.96	21.35	118.84

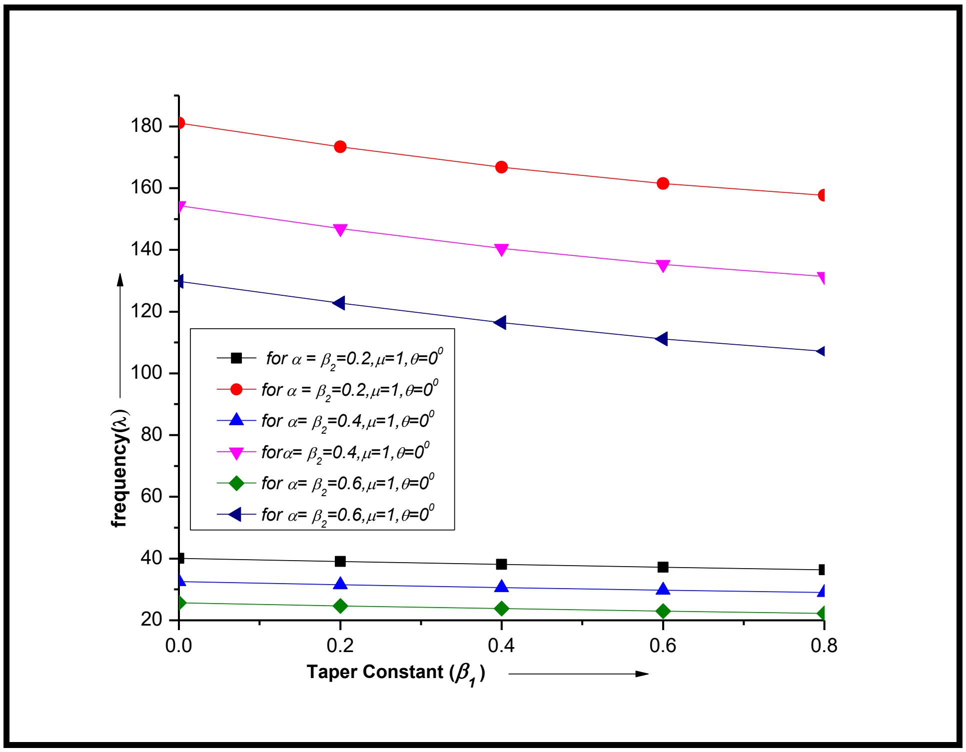

Figure 1.2 Variation of frequency with taper constant (β_1) for triangular plate

Table 1.3 includes the frequency parameter of S-C-S-C triangular plate for various values of taper constant (β_2) and the fixed values of θ=0.0, μ=1.0 for the first two modes of vibration. From the data presented, it is clear that when the values of the thermal gradient are raised, the frequency parameter (λ) falls.

Table 1.3

β_2	$\alpha = \beta_1 = 0.2$		$\alpha = \beta_1 = 0.4$		$\alpha = \beta_1 = 0.6$	
	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	45.92	194.79	43.70	181.12	41.57	168.83
0.2	39.05	173.38	36.99	160.38	35.03	148.65
0.4	32.54	152.91	30.60	140.52	28.76	129.31
0.6	26.65	133.76	24.75	121.93	22.97	111.16
0.8	21.83	116.54	19.86	105.19	18.04	94.82

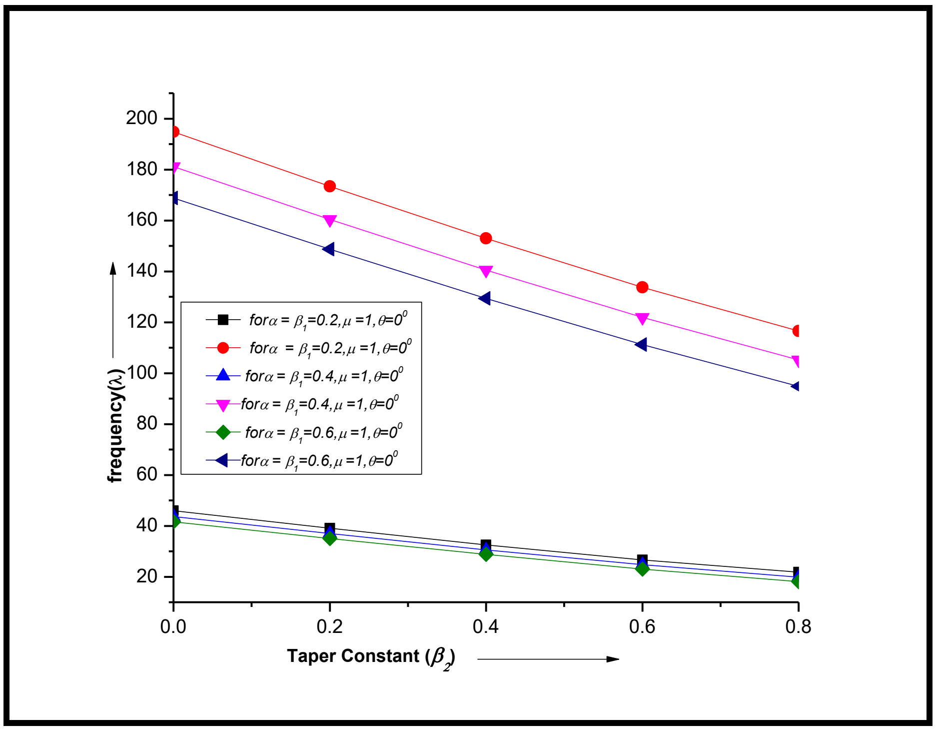

Figure 1.3 Variation of frequency with taper constant (β_2) for triangular plate

Table 1.4-1.6

Frequency parameter (λ) for First two Mode of S-C-S-C triangular plate with varying values of taper constants (β_1, β_2), thermal gradient (α), $\theta = 1/\sqrt{3}$ and $\mu = 1.5$

Table 1.4 includes the value of the frequency parameter of S-C-S-C triangular plate for different combinations of thermal gradient from 0.0 to 0.8 with a common ratio of 0.2 and fixed value of θ= $1/\sqrt{3}$ and μ=1.5. From the tables and figures, it has been observed that the frequency parameter (λ) decreases on increases the value of thermal gradient (α).

Table 1.4

α	$\beta_1 = \beta_2 = 0.2$		$\beta_1 = \beta_2 = 0.4$		$\beta_1 = \beta_2 = 0.6$	
	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	34.34	186.29	31.07	160.94	25.41	137.45
0.2	36.12	179.22	29.98	154.53	24.43	131.65
0.4	34.86	171.85	28.76	129.31	23.40	125.59
0.6	33.55	164.16	27.67	140.86	22.33	119.21
0.8	32.19	156.09	18.04	94.82	21.70	112.41

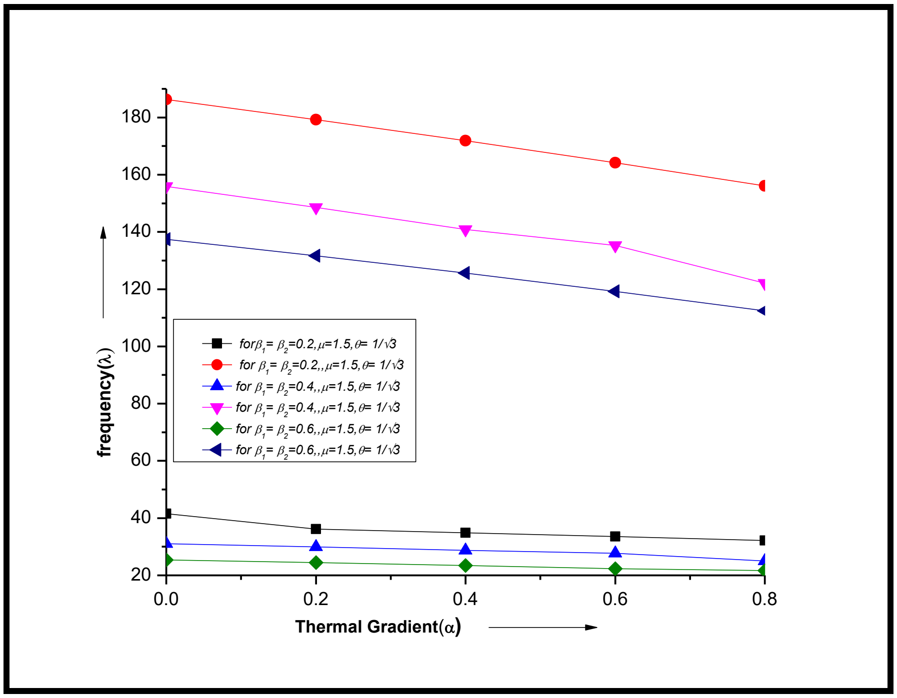

Figure 1.4 Variation of frequency with thermal gradient (α) for triangular plate

Table 1.5 includes the value of the frequency parameter of S-C-S-C triangular plate for different combinations of taper constants (β_1) from 0.0 to 0.8 with a common ratio of 0.2 and fixed value of θ= $1/\sqrt{3}$ and μ=1.5. From the tables and figures, it has been observed that the frequency parameter (λ) decreases on increases the value of taper constants (β_1).

Table 1.5

β_1	$\alpha = \beta_2 = 0.2$		$\alpha = \beta_2 = 0.4$		$\alpha = \beta_2 = 0.6$	
	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	37.46	182.26	31.26	153.55	25.61	127.15
0.2	36.12	179.22	30.03	150.17	24.48	123.50
0.4	34.83	177.25	28.76	129.31	23.38	120.81
0.6	33.58	176.60	27.71	146.75	22.33	119.21
0.8	32.41	174.43	26.64	145.96	21.35	118.84

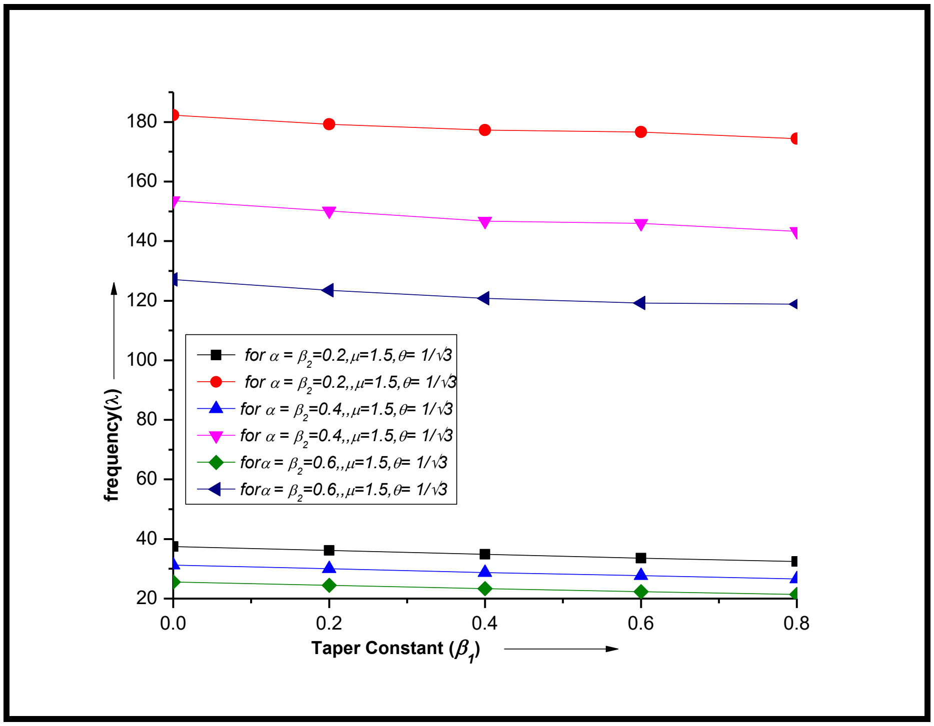

Figure 1.5 Variation of frequency with taper constants (β_1) for triangular plate

Table 1.6 includes the value of the frequency parameter of S-C-S-C triangular plate for different combinations of taper constants (β_2) thermal gradient from 0.0 to 0.8 with a common ratio of 0.2 and fixed value of $\theta = 1/\sqrt{3}$ and μ=1.5. From the tables and figures, it has been observed that the frequency parameter (λ) decreases on increases the value of taper constants (β_2).

Table 1.6

β_2	$\alpha = \beta_1 = 0.2$	$\alpha = \beta_1 = 0.4$	$\alpha = \beta_1 = 0.6$

	λ_1	λ_2	λ_1	λ_2	λ_1	λ_2
0	41.34	202.53	38.66	192.74	36.10	183.83
0.2	36.12	179.22	33.62	169.86	31.22	161.36
0.4	31.21	156.83	28.85	147.86	26.58	139.68
0.6	26.77	135.78	24.50	127.12	22.33	119.21
0.8	23.06	116.74	20.83	108.30	18.70	100.57

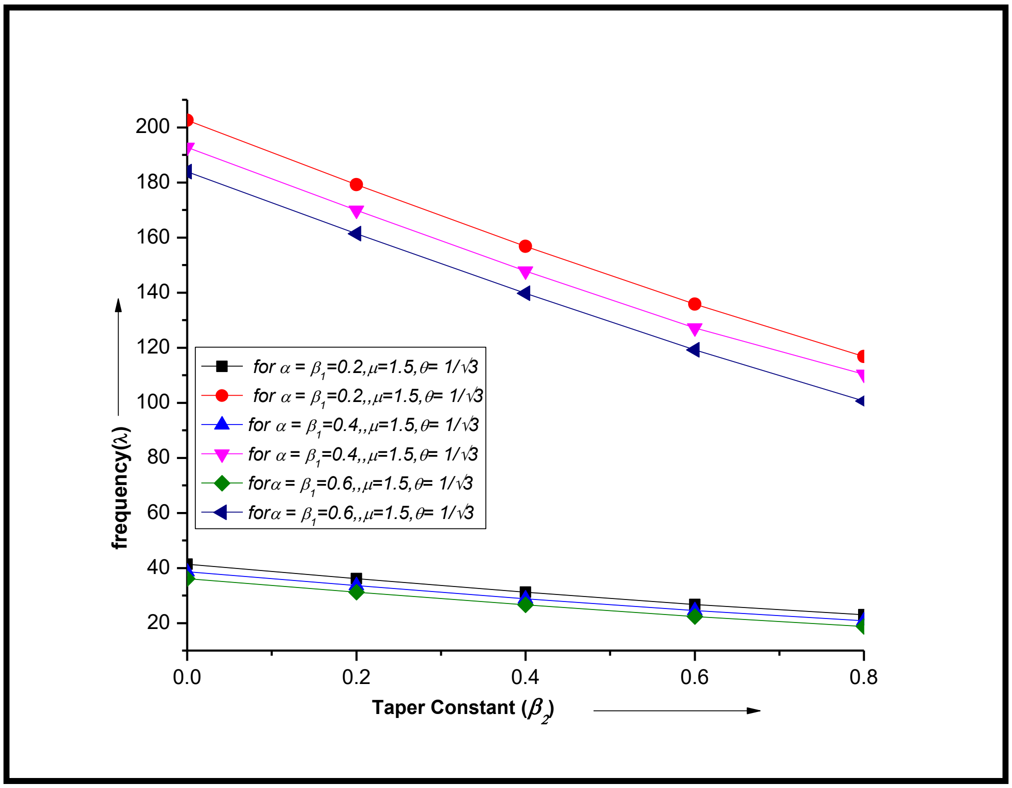

Figure 1.6 Variation of frequency with taper constants (β_2) for triangular plate

Conclusion

Based on the graphs presented, the author concluded that under non-zero thermal gradients, the frequency parameter for both vibration modes is lower than that observed under zero thermal gradients. This highlights the influence of temperature on the vibrational behavior of a triangular plate. Additionally, when comparing different plate thickness profiles, the author found that the frequency parameter reaches its maximum for plates with uniform thickness. This suggests that appropriate tapering of the plate can be used to achieve desired frequency parameter values.

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