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Comparison of Vibration Response from Various Annulated Circular Plates attached with Holes and Concentrated Stiffened Patches under Thermal Environment

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Abstract: This study investigated the comparative vibration characteristics of annulated circular plates, annulated circular concentrated stiffened plates, and annulated circular hole plates through finite element analysis (FEA), while maintaining a constant mass for the plates. The temperature is held constant across all scenarios. The analysis of these plates is conducted under both free-free and clamped-free boundary conditions. Parametric studies are performed to assess the impact of Eigen frequencies, aspect ratio, and radius ratio on various stiffened plates, with the results being thoroughly examined and outlined.
Keywords: Circular annulated plate, Concentrated stiffened plate, Plate with holes, Aspect ratio, Radii ratio, FEA.

I. INTRODUCTION

Circular annulated stiffened plates are utilized in numerous structural applications, such as in architectural design, diaphragms and deck plates for launch vehicles, turbine diaphragms, aircraft, and missiles, as well as in naval architecture, nuclear reactors, shipbuilding, automotive manufacturing, and the space shuttle. While there exists a substantial amount of research regarding the structural dynamics of circular annulated plates in varying temperature conditions, the study of vibrations in circular annulated stiffened plates subjected to thermal environments remains insufficiently investigated. Numerous researchers have proposed different theories to elucidate the vibration response of a circular annular stiffened plate in out-of-plane modes. Abbasi et al. [1] employed the differential transforms method (DTM) subjected to non-uniform axisymmetric transverse loading to perform the static analysis of a circular plate situated on a Winkler elastic foundation. Abolghasemi et al. [2] discovered the axisymmetric buckling of an annular plate utilizing an asymptotic approximation technique grounded in the perturbation method. Askari et al. [3] applied closed-form geometric equations along with the chain rule to examine the free vibration of eccentric annular plates as a novel mathematical approach. Lee et al. [4] utilized the Multipole Trefftz Method to conduct a free vibration analysis of a circular plate featuring multiple circular holes. Thakre et al. [5] explored the free vibration of circular plates with holes and cutouts through finite element analysis (FEA). Lee et al. [6] assessed the free vibration of circular plates with several circular holes using indirect Boundary Integral Equations Methods (BIEMs). Lee and Chen [7] employed Indirect BIEM and the Addition Theorem to study the free vibration of a circular plate with multiple circular holes. Ghosh S. et al. [8] utilized FEA to evaluate the influence of radius ratio and aspect ratio on both free and forced vibration of a uniform annular circular plate in a thermal environment. Lee and Singh [9] applied thin and thick plate theories to ascertain the sound radiation from out-of-plane modes of a uniformly thick annular circular plate. This article presents a study investigating the comparative vibration characteristics of annulated circular plates, annulated circular concentrated stiffened plates, and annulated circular hole plates through finite element analysis (FEA), while ensuring a constant mass for the plates. The vibration behaviour of these plates is examined specifically in relation to out-of-plane flexural modes. The temperature is maintained consistently across all scenarios. The existing literature demonstrates the flexural response of circular annulated plates but limits the exploration of vibration sensitivity in circular annulated stiffened plates at a constant temperature. Consequently, a research gap has been identified. Thus, the literature emphasizes the vibration response of out-of-plane modes of circular annulated stiffened plates using finite element analysis (FEA) within a thermal environment.

The primary aim of this research is to examine the impact of Eigen frequency on the radii and aspect ratios of various circular annulated stiffened plates within a thermally sensitive environment to vibrations. The findings are thoroughly analyzed and documented.

II. METHODOLOGY

A. Plate Modeling Equation

The natural frequency along with the mode shape of the plate during modal analysis is obtained as

$$\left([K]-\omega^2[M]\right)\psi_{mn}=0 \quad (1)$$

In the above formulation, $[M]$is said to be mass matrix and $[K]$ is said to be the stiffness matrix. The mode shape is represented by ψ_{mn} and the corresponding natural frequency of the plate is represented by **ω** denoted as rad/sec. Further, (λ^2) known as the non-dimensional frequency parameter which is obtained as

$$\lambda^2=\omega a^2\sqrt{\frac{\rho h}{D}} \quad (2)$$

Where **D** is said to be the flexure rigidity$=\frac{Eh^3}{12\left(1-\upsilon^2\right)}$, '**a**' is said to be the outer radius, '**E**' is said to be the Young's modulus of elasticity, 'υ' is said to be Poisson's ratio, 'h' is said to be the thickness of the plate and 'ρ is said to be the density of plate.

B. Plate Geometric Modelling

The vibration characteristics of the plate are examined through Finite Element Analysis (FEA). The modelling of the plate is conducted in ANSYS utilizing Plane 185, which consists of eight brick nodes, each possessing three degrees of freedom. It is assumed that the thickness remains uniform. This research employs an annular plate with an outer radius of 151.5 mm, an inner radius of 82.5 mm, and a thickness of 31.5 mm for the analysis. Various boundary conditions, including free-free and clamped-free, are applied during the plate's examination. Structural modes that occur out-of-plane are taken into account. The temperature is maintained at a constant level. The specifications of the stiffener plate and additional material properties are presented in Table 1. Figure 1 illustrates the reference and coordinate systems of the plate.

TABLE I

An annular circular stiffener plate with consistent thickness: specifications and material qualities

Dimension of the plate	**Values**
Outer radius (a) m	0.1515
Inner radius (b) m	0.0825
Radii ratio, (b/a)	0.54
Thickness ratio, (h/a),	0.21
Density, ρ (kg/m^3)	7905.9
Young's modulus, E (GPa)	218
Poisson's ratio, υ	0.305
Temperature, K	273
Diameter of hole, m	0.01
Diameter of patch, m	0.01
Temperature, T, Kelvin (K)	273

The plate is classified as a circular annular stiffened plate for both free and forced vibrations. The mass and overall volumes of the plate are kept constant. It is assumed that the thickness is uniform across all scenarios. Three distinct configurations of plates are analysed. Figures 1, 2, and 3 illustrate the coordinate system and geometry of the plate for various cases, with 'Z' representing the estimated out-of-plane mode vibration coordinates. Figure 1 depicts a simple annular circular plate. Figure 2 presents the circular annulated plate featuring configurations with 2-holes and 4-holes, while Figure 3 illustrates the annulated circular plate equipped with 2-concentrated and 4-concentrated stiffener patches. A uniform temperature of T = 2730K is consistently maintained during the investigation of vibration sensitivity in this study. Any errors due to temperature variation are disregarded in these instances. The first five structural out-of-plane modes are taken into account for the estimation of Eigen frequencies across all scenarios.

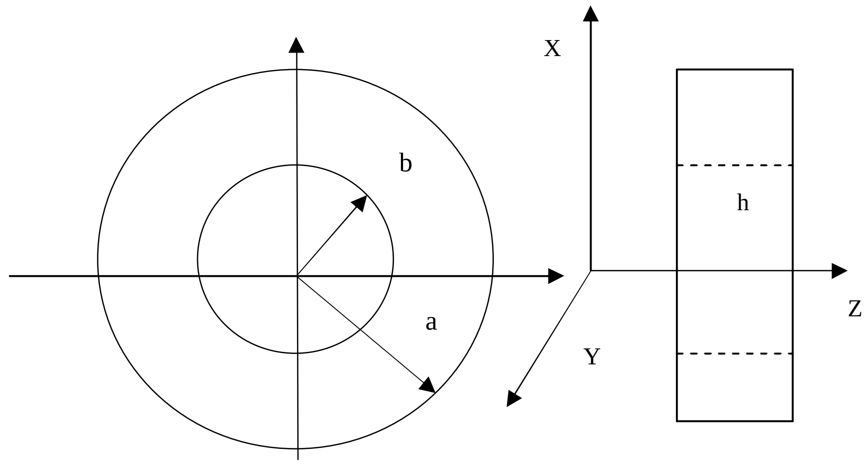

Fig.1. Coordinate and geometrical shape of annular circular plate with uniform thickness

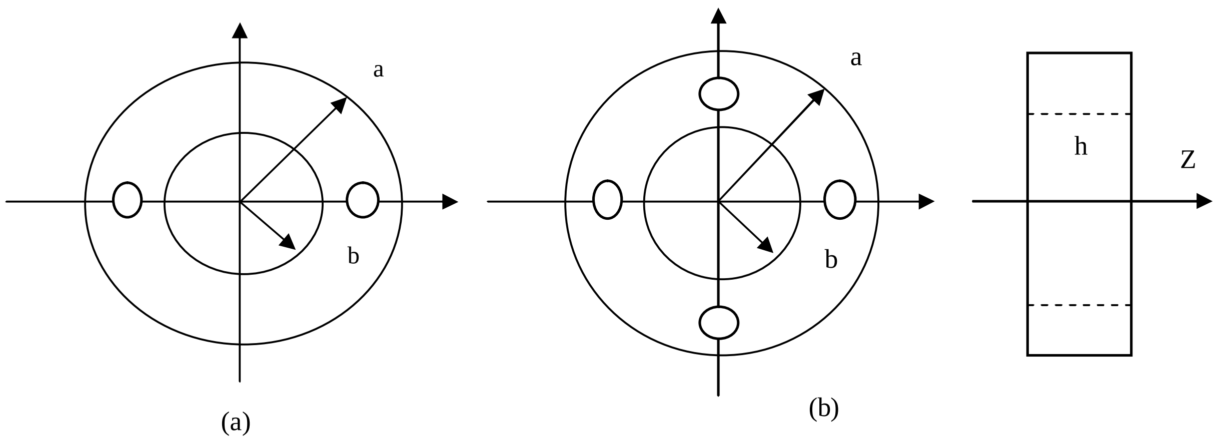

Fig. 2. Coordinate and geometrical shape of annular circular plate with varying hole layouts (a) 2-holes (b) 4-holes

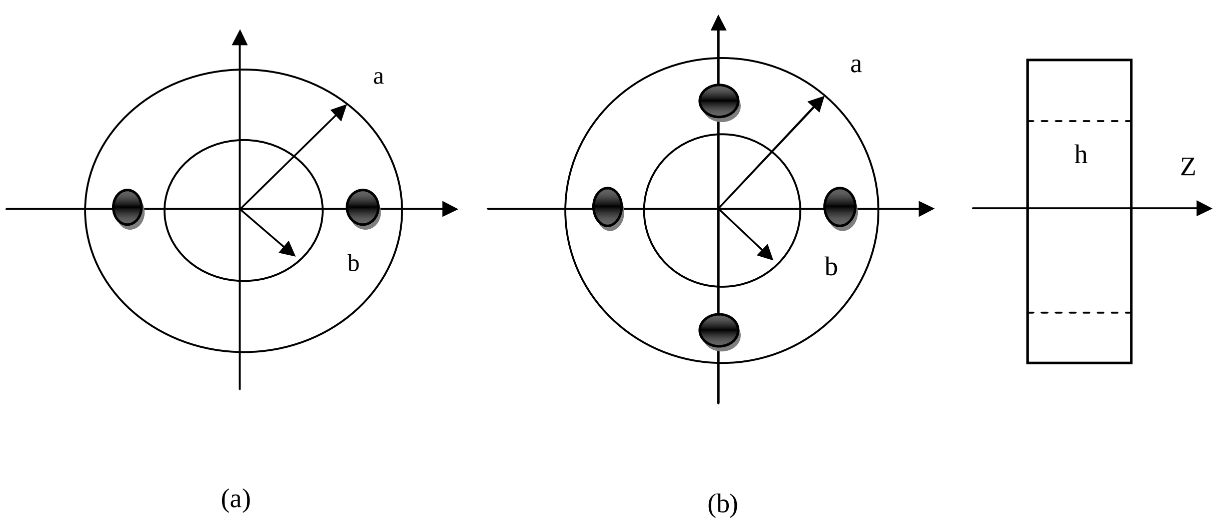

Fig. 3. Coordinate and geometrical shape of annular circular plate with varying patch layouts (a) 2-patches (b) 4-patches

C. *Validation of Study*

In this paper the Eigen frequencies (ω) of a uniform annular circular plate without stiffener is validated with the result of Lee & Singh [9]. It is clear form Table II and Table III that the result obtained in this case is similar that of the published result.

TABLE II
Comparison And Validation Of Natural Frequency (Ω) Of A Uniform Annular Circular Plate With Free-Free Boundary Condition Obtained In The Present Work With That Of Lee And Singh [9]

Plate	Mode	Frequency parameter, $\lambda^2 = \omega a^2 \sqrt{\frac{\rho h}{D}}$	
		Lee and Singh [9]	Present work
Annulated un-	(0,2)	3.82	3.83
stiffened plate	(1,0)	8.85	8.82
b/a = 0.54	(0,3)	10.59	10.02
h/a = 0.21	(1,1)	15.42	13.70

TABLE III
Comparison And Validation Of Natural Frequency (Ω) Of A Uniform Annular Circular Plate With Clamped–Free Boundary Condition Obtained In The Present Work With That Of Lee And Singh [9]

Plate	Mode	Frequency parameter, $\lambda^2 = \omega a^2 \sqrt{\frac{\rho h}{D}}$	
		Lee and Singh [2005]	Present work
Annulated un-	(0,0)	13.61	13.49
stiffened plate	(0,1)	13.43	13.50
b/a = 0.54	(0,2)	15.28	14.12
h/a = 0.21	(0,3)	16.81	16.67

III. RESULTS AND DISCUSSION

A. *Effect of Radii Ratio on Plates*

This section illustrates the impact of natural frequency on the vibration response of a uniform plate lacking a stiffener, influenced by various radius ratios and boundary conditions. The mass and volume of the plate remain constant. Tables IV - IX present the influence of different radius ratios on both uniform annulated circular plates that are un-stiffened and those that are stiffened. Tables IV and V detail how varying radius ratios affect the first five Eigen frequencies (ω) of an annulated circular un-stiffened plate under free-free and clamped-free boundary conditions. According to Table IV, it is observed that as the radius ratio increases from β = 0.1 to 0.3, the Eigen frequencies (ω) decrease with higher modes for an annulated circular un-stiffened plate under free-free boundary conditions. This phenomenon may be attributed to the reduced stiffness of the plate, which results in lower natural frequencies across various mode shapes. Conversely, for clamped-free boundary conditions, an increase in radius ratios leads to a rise in the Eigen frequency (ω) of the plate. This is due to the enhanced rigidity of the plate as the radius ratio increases, causing it to vibrate at a higher frequency. Additionally, Table V indicates that the clamped-free boundary condition exhibits greater rigidity compared to the free-free boundary condition. Consequently, in comparison to the free-free boundary condition, the plate exhibits a higher natural frequency during vibration. Tables VI and VII illustrate the influence of varying radius ratios on the first five Eigen frequencies (ω) of an annulated circular stiffened plate with holes, analyzed under both free-free and clamped-free boundary conditions. It is evident from Table VI that as the radius ratios increases from β=0.1 to 0.2, the Eigen frequency (ω) decreases with the rising modes for an annulated circular stiffened plate with holes under free-free boundary conditions. This phenomenon may be attributed to the reduced stiffness of the plate as the number of perforations increases. Conversely, for clamped-free boundary conditions, the Eigen frequency (ω) of the perforated plate increases with higher radius ratios. This is due to the fact that clamped-free vibrations demonstrate greater rigidity compared to free-free vibrations. Consequently, the plate vibrates at a higher frequency. Furthermore, Tables VI and VII indicate that an increase in the number of holes results in a decrease in the plate's mass, which subsequently leads to a lower frequency of vibration across all mode shapes. It is also revealed that as the number of holes increases, the rigidity of the plate diminishes. Therefore, a plate with four holes will vibrate at a lower frequency than one with two holes. Additionally, Tables VIII and IX present the impact of different radius ratios on the first five Eigen frequencies (ω) of an annulated circular stiffened plate with concentrated patches, evaluated under both free-free and clamped-free boundary conditions. Table VIII clearly indicates that as the radius ratios increase from $\beta = 0.1$ to 0.2, the Eigen frequency (ω) decreases with the rising modes for an annulated circular stiffened plate featuring concentrated patches under free-free boundary conditions. This phenomenon may be attributed to the reduced stiffness of the plate as the number of patches increases. Conversely, for clamped-free boundary conditions, the Eigen frequency (ω) of the plate with patches rises with increasing radius ratios. This is due to the fact that clamped-free vibration demonstrates greater rigidity compared to free-free vibration, resulting in the plate vibrating at a higher frequency. Furthermore, both Table VIII and Table IX illustrate that as the quantity of patches increases, the stiffness of the plate decreases, leading to a lower frequency of vibration across all mode shapes. It is also evident that as the number of patches increases, the rigidity of the plate diminishes. Consequently, the plate with four patches will vibrate at a lower frequency than the plate with two patches.

Figures 4-9 illustrate the comparison of the impact of various radius ratios (β) on both un-stiffened and stiffened circular plates under free-free and clamped-free boundary conditions. As shown in Figure 4, it is evident that as the radius ratio ($\beta = 0.1$) increases, the Eigen frequency rises with the increasing modes for both the un-stiffened and stiffened plates under free-free boundary conditions. Furthermore, Figure 4 indicates that the unloaded plate (annulated circular plate) achieves the highest frequency across all mode shapes, whereas the plate featuring four holes and four patches records the lowest frequency among all scenarios. In comparison to other plates, the plate with four holes is lighter and less rigid, leading to a reduced frequency for all mode shapes. The lower rigidity of the four patches also results in a diminished frequency in this instance. Figure 5 presents the comparison of natural frequencies under free-free boundary conditions with a radius ratio of 0.2. The graphics clearly demonstrate that the plate with the most holes, in contrast to patches, shows the lowest Eigen frequencies for each mode shape, while the unloaded plate (annulated circular plate) exhibits the highest natural frequencies for all mode shapes. According to Figure 6, the plate with four holes has the lowest frequency when compared to the (0, 2), (0, 1), (0, 3), and (1, 1) mode shapes. However, in the (0, 4) mode, the plate with four patches has a marginally lower natural frequency than the plate with four holes.

TABLE IV
Effect of natural frequencies ON RADII ratio AND modes for circular annular uniform plate (unloaded plate) under free-free boundary conditions

BCS	Mode	Radii Ratio		
		b/a=0.1	b/a=0.2	b/a=0.3
F-F	(0,2)	2335.3	2263.3	2198.2
	(0,1)	3601	3530	3656.7
	(0,3)	5561.6	5472.8	5466.7
	(1,1)	7510.4	7191.5	6826.1
	(0,4)	9397.6	9260.9	9395.5

TABLE V
Effect of natural frequencies ON RADII ratio AND modes for circular annular uniform plate (unloaded plate) under clamped-free boundary condition

BCS	Mode	Radii Ratio		
		b/a=0.1	b/a=0.2	b/a=0.3
C-F	(0,0)	982.1	1621.1	2374.4
	(0,1)	1519.3	1930	2581.9
	(0,2)	2404.2	2598.7	2912.2
	(0,3)	5564.6	5516.6	5760.3
	(0,4)	9407.5	9264.9	9476.9

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Figure 7 clearly illustrates that the clamped-free vibration curve is shorter than that of the free-free vibration curve. This difference arises from the greater rigidity of clamped-free boundary conditions compared to free-free conditions. Furthermore, the graph indicates that among the various plates, the unloaded plate (annulated circular plate) demonstrates the highest natural frequencies across all mode shapes, whereas the plate with four holes displays the lowest natural frequencies for all mode shapes. Figure 8 depicts a similar increase in Eigen frequencies, which rise concurrently with the radius ratio. It is apparent from the figure that the clamped-free boundary condition exerts a more significant enhancing effect than the free-free condition. Moreover, it is observed that the unloaded plate maintains the highest natural frequencies for all mode shapes in comparison to the other plates, while the four-hole plate consistently shows the lowest natural frequencies across all mode shapes. Figure 9 clearly indicates that for all mode shapes, the number of natural frequencies increases as the radius ratio progresses from 0.1 to 0.3. Additionally, the four holes and four patches exhibit the lowest sensitivity across all mode shapes.

Table VI
Effect of natural frequencies ON RADII ratio AND modes for circular annular holed plate under free-free boundary conditions

BCS	No. of Holes	Mode	Radii Ratio		
			b/a=0.1	b/a=0.2	b/a=0.3
F-F	2	(0,2)	2220	2155.7	2098.5
		(0,1)	3478	3464.4	3639
		(0,3)	5327.8	5326.4	5342.6
		(1,1)	7211.8	6925.4	6654.2
		(0,4)	8963.9	9036.1	9111.4
F-F	4	(0,2)	2051.2	1989.6	1973.9
		(0,1)	3156.8	3093.9	3269.6
		(0,3)	4885.3	4812.1	4901.3
		(1,1)	6591.7	6314.7	6077
		(0,4)	8356.5	8263	8454.4

Table VII
Effect of natural frequencies ON RADII ratio AND modes for circular annular holed plate under clamped-free boundary conditions

BCS	No. of Holes	Mode	Radii Ratio		
			b/a=0.1	b/a=0.2	b/a=0.3
C-F	2	(0,0)	437.2	814.8	1150.9
		(0,1)	1261.4	1316.1	1459.1
		(0,2)	2257.1	2247.9	2367.6
		(0,3)	5327.4	5329.7	5388.2
		(0,4)	8973.4	9075.3	9118.1
C-F	4	(0,0)	426.83	776.6	1077.4
		(0,1)	1201.5	1238.7	1365.9
		(0,2)	2056.8	2039.7	2159.7
		(0,3)	4885.3	4819.6	4951.4
		(0,4)	8361	8264.2	8469.9

Table VIII
Effect of natural frequencies ON RADII ratio AND modes for circular annular stiffened plate under free-free boundary conditions

BCS	No. of Patches	Mode	Radii Ratio		
			b/a=0.1	b/a=0.2	b/a=0.3
F-F	2	(0,2)	2250.9	2181.9	2129.9
		(0,1)	3534	3498	3649.8
		(0,3)	5396.6	5336	5423.7
		(1,1)	7308.5	6999.9	6707.9
		(0,4)	9114.7	9059.9	9236.1
F-F	4	(0,2)	2085	2037.9	1975
		(0,1)	3219.9	3187.7	3282.3
		(0,3)	4984.7	4937.4	4911.8
		(1,1)	6749.9	6440.8	6157.8
		(0,4)	8524.4	8431.5	8449.5

Table IX
Effect of natural frequencies ON RADII ratio AND modes for circular annular stiffened plate under clamped-free boundary conditions

BCS	No. of Patches	Mode	Radii Ratio		
			b/a=0.1	b/a=0.2	b/a=0.3
C-F	2	(0,0)	937.7	1595.8	2332.2
		(0,1)	1490.2	1902	2548.8
		(0,2)	2352.9	2537	3076.1
		(0,3)	5397.9	5380.1	5684
		(0,4)	9127.8	9058.8	9324.7
C-F	4	(0,0)	948.6	1537.1	2244
		(0,1)	1419.4	1814.1	2411
		(0,2)	2138.6	2350.8	2851.6
		(0,3)	4974.5	4981.1	5167.2
		(0,4)	8483.9	8438.9	8525.1

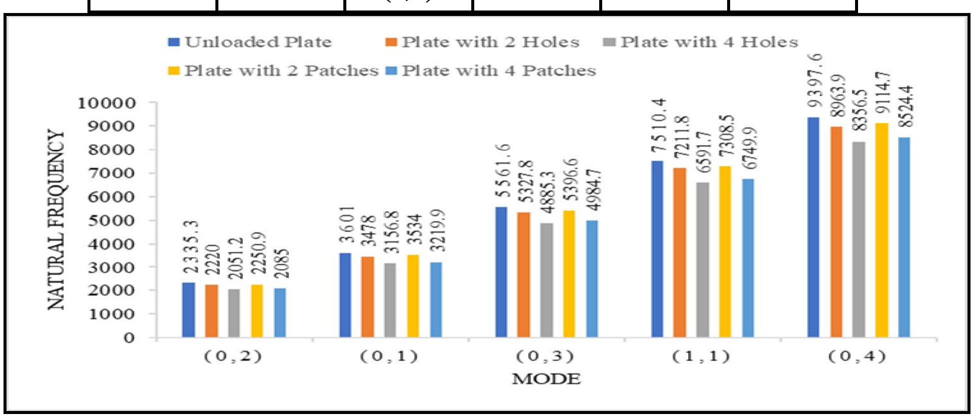

Fig.4. Effect of radii ratio (β = b/a= 0.1) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary conditions

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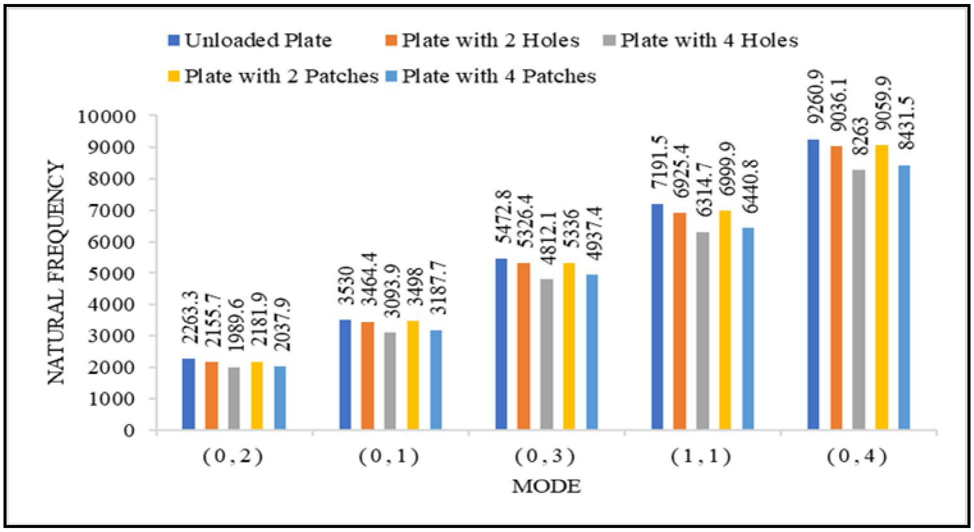

Fig.5. Effect of radii ratio (β = b/a= 0.2) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary conditions

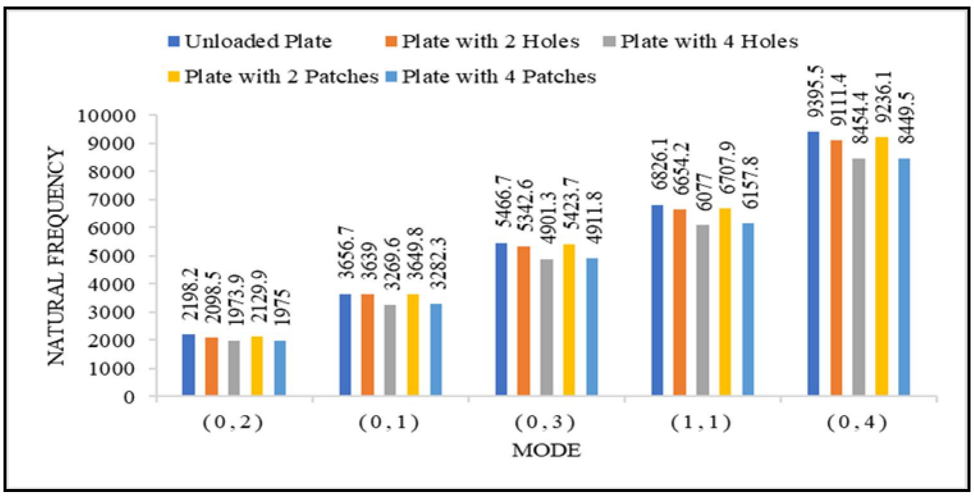

Fig.6. Effect of radii ratio (β = b/a= 0.3) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary conditions

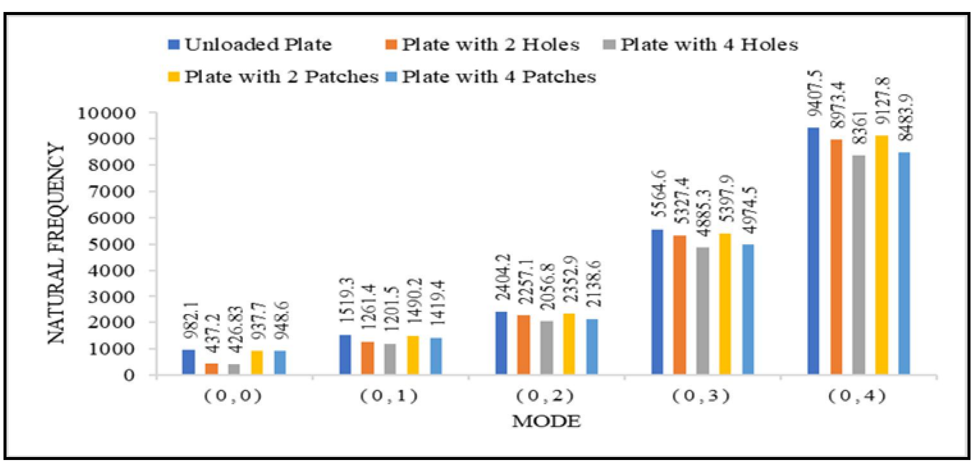

Fig.7. Effect of radii ratio (β = b/a= 0.1) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

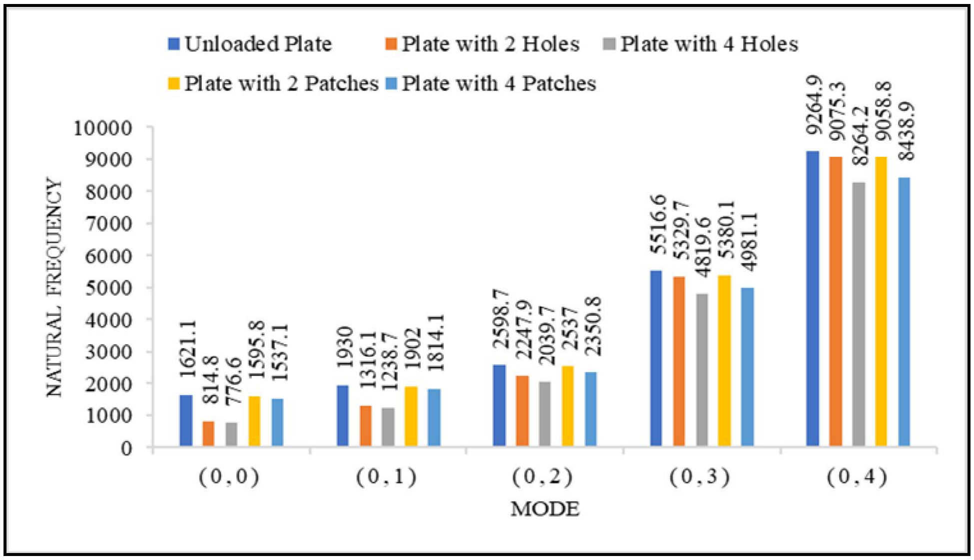

Fig.8. Effect of radii ratio (β = b/a= 0.2) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

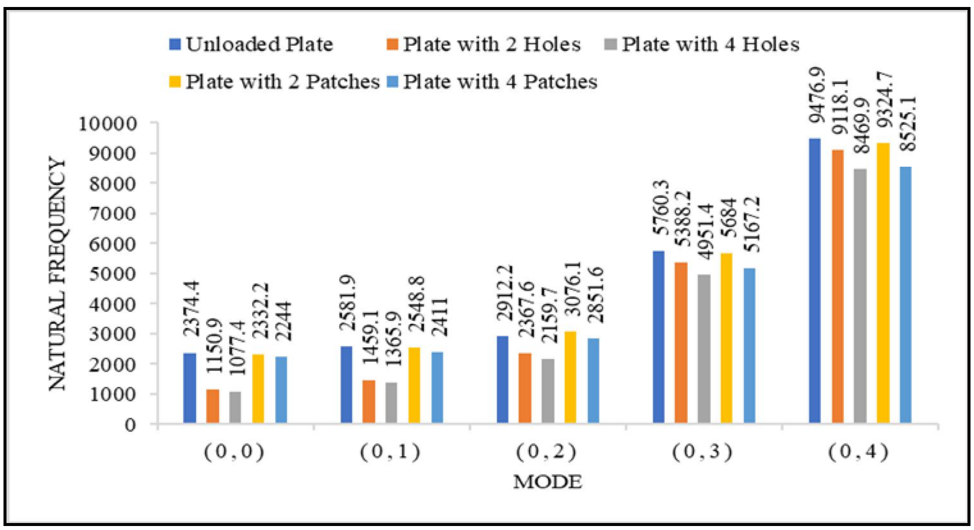

Fig.9. Effect of radii ratio (β = b/a= 0.1) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

B. Effect of Aspect Ratio on Plates

This section illustrates the impact of natural frequency on the vibration response of a uniform plate lacking a stiffener, influenced by various aspect ratios and boundary conditions. The mass and volume of the plate remain constant. Tables X through XV presents the effects of different aspect ratios on both uniform annulated circular un-stiffened plates and stiffened plates. Specifically, Tables X and XI detail how varying aspect ratios affect the first five Eigen frequencies (ω) of an annulated circular un-stiffened plate under free-free and clamped-free boundary conditions. According to Table X, it is observed that as the radius ratio increases from € = 0.1 to 0.3, the Eigen frequencies (ω) rise with increasing modes for an annulated circular un-stiffened plate under free-free boundary conditions. This phenomenon may be attributed to the enhanced stiffness of the plate, which enables it to vibrate at higher natural frequencies across various mode shapes. Conversely, for clamped-free boundary conditions, an increase in aspect ratios also leads to a rise in the Eigen frequency (ω) of the plate. This occurs because the plate's rigidity improves with a higher radius ratio, resulting in vibrations at a greater frequency. Furthermore, Table XI indicates that the clamped-free boundary condition exhibits greater rigidity than the free-free boundary condition. Consequently, the plate vibrates at a higher natural frequency compared to the free-free boundary condition.

Tables XII and XIII illustrate the impact of varying aspect ratios on the first five Eigen frequencies (ω) of an annulated circular stiffened plate with holes, under both free-free and clamped-free boundary conditions. Table XII clearly indicates that as the radius ratios increase from € = 0.1 to 0.3, the Eigen frequency (ω) rises with the increasing modes for the annulated circular stiffened plate with holes under free-free conditions. This phenomenon may be attributed to the enhanced stiffness of the plate resulting from the increased perforations. Conversely, for clamped-free boundary conditions, the Eigen frequency (ω) of the perforated plate also increases with higher aspect ratios. This can be explained by the fact that clamped-free vibrations demonstrate greater rigidity compared to free-free vibrations, leading to higher frequency vibrations of the plate. Furthermore, both Table XII and Table XIII indicate that an increase in the number of holes results in a decrease in the plate's mass, which consequently leads to a lower frequency of vibration across all mode shapes. It is also noted that as the number of holes increases, the rigidity of the plate diminishes. Therefore, a plate with four holes will vibrate at a lower frequency than one with two holes.

Tables XIV and XV illustrate the impact of varying aspect ratios on the first five Eigen frequencies (ω) of an annulated circular stiffened plate featuring concentrated patches, under both free-free and clamped-free boundary conditions. Table XIV clearly indicates that as the radius ratios increase from € = 0.1 to 0.3, the Eigen frequency (ω) rises with the increasing modes for the annulated circular stiffened plate with concentrated patches under free-free conditions. This phenomenon may be attributed to the enhanced stiffness of the plate resulting from the addition of patches. Conversely, for clamped-free boundary conditions, the Eigen frequency (ω) of the plate with patches also increases with higher aspect ratios. This can be explained by the fact that clamped-free vibrations demonstrate greater rigidity compared to free-free vibrations, leading to higher frequency vibrations of the plate. Furthermore, both Table XIV and Table XV indicate that an increase in the number of patches results in a reduction of the plate's stiffness, consequently causing a decrease in the vibration frequency across all mode shapes. It is also noted that as the number of patches increases, the rigidity of the plate diminishes. Therefore, a plate with four patches will exhibit a lower frequency of vibration than a plate with two patches.

Figures 10-15 illustrate the comparison regarding the impact of various aspect ratios (€) on both un-stiffened and stiffened circular plates under free-free and clamped-free boundary conditions. From Figure 10, it is evident that as the aspect ratio (β = 0.1) increases, the Eigen frequency rises with the increasing modes (0, 2), (0, 1), (0, 3), (1, 1), and (0, 4) for both the un-stiffened and stiffened plates under the free-free boundary condition. Additionally, Figure 10 indicates that the unloaded plate (annulated circular plate) exhibits the highest frequency across all mode shapes, whereas the plate featuring four holes and four patches displays the lowest frequency among all scenarios.

Table X

Effect of natural frequencies ON ASPECT ratio AND modes for circular annular uniform plate (unloaded plate) under free-free boundary conditions

BCS	Mode	Aspect Ratio		
		h/a=0.1	h/a=0.2	h/a=0.3
F-F	(0,2)	1177.3	1861.9	2301.3
	(0,1)	3278.9	4725.9	5151.5
	(0,3)	3452	5017	5854.8
	(1,1)	5157.2	7109.1	7648.3
	(0,4)	6452.9	8988.4	10235

In comparison to other plates, a plate featuring four holes is lighter and less rigid, leading to a reduced frequency across all mode shapes. The diminished rigidity of the four patches results in a lower frequency in this scenario. Figure 11 illustrates the comparison of natural frequencies under free-free boundary conditions with an aspect ratio of 0.2. The graphics clearly indicate that the plate with the most holes and the highest patches demonstrates the highest Eigen frequencies for each mode shape, whereas the unloaded plate (annulated circular plate) exhibits the highest natural frequencies for all mode shapes. Nevertheless, in the (0, 4) mode, the frequency of the plate with four holes is slightly lower than that of the plate with four patches. Furthermore, Figure 12 reveals that both the plate with four holes and the one with four patches show increasing frequency trends with the (0, 2), (0, 1), (0, 3), and (1, 1) mode shapes. The unloaded plate also displays similar increasing trends. Thus far, no sudden increases or decreases in the graph have been observed.

Figure 13 clearly illustrates that the clamped-free vibration curve is shorter than that of the free-free vibration curve. This difference arises from the greater rigidity of clamped-free boundary conditions compared to free-free conditions. Furthermore, the graph indicates that among the various plates, the unloaded plate (annulated circular plate) demonstrates the highest natural frequencies across all mode shapes, whereas the plate with four holes displays the lowest natural frequencies for all mode shapes. Figure 14 depicts a similar increase in Eigen frequencies, which rise in correlation with the aspect ratio. It is apparent from the figure that the clamped-free boundary condition exerts a more significant enhancing effect than the free-free condition. Moreover, it is observed that the unloaded plate exhibits the highest natural frequencies for all mode shapes when compared to the other plates, while the four-hole and four-patch plates present the lowest natural frequencies for all mode shapes. Figure 15 clearly indicates that for all mode shapes, the number of natural frequencies increases as the aspect ratio progresses from 0.1 to 0.3 across all plate cases. Additionally, the 2-holes & 2-patches, as well as the 4-holes and 4-patches, show the least sensitivity for all mode shapes. A notable sensitivity of frequency is observed for the unloaded plate.

Table XI
Effect of natural frequencies ON ASPECT ratio AND MODES for circular annular uniform plate (unloaded plate) under clamped-free boundary conditions

BCS	Mode	Aspect Ratio		
		h/a=0.1	h/a=0.2	h/a=0.3
C-F	(0,0)	3983	5649.2	7042.6
	(0,1)	4008.7	5750.3	7105.5
	(0,2)	4846.2	6442.9	7754.8
	(0,3)	6067.2	8100	9363.2
	(0,4)	7932.2	10926	12490

Table XII
Effect of natural frequencies ON ASPECT ratio AND modes for circular annular holed plate under free-free boundary conditions

BCS	No. of Holes	Mode	Aspect Ratio		
			h/a=0.1	h/a=0.2	h/a=0.3
F-F	2	(0,2)	1070.1	1746.3	2213.3
		(0,1)	3200.7	4500.9	4961.6
		(0,3)	3355.9	4896.3	5654.3
		(1,1)	4922.7	6783.7	7385.4
		(0,4)	6213.3	8658.5	9834.3
F-F	4	(0,2)	1034.7	1631.5	2209.4
		(0,1)	2796.5	3891.2	4892.2
		(0,3)	3056.4	4420.6	5629.3
		(1,1)	4588.8	6151.2	7274.8
		(0,4)	5812.8	6303.3	9757

Table XIII
Effect of natural frequencies ON ASPECT ratio AND modes for circular annular holed plate under clamped-free boundary conditions

BCS	No. of Holes	Mode	Aspect Ratio		
			h/a=0.1	h/a=0.2	h/a=0.3
C-F	2	(0,0)	1614.1	2193.2	2481.9
		(0,1)	1481.2	2136.4	2515.9
		(0,2)	2501.8	3189.5	3574.1
		(0,3)	4353.3	5747.6	6466.1
		(0,4)	6890.5	9277.6	10373.5
C-F	4	(0,0)	1455.3	1878.3	2453.5
		(0,1)	1327.1	1965.1	2497.6
		(0,2)	2172.3	2808.2	3481.4
		(0,3)	4030.4	5432.4	6448.7
		(0,4)	6410.9	8604.9	10308.9

Table XIV
Effect of natural frequencies ON ASPECT ratio AND MODES for circular annular stiffened plate under free-free boundary conditions

BCS	No. of Patches	Mode	Aspect Ratio		
			h/a=0.1	h/a=0.2	h/a=0.3
F-F	2	(0,2)	1106.8	1786.1	2227.1
		(0,1)	3178.9	4512.2	4987.9
		(0,3)	3314.8	4926.4	5670.3
		(1,1)	4974.2	6871	7418.2
		(0,4)	6203.3	8760	9921.8
F-F	4	(0,2)	1044.4	1656.7	2215.4
		(0,1)	2869	3946.9	4930.7
		(0,3)	3093.4	4474.4	5656.5
		(1,1)	4597.5	6216	7344.7
		(0,4)	5755.6	8108.3	9756.6

Table XV
Effect of natural frequencies ON ASPECT ratio AND modes for circular annular stiffened plate under clamped-free boundary conditions

BCS	No. of Patches	Mode	Aspect Ratio		
			h/a=0.1	h/a=0.2	h/a=0.3
C-F	2	(0,0)	1725.6	2268.4	2475.7
		(0,1)	1474.8	2217.5	2517.8
		(0,2)	2575.4	3335.1	3604.5
		(0,3)	4396.4	5962.2	6509.6
		(0,4)	6968.7	9583.9	10434.2
C-F	4	(0,0)	1556.9	1989.6	2470.5
		(0,1)	1334.4	1893.4	2517.2
		(0,2)	2235.9	2891.6	3546.7
		(0,3)	3989.2	5461	6492.6
		(0,4)	6391.2	8764.3	10352.7

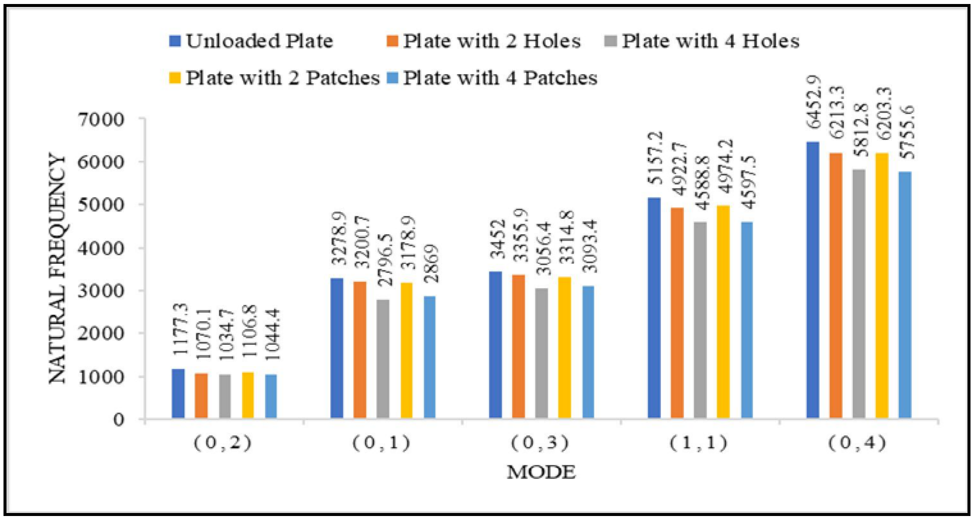

Fig.10. Effect of aspect ratio (€ = h/a= 0.1) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary conditions

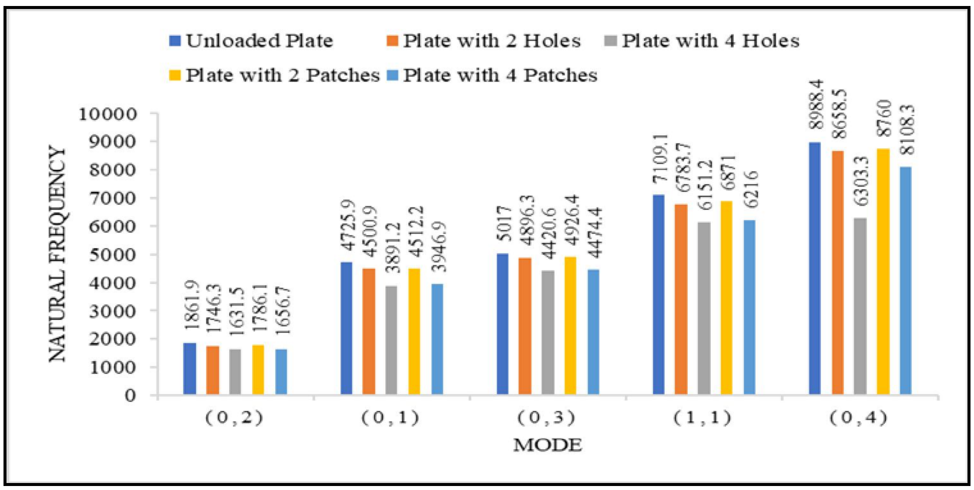

Fig.11. Effect of aspect ratio (€ = h/a= 0.2) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary conditions

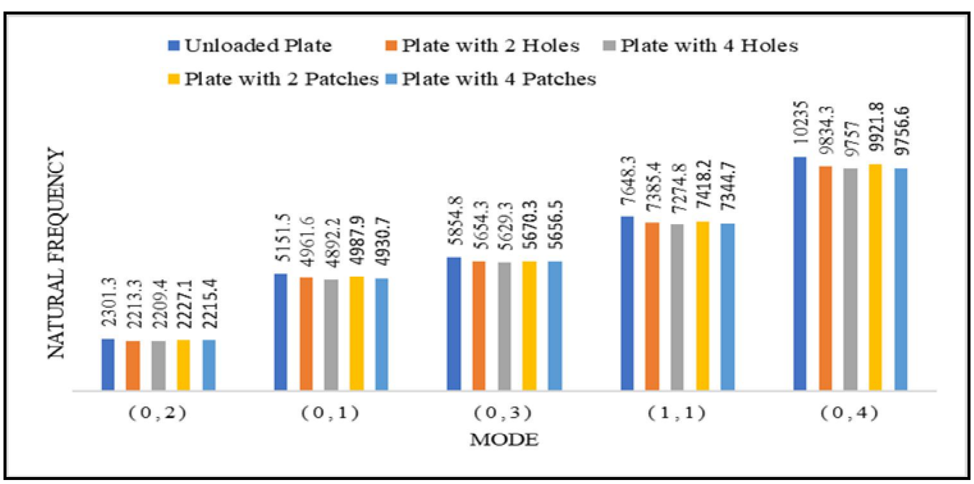

Fig.12. Effect of aspect ratio (€ = h/a= 0.3) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under free-free boundary condition

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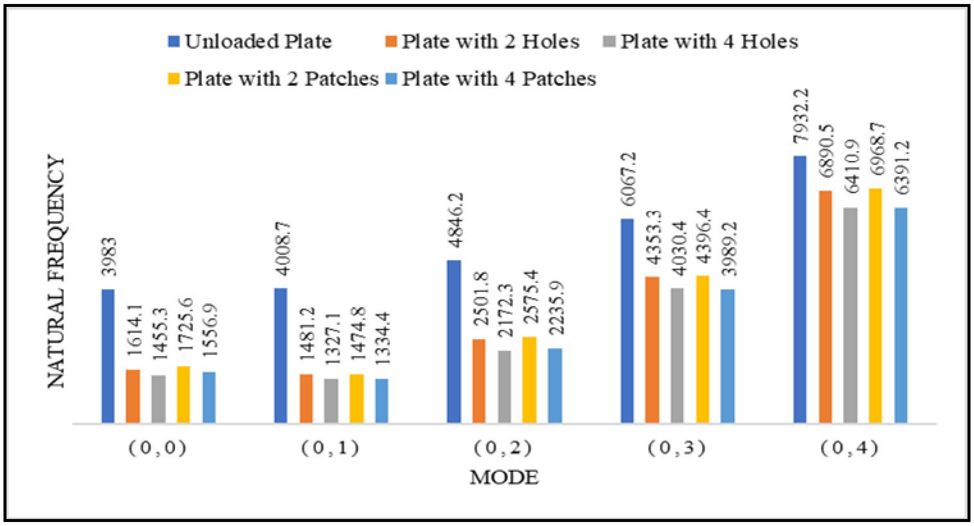

Fig.13. Effect of aspect ratio (€ = h/a= 0.1) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

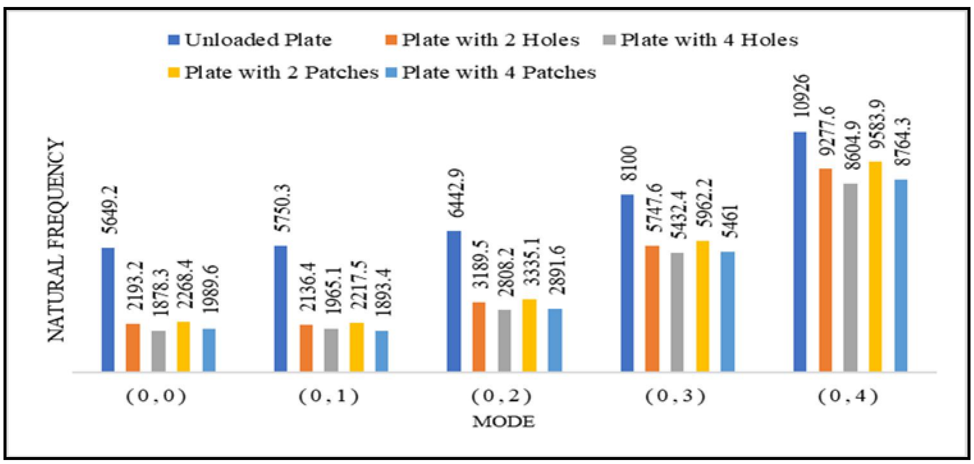

Fig.14. Effect of aspect ratio (€ = h/a= 0.2) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

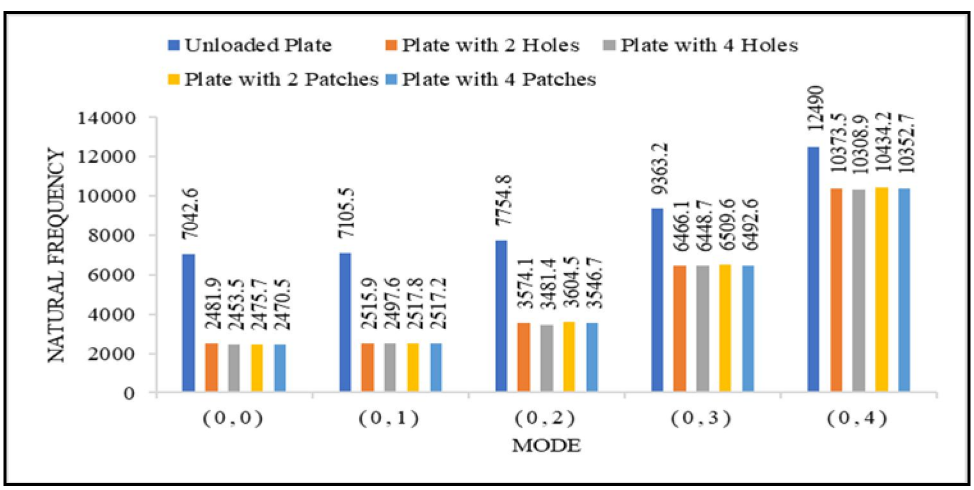

Fig.15. Effect of aspect ratio (€ = h/a= 0.3) on the Eigen frequencies (ω) of various stiffened less as well as stiffened plates under clamped-free boundary conditions

IV. CONCLUSIONS

This research examined the comparative vibration characteristics of annulated circular plates, annulated circular concentrated stiffened plates, and annulated circular hole plates using finite element analysis (FEA), while ensuring a consistent mass for the plates. The temperature remains constant across all scenarios. The evaluation of these plates is performed under both free-free and clamped-free boundary conditions. It is noted that the Eigen frequency (ω) decreases as the radius ratio increases for free-free boundary conditions, while the Eigen frequency rises with an increasing radius ratio for clamped-free boundary conditions. Notably, the Eigen frequency for the clamped boundary condition demonstrates greater vibration sensitivity compared to the free-free boundary condition. Nevertheless, as the aspect ratio increases, the Eigen frequency also increases for both free-free and clamped-free boundary conditions. Furthermore, it is found that the un-stiffened annulated circular plate possesses the highest Eigen frequency across all cases. The analysis indicates that as the number of holes and patches increases, the natural frequency decreases. The configuration with 4-holes and 4-patches shows the lowest frequency under all conditions. Additionally, it has been observed that the clamped-free boundary condition results in a shorter plate compared to the free-free boundary condition.

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