Buckling Analysis of Rectangular Plate Element Subjected to In-Plane Loading Using Finite Element Method

[1]Saumit Kumar Mandal [2]Pradeep Kumar Mishra

Abstract- The usage of heterogeneous materials in situations where large strength to weight ratio is required has been increased substantially over the world in all construction aspects. The behavior of the plate under each loading is different. The buckling factors are evaluated by changing the position of the holes, length to thickness ratio. The effect of changing the position of holes, a/b ratio, thickness and buckling load per unit length is discussed. This article summarizes the numerical study carried out using finite element software ANSYS and Timoshenko's methodology to examine the buckling behavior of homogeneous and heterogeneous rectangular plate element with and without hole. Also the effect of aspect ratio on the buckling behavior with varying plate thickness for different material and different boundary conditions was also examined. The results shows that buckling load per unit length is in simply-supported boundary conditions and the laminated composite plates have varying aspect ratio, varying thickness (t), cut out edge, centre of hole and without cut plate.

Index Terms- Buckling Analysis, Plate Element, Finite Element Analysis, Aspect ratio, Buckling load

1 INTRODUCTION

CONSIDERING engineering structures like's columns, beams or plates that did not develop not only from buckling but also excessive stresses. As per the change of buckling behavior the aspect ratio also changes, at different end the thickness of the element is subjected. Plate identifies to work in column at high aspect ratio. Whenever the aspect ratio decreases, the limit of elastic buckling does not take place.

This paper deals with the analysis of a rectangular element being considered as a plane stress condition under various boundary conditions and loadings. Throughout the analysis, the master element which is plane 183 is used to perform buckling analysis using ANSYS 17.2. Finally, results have been checked with exact results obtained from Timoshenko's plate buckling equation for different end conditions.

Jana et al. [1] considering simply supported rectangular plate without cut done buckling analysis taking subjected to various types of non-uniform compressive loads. Chai et al. [2] taking different boundary condition without cut out under various linearly varying in plane loading done the influence of boundary conditions, plate aspect ratios on the optimal ply angle and associated optimal buckling loads of anti-symmetrically laminated composite plates.

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Hu et al. [3] experimentally using the symmetrically laminated composite rectangular plate without cut out under parabolic variation of axial loads for the buckling behavior of graphite/epoxy. Jain et al. [4] taking parabolic elliptical cut plate for find out the effects of the cut out shape, size and the alignment on the buckling and the first-ply failure loads of square laminates subjected to uni-axial compression load. Aydin Komur et al. [5] by taking the effects of an elliptical/circular cut out on the buckling load of symmetric cross-ply and angle-ply laminate square composite plates. Srivatsa and Murti [6] by considering a parametric study of plate compression buck-ling behavior of stress loaded composite plate with a central circular cutout.

2 BUCKLING ANALYSIS

According to Buckling analysis we able to determine buckling loads - critical loads should unstable and buckled mode shapes it must associated with a structure's buckled response. Using different methods, like as energy and equilibrium methods, have been used to calculate the lowest Eigen value, or the actual buckling load. By taking above these methods are given in this work and the reader is referred to Timoshenko's' Theory of Elastic Stability [6] for a more comprehensive treatment of homogeneous plate buckling. For calculating a homogeneous plate the following formula is used to the critical buckling load per unit length:

$$(N_X)_{cr} = \frac{K\pi^2 E\, t^3}{12(1-\vartheta^2)b^2}$$

For a homogeneous plate the following formula is used to calculate the critical buckling load per unit length. Where, E is Young's Modulus, ɣ is Poisson's ratio, t is the plate thickness, b is the width of the plate, and k is a constant determined by

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the boundary condition and aspect ratio of the plate.

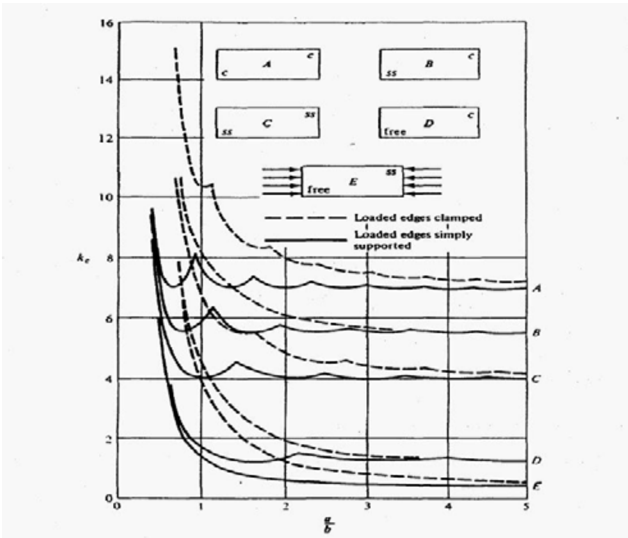

Fig.1. Influence of boundary conditions on the buckling coefficients of plates subjected to in-plane compressive loading [6].

3 NUMERICAL ANALYSIS

This work is to find critical buckling load of homogeneous and heterogeneous rectangular plate element subjected to in plane loading using finite element analysis **ANSYS 17.2**. The plate has length a, width b & thickness t. The length of plate is taken as constant b= 1000 mm. The analysis is done in the following cases:

Case1: The analysis is done by edge cut in the rectangular plate and its radius (R) is 50mm. The Nature of critical buckling load factor with respect to aspect ratio (a/b).

Case2: Further the work is extended to the analysis is done by placing centre hole in the rectangular plate and its radius (R) is 50 mm. The Nature of critical buckling load factor with respect to aspect ratio a/b.

Case3: Next the analysis is done by different materials like stainless steel, mild steel, copper and cast iron are without hole in the rectangular plate. The Nature of critical buckling load is factor with respect to aspect ratio (a/b).

4 ELEMENT DESCRIPTIONS

In this study, the element used in the analysis is PLANE183. PLANE183 is a higher order version of the 2-D, four node element (PLANE42). It provides more accurate results for mixed (quadrilateral-triangular) automatic meshes and can tolerate irregular shapes without affect upon accuracy. The 8-node elements have compatible displacement shapes and are well suited to model curved boundaries. The 8-node element is defined by eight nodes having two degrees of freedom at each node: translations in the nodal x and y directions. The element may be used as a plane element or as an axisymmetric element. The element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities.

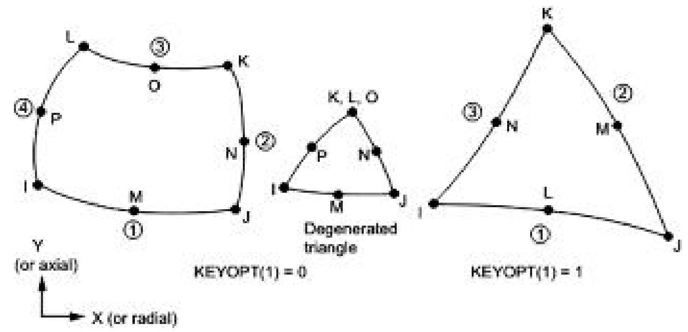

Fig 2: Element geometry of solid Plane 183

5 RESULTS AND DISCUSSION

5.1 Homogeneous Plate

5.1.1 Analytical detail

Homogenous plate is assumed as Aluminum Alloy which in nonferrous metal.

Isotropic properties of aluminum alloy material

Material	Young's modulus	Poisson's ratio	Density
Aluminum	$7x10^{10}$ pa	0.30	2700 kg/m^3
Cast iron	91.189 Gpa	0.22	7207 kg/m^3
Mild steel	210 Gpa	0.30	7850 kg/m^3
Copper	128 Gpa	0.36	8960 kg/m^3
Stainless steel	203 Gpa	0.27	8030 kg/m^3

The width of rectangular plate (b) is varying from 1000mm, 900mm, 800mm, 700mm, 600mm, and 500mm and length of the rectangular plate (a) is 1000mm, which is fixed. Then the corresponding "a/b" ratios are 1, 1.11, 1.25, 1.43, 1.67, 2.0. The thickness of the plate (t) is varied as 3mm, 4mm and 5mm. Then finally the radius of the circular hole is 50mm, which is positioned at different locations like centre and edge cut of the plate. The modeling includes defining the element type, real constants, and material property for isotropic, meshing and it is followed by solution includes buckling analysis. In this study, solid plane 183 selected as the element type.
To model is created with area then the plate is meshed with solid elements after that the load is applied on the plate. The plate is subjected to simply supported boundary conditions.

5.1.2 Modeling

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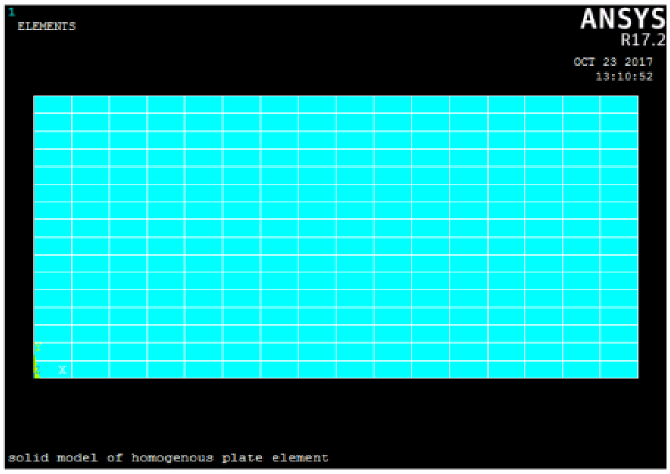

Fig. 3 Solid model of homogenous plate element

Figure 2 shows metal block of size 1m x 0.5m which was modeled as 2-D model and meshed with approximately 16 elements across the width of the metal block. To model is created with area then the plate is meshed with solid elements after that the load is applied on the plate. The plate is subjected to simply supported boundary conditions. The critical buckling load results are tabulated in tables.

Table:-1 Critical buckling loads of validation paper

a (mm)	b (mm)	Aspect Ratio (a/b)	Plate thick ness (mm)	Critical Buckling Load (N/mm)	ANSYS Critical Buckling Load (N/mm)
1000	1000	1	3	6.8383	6.8683
1000	900	1.11	3	8.5364	8.5470
1000	800	1.25	3	11.2258	11.834
1000	700	1.43	3	15.8077	16.026
1000	600	1.67	3	24.3984	24.814
1000	500	2	3	42.7394	42.735
1000	1000	1	4	16.2093	16.367
1000	900	1.11	4	20.2345	20.243
1000	800	1.25	4	26.6092	26.525
1000	700	1.43	4	37.4071	36.630
1000	600	1.67	4	57.8382	59.172
1000	500	2	4	101.3082	109.890
1000	1000	1	5	31.6588	32.001
1000	900	1.11	5	39.5204	38.462
1000	800	1.25	5	51.9712	51.282
1000	700	1.43	5	73.1838	69.930
1000	600	1.67	5	112.9555	109.890
1000	500	2	5	197.8676	192.308

The width of rectangular plate (b) is varying from 1000mm, 900mm, 800mm, 700mm, 600mm, and 500mm and length of the rectangular plate (a) is 1000mm, which is fixed. Then the corresponding "a/b" ratios are 1, 1.11, 1.25, 1.43, 1.67, 2.0. The thickness of the plate (t) is varied as 3mm, 4mm and 5mm.
The value of length (a), width (b), aspect ratio (a/b) and thickness (t) of the plate are same as per the reference of the Buckling Analysis of Plate Element Subjected to In Plane Loading Using ANSYS by Monica S Swamy, Ranjith A, Sandya D S, Shrithi S Badami.I have done the validation of the critical buckling load is approximately same.

Case:-1

The plate has an edge cut circular hole of radius (R). Here aspect ratio varies from 1 to 2. Nature of critical Buckling load per unit length with respect to aspect ratio was studied.

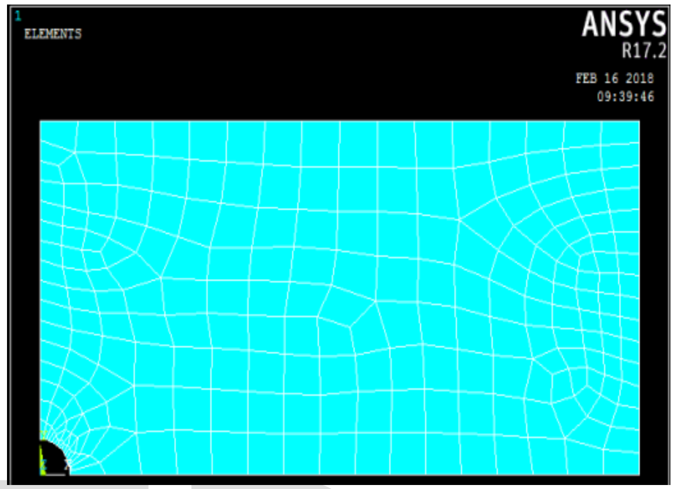

Fig. 4 Meshed model of rectangular plate in edge cut

Table:-2 ANSYS critical buckling loads of edge cut in rectangular plate

a (mm)	b (mm)	Aspect Ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load Edge Cut in plate
1000	1000	1	3	1.69
1000	900	1.11	3	2.23
1000	800	1.25	3	2.454
1000	700	1.43	3	3.667
1000	600	1.67	3	5.751
1000	500	2	3	9.259
1000	1000	1	4	4.027
1000	900	1.11	4	5.281
1000	800	1.25	4	5.5
1000	700	1.43	4	8.381
1000	600	1.67	4	13.713
1000	500	2	4	23.819
1000	1000	1	5	7.887
1000	900	1.11	5	10.035
1000	800	1.25	5	10.634
1000	700	1.43	5	16.455
1000	600	1.67	5	25.468
1000	500	2	5	41.666

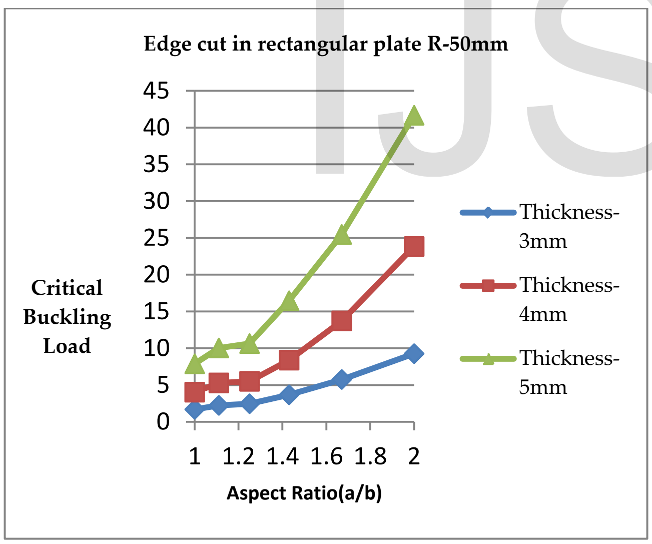

Fig:-5 Comparison of critical buckling load with aspect ratio for plate with edge cut at various plate thickness.

The fig.5 shows variation of aspect ratio a/b for plate with edge cut. It is observed that as a/b ratio increases the critical buckling load increases.

Case:-2

The plate has a central circular hole of radius (R). Here aspect ratio varies from 1 to 2.Nature of critical buckling load per unit length with respect to aspect ratio was studied.

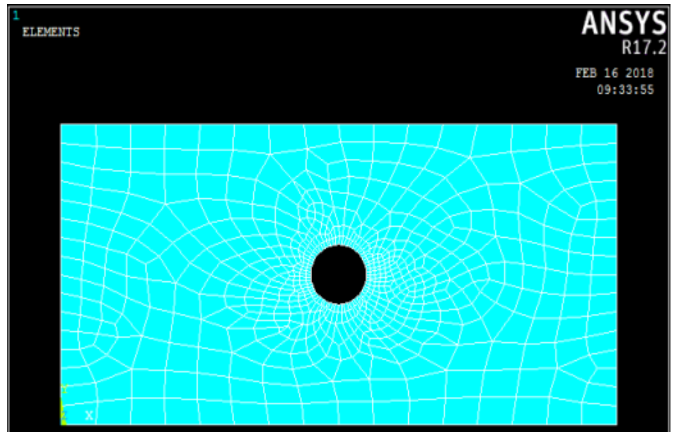

Fig. 6 Meshed model of rectangular plate in centre circular hole

Table:-3 ANSYS critical buckling loads of central circular hole in rectangular plate

a (mm)	b (mm)	Aspect Ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load Hole of the Centre in Plate
1000	1000	1	3	1.339
1000	900	1.11	3	1.656
1000	800	1.25	3	2.293
1000	700	1.43	3	3.104
1000	600	1.67	3	4.774
1000	500	2	3	8.42
1000	1000	1	4	3.191
1000	900	1.11	4	3.992
1000	800	1.25	4	5.139
1000	700	1.43	4	7.097
1000	600	1.67	4	11.385
1000	500	2	4	21.653
1000	1000	1	5	6.249
1000	900	1.11	5	7.453
1000	800	1.25	5	9.936
1000	700	1.43	5	13.548
1000	600	1.67	5	21.114
1000	500	2	5	37.894

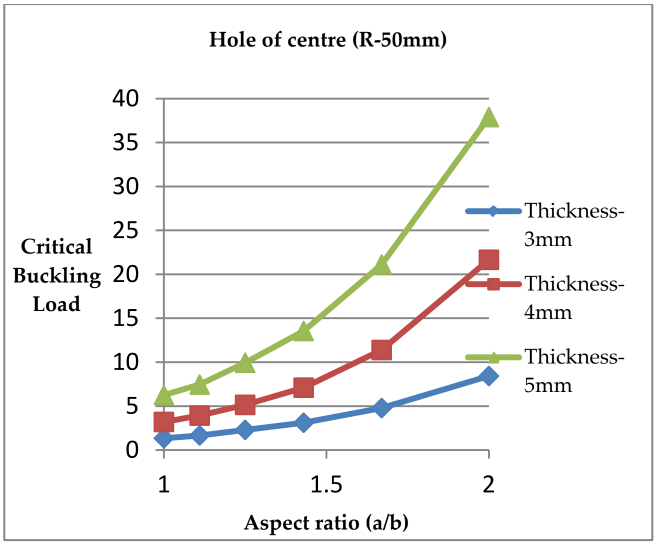

Fig:-7 Comparison of critical buckling load with aspect ratio for plate central circular hole at various Plate thickness.

The fig.7 shows variation of aspect ratio a/b for plate with central circular hole. It is observed that as a/b ratio increases the critical buckling load increases.

Case:-3

The plate has a without hole for different materials like stainless steel, mild steel, copper and cast iron. The Nature of critical buckling load is factor with respect to aspect ratio (a/b).

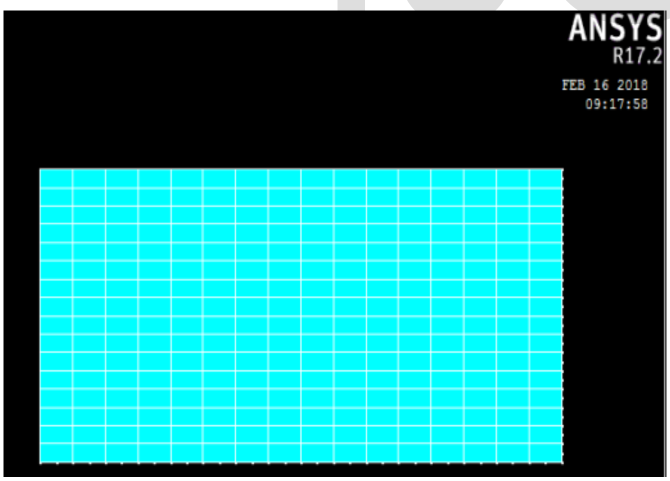

Fig. 8 Meshed model of rectangular plate in without hole

Table:-4 ANSYS critical buckling loads of different material in without hole

a (mm)	b (mm)	Aspect ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load in Cast Iron (N/mm)	ANSYS Critical Buckling Load in Copper (N/mm)
1000	1000	1	3	8.555	13.13
1000	900	1.11	3	10.647	16.34
1000	800	1.25	3	14.742	22.62
1000	700	1.43	3	19.964	30.64
1000	600	1.67	3	30.911	47.44
1000	500	2	3	53.237	81.7
1000	1000	1	4	20.388	31.29
1000	900	1.11	4	25.217	38.7
1000	800	1.25	4	33.043	50.71
1000	700	1.43	4	45.611	70.03
1000	600	1.67	4	73.713	113.1
1000	500	2	4	136.896	210.1
1000	1000	1	5	39.927	61.27
1000	900	1.11	5	47.913	73.53
1000	800	1.25	5	63.884	98.04
1000	700	1.43	5	87.115	133.7
1000	600	1.67	5	136.896	210.1
1000	500	2	5	239.568	367.6

Table:-5 ANSYS critical buckling loads of different material in without hole

a (mm)	b (mm)	Aspect ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load in Stainless Steel (N/mm)	ANSYS Critical Bucklin g Load in Mild Steel (N/mm)
1000	1000	1	3	19.55	20.604
1000	900	1.11	3	24.329	25.641
1000	800	1.25	3	33.686	35.503
1000	700	1.43	3	45.617	48.076
1000	600	1.67	3	70.633	74.441
1000	500	2	3	121.646	128.205
1000	1000	1	4	46.587	49.099
1000	900	1.11	4	57.621	60.728
1000	800	1.25	4	75.504	79.575
1000	700	1.43	4	104.268	109.89
1000	600	1.67	4	168.433	177.515
1000	500	2	4	312.803	329.67
1000	1000	1	5	91.234	96.153
1000	900	1.11	5	109.481	115.385
1000	800	1.25	5	145.975	153.846
1000	700	1.43	5	199.057	209.79
1000	600	1.67	5	312.803	329.67

1000	500	2	5	547.406	576.923

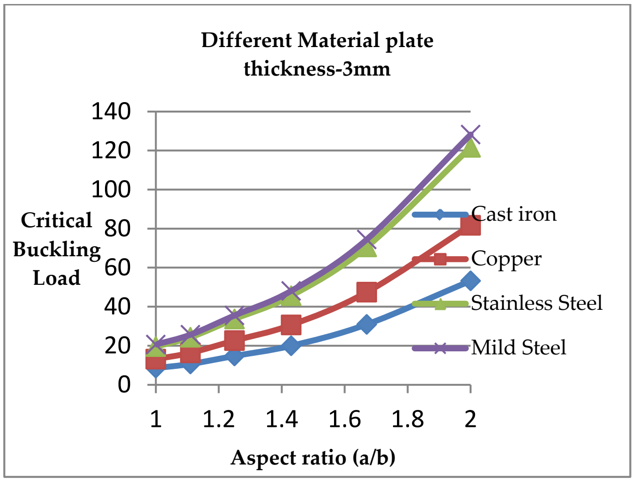

Fig:-9 Comparison of critical buckling load with aspect ratio for different plate at thickness 3mm.

The fig.9 shows variation of aspect ratio a/b for plate with different material. It is observed that as a/b ratio increases the critical buckling load increases at plate thickness 3mm

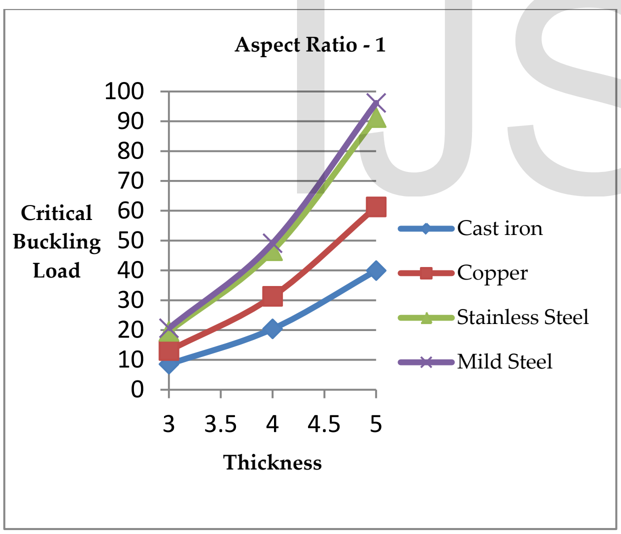

Fig:-10 Comparison of critical buckling load with thickness for different plate at aspect ratio 1.

The fig.10 shows variation of thickness for plate with different material. It is observed that as thickness increases the critical buckling load increases at aspect ratio 1.

5.2 Laminated Heterogeneous Plate

5.2.1 Analytical detail

Laminated heterogeneous plate is analyzed in ANSYS as the combination of Aluminum and Low Carbon Steel with following isotropic material properties.

material	Young's modulus	Poisson's ratio	Density
Aluminum	$7x10^{10}$pa	0.30	2700kg/m^3
low carbon steel	$2x10^{11}$pa	0.29	7872 m^3

5.2.2 Modeling

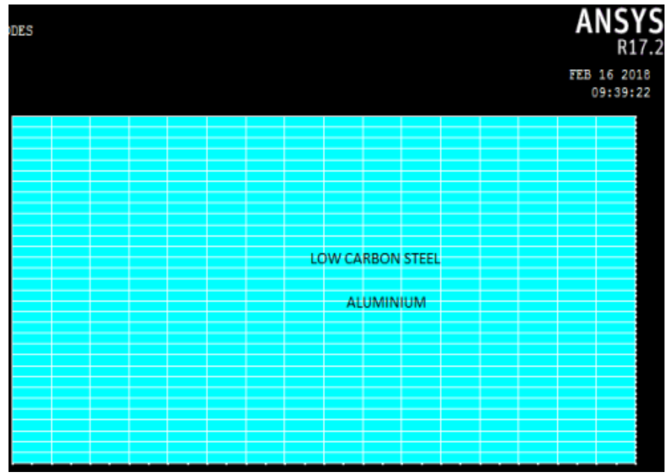

Fig.11 Solid model of laminated heterogeneous plate

The analysis is carried out for different plate thickness, end condition, aspect ratios. The corresponding critical buckling loads are tabulated.

Case:-1

The composite plate has an edge cut circular hole of radius (R). Here aspect ratio varies from 1 to2. Nature of critical Buckling load per unit length with respect to aspect ratio was studied.

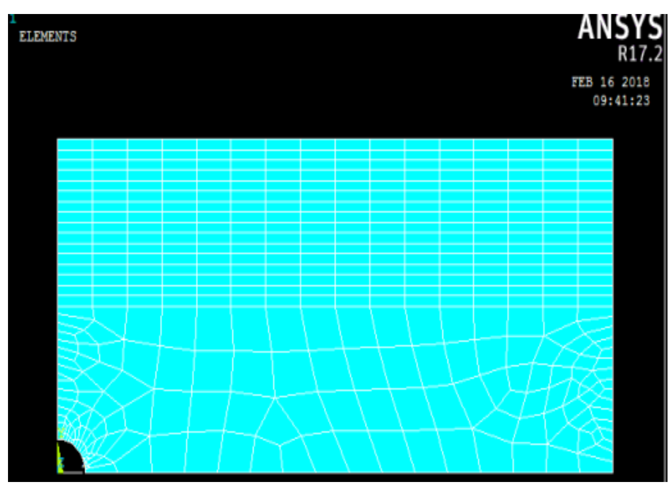

Fig. 12 Meshed model of composite rectangular plate in edge cut

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Table:-6 ANSYS critical buckling loads of edge cut in composite rectangular plate.

a (mm)	b (mm)	Aspect ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load for Edge Cut in composite plate
1000	1000	1	3	2.1627
1000	900	1.11	3	2.9079
1000	800	1.25	3	4.0313
1000	700	1.43	3	5.9096
1000	600	1.67	3	8.9224
1000	500	2	3	16.1416
1000	1000	1	4	4.8395
1000	900	1.11	4	6.8872
1000	800	1.25	4	9.0357
1000	700	1.43	4	13.5078
1000	600	1.67	4	21.2766
1000	500	2	4	41.507
1000	1000	1	5	10.0927
1000	900	1.11	5	13.0858
1000	800	1.25	5	17.4691
1000	700	1.43	5	25.7876
1000	600	1.67	5	39.5136
1000	500	2	5	72.6373

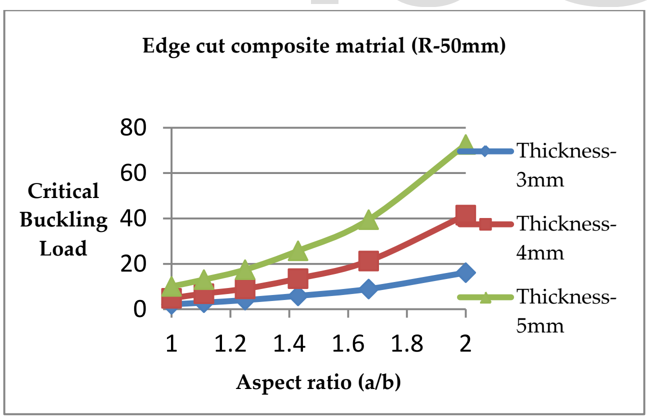

Fig:-13 Comparison of critical buckling load with aspect ratio for composite plate with edge cut at various plate thicknesses.

The fig.13 shows variation of aspect ratio a/b for composite plate with edge cut. It is observed that as a/b ratio increases the critical buckling load increases.

Case:-2

The composite plate has a central circular hole of radius (R). Here aspect ratio varies from 1 to 2.Nature of critical buckling load per unit length with respect to aspect ratio was studied.

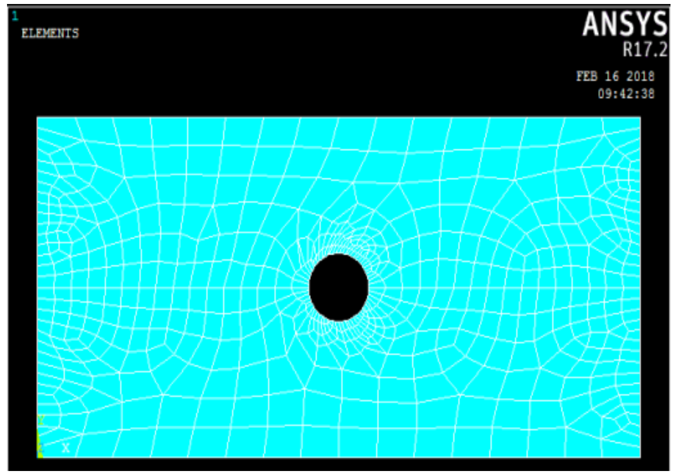

Fig. 14 Meshed model of composite rectangular plate in centre circular hole

Table:-7 ANSYS critical buckling loads of central circular hole in composite rectangular plate

a (mm)	b (mm)	Aspect ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load for Hole of the Centre in composite plate
1000	1000	1	3	2.1189
1000	900	1.11	3	2.7734
1000	800	1.25	3	3.7375
1000	700	1.43	3	5.2395
1000	600	1.67	3	8.6844
1000	500	2	3	14.7986
1000	1000	1	4	3.1750
1000	900	1.11	4	6.5685
1000	800	1.25	4	8.3772
1000	700	1.43	4	11.9762
1000	600	1.67	4	20.3731
1000	500	2	4	38.0536
1000	1000	1	5	9.8885
1000	900	1.11	5	12.4803
1000	800	1.25	5	16.196
1000	700	1.43	5	22.8636
1000	600	1.67	5	37.8358
1000	500	2	5	66.5938

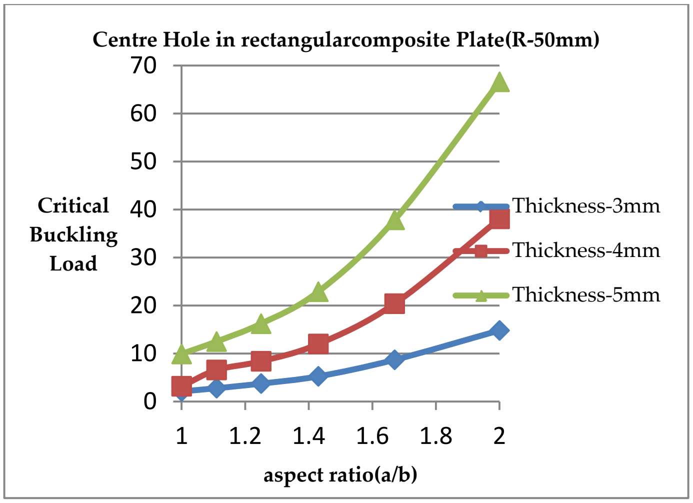

Fig:-15 Comparison of critical buckling load with aspect ratio for composite plate central circular hole at various Plate thickness.

The fig.15 shows variation of aspect ratio a/b for composite plate with central circular hole. It is observed that as a/b ratio increases the critical buckling load increases.

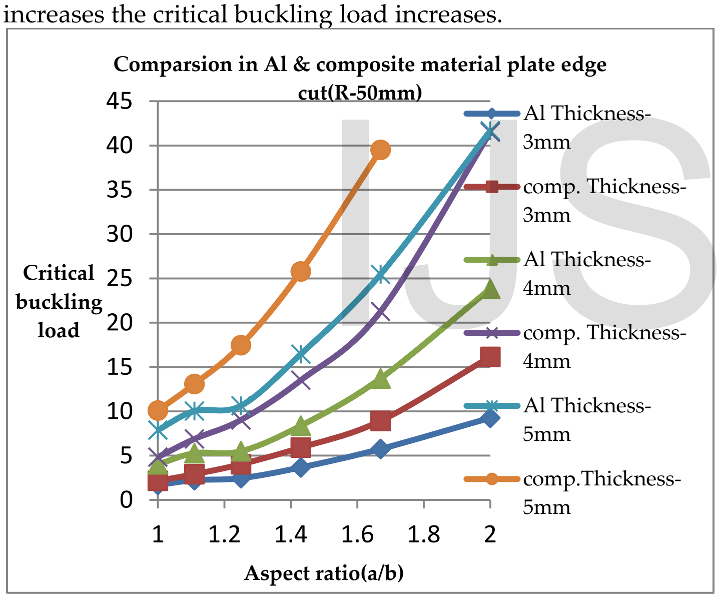

Fig. 16: Comparison of critical buckling load with aspect ratio at edges cut in different thickness.

From Fig.16 the edge cut by taking difference thickness of Homogeneous and heterogeneous plate the aspect ratio will be increase, when the critical buckling load increases.

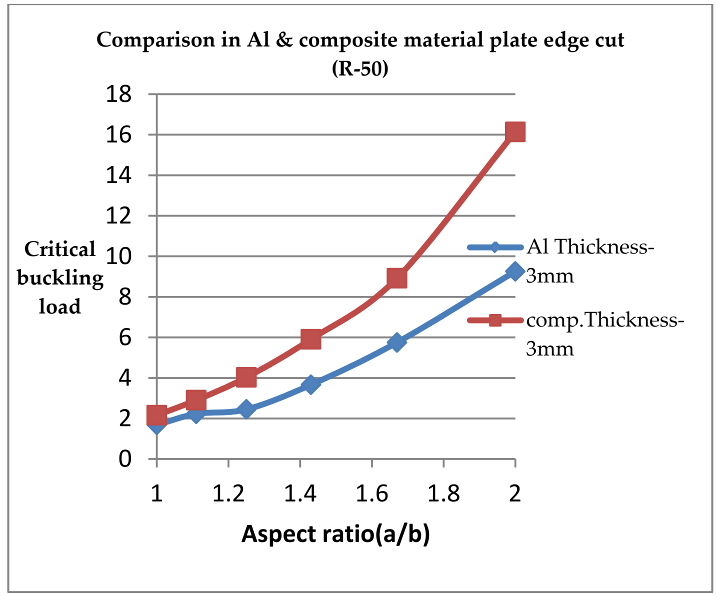

Fig. 17: Comparison of critical buckling load with aspect ratio at edge cut in thickness-3mm

From Fig:-17 the critical buckling load of heterogeneous is higher then to homogeneous plate as taking plate thickness 3mm.

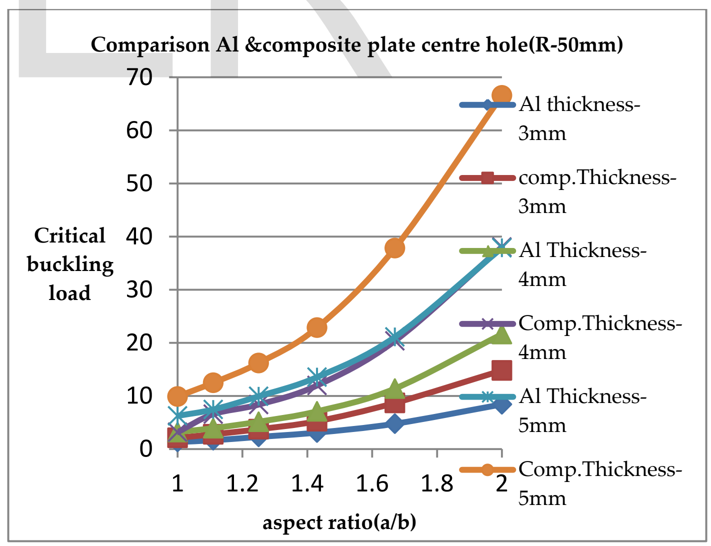

Fig. 18: Comparison of critical buckling load with aspect ratio at central circular hole in different thickness.

From Fig.18 the central circular hole by taking difference thickness of homogeneous and heterogeneous plate the aspect ratio will be increase, when the critical buckling load increases.

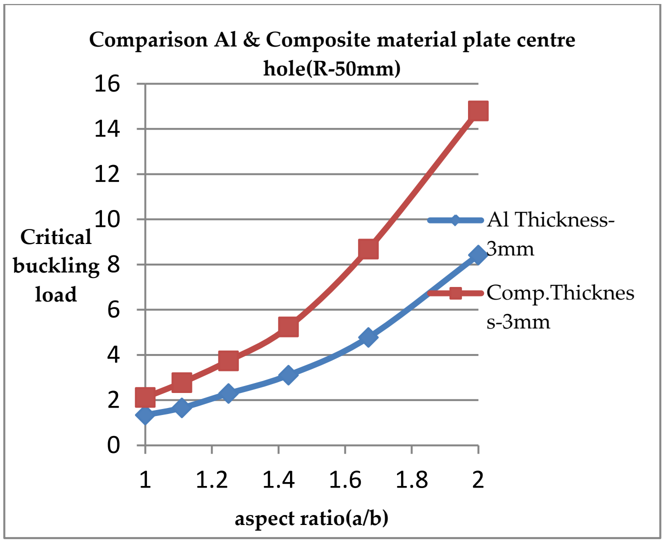

Fig. 19: Comparison of critical buckling load with aspect ratio at central circular hole in thickness-3mm

From Fig:-19 the critical buckling load of heterogeneous is higher then to homogeneous plate as
Taking plate thickness 3mm

Case:-3

The plate has a without hole for composite material in aluminum and law carbon steel. The Nature of critical buckling load is factor with respect to aspect ratio (a/b).

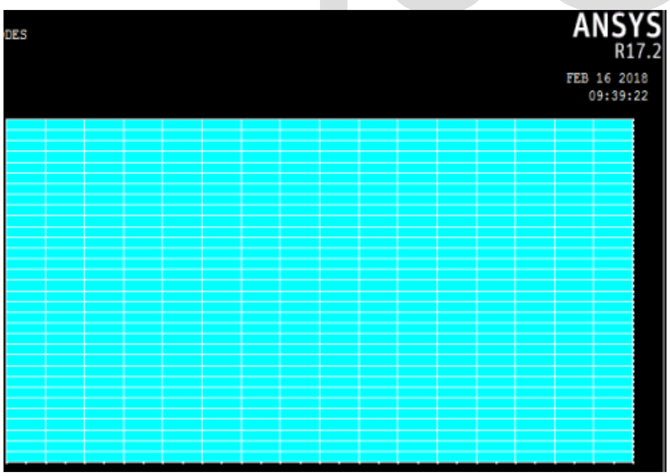

Fig. 20 Meshed model of rectangular composite plate in without hole

Table:-8 ANSYS critical buckling loads of composite material in without hole

a (mm)	b (mm)	Aspect ratio (a/b)	Plate thickness (mm)	ANSYS Critical Buckling Load for Composite plate
1000	1000	1	3	5.6415
1000	900	1.11	3	7.0756
1000	800	1.25	3	10.0112
1000	700	1.43	3	13.7121
1000	600	1.67	3	21.8191
1000	500	2	3	38.2658
1000	1000	1	4	13.4438
1000	900	1.11	4	16.758
1000	800	1.25	4	22.4388
1000	700	1.43	4	31.3418
1000	600	1.67	4	52.0314
1000	500	2	4	98.3976
1000	1000	1	5	26.3274
1000	900	1.11	5	31.8402
1000	800	1.25	5	43.3817
1000	700	1.43	5	59.8344
1000	600	1.67	5	96.6298
1000	500	2	5	172.196

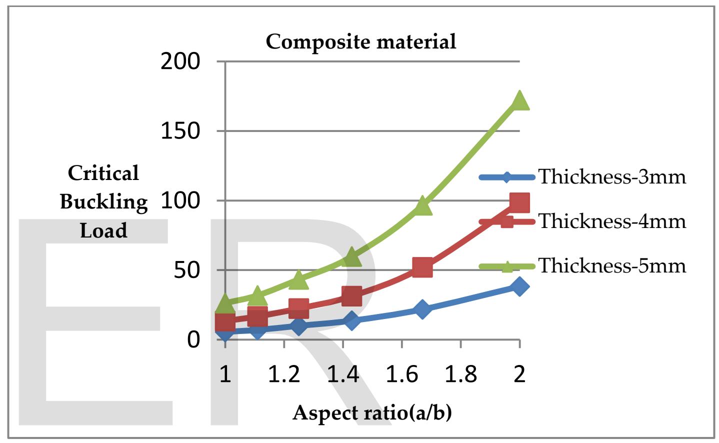

Fig:-21 Comparison of critical buckling load with aspect ratio for composite plate without cut hole at various plate thickness.

The fig.21 shows variation of aspect ratio (a/b) for composite plate without circular hole. It is observed that as a/b ratio increases the critical buckling load increases.

6 CONCLUSIONS

This study considers the critical buckling load response of homogeneous and laminated heterogeneous rectangular with simply supported boundary conditions. The laminated composite plates have varying aspect ratio, varying thickness t ratio, cut out edge, centre of hole and without cut plate are considered. From the present analysis, the following conclusions are made:

- It was noted that the buckling load/unit length decreases with increases of aspect ratio (a/b).
- As the b/t ratio increases the buckling load decreases.
- The critical buckling load is increases with the different materials in plate thickness 3mm.
- The critical buckling load for homogeneous plate is more compare to laminated heterogeneous plate.

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8 REFERENCES

[1] Jana P and Bhaskar K (2006), "Stability analysis of simply-supported rectangular plates under non- uniform uniaxial compression using rigorous and approximate plane stress solutions". Thin-Walled Struct. 44: 507–516

[2] Chai G B, Ooi K T and Khong P W (1993)," Buckling strength optimization of laminated composite plates" Comput. Struct. 46: 77–82

[3] Hu H, Badir A and Abatan A (2003), "Buckling behavior of graphite/epoxy composite plate under parabolic variation of axial loads". Int. J. Mech. Sci. 45: 1135–1147

[4] Jain P and Ashwin K (2004), "Post buckling response of square laminates with a central Circular/elliptical 0". Compos. Struct. 65: 179– 185

[5] ANSYS Documentation.Timoshenko S. P., (1961) *Theory of Elastic Stability*, McGraw-Hill, New York.Choudary R. B. "Introduction to ANSYS 10.0" *I. K. International Publishers Pv .Ltd.*

[6] Dr. A. V. Phan(1981) "ANSYS tutorial- 2-D Fracture analysis" *University of SouthAlabama.* Ugural,A. C. "Stresses in Plates and Shells", U.S.A.: McGraw-Hill Book Company

[7] Khan MZ, Walker AC(1972) Buckling of plates Subjectedtolocalized edge loading. Struct Engi;50(6):225–32.

[8] Spencer HH, Surjanhata H.(1985) Plate buckling under partial edge loading, In: Developments in mechanics , Proceedings of the 19th mid-western mechanics conference, vol., 13; p. 83–4.

[9] Bert CW, Devarakonda KK(2003). Buckling of rectangular plates subjected to nonlinearly distributed in-plane loading. Int J Solids Struct ;40:4097–106.

[10] Van der Neut A(1958). Buckling caused by thermal stresses. In: High temperature effects in aircrafts structure. AGARD report, vol. 28, p. 215–47.

[11] Benoy MB.(1969) An energy solution for the buckling of rectangular plates under non-uniform in-plane loading. Aeronaut J ;73:974–7.

[12] Little RW(1973). Elasticity. New Jersey: Prentice-Hall;

[13] Gorman DJ, Singhal RK(1993). A superposition Rayleigh–Ritz method for free vibration analysis of non- uniform tensioned members. J Sound Vib ;162:489–501.