Buckling Analysis of Composite Cylindrical Shells with Multiple Cutouts

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Declaration

I certify that this research work titled "*Buckling Analysis of Composite Cylindrical Shells with Multiple Cutouts"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

Signature of Student

Adil Abid Raja

2016-NUST-Ms-Mech-205969

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This thesis has been checked for Plagiarism. Turnitin report endorsed by Supervisor is attached.

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Abstract

The aim of the study is to analyze the effect on the critical buckling load for the cylindrical shell with multiple cutouts which are subjected to compressive loading. The study will focus on the effect of the shape of the cutout including circular, rectangular and the elliptical. The effect of the cross sectional area of the cutout will be analyzed using the buckling analysis and the position of the cutout will be changed in axial direction to see the effect on the critical buckling load value. The research will focus primarily upon the effects of the geometry of the cutouts and how these factors will affect the critical buckling strength of the cylindrical shell. The material used in the cylindrical shell is composite which has fixed number of plies and the orientation. The composite material has been modelled using the Ansys Composite PrepPost and the Linear Eigenvalue buckling analysis has been carried out in the Ansys. The buckling analysis in this case uses the linear assumptions and therefore is suitable for small deformations. The results obtained from the Finite Element analysis gives more buckling stability without any cutout on the cylinder but there has been a reduction in the buckling strength when the cutout is introduced. The rectangular cutout has the lowest buckling strength while the elliptical and circular cutouts are more resistive to buckling at lower modes of failure. By increasing the cross sectional area the circular cutout is not suitable for large cross section since there is significant reduction in the buckling strength is observed. Similarly by changing the position of the cutout along the axial axis the percentage reduction in buckling strength is highest for the circular cutout. The main conclusion is that for small cross sectional area the elliptical cutout has more buckling strength as well as the circular cutout. By increasing the cross sectional area or changing the axial position of the cutout the circular cross section has less stiffness to withstand the axial compressive forces.

Key Words: *Critical buckling, Compressive loading, buckling failure, Cutouts, Shells*

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Chapter 1: Introduction

The cylindrical shells are most widely used in the mechanical engineering, aerospace engineering and civil engineering, etc. In most of the designs we need to introduce the cutouts for various reasons and the shape of the cutout is in rectangular, circular and in some cases has elliptical geometry. The cutouts are made in the design to serve as windows, doors or some sort of access points. It can be analyzed from many of the authors that introduction of a particular cutout subjected to compressive loading can result in a significant reduction in the critical buckling load of the shell. Buckling refers to the failure of the structure when the applied load exceeds the critical value and can no more sustain the load. The material used in the structure must have strength to withstand the deformations without yielding. Failures due to buckling can cause huge catastrophic damage since buckling can occur without any prior notice as it happens very suddenly.

The buckling failure in the structure usually occur in the form large deformations but from the engineering perspective the buckling phenomena usually occurs before the large deformations being developed and the structure is slightly deformed. The study of buckling is an important phenomena in the structural and mechanical design since the buckling can lead to complete failure of the structures. So for the shells structure the buckling phenomena can happen any time without any prior warning and therefore can have devastating effects. The mechanical properties of the composites, which include higher strength, low weight, more performance and enhanced service life, make them very suitable as a design material to be used in making cylindrical shells.

In this research we will be focusing on the composite cylindrical shells with various configurations of the cutouts of various cross sections and how they affect the buckling mode and the critical buckling value. With the introduction of the cutout the stress concentration around the region of the cylinder increases which thereby reduces the buckling strength of the composite cylinder under axial loadings. In this research we will be focus on finding the buckling strength using the FEA in the Ansys.

1.1 Objective

The aim of the study is to understand the effect of the cutout on the composite cylindrical shell under axial compressive loading. In this research we will be using the linear assumptions for the eigenvalue buckling analysis to predict the failure mode for buckling and how the strength is changed as the cutout cross sectional area and the relevant position of the cutout is changed across the axial length of the cylinder. The cutouts made will be circular, elliptical and rectangular under the analysis will be observed to see the affect produced on the buckling strength under same boundary conditions.

Chapter 2: Literature Review

The cylindrical shells which are subject to compressive loading have been studied by many authors by considering analytical methods and finite element analysis. The buckling load estimated using the classical buckling theory is much greater than the actual buckling load acting upon the cylindrical shell. Hilburger et al. [1] has studied the effect of the imperfections in the form of localized stress with the critical load. Study done by Wang et al. [2] was on the stiffened cylindrical shells with various cutouts. The geometrical shape of the cutout is in the form of circular, rectangular or elliptical.

Allahbakhsh et al.[3] has done research on the buckling analysis of the composite panels made of lamination having elliptical cutout under the action of axail compressive loading. Their study has made three important considerations which icludes effect of lamiantion of the buckling strength, optimizing the oreintation of the aligned stacked fibres in such a manner as to have more buckling strength and also optimizng the design parameters fo the cutout to have more buckling resistacne. Haslin et al.[4] did a numerical simulation on the composite cylinders to see the effect of the buckling strength subjected to internal pressure. In his research he studied the how changing the dimensions of the circualr and rectangular cutout can change the critical buckling load for the cylinder. In his study there is no correlation established between the geometric imperfrction and the cutout size in the cylinder.

Gangadhar et al.[5] has done research where he study the effect of the circular cutouts with the introduction of the imperfection under the buckling of cylinder. The results obtained from the simualtion shows that the critical buckling load under the axial compressive forces in three zones. There is an impact of imperfections on the size of the cutout and it is function of the radius of the cylidner. Fesharaki et al.[6] investigated the axial compressive behavior of the composite cylindrical shell with opening using the analytical and the finite element analysis. The post buckling behaviour of the composite cylinder under the external pressure based on the deformation theory of shear has been carried out by Sonmez et al.[7]

Shokrieh et al.[8] has done study which is based upon the numerical method to evaluate the buckling strength and behaviour of cylinder and the tensile strength of the structure.

The buckling strength subject to compressive axial loading for a cylindrical shell is more susceptible to imperfections than the perfect shells subject to other type of loading conditions. The behavior of the buckling phenomena varies significantly due to changes in the geometrical pattern, type of boundary conditions being imposed and the magnitude of imperfections in the geometry has been discussed by Omidi et al. [9]

The simultaneous effect of the buckling design and the stiffened cylindrical shells with multiple cutout have been studied by Farshad et al [10]. The buckling optimization of the fibre composite laminated shells using the in-plane nonlinearity was investigated by the Walt et al. [11]

In the study done by Ramaniah,J et al.[12] he has compared the effect of cutout and with no cutout on the orthotropic composite cylindrical shell. In his research he concluded two important results; one is that the cutout has reduced the buckling strength of the cylinder significantly under the same conditions. The other important consideration is that the cutout has impact on the buckling strength of greater magnitude when the position is changed.

Wiedemann et al [13] has investigated the buckling analysis based on the deformation energy of the straight composite z-frames subjected to critical buckling loads and the effect on increasing the size of the cylinder to the buckling load.

Shirkavand et al [14] in his paper makes research on the orientation and the size of the rectangular cutout on the buckling analysis of the cylinder made of composite under axial compressive loadings. His study doesn't take any nonlinerity into consideration and the impact of changing the position of the cutout on the cylidenrical shell.

Chapter 3: Methodology

To model the composite structure for the cylindrical shell we have use the Ansys Composite PrepPost (ACP) which allows the creation of the fabric, stacking sequence and the ply orientation and number of thickness of the plies made. The Finite Element analysis has been done in the Ansys Workbench where the model has been created, and then analyzed using the Linear Eigenvalue buckling.

In the Linear Eigenvalue buckling the critical value for the load of the structure is determined on the linear approximation. The system uses the standardized equation which has the form: This method is very good for small deformations and can yield results with less computational time due to its assumptions of linear solution.

The results for the buckling modes failure and the critical buckling loads were then computed and the comparison has been carried out for each cutout configuration.

$$|K_{con} + \lambda K_{\sigma}|x = 0$$

Where the

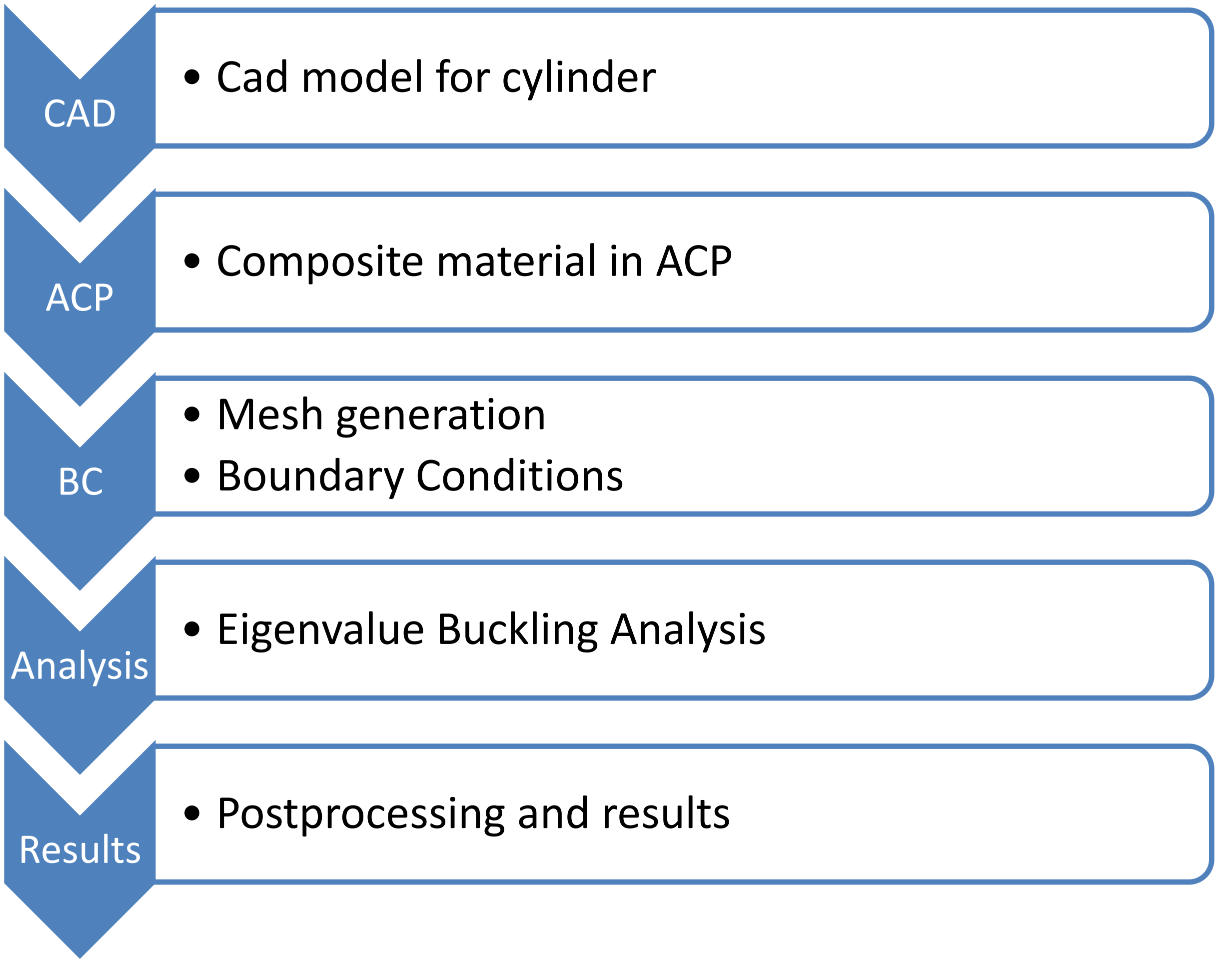
$$K_{con}\ refers\ to\ the\ constitutive\ stiffness\ matrix$$

$$K_{\sigma}\ is\ the\ stiffness\ matrix\ for\ the\ geomter$$

$$\lambda\ is\ the\ Eigenvalue$$

$$x\ is\ the\ direction\ vector\ for\ Eigenvalue$$

3.1 Schematic of Work



Chapter 4: Results

The design and analysis of the cylindrical shell subjected to axial compressive force has been carried out in a proper method comprising of making geometry, followed by applying the composite material. The model is then meshed with appropriate number of elements to yield the best possible results. Then the model is applied with the boundary conditions to evaluate the shell for eigenvalue buckling. In the end post processing is done to compute the contours for bucking mode and buckling load factor at which failure can occur.

4.1 Geometry

To start the analysis we need to first model the design in the Ansys Geometry design modeler. There are various configurations of the model being generated where we have varied the parameters for the cutouts including circular, elliptical and rectangular. Following are the details and dimensions for the cylindrical shell in which the parameters are fixed for all the configurations and the dimensions of the cutouts have been varied to study the effect on the buckling load factor under axial compressive loading.

$$Outer\ diameter\ for\ shell = 2.5\ m$$

$$Inner\ diameter\ for\ shell = 2.492\ m$$

$$Thickness\ of\ shell = 0.008\ m$$

$$Length\ of\ circular\ shell = 8\ m$$

CAD Model without Cutout

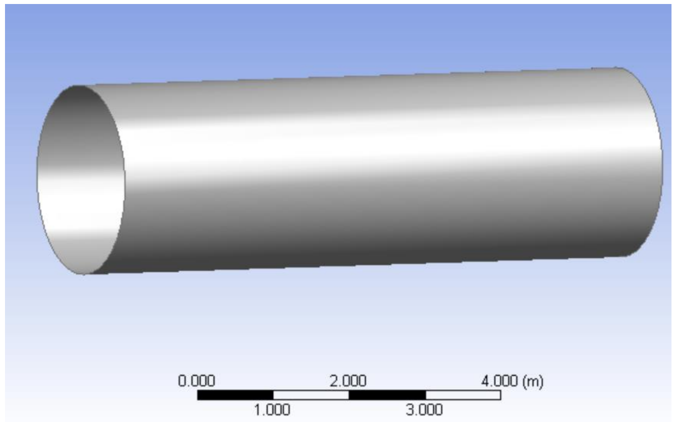


Figure 1 Cad model for the Cylinder

Circular Cutout

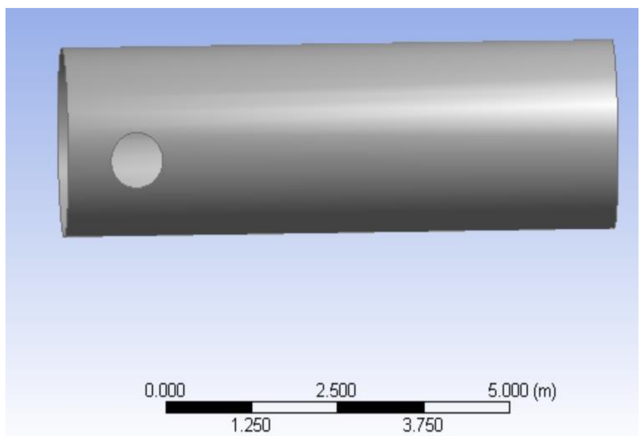


Figure 2 Cad model for the circular cutout

Rectangular Cutout

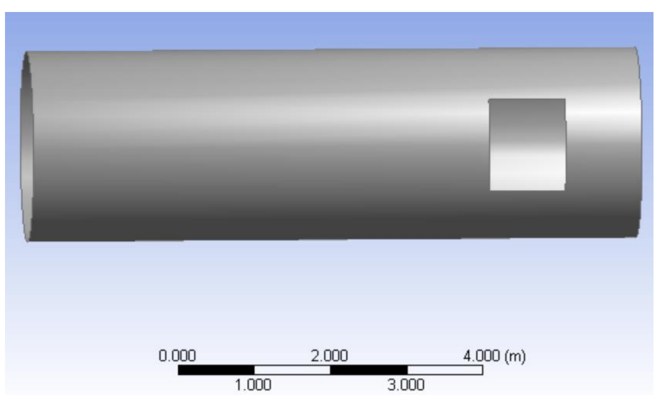


Figure 3 Cad model for the rectangular cutout

Elliptical Cutout

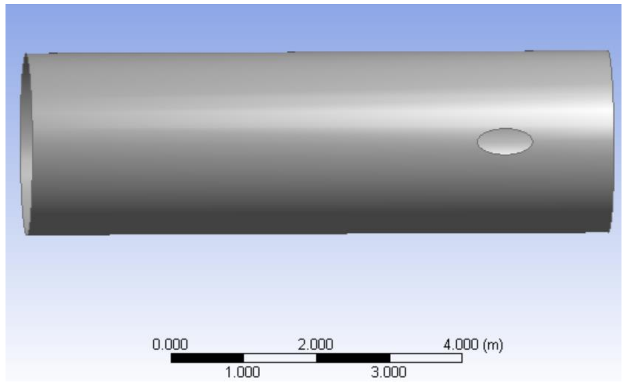


Figure 4 Cad model for the elliptical cutout

All the cutouts are made in a way so that the cross sectional area of the cutout for the circular, elliptical and the rectangular are consistent so we can do the parametric study by changing the size of the cutout.

4.2 Sample Calculation (Single Cutout)

For Circular Cutout

$$Diameter\ for\ the\ cutout = 0.5\ m$$

$$Cross\ sectional\ area\ of\ circular\ cutout = \pi * \frac{d^2}{2}$$

$$Cross\ sectional\ area\ of\ circular\ cutout = 3.142 * \frac{0.5^2}{2}$$

$$Cross\ sectional\ area\ of\ circular\ cutout = 0.1963\ m^2$$

Rectangular Cross Section

$$Length\ for\ the\ cutout = 0.4\ m$$

$$Width\ for\ the\ cutout = 0.49\ m$$

$$Cross\ sectional\ area\ of\ circular\ cutout = L * B$$

$$Cross\ sectional\ area\ of\ circular\ cutout = 0.4 * 0.49$$

$$Cross\ sectional\ area\ of\ circular\ cutout = 0.196\ m^2$$

Elliptical Cross Section

$$Major\ Radius(a) = 0.35\ m$$

$$Minor\ Radius(b) = 0.18\ m$$

$$Cross\ sectional\ area\ of elliptical\ cutout = \pi * a * b$$

$$Cross\ sectional\ area\ of elliptical\ cutout = 3.142 * 0.35 * 0.18$$

$$Cross\ sectional\ area\ of elliptical\ cutout = 0.198\ m^2$$

In the similar manner we increased the cutout diameter for the circular shape and find the corresponding cross sectional area which is then used as a base value to find the relevant dimensions for the rectangular cutout where the length and the breadth is adjusted so to give the same cross sectional area as incase of the circular cutout. In the same way we did adjust the values for the major radius and the minor radius and find the cross sectional area of the elliptical cutout which must be consistent with the values obtained from the circular and rectangular cross sections.

Table 1 Details for the dimensions of the Cutout

Circular		Rectangular			Elliptical		
Cutout diameter/m	Cross sectional Area/m^2	Length/m	Width/m	Cross sectional Area/ m^2	Major Radius/m	Minor Radius/m	Cross sectional Area/ m^2
0.5	0.196375	0.49	0.4	0.196	0.35	0.18	0.2
0.65	0.33187375	0.55	0.6	0.33	0.5	0.2	0.33
0.8	0.50272	0.67	0.75	0.5025	0.65	0.25	0.5
0.95	0.70891375	0.8	0.88	0.704	0.7	0.3	0.71
1.1	0.950455	0.95	1	0.95	0.85	0.35	0.95

The above table summarizes all the case studies being generated by changing the parameters for all the cutouts and the relevant dimension for each of the cutout to yield the same cross sectional area for the study of buckling strength of the composite cylindrical shell under compressive axial loading. As we can clearly see that the cross sectional area for the circular cutout, elliptical cutout and the rectangular cutout are almost same in magnitude which makes the comparative analysis easier and the approach can generate some reliable results as to which cutout is more resistant to deformation and failure under buckling modes.

4.3 Material Selection and Properties

After the model is designed we need to apply the specific material to the cylindrical shell so that we can carry out the structural analysis followed by the buckling analysis in the Ansys software. There are many available materials which can be assigned such as metals, composites, plastics, ceramics and other alloys. Each of the designed material has certain mechanical properties due to which it is suited under those applications, Metals are very strong in terms of resistance to buckling, higher strength, more resistant to higher temperature and are considered very rigid. But metals can be very costly in terms of manufacturing and weight can be a big issue due to which we can shift to composites which are very strong and cost effective as well.

4.3.1 Composites

Composites are the material which are composed of two or more than two constituent materials with significantly different chemical and the physical composition. And when they are combined the final material has the properties which are different from the individual constituents materials from which the composite is made up of. Composites which are known as CFRP refer to the Carbon fibre-reinforced plastics which are a mixture of reinforcements and matrix. If the individual materials are considered none of them have the strength or rigidity but when they are combined the resultant product has very strong structural strength and is considered as a rigid material which can’t be deformed very easily. Mostly the composites are used for the buildings, large and massive structures, spacecraft and defense industries.

4.3.2 Overview of Composite Structure

The composites are made from the 2 main constituent materials which includes matrix, which serves as binder, and the reinforcement. The main function of the matrix material is to surround and support the reinforced material to its specific position. On the other side the reinforcement material impacts and enhance the mechanical properties of the matrix. In most of the composites the matrix is composed of a polymer matrix material which is in the form a resin solution. Among them are epoxy, polyester, vinyl ester and etc. The reinforcement materials mostly comprises of the fibres. The fibre-reinforced composite are most commonly divided into 2 main categories which are classified as

1. Continuous fibre-reinforced composites: They are usually in the form of constitute layering or laminated structure.
2. Short fibre-reinforced composites: The short fibres are usually made from the compression molding and sheet forming.

Common fibres used in case of reinforcement consists of the glass fibre, carbon fibre and the polymer of higher strengths.

4.3.4 Advantages of the composites

1. Very high resistance to the chemicals
2. High strength to weight ratio
3. Very effective for fatigue strength
4. Impact strength is very high
5. Design flexibility and more efficient under harsh environments
6. Lighter and stronger

4.4 Material Assigned

Epoxy Carbon UD Prepregs

Prepegs are the reinforced fabric which is already pre-impregnated with a resin structure where the resin is composed of epoxy. The advantages of the Prepregs can provide maximum strength enhancement properties, structural uniformity and regularity in the structure, lesser curing time and better manufacturing ability to make the composite.

Table 2 Properties for the assigned composite material

Orthotropic Elasticity	**Value**
Young's Modulus X direction	1.21E+05 MPa
Young's Modulus Y direction	8600 MPa
Young's Modulus Z direction	8600 MPa
Poisson's Ratio XY	0.27
Poisson's Ratio YZ	0.4
Poisson's Ratio XZ	0.27
Shear Modulus XY	4700 MPa
Shear Modulus YZ	3100 MPa
Shear Modulus XZ	4700 MPa

The fabric is made from the epoxy carbon with the thickness of 8mm. The stack up sequence for the composites comprises of following sequence -45°/45°/90°/0°. The sequence is symmetrical and the top down layup sequence is followed. The number of plies modelled for the composites are 5.

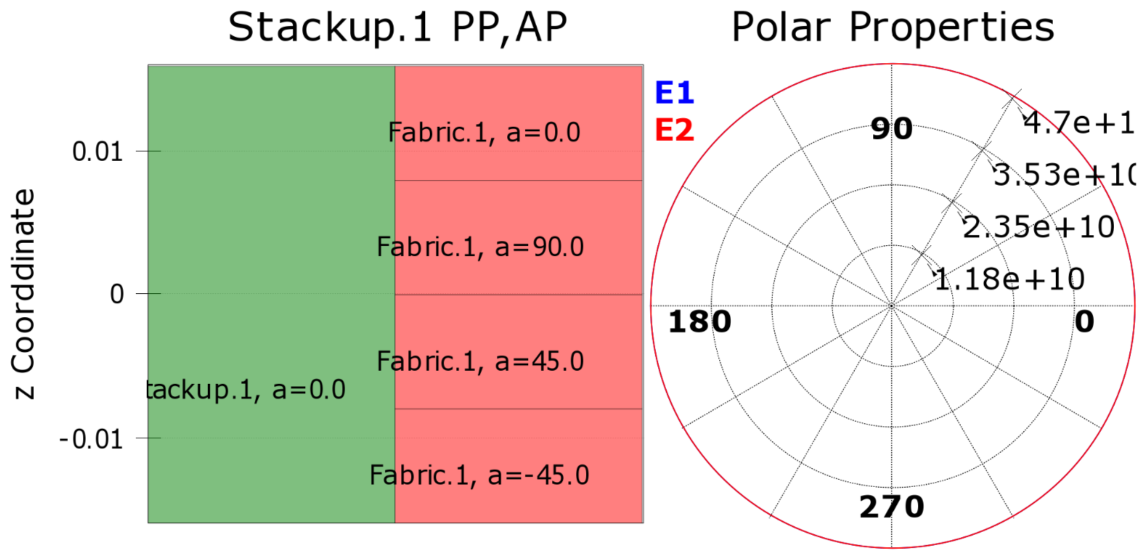


Figure 5 Stacking sequence for the composite

4.5 Meshing

The design modelled is then meshed with appropriate number of elements and nodes so to satisfy the obtained results. The meshing details must consider and ensure proper meshing density which can be very effective in producing the results.

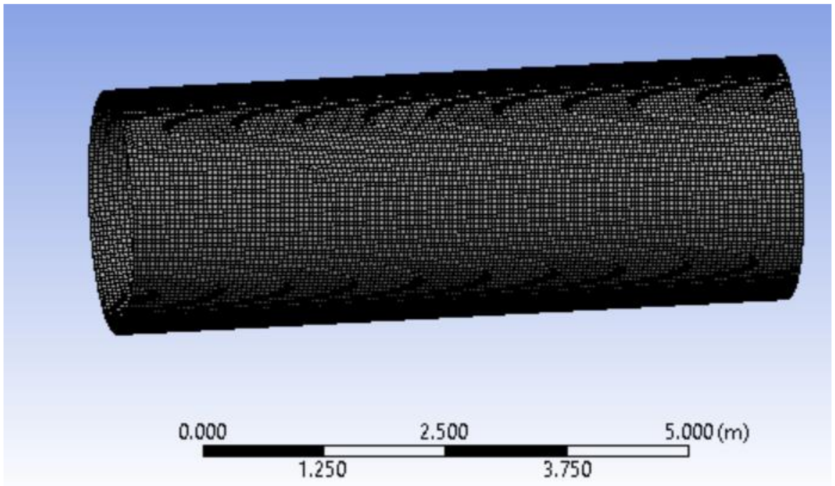


Figure 6 Meshing for the Cylinder

In the mesh options the size function is set to curvature, relevance center is set to Coarse and the growth rate is set to normal. The number of meshed nodes are 23473 and the Elements are 23318.

Body Sizing

The body sizing is a technique where we select the complete part and divide the element size to a minimum value. In this case the element size is set to 5mm.

4.6 Boundary Condition

Before carrying out the analysis we need to apply the specific boundary conditions which can lead to proper results. There are many different boundary conditions available and one must select appropriately to model the physics and dynamics of the problem according to the real working situation.

1. Fixed boundary Condition:

In the fixed boundary condition we have to fix the part to restrict translational movement as well as the rotational movement in all axes. This boundary condition is applied to restrict the motion in that particular region. In our case of analysis we have fixed the lower end of the composite cylindrical shell in restrict the motion in following:

$$Rx = Ry = Rz = 0$$

$$Mx = My = Mz = 0$$

This means that the rotation is fixed in all directions and the translation is fixed in all directions.

Fixed Boundary Condition

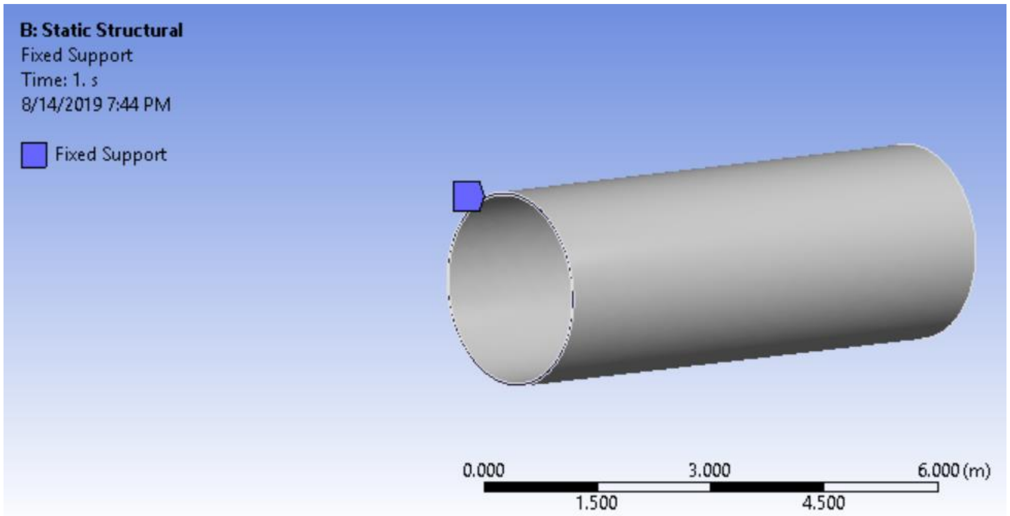


Figure 7 Fixed boundary condition

2. Force

To simulate the model for the Eigen value buckling we will apply the force on the other end. This is in accordance to the Euler column buckling where one end is fixed in all types of motions and the free end is then given a force to simulate the buckling modes and the magnitude of the critical buckling load factor is then found using the analysis.

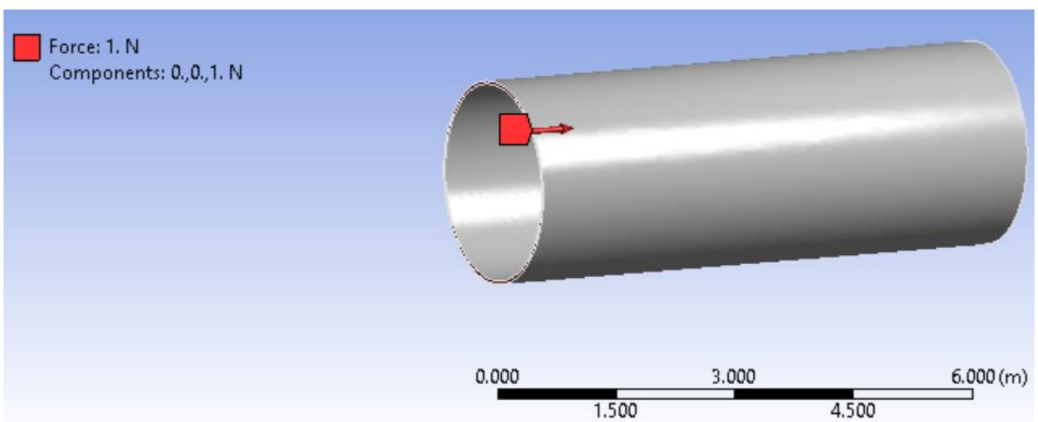


Figure 8 Force on the side of the cylinder

4.7 Simple Composite Cylindrical Shell Contours

1st Buckling Mode

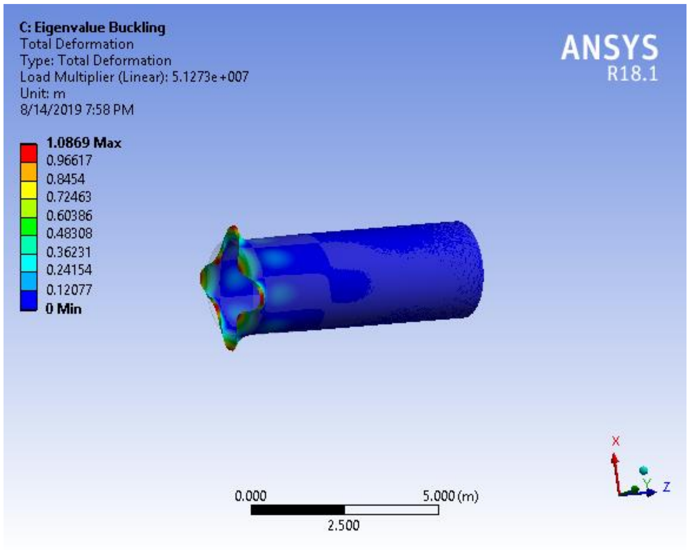


Figure 9 1st Buckling mode for simple cylinder

2nd Buckling Mode

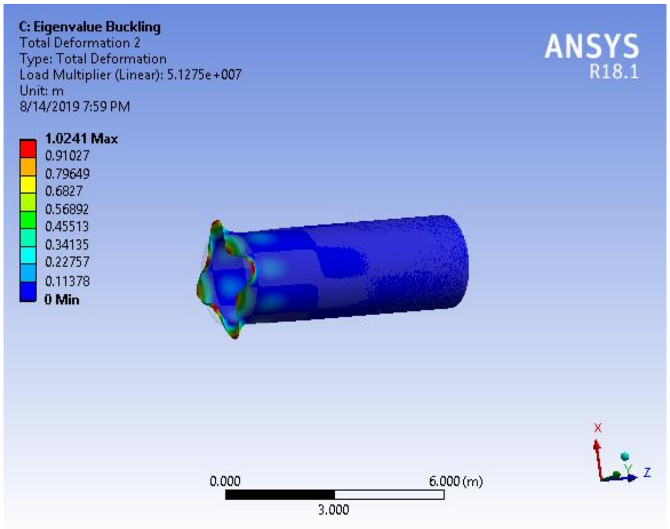


Figure 10 2nd Buckling mode for simple cylinder

3rd Buckling Mode

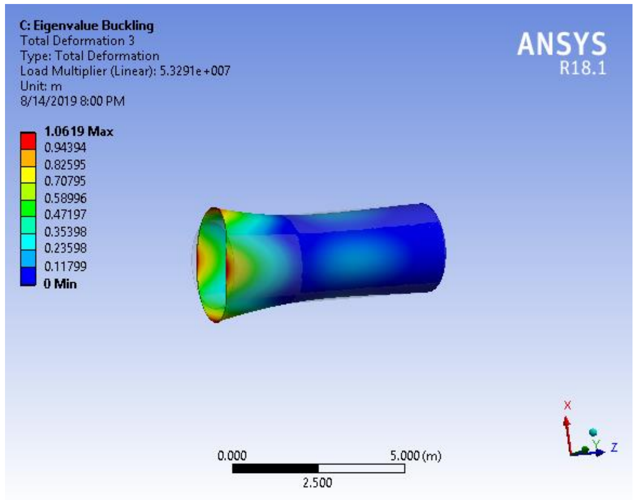


Figure 11 3rd Buckling mode for simple cylinder

4th Buckling Mode

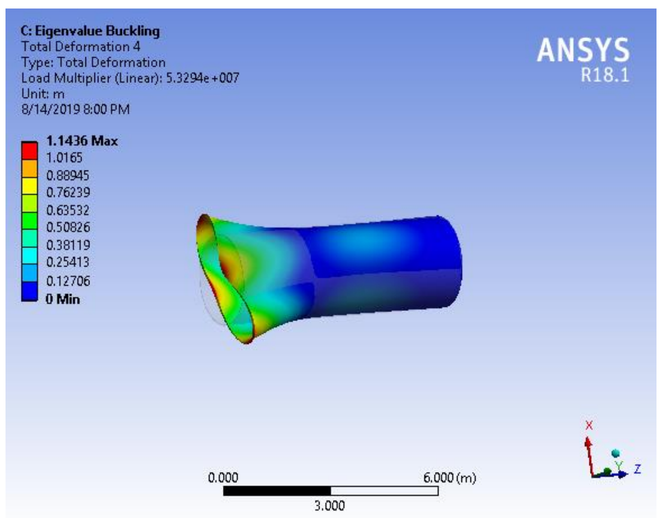


Figure 12 4th Buckling mode for simple cylinder

Critical Buckling load (1st Mode)

$$Force\ applied = 1\ N$$

$$Load\ Multiplier = 5.1273 * 10^7$$

$$Critical\ buckling\ load = Force\ applied * Load\ Multiplier$$

$$Critical\ buckling\ load = 1 * 5.1273 * 10^7 = 5.1273 * 10^7 N$$

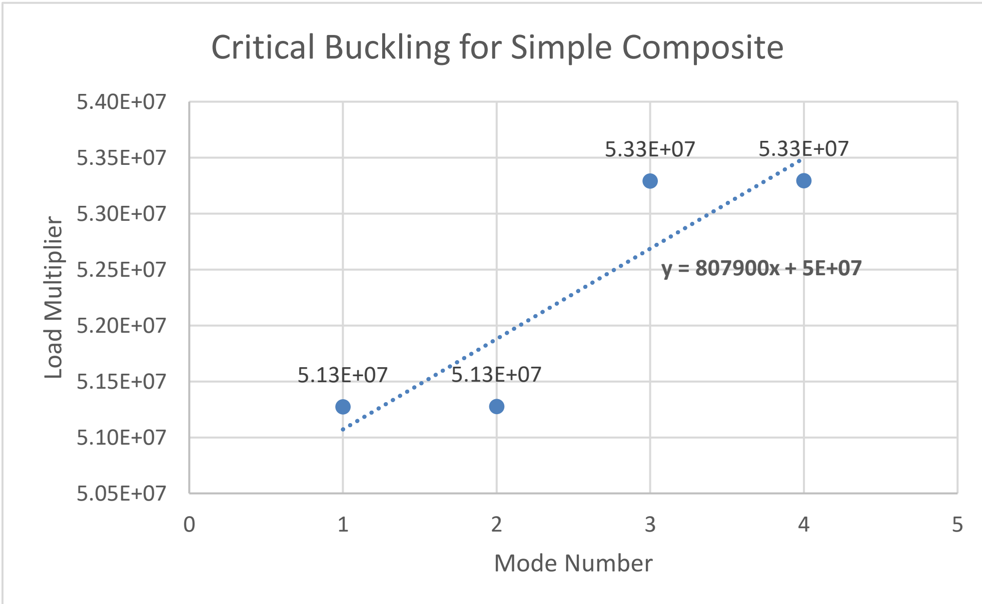


Figure 13 Critical buckling load vs mode number

The first 2 modes for the buckling failure are more susceptible to failure so we must consider the critical buckling load factor for the first 2 Modes to evaluate all the designs under Eigen value buckling analysis. From the graph we can observe that the first 2 buckling modes are occurring at lower frequencies and both modes have the similar magnitude, and in the similar manner the mode number 3 and 4 have same values. So in our proceeding case studies we are more focused on the first 2 buckling mode failures.

Linear Correlation

$$y = 807900\,x + 5 * 10^7$$

$$y\ is\ the\ buckling\ load\ multiplier$$

$$x\ is\ the\ Buckling\ Mode$$

This is the linear trend line equation to correlate the mode number with the load multiplier for the buckling.

Now we will introduce the single cutouts to the composite cylinder in the shape of circular, rectangular and the elliptical and model them in the same manner. Then we will introduce another cutout so the total number of cutouts will be 2 and will be equidistant from each other in the arrangement.

Circular Single Cutout

1st Mode

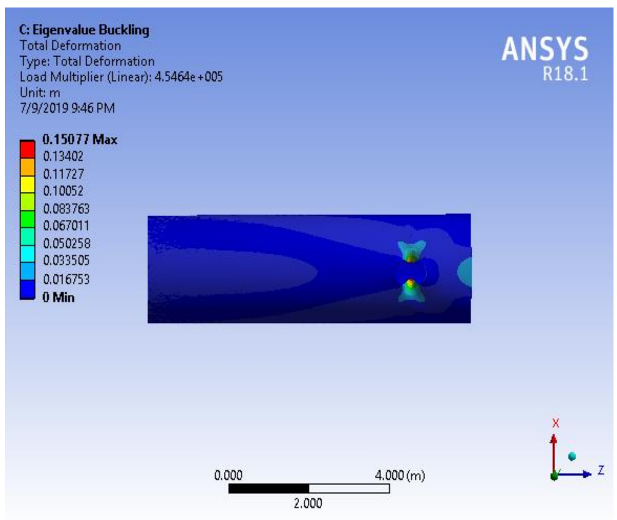


Figure 14 1st buckling mode for circular cutout

2nd Mode

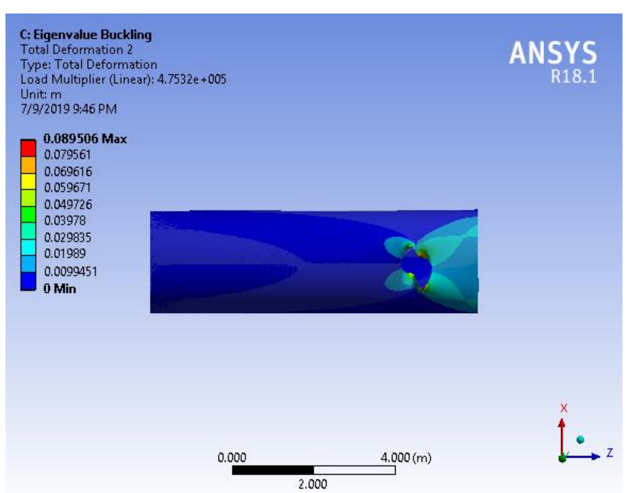


Figure 15 2nd buckling mode for circular cutout

Elliptical Single Cutout

1st Mode

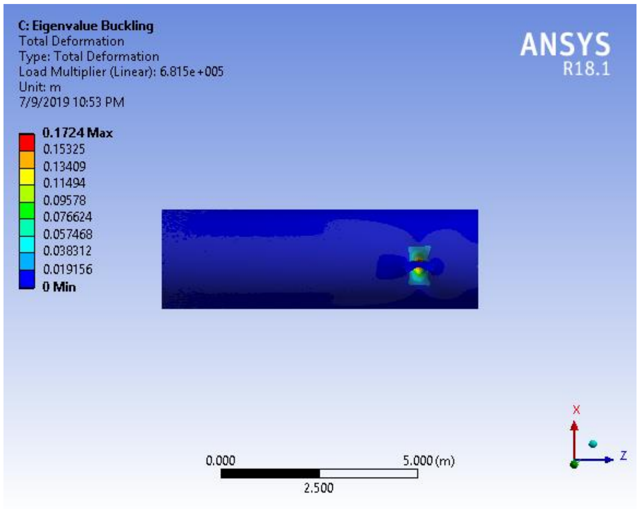


Figure 16 1st buckling mode for elliptical cutout

2nd Mode

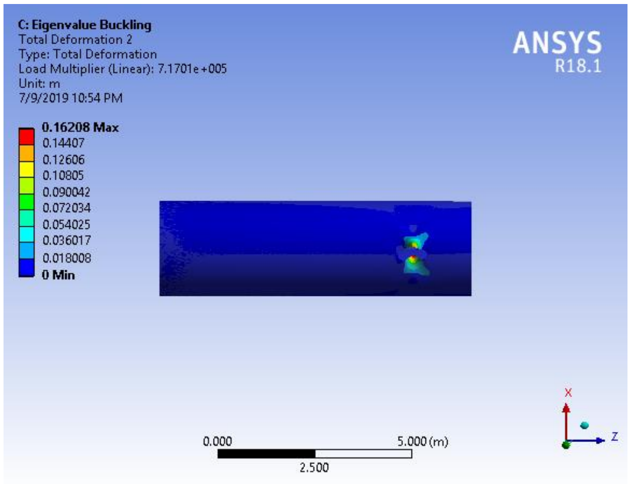


Figure 17 2nd buckling mode for elliptical cutout

Rectangular Single Cutout

1st Mode

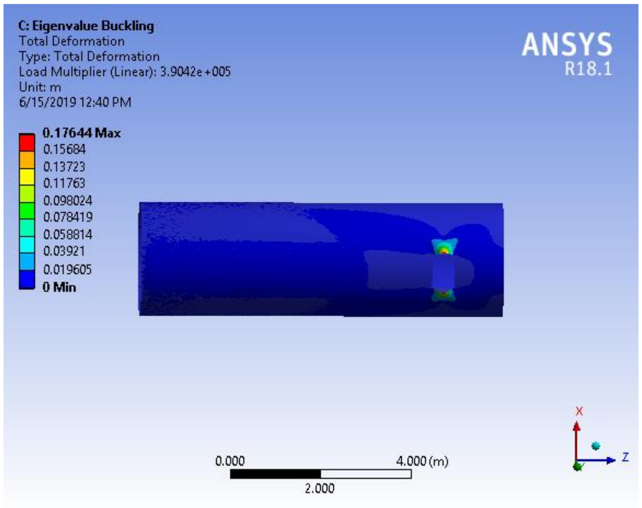


Figure 18 1st buckling mode for rectangular cutout

2nd Mode

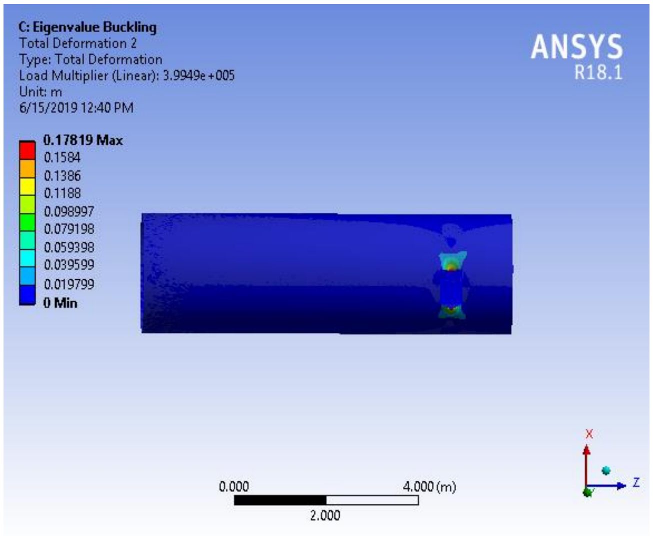


Figure 19 2nd buckling mode for rectangular cutout

Comparison of all Cutouts

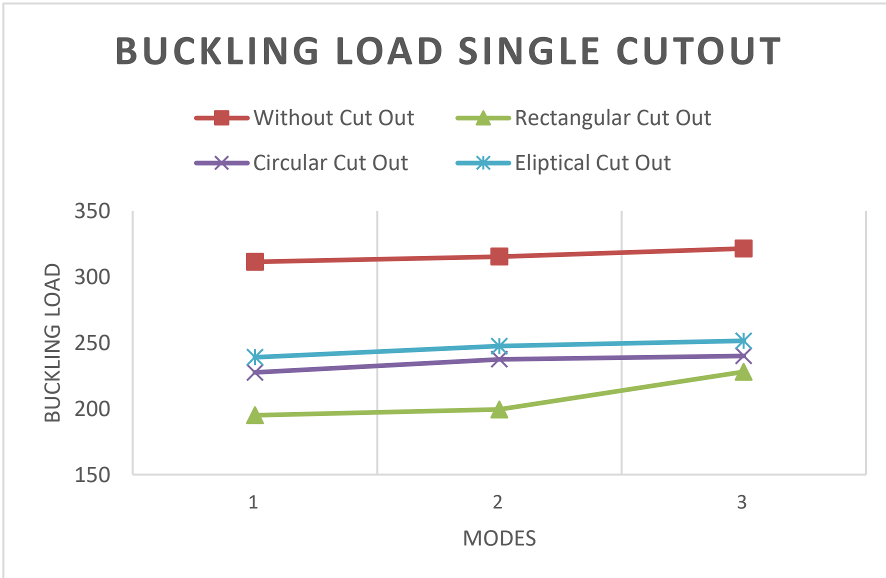


Figure 20 Comparison for buckling load for single cutout

$$Applied\ Load = 2000\ N$$

$$Critical\ buckling\ load = Force\ applied * Load\ Multiplier$$

For Circular Cutout

$$Critical\ buckling\ load = 2000 * 227.5 = 4.5 * 10^5\ N$$

From the above graph we can see that the cylindrical shell without any cutout is more stable and resistant to the buckling failure as compared to shells with the cutouts. Among the cutouts the rectangular cutouts are most likely to buckle at earlier loads which means the rectangular cutouts have less structural strength as compared to circular and elliptical cutouts. On the other side the circular and elliptical cutouts have much higher strength and are more resistant to the buckling failure under same loading conditions. Among the three cutouts elliptical cutouts are most efficient in providing more strength and resistance to the buckling mode failures.

Percentage Reduction in Buckling Load

Table 3 Percentage reduction in buckling strength

Modes	Percentage Reduction-Rectangular (%)	Percentage Reduction-Circular (%)	Percentage Reduction-Elliptical (%)
1	37.37253705	26.93462656	23.24121208
2	36.74197384	24.692826	21.52199762
3	29.07249848	25.33947209	21.76198846

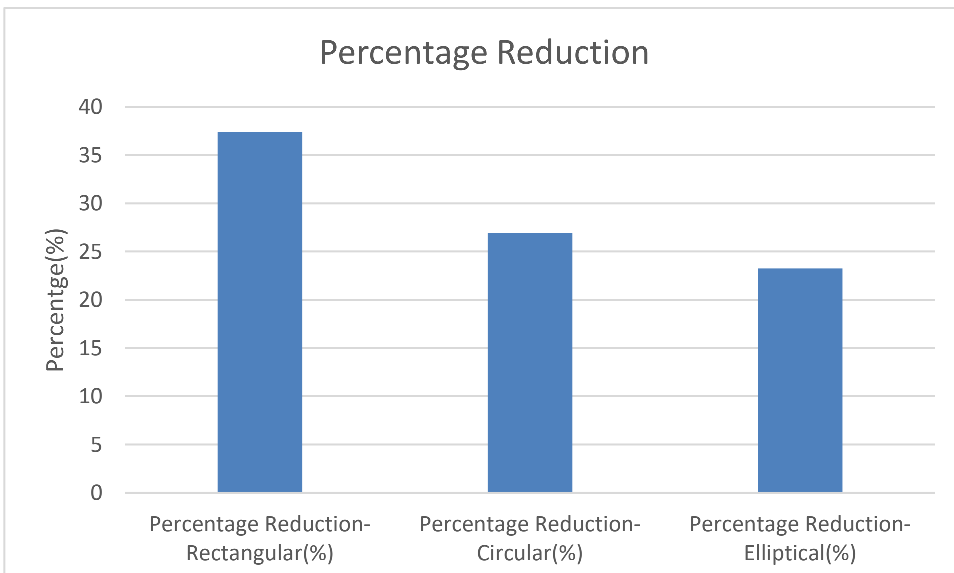


Figure 21Percentage Reduction in Buckling Load

The above table summarizes the percentage reduction in the value for the buckling load in comparison with the cylindrical shell without cutout. This is to analyze the strength of the shell when particular cutout is made. For the rectangular cutout the percentage reduction in the buckling load for the first mode is 37.37 % which is highest among all cutouts. For the same mode the percentage reduction for the circular is 26.93 % and for the elliptical the percentage is 23.24 %. So form the results we can conclude that the strength is highest of the cutout is of elliptical shape and lowest with the rectangular cutout.

Double Cutout-Circular Cutout

1st Mode

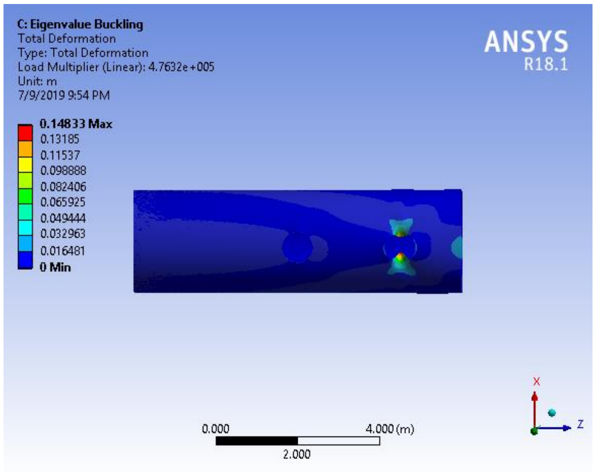


Figure 22 1st Buckling mode failure for double circular cutouts

2nd Mode

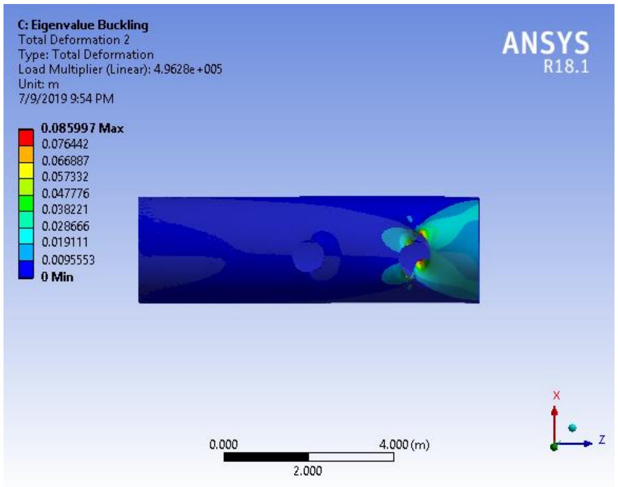


Figure 23 2nd Buckling mode failure for double circular cutouts

Double Cutout-Elliptical Cutout

1st Mode

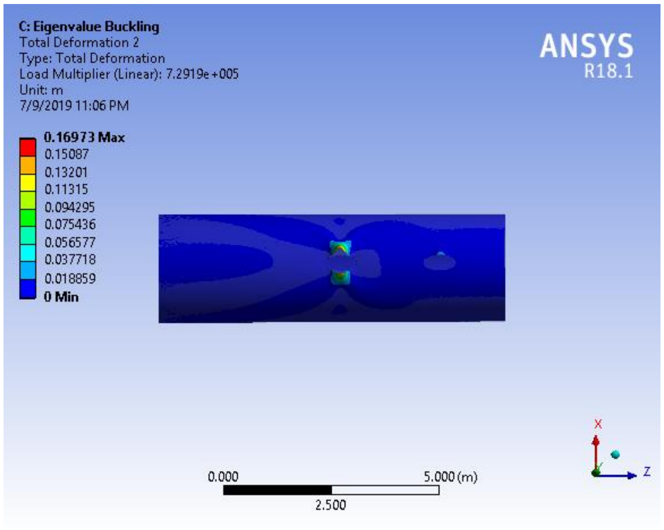


Figure 24 1st Buckling mode failure for double elliptical cutouts

2nd Mode

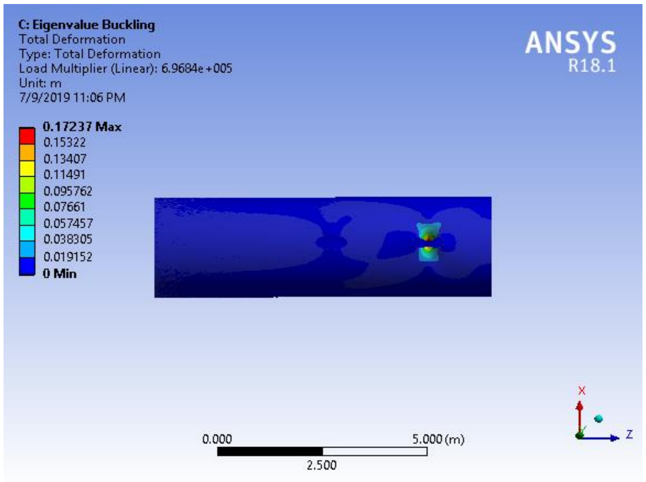


Figure 25 12nd Buckling mode failure for double elliptical cutouts

Double Cutout-Rectangular Cutout

1st Mode

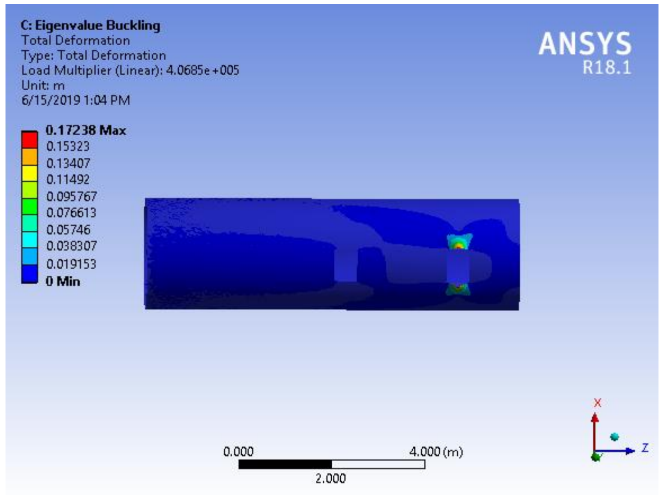


Figure 26 1st Buckling mode failure for double rectangular cutouts

2nd Mode

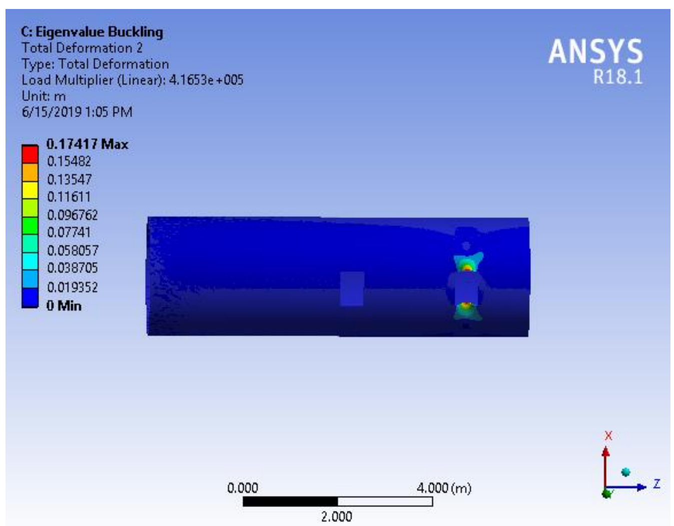


Figure 27 2nd Buckling mode failure for double rectangular cutouts

Comparison of Double Cutouts

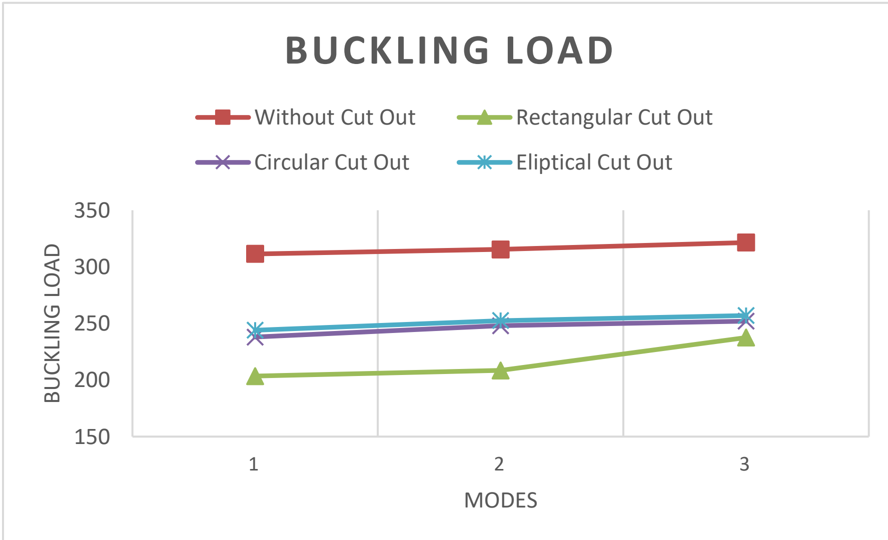


Figure 28 Comparison for buckling strength with double cutouts

$$Applied\ Load = 2000\ N$$

$$Critical\ buckling\ load = Force\ applied * Load\ Multiplier$$

For Circular Cutout

$$Critical\ buckling\ load = 2000 * 238 = 4.7 * 10^5\ N$$

Table 4 Percentage reduction for double cutouts

Modes	Percentage Reduction-Rectangular (%)	Percentage Reduction-Circular (%)	Percentage Reduction-Elliptical (%)
1	34.642622	23.56237856	21.6353797
2	33.8882283	21.3634562	19.93658343
3	26.11718592	21.60644569	20.05101803

The above table summarizes the percentage reduction in the value for the buckling load with double cutout in comparison with the cylindrical shell without cutout. This is to analyze the strength of the shell when particular cutout is made. For the rectangular cutout the percentage reduction in the buckling load for the first mode is 34.62 % which is highest among all cutouts. For the same mode the percentage reduction for the circular is 23.52 % and for the elliptical the percentage is 21.63 %. So form the results we can conclude that the strength is highest of the cutout is of elliptical shape and lowest with the rectangular cutout.

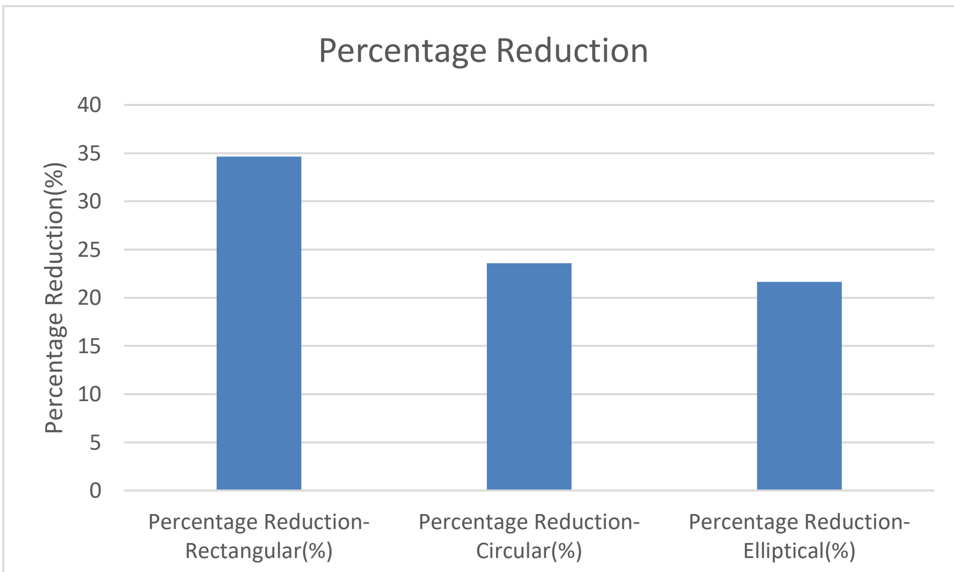


Figure 29 Percentage reduction for double cutout

Area ratio for Circle Cutout

In the next step we will model the ratio of the cross sectional area of the cutout with the original cross sectional for the cylindrical composite to a specific range and see the effect on the buckling modes and the critical buckling load value.

Table 5 Area ratio details for the circular cutout

Cutout diameter	Cross-sectional area	C(A/A1)	Mode 1	Mode 2
0.5	0.196375	0.04	358.825	412.54
0.65	0.33187375	0.0676	283.12	296.34
0.8	0.50272	0.1024	207.12	215.34
0.95	0.70891375	0.1444	162.88	166.74
1.1	0.950455	0.1936	121.23	128.88
1.25	1.22734375	0.25	100.7	102.17

Where

A is the cut out cross sectional area

A1 is the cross sectional area of cylinder = 4.909 m^2

C= Ratio of A/A1

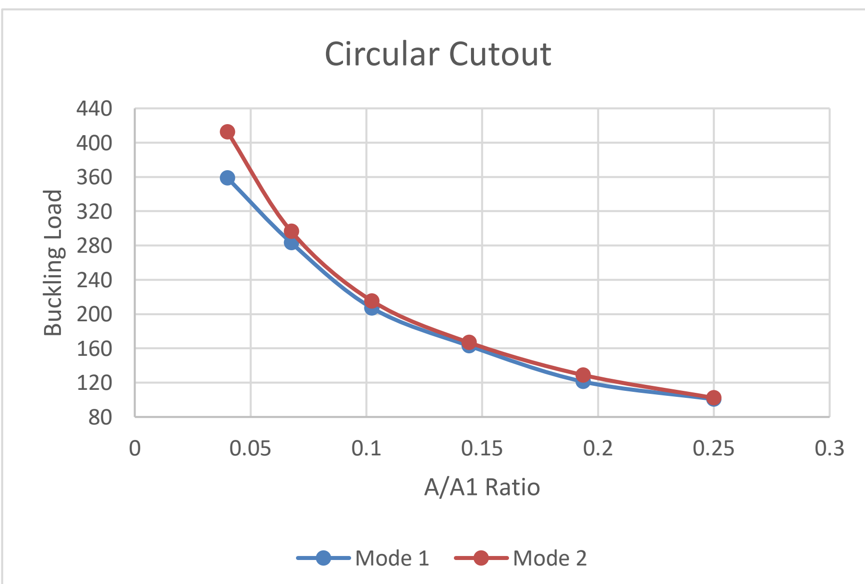


Figure 30 Buckling load variation with area ratio for circular cutout

So from the above graph we can clearly see that as the cutout cross section is being increased the critical buckling load is significantly being reduced, which means the area of the cut out is inversely proportional to the buckling load.

Area ratio for rectangular Cutout

In the same manner we will change the cross sectional area for the rectangular cutout and take the ratio for areas and find the critical buckling value for various modes.

Table 6 Area ratio details for rectangular cutout

Width	Length	Cross-sectional area	C(A/A1)	Mode 1	Mode 2
0.4	0.49	0.196	0.039923616	152.34	161.025
0.6	0.55	0.33	0.067218332	141.81	152.12
0.75	0.67	0.5025	0.102355188	129.84	139.22
0.88	0.8	0.704	0.143399109	112.84	127.98
1	0.95	0.95	0.19350732	103.3	108.7
1.2	1	1.2	0.244430299	95.34	96.925

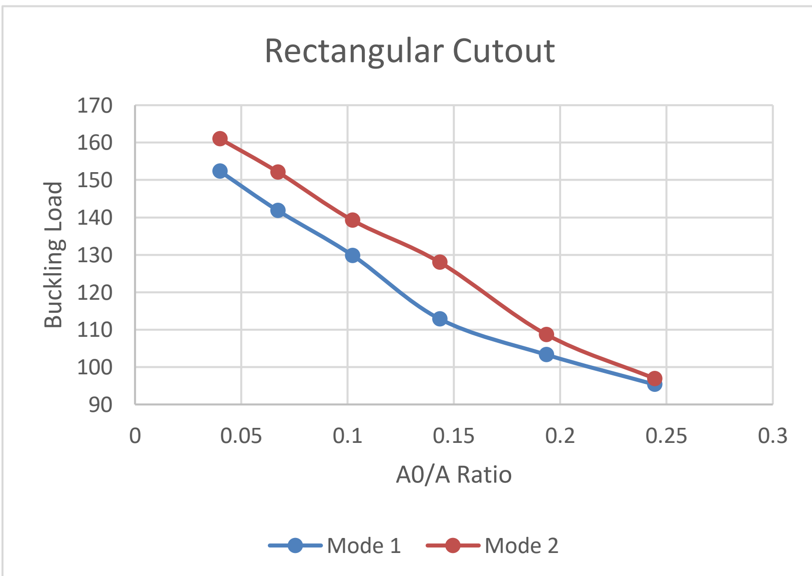


Figure 31buckling load variation with area ratio for rectangular cutout

Area ratio for Elliptical Cutout

Table 7 Area ratio details for elliptical cutout

Major radius(a)	Minor radius	Cross-sectional area	C(A/A1)	Mode 1	Mode 2
0.35	0.18	0.2	0.040738383	348.175	368.055
0.5	0.2	0.33	0.067218332	330.945	349.055
0.65	0.25	0.5	0.101845958	313.715	330.055
0.7	0.3	0.71	0.14462126	296.485	311.055
0.85	0.35	0.95	0.19350732	279.255	292.055
0.9	0.44	1.22	0.248504137	276.695	278.26

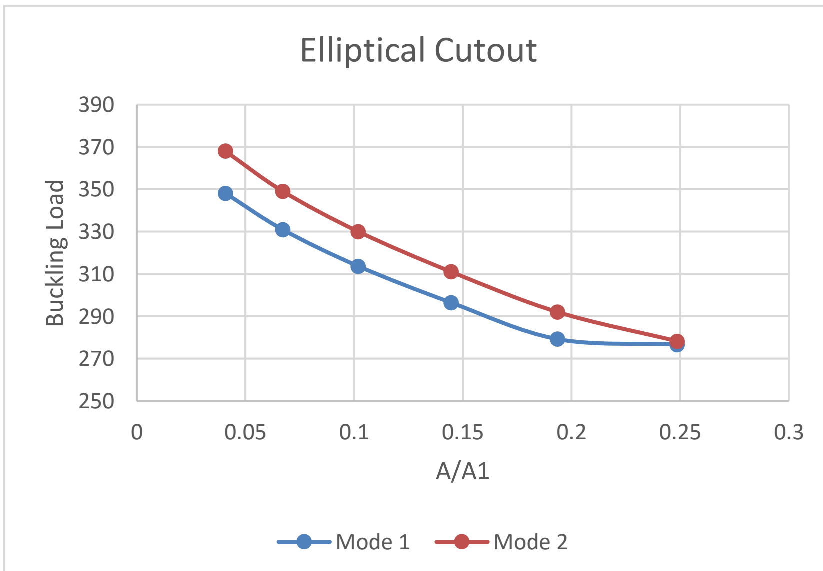


Figure 32 buckling load variation with area ratio for elliptical cutout

Comparison of buckling loads with Area Ratio (1st Mode)

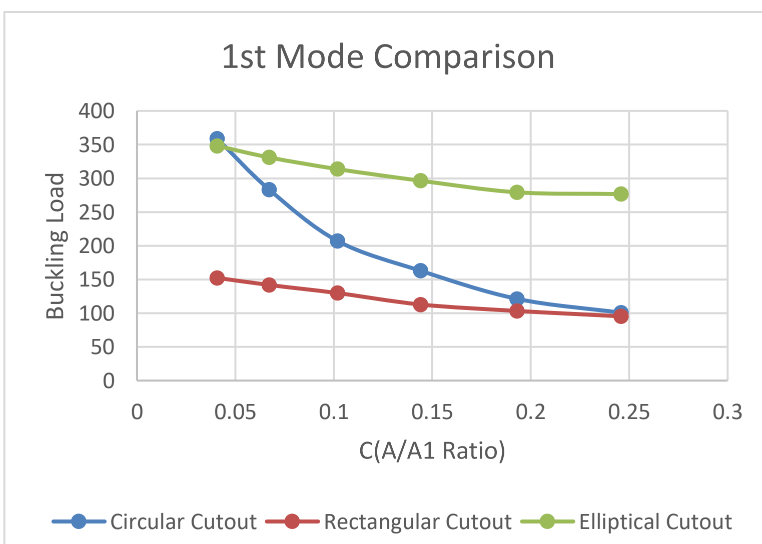


Figure 33 Comparison for cutouts

Percentage decrease

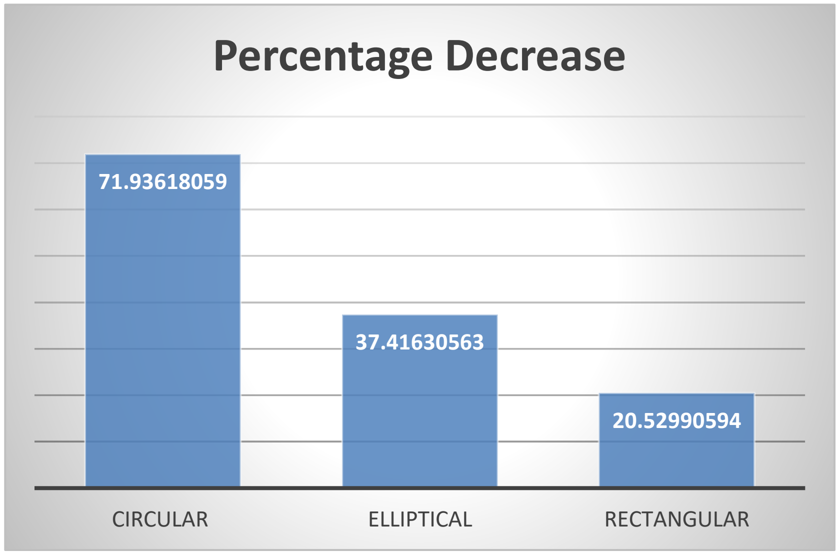


Figure 34 Percentage decrease in buckling strength

Comparison of buckling loads with Area Ratio (2nd Mode)

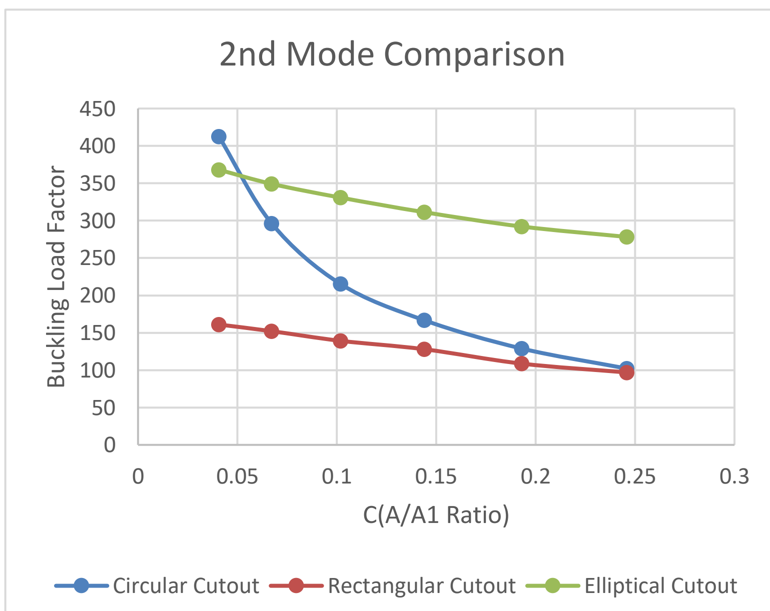


Figure 35 2nd mode buckling strength comparison

Percentage Decrease

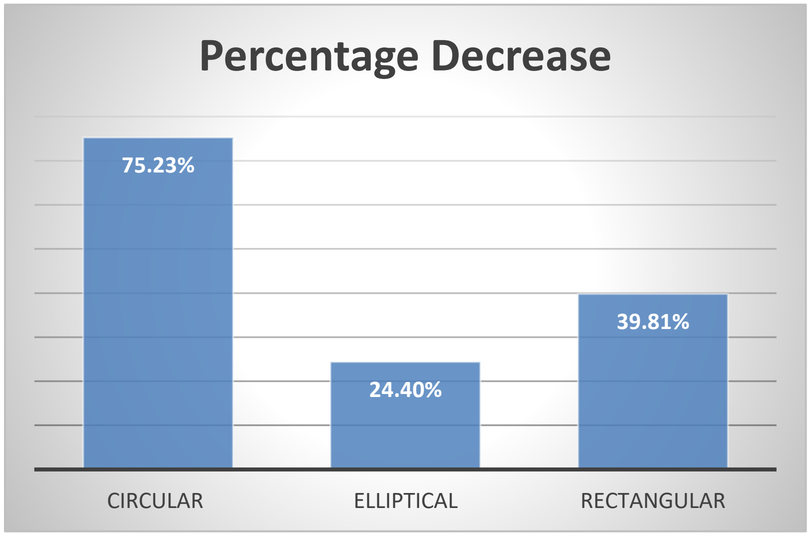


Figure 36 Percentage decrease in buckling strength

Effect of Cutout location on Critical Buckling Strength

To analyze the effect of the location of the cutout on the critical buckling strength, we will be doing the buckling analysis by varying the position of the cutout along the axial direction or the centerline of the cylinder. We will change the distance from the fixed end and increase the incremental length after every simulation. So the length will be varied from 2m to 7m and the respective length ratio will be used to see the effect on the critical buckling load of the composite cylinder. The cutouts were made at 2m and then followed by an increment of 1m in the proceeding cases. To illustrate the effect we took the ratio of the original full length of the cylinder and the location of length where the cutout has been made.

Circular Cutout

Table 8 Position of the circular cutout along axial length

Position of Cutout	Lc/L	Buckling Load
2	0.25	282.535
3	0.375	262.535
4	0.5	244.535
5	0.625	228.535
6	0.75	214.535
7	0.875	201.915

Where

$$L = full\ length\ of\ the\ cylinder\ (8\ m)$$

$$Lc = length\ where\ cutout\ is\ made\ (2\ m)$$

$$\frac{Lc}{L} = \frac{2}{8} = 0.25$$

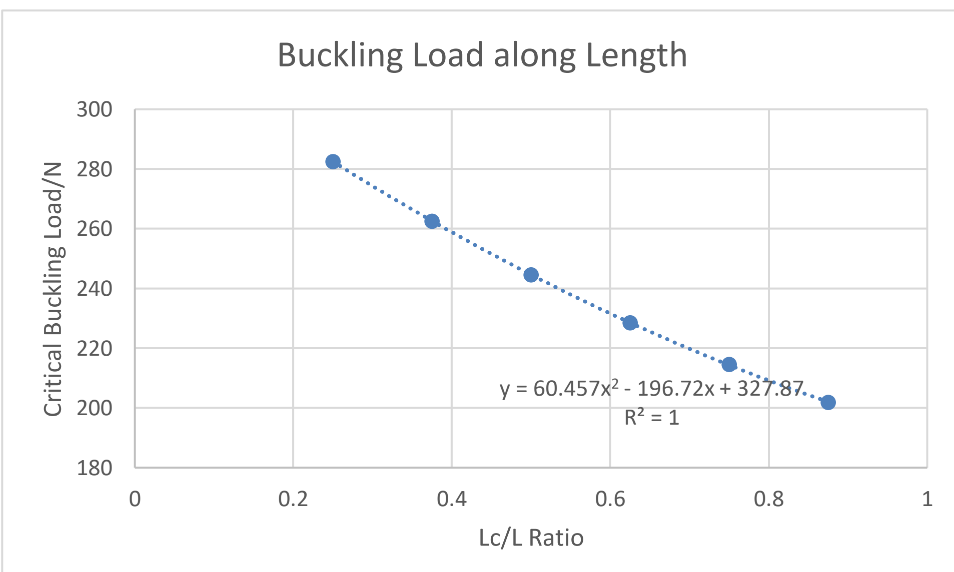


Figure 37 Variation of buckling value with circular cutout position

Elliptical Cutout

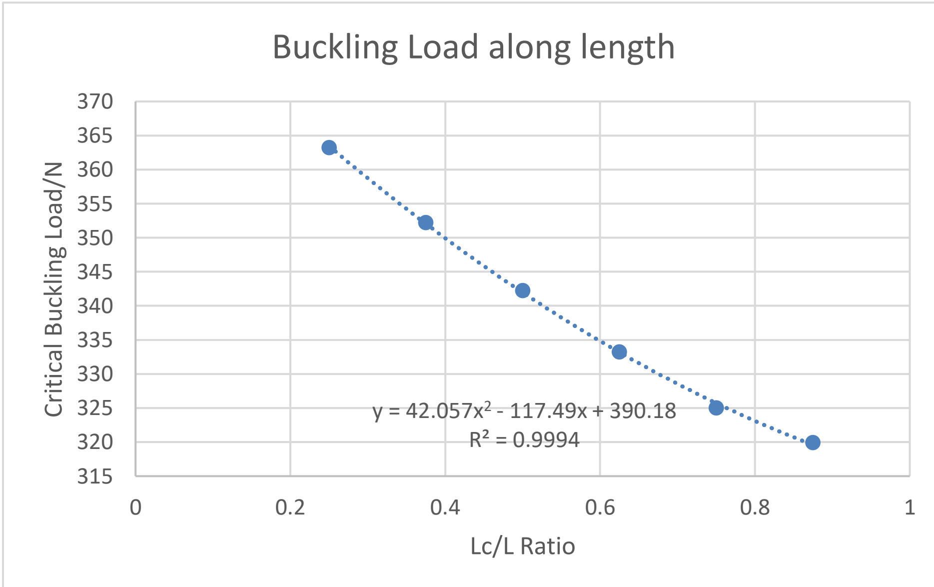


Figure 38 Variation of buckling value with elliptical cutout position

Rectangular Cutout

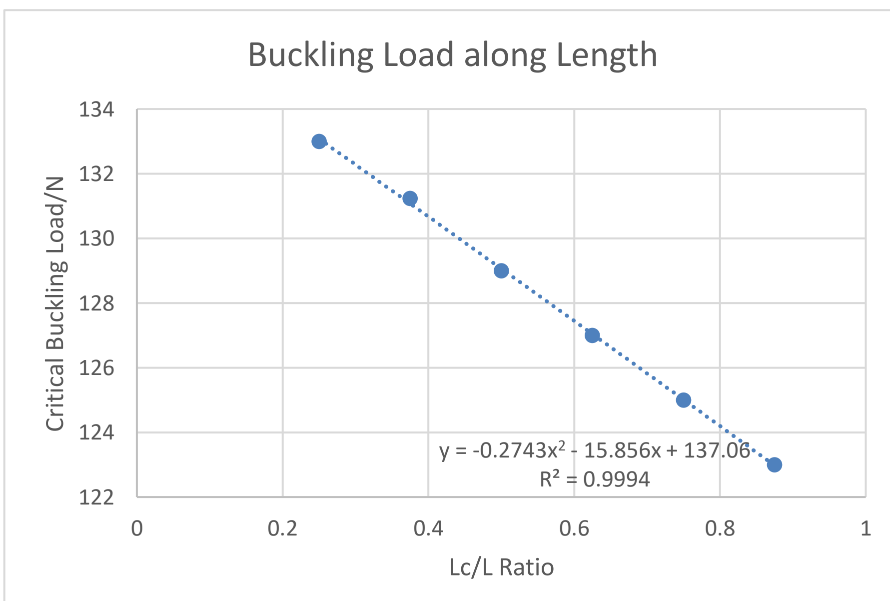


Figure 39 Variation of buckling value with circular rectangular position

Comparison of Cutouts Position

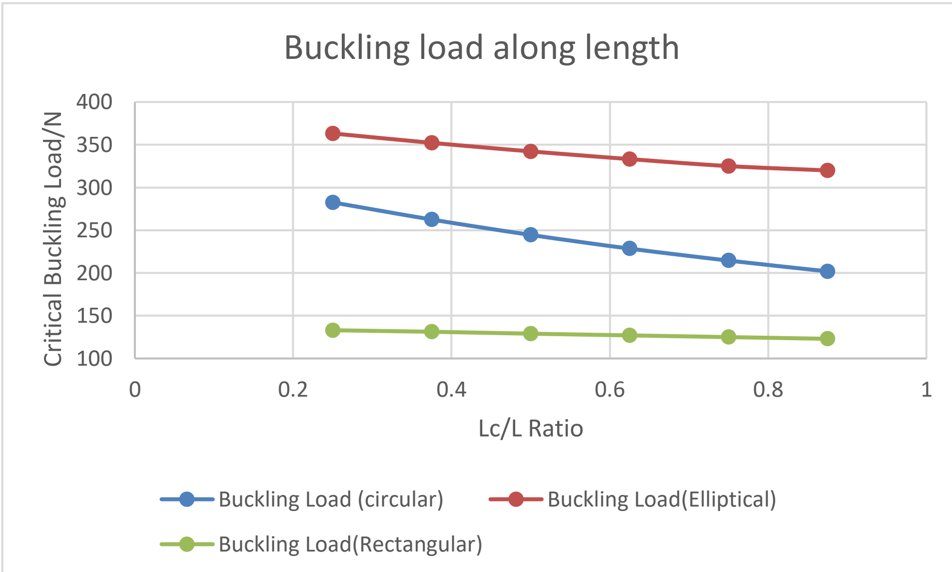


Figure 40 Buckling strength comparison for cutout position

Percentage Decrease

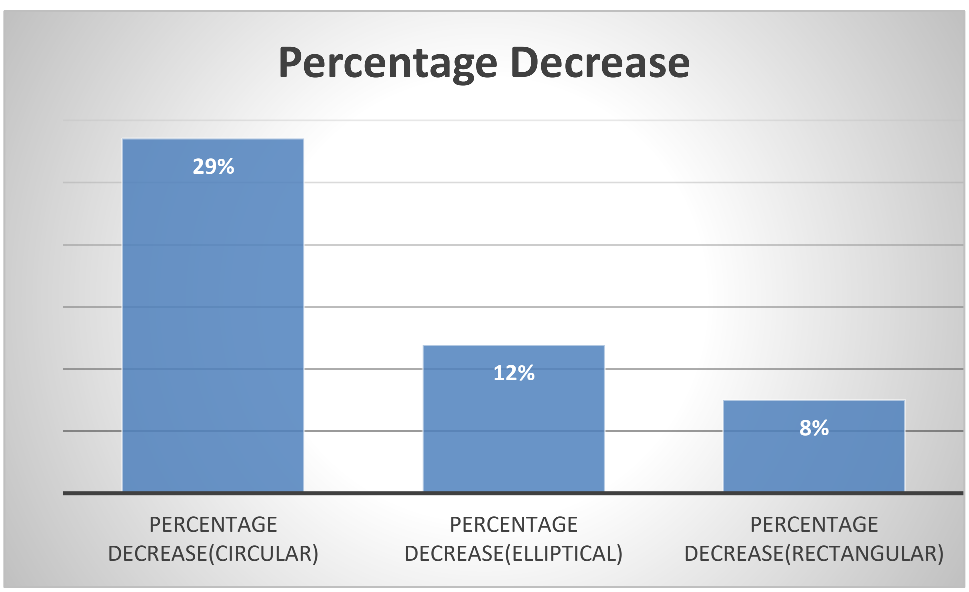


Figure 41 Percentage decrease in buckling strength

The composite cylinders without having any cutout is considered very effective and has the highest buckling resistive strength for all modes of buckling failure. The structures in real applications are tend to have cutouts, from single to multiple so we need to analyze them under compressive loadings so that they don't fail under buckling. The results obtained from the Finite element analysis in the Ansys yields that the elliptical cutout configuration is most effective among the circular and rectangular cutout. Among the cutouts the rectangular cutouts are most likely to buckle at earlier loads which means the rectangular cutouts have less structural strength as compared to circular and elliptical cutouts. On the other side the circular and elliptical cutouts have much higher strength and are more resistant to the buckling failure under same loading conditions. Among the three cutouts elliptical cutouts are most efficient in providing more strength and resistance to the buckling mode failures. The percentage reduction in the value for the buckling load in comparison with the cylindrical shell without cutout. This is to analyze the strength of the shell when particular cutout is made. For the rectangular cutout the percentage reduction in the buckling load for the first mode is 37.37 % which is highest among all cutouts. For the same mode the percentage reduction for the circular is 26.93 % and for the elliptical the percentage is 23.24 %. So form the results we can conclude that the strength is highest of the cutout is of elliptical shape and lowest with the rectangular cutout. The percentage reduction in the value for the buckling load with double cutout in comparison with the cylindrical shell without cutout. This is to analyze the strength of the shell when particular cutout is made. For the rectangular cutout the percentage reduction in the buckling load for the first mode is 34.62 % which is highest among all cutouts. For the same mode the percentage reduction for the circular is 23.52 % and for the elliptical the percentage is 21.63 %. So form the results we can conclude that the strength is highest of the cutout is of elliptical shape and lowest with the rectangular cutout.

In the same manner the cross sectional area of the cutout has been varied to see the effect for all three configurations. The results are indicating that circular cutout has the maximum percentage decrease in the buckling strength as the area goes to maximum value which means for large cutouts the best performance can be obtained from the rectangular and elliptical cutouts. Since the percentage decrease in the value of critical buckling is around 72% which is very high and for the elliptical configuration the percentage decrease is almost 32% and lowest for rectangular when the first mode of failure is being considered. In the same manner the second mode of buckling failure is also analyzed where the circular has still the highest percentage decrease but the elliptical is more stable as compared to rectangular.

To see the effect of the position we can observe that the largest decrease in the buckling strength is given by the circular cutouts since the critical buckling load value decreases from 283 N to almost 201 N as the length is changed from 2 m to 7m. And for the rectangular cutout the decrease in the buckling strength is 133 N to 123 N which is the least among other cutouts configuration. The percentage decrease for the circular is 29 % which is the maximum, followed by the elliptical which has almost 12% decrease in strength and the rectangular cutout has the lowest which corresponds to the 8%.

Chapter 5: Conclusion

In this study the buckling analysis of the composite cylindrical shell under axial compressive loading is carried out with multiple cutouts. The study focused on the effect of the shape of the cutout including circular, rectangular and the elliptical. The cylindrical shell has the composite as a material which will have fixed orientation and number of plies. The various cases have been run so to study the effect of the position of the cutout and the cross sectional area of the cutout and the decrease in the value of buckling strength for particular cutout.

For the single cutout, the elliptical cutout is most effective and provides the maximum buckling resistive strength to the composite cylinder. The rectangular cross sectional cutout has the least buckling strength. For the double cutout the same trend has been observed in terms of the buckling strength and an important conclusion can be drawn. For the double cutout the buckling failure and the maximum displacement is occurring at the first single cutout at the lower modes which means when 2 cutouts are made one acts as a stress reliever for the second cutout which means the first cut is taking all the loadings. At the higher buckling mode failures both cutouts have impact on the buckling failure.

The cross sectional area of the cutout has been varied to see the effect for all three configurations. The results are indicating that circular cutout has the maximum percentage decrease in the buckling strength as the area goes to maximum value which means for large cutouts the best performance can be obtained from the rectangular and elliptical cutouts. Since the percentage decrease in the value of critical buckling is around 72% which is very high and for the elliptical configuration the percentage decrease is almost 32% and lowest for rectangular when the first mode of failure is being considered. In the same manner the second mode of buckling failure is also analyzed where the circular has still the highest percentage decrease but the elliptical is more stable as compared to rectangular.

5.1 Future Work

There are many ways in which we can further improve the critical buckling strength of the composite shells under compressive loadings. One of the possible research would be to introduce the stiffeners in the cylinder and make the comparative study by using the various stiffeners configurations and combination of arrangements. Also the cutouts can be made along circumference of the cylinder or at some angle so to study their effect on the critical buckling strength. The composition of composite and the arrangement can be altered such as the number of plies and their stacking sequence can be increased to see how the cylinder will behave under buckling failure.

References

[1] Allahbakhsh, H., & Dadrasi, A. (2012). Buckling analysis of laminated composite panel with elliptical cutout subject to axial compression. *Modelling and Simulation in Engineering*, *2012*.

[2] B, N., & Haslin S.M, M. (2015). Buckling Analysis on Composite Cylindrical Shells with Cut-outs. *International Journal of Engineering Trends and Technology*, *29*(2), 68–72.

[3] Gangadhar, L., & Sunil Kumar, T. (2016). Finite element buckling analysis of composite cylindrical shell with cutouts subjected to axial compression. *International Journal of Advanced Science and Technology*, *89*, 45–52.

[4] Hao, P., Wang, B., Tian, K., Liu, H., Wang, Y., Niu, F., & Zeng, D. (2017). Simultaneous buckling design of stiffened shells with multiple cutouts. *Engineering Optimization*, *49*(7), 1116–1132.

[5] Hilburger, M. W. (2007). Buckling and failure of compression-loaded composite laminated shells with cutouts. *Collection of Technical Papers - AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics and Materials Conference*, *6*, 6366–6378.

[6] Meyer-Piening, H. R., Farshad, M., Geier, B., & Zimmermann, R. (2001). Buckling loads of CFRP composite cylinders under combined axial and torsion loading - experiments and computations. *Composite Structures*, *53*(4), 427–435.

[7] Optimization, S., & Engineering, C. (1994). *Buckling optimization of fiber-composite laminate shells considering in-plane shear nonlinearity*. *173*(4), 168–173.

[8] Ramaniah, J., Jagan, K., & Rao, M. (2013). Buckling analysis of orthotropic composite shell with and without cutouts using fem. *International Journal of Engineering Research T Technology*, *2*, 2042–2049.

[9] Renani, M. J. K., & Fesharaki, J. J. (2019). Nonlinear Buckling Analysis of Cones with Rectangular Cutouts, Numerical and Experimental Investigation. *International Journal of Steel Structures*, (2013).

[10] Shirkavand, A., Taheri-Behrooz, F., & Omidi, M. (2019). Orientation and size effect of a rectangle cutout on the buckling of composite cylinders. *Aerospace Science and Technology*, *87*, 488–497.

[11] Shokrieh, M. M., & Nouri Parkestani, A. (2017). Post buckling analysis of shallow composite shells based on the third order shear deformation theory. *Aerospace Science and Technology*, *66*(January), 332–341.

[12] Sonmez, F. O. (2017). Optimum design of composite structures: A literature survey (1969-2009). *Journal of Reinforced Plastics and Composites*, *36*(1), 3–39.

[13] Taheri-Behrooz, F., Esmaeel, R. A., & Taheri, F. (2012). Response of perforated composite tubes subjected to axial compressive loading. *Thin-Walled Structures*, *50*(1), 174–181.

[14] Weiß, L., Köke, H., Rohwer, K., Hühne, C., & Wiedemann, M. (2015). Analysis of the deformation energy of straight composite z-frames subjected to critical buckling loads. *Finite Elements in Analysis and Design*, *98*, 50–60.