INFLUENCE OF MULTIPLE CUT-OUTS ON THE BUCKLING OF THIN CYLINDRICAL SHELLS OF AEROSPACE BODY UNDER AXIALLY COMPRESSIVE LOAD USING FINITE ELEMENT ANALYSIS

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Abstract-Thin cylindrical shells are most widely used structural forms in aerospace and missile applications.in designing efficient and optimized shell structures they become increasingly sensitive to buckling. The catastrophic nature of buckling failure often brings surprises in testing by failure at staggeringly low loads as compared to theoretical predictions.it is well known that the experimental disparity is mainly attributed to geometrical imperfections like damage in the structure.in missile airframe the cylindrical shell structures are generally provided with cutouts for accessing internal components during integration. The cutouts invariably reduce the strength of the thin cylindrical shell more specifically the buckling load.to improve strength stiffeners provided around the cutout. Due to this reduces buckling load. This study is aimed at generating elaborate data on buckling load with different cutout sizes and the strengthening effect by providing stiffener around the cutout. With efficient general purpose FEA packages, for the parametric study of the problems of this nature is relatively inexpensive.

Keywords-cylindrical shells, cutouts, stiffener, buckling, ANSYS.

I. INTRODUCTION

In the field of material science and metallurgy need for lighter and stronger material to perform the required functions of the specific members or elements in any system or sub system.as the member may be in the form of shells, piston and hatch covers etc., having different geometric features and dimensions. The development of different software packages like ANSYS, FEMAP, CATIA etc., is great use of analyzing the results. Thin shells are subjected to axial compressive loads acting along the perimeter of the shells.in addition if the shell contains any cutouts the effect is not known. Hence it is necessary to investigate the effect of cutouts in the shells on buckling strength, von-mises stresses and nodal displacements.

By generating required data the relevant information obtained for shells with cutouts, without cutouts, with stiffener and without stiffeners.in the application of thin shells with circular holes on curved surface are provided as an access port in a missile skin or aircraft, a ship hatch or for numerous other reasons. Several investigations have studied the effect of circular hole on the stresses in a cylindrical structures might be required to carry static compressive loads or in the case of an aircraft missile, fluctuating flight loads which produce compressive components.

Whenever a compressive load is applied to thin shell structures then there is possibility of buckling of the structure.to predict the effect of circular cutouts on the buckling load of shells under compressive loading, ANSYS package is used to investigate the effect on shell of with and without adding stiffener around the cutouts.

II. LITERATURE REVIEW

Thin cylindrical shells are most sensitive to buckling.as compressive load applied on the shell; it leads to failure certain load known as critical load.as the cylindrical shells have wide range of applications in variety of fields, its structural design for safety is very much essential.in the application of thin shell structures it is often to design a shell with cutouts on its circumference. The cutout may be circular, square, or rectangular etc.,

Purpose of the hole is to access interior parts of missile or any specific applocations.the effect of a single circular hole on the buckling of thin circular cylinder under axial compression was studied experimentally and theoretically.

The buckling of annular circular plates has been receiving a lot of attention for quite some time. Majmudar (1970) studied both theoretically and experimentally, the buckling of an annular plate under uniform compression.

Pardon (1973, 1978) used finite elements to analyze axisymmetric and symmetric buckling in isotropic annular plates subjected to external pressure. Radicliff and Mote (1977) studied buckling in stationary and rotating discs under an edge load. A similar problem was studied by Srinivasan and Rammurthy (1980).Tani and Nakumara (1918, 1980) studied buckling of annular plates under periodic radial and torsion loads. Laura and Ficcadenti (1981) used a Ritz approach to calculate the buckling loads circular plates of varying thickness under external pressure. Nariti (1985) applied Galerkin's method to determine the buckling loads of circular plate under a partially distributed and concentrated in plane loads. Hammed and Harima (1986) solved the buckling problem of annular plate under plane shearing stress.Laura,Etal (1982) studied the buckling of annular plates under hydrostatic in plane forces. Chen,Etal (1989) studied the dynamic stability of thick annular plates using FEM.

The finite element method using semi analytical annular disc elements and sector elements was used for analysis. The present study is to investigate the sensitivity of the cylindrical shells with and without cutouts and adding the stiffeners when it is subjected to buckling load.

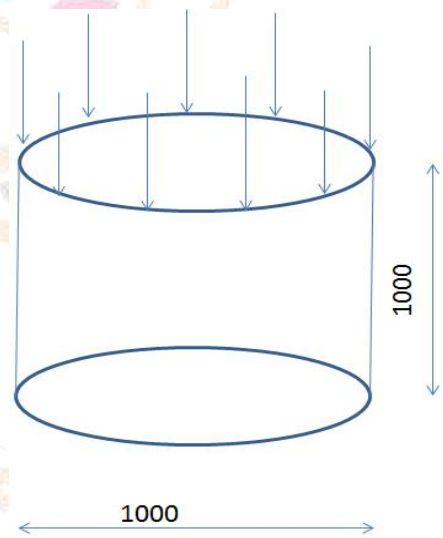
III. BUCKLING ANALYSIS

If a cylindrical shell is uniformly compresses in the axial direction, buckling is symmetrical with respect to the axis of the cylinder. The buckling may occur at a particular value of the compressive load. If the wall thickness is less than about 7%of the inner diameter then the cylinder may treated as thin one. The critical value of compressive force σ_{cr} per unit area of the edge of the shell.as the shell is compressed in the axial direction, we must consider of the middle surface in circumferential direction and also bending of shell. Thus the strain energy of the shell is increased.at the critical value of the load, this increase in the energy must be equal to the work done by the compressive load as the cylinder shortens owing to buckling.

$$\sigma_{cr} = Et/[r\sqrt{3(1-v^2)}]$$

Where E=young's modulus, t = thickness of the shell , r = radius of the shell v = poisson's ratio

Outside diameter of shell = 1000 mm
Length of the shell = 1000 mm
Thickness of the shell = 5 mm
Material of the shell is Aluminum
Young's modulus = 70.16 Gpa
Poisson's ratio = 0.33



$$\sigma_{cr} = Et/[r\sqrt{3(1-v^2)}]$$

$= \frac{70.16 \times 10^9 \times 0.05}{0.5 \times [\sqrt{3}\,(1-0.33^2)]}$

$= 429107130.4$ N/m^2

$\sigma = \frac{p}{\pi d t}$

$= 1000/\pi \times 1 \times 0.005$

$= 63661.98$ N/m^2

Buckling factor = $\frac{\sigma_{cr}}{\sigma}$ = 429107130.4/63661.98 = 6740.40

IV. BUCKLING OF SHELLS WITH CIRCULAR CUTOUTS

The objective of this project work is to develop shell with circular cutouts on the circumference of the cylindrical shell with various parameters. Also this study aimed to finding elaborate effect by providing stiffener around the cutout. The buckling factor depends on the structure and nature of loading.

In the first phase geometric modeling of the shell, shell without cutout and shell with cutout and then stiffener around it has been developed. For this work finite element analysis package ANSYS has been used for both geometry modeling, meshing, static and buckling analysis. The number elements and nodes on the model will be varying and selected appropriate element divisions (85,35) by keeping in view accuracy of values and the cylindrical shell is subjected to compressive loads only.

Start with one cutout is chosen on circumferential surface of the shell, at mid span and solved for buckling analysis by varying the size of the cutout in terms of percentage of circumference of the shell, and observed the buckling factor and mode shape and behavior of shell at various cutout sizes.

Similarly buckling analysis has been carried with two cutouts placed diametrically opposite on circumference of shell with same boundary conditions and also by varying the cutout sizes. The width of the stiffener is changed depending upon the size of cutout.by varying the thickness of the stiffener the improvement in buckling load is observed.

V. RESULTS AND DISCUSSIONS

The results are studied to understand the influence of cutout on buckling strength of the shell and also the extent of improvement by providing stiffener around the cutouts. The shell model is generated as shown in figure1 is elastic 4-node shell 63 as specified in ANSYS package. Thin cylindrical shell with boundary conditions i.e. one end is fixed and other end is subjected to axial compressive load as shown in figure2 without circular cutout. Meshing also generated on the shell.

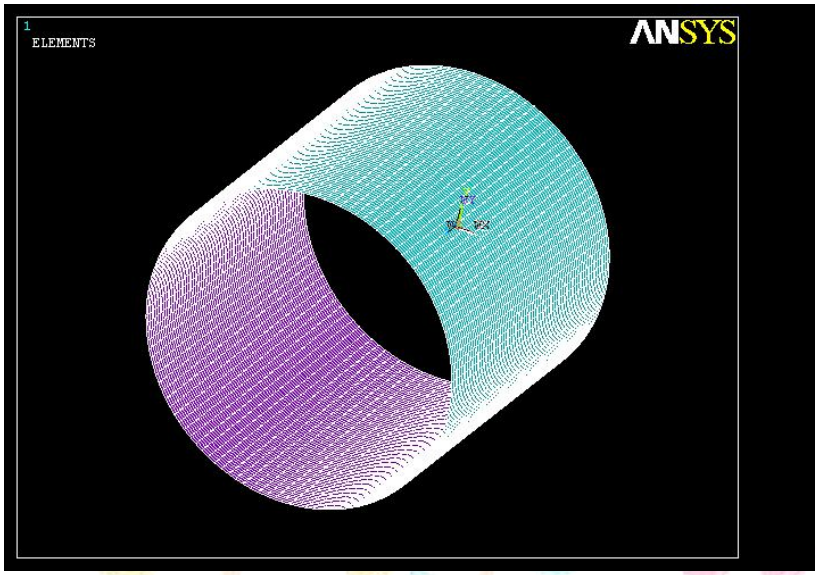


Figure1: Thin cylindrical shell model with meshing

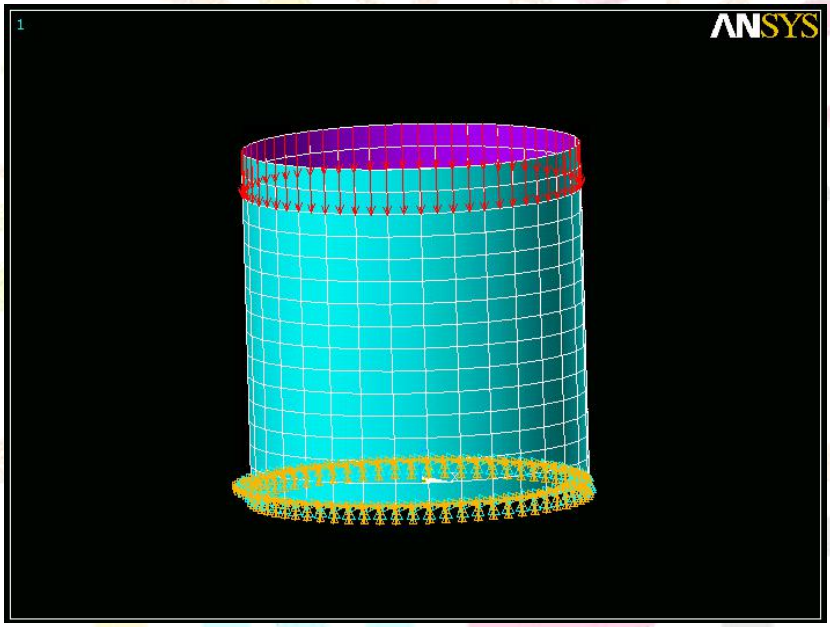


Figure2: Thin cylindrical shell with boundary conditions

The analysis of the cylindrical shell without cutout is subjected to the loading and fixed conditions as specified revealed that the buckling factor is about 2499.51 which can be approximated to 2500.the corresponding analysis with deformed shape shown in figure3.

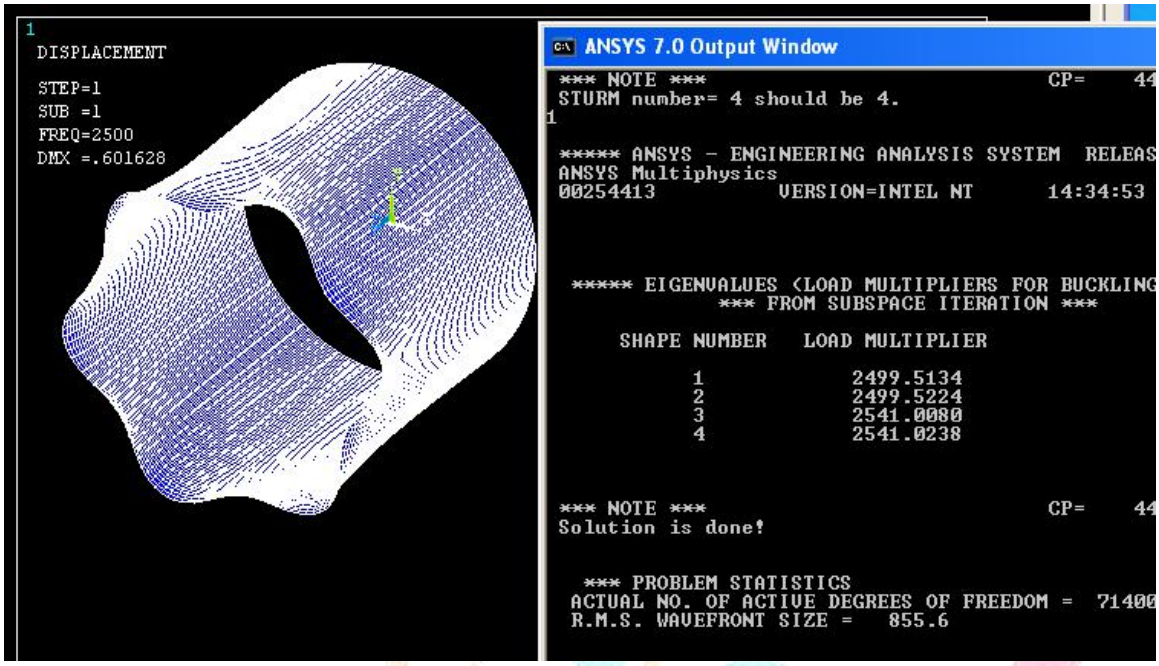


Figure3: Buckling factor of thin cylindrical shell without cutout

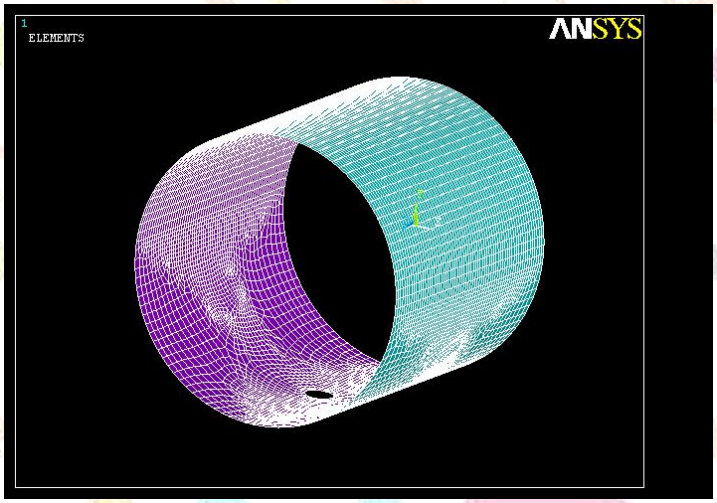


Figure4: Thin cylindrical shell with one circular cutout

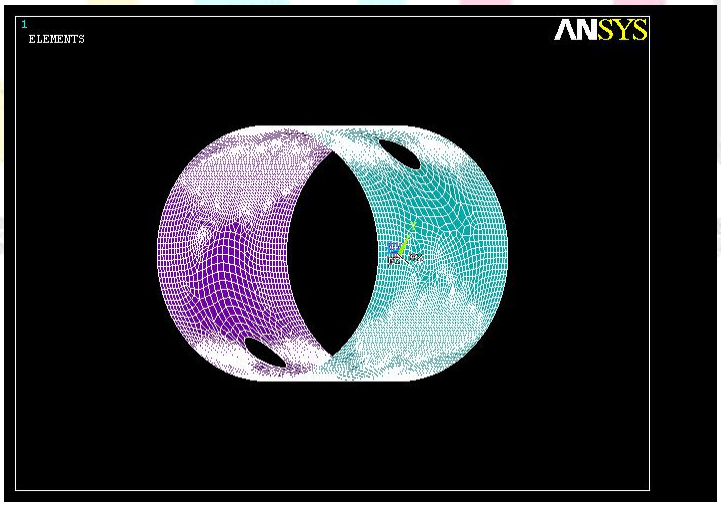


Figure5: Thin cylindrical shell with two circular cutouts

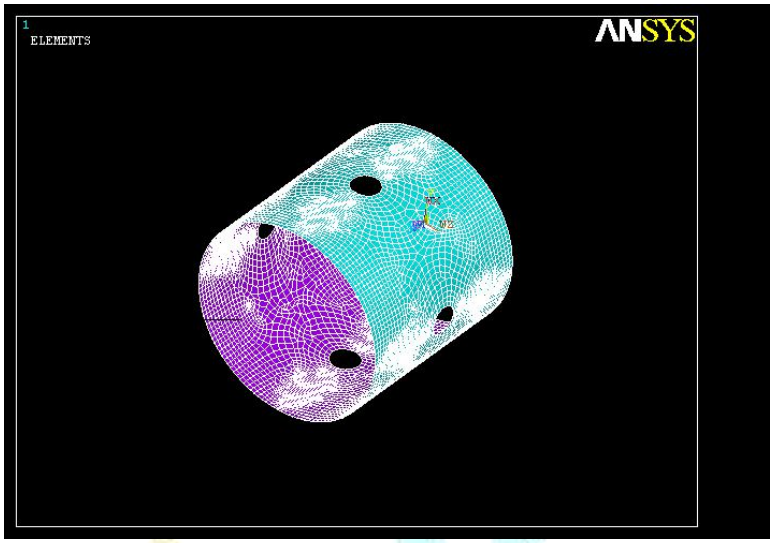


Figure6: Thin cylindrical shell with four circular cutouts

The shell is meshed with 85(straight lines),35 (curved lines) of element divisions.as per the applied boundary conditions of the shell i.e., one side of the shell keeping fixed constraints and other side subjected to axial compressive load. Under these conditions the results obtained from the analysis for static loading and buckling of the shell. The values of buckling factor, Von-mises stresses and displacement of the shell are shown in the table1. Similarly analysis has been carried out for 2 and 4 cutouts for different sizes and the results of buckling factors, Von-mises stresses and displacements are shown in table 2 and 3.

Table 1: Shell with one cutout for various cutout sizes

S.No	Cutout Size (m)	Buckling Factor	Von Mises Stress (N/m2)	Displacement (m)
1	0.10	2467.50	189737	0.118e-5
2	0.15	2382.84	214461	0.199e-5
3	0.20	1631.68	346569	0.384e-5
4	0.25	1218.22	431218	0.673e-5
5	0.30	980.83	492501	0.103e-4
6	0.35	711.63	621607	0.171e-4

Table 2: Shell with two cutouts for various cutout sizes

S.No	Cutout Size (m)	Buckling Factor	Von Mises Stress (N/m2)	Displacement (m)
1	0.10	2457.1	225324	0.124e-5
2	0.15	2307.6	288983	0.210e-5
3	0.20	1620.54	357353	0.382e-5
4	0.25	1202.98	411503	0.666e-5
5	0.30	911.525	490675	0.11e-4
6	0.35	706.91	551764	0.179e-4

Table 3: Shell with four cutouts for various cutout sizes

S.No	Cutout Size (m)	Buckling Factor	Von Mises Stress (N/m2)	Displacement (m)
1	0.10	2449.10	233794	0.130e-5
2	0.15	2254.81	299685	0.229e-5
3	0.20	1613.14	350473	0.424e-5
4	0.25	1164.26	435461	0.752e-5
5	0.30	872.67	510272	0.122e-4
6	0.35	681.06	538822	0.190e-4

In buckling of shell with one cutout shown in Figure4 it has been observed from the table1 that the size of the cutout is increased gradually, the reduction in buckling load factor seem to be very less for small sized cutouts.table2 and table 3 shows data obtained by carrying out the analysis of shell with two and four cutouts respectively. These tables also show that Von-mises stresses and displacements of varied cutout sizes with increased number of cutouts.

Table 4: Buckling factors of shell with one, two and four cutouts for different cutout sizes

S.No	Cutout Size(m)	Buckling Factor One Cutout	Buckling Factor Two Cutouts	Buckling Factor Four Cutouts
1	0.10	2467.58	2477.1	2449.10
2	0.15	2382.84	2307.6	2254.81
3	0.20	1620.54	1631.67	1613.14
4	0.25	1218.22	1202.98	1164.26
5	0.30	980.83	911.525	872.67
6	0.35	706.91	711.63	681.06

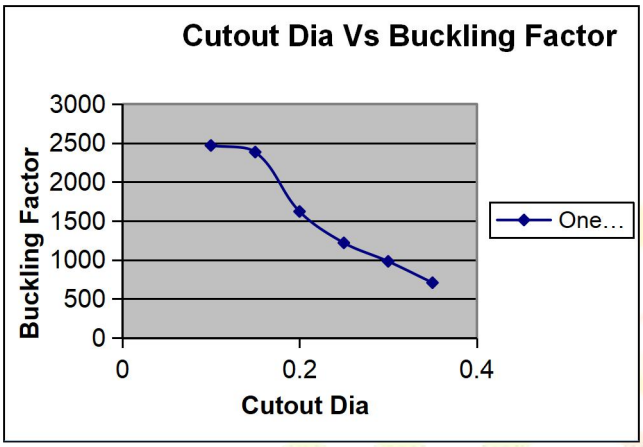


Figure7: Buckling factors Vs. cutout sizes

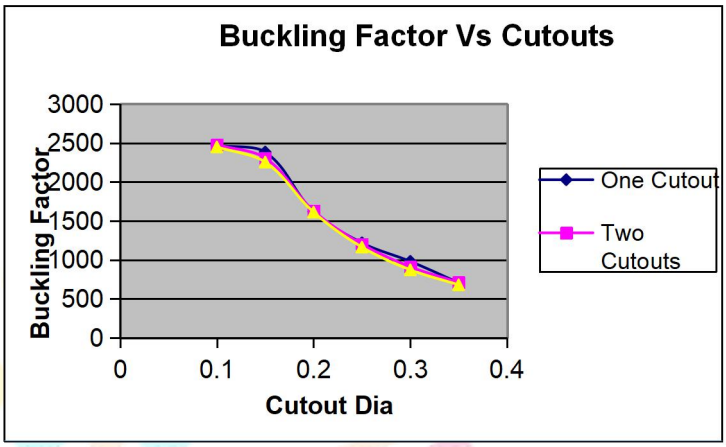


Figure8: Buckling factor Vs. cutout sizes

Table5: Displacement Vs. one, two and four cutouts

S.No	Cutout Size (m)	Displacement One Cutout (m)	Displacement Two Cutouts (m)	Displacement Four Cutouts (m)
1	0.10	0.118e-5	0.124e-5	0.130e-5
2	0.15	0.199e-5	0.210e-5	0.229e-5
3	0.20	0.384e-5	0.382e-5	0.424e-5
4	0.25	0.673e-5	0.666e-5	0.752e-5
5	0.30	0.103e-4	0.110e-4	0.122e-4
6	0.35	0.171e-4	0.170e-4	0.190e-4

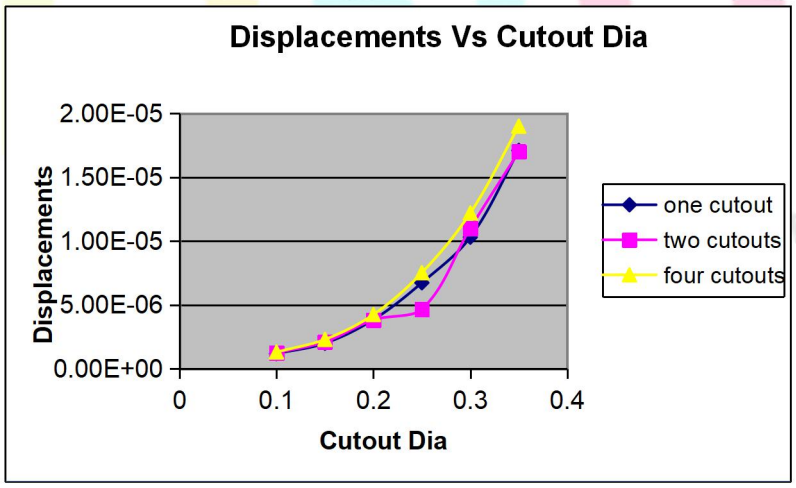


Table6: Von mises stresses Vs. one, two and four cutouts

Cutout size (m)	Von Mises Stress for one cutout (N/m2)	Von Mises Stressfor two cutouts (N/m2)	Von Mises Stress for four cutouts (N/m2)
0.10	189737	225324	233794
0.15	214461	288983	299685
0.20	346569	357353	350473
0.25	431218	411503	435461
0.30	492501	490675	510272
0.35	621607	551764	538822

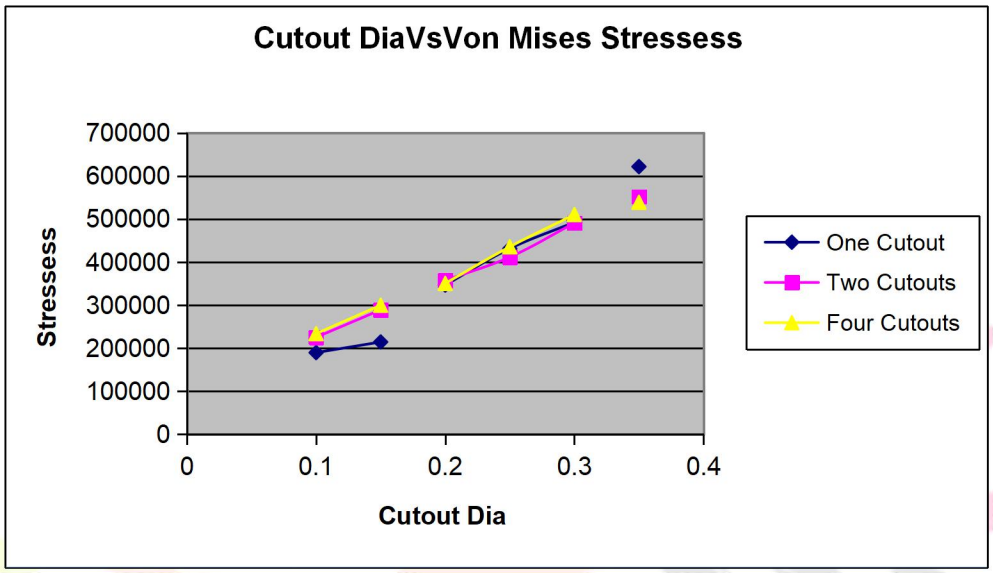


Figure9: Cutout Dia Vs. Von mises stresses for one, two and four cutouts

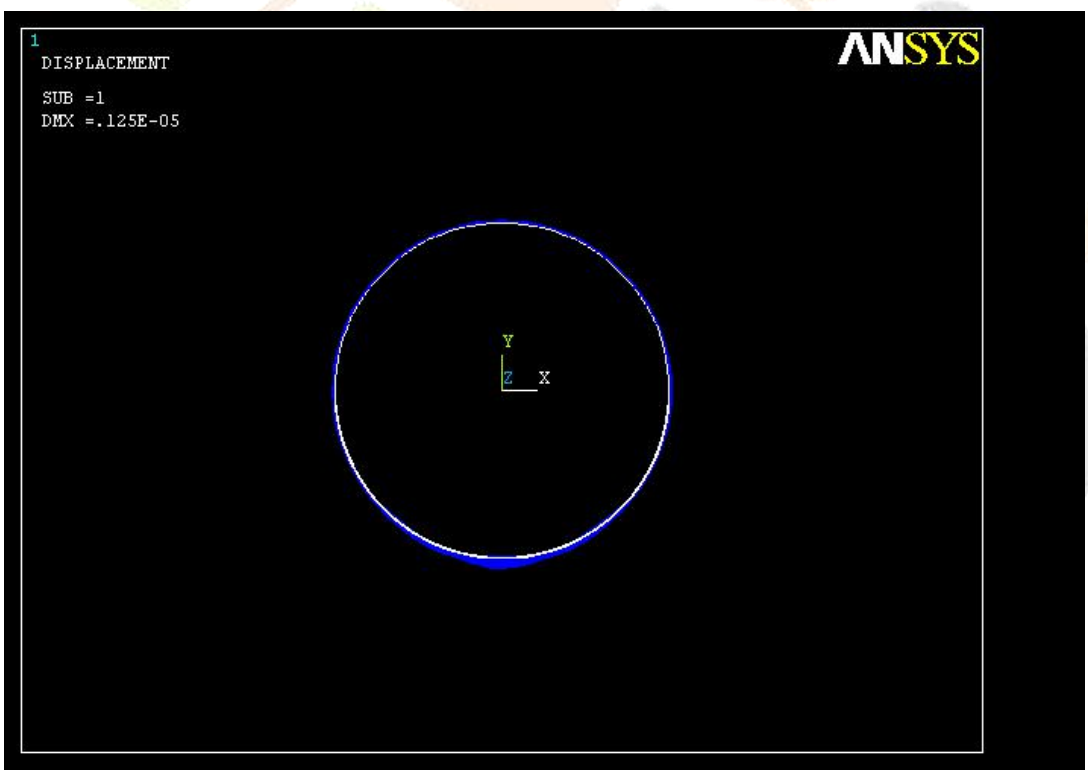


Figure10: Displacement of single cutout

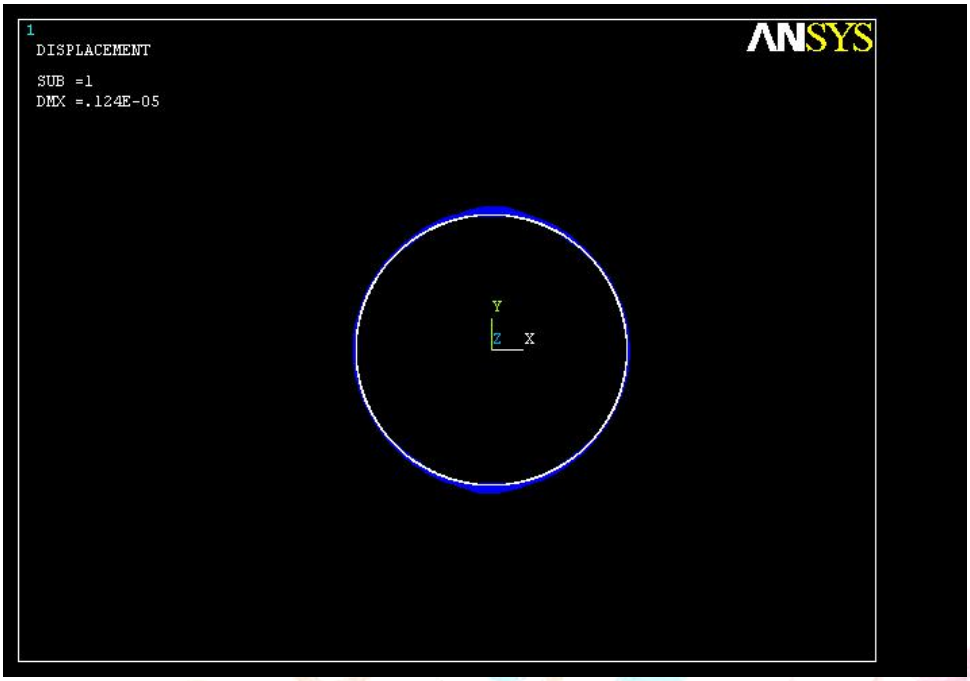


Figure11: Displacement of two cutouts

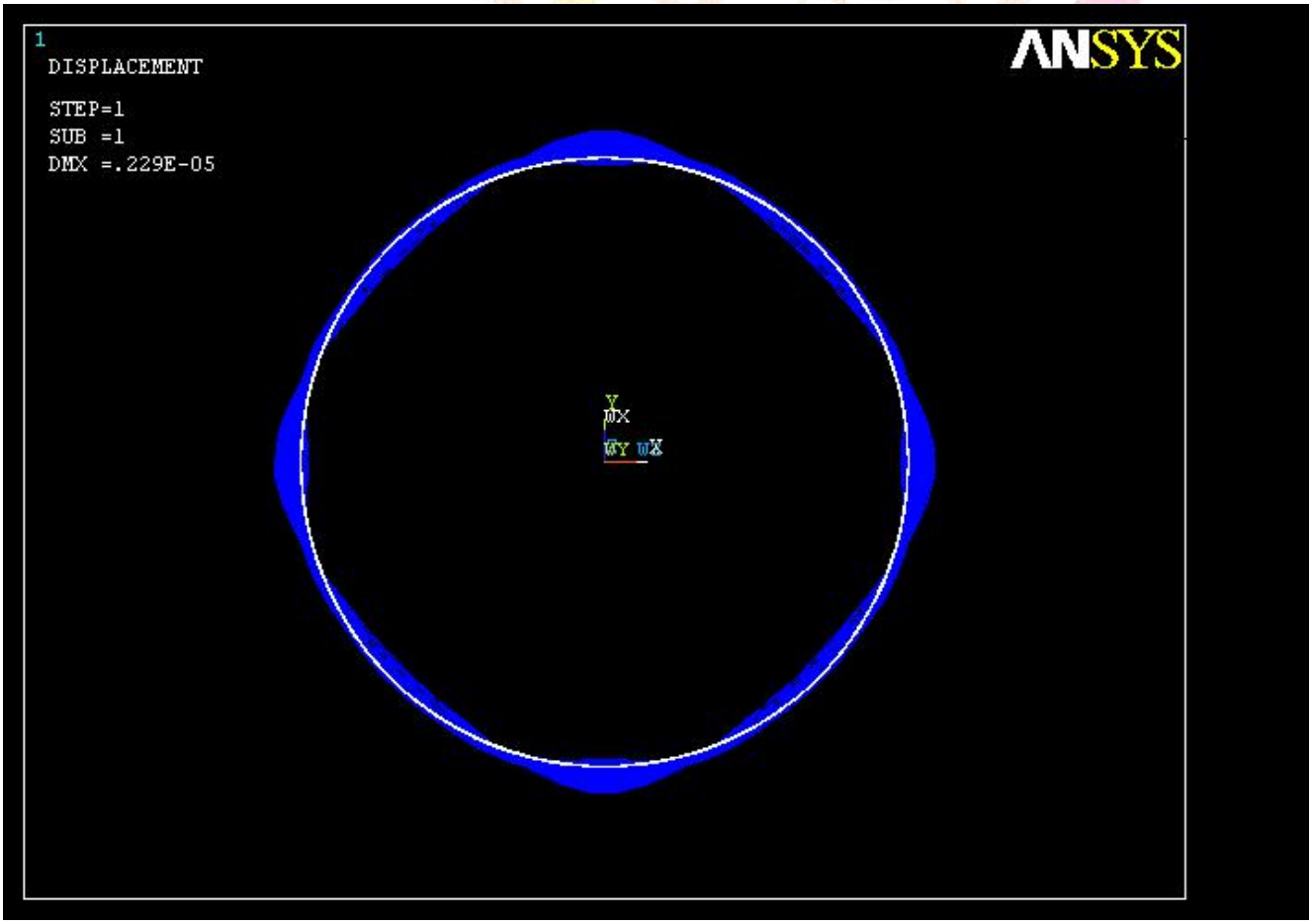


Figure12: Displacement of four cutouts

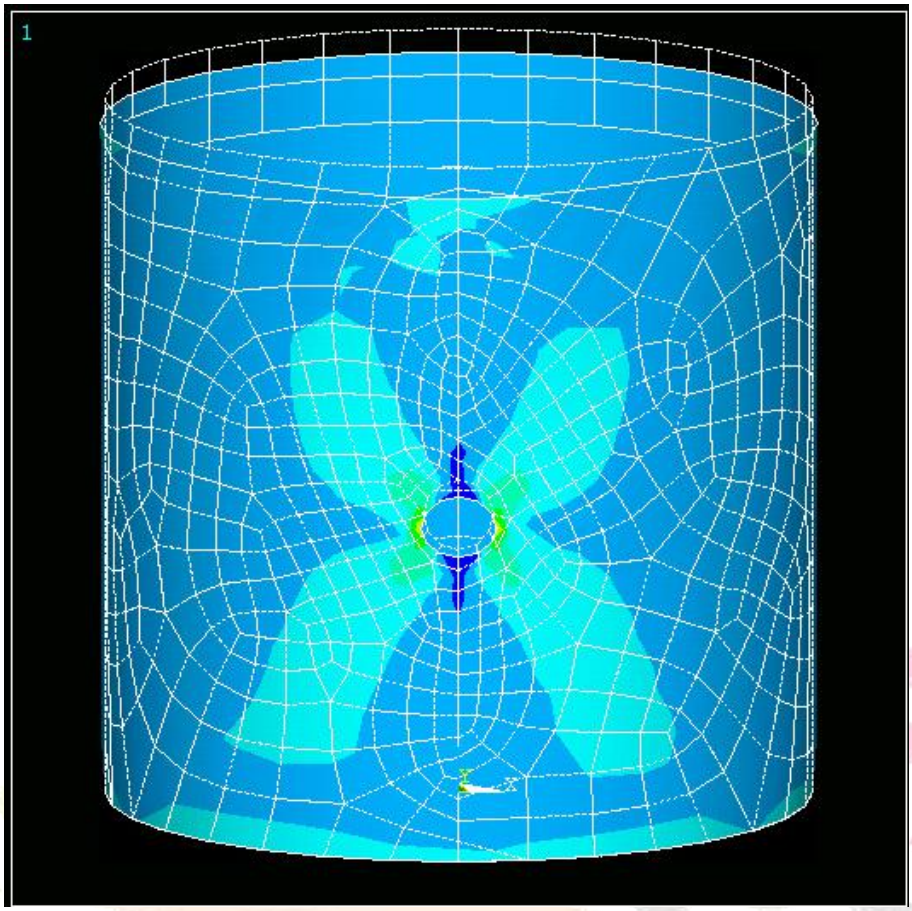


Figure13: Stress contour of single cutout

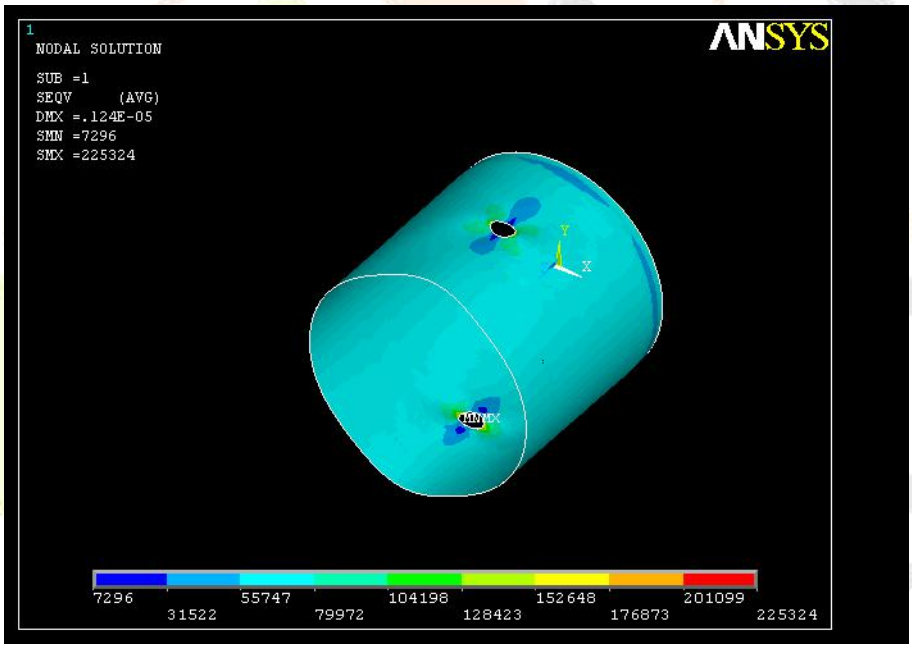


Figure14: Stress contour of two cutouts

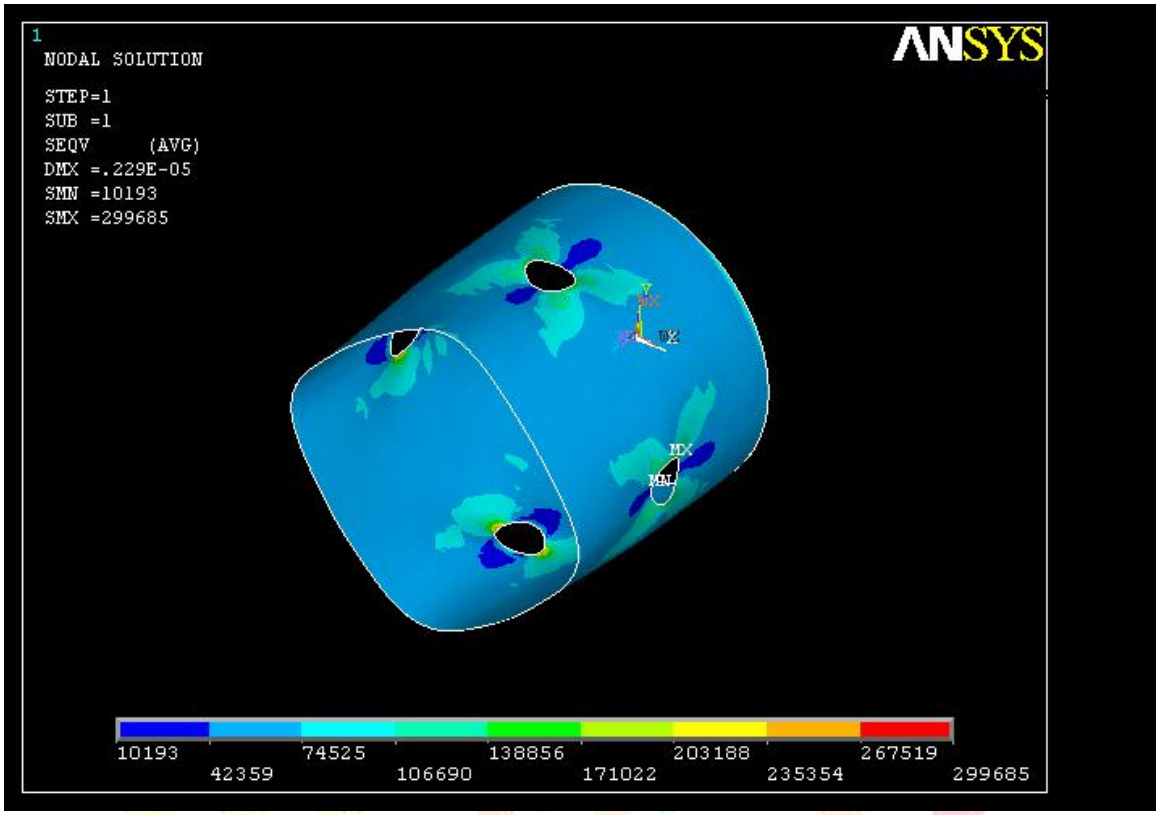


Figure15: Stress contour of four cutouts

VI. Stiffeners

If stiffeners are provided around the cutout, there is a possibility to minimize the stress concentration around the cutout. The stiffener shown in figure16 is provided around the cutout at constant width. The converged element shell model with two and four cutouts with stiffener shown in figure17 and figure18.the analysis has been carried out by varying the thickness of the stiffener by keeping the thickness of the shell and size of the cutout is constant.

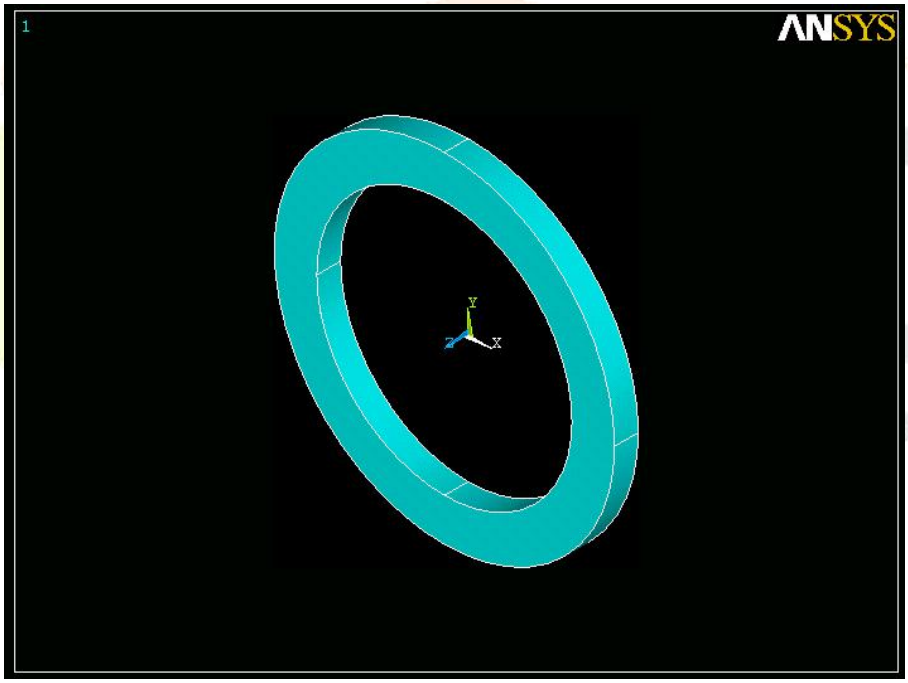


Figure16: Stiffener

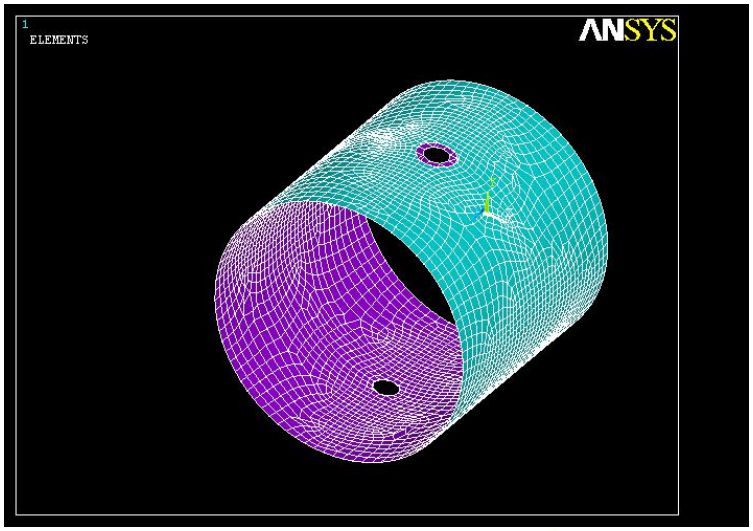


Figure17: Meshed shell with two cutouts with stiffener

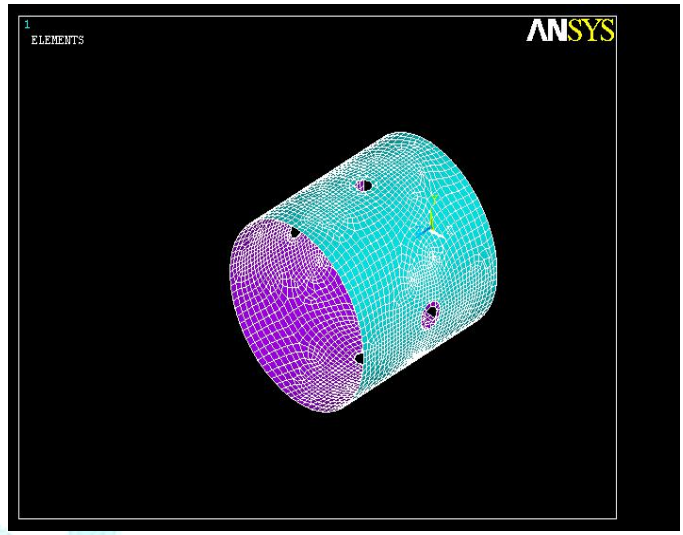


Figure18: Meshed shell with four cutouts with stiffener

Table7: Estimated values for shell with stiffener around two cutouts
Cutout size =0.1 m dia, Stiffener size= 0.2 m dia

S.NO.	Stiffener thickness (m)	Buckling factor	Von Misess stress (N/m2)	Displacement (m)
1	0.0075	2543.79	208769	0.106e-5
2	0.0080	2558.16	199260	0.105e-5
3	0.0090	2561.15	183062	0.973e-6
4	0.0100	2569.13	169761	0.867e-6
5	0.0120	2580.15	149148	0.795e-6
6	0.0140	2587.89	133817	0.720e-6

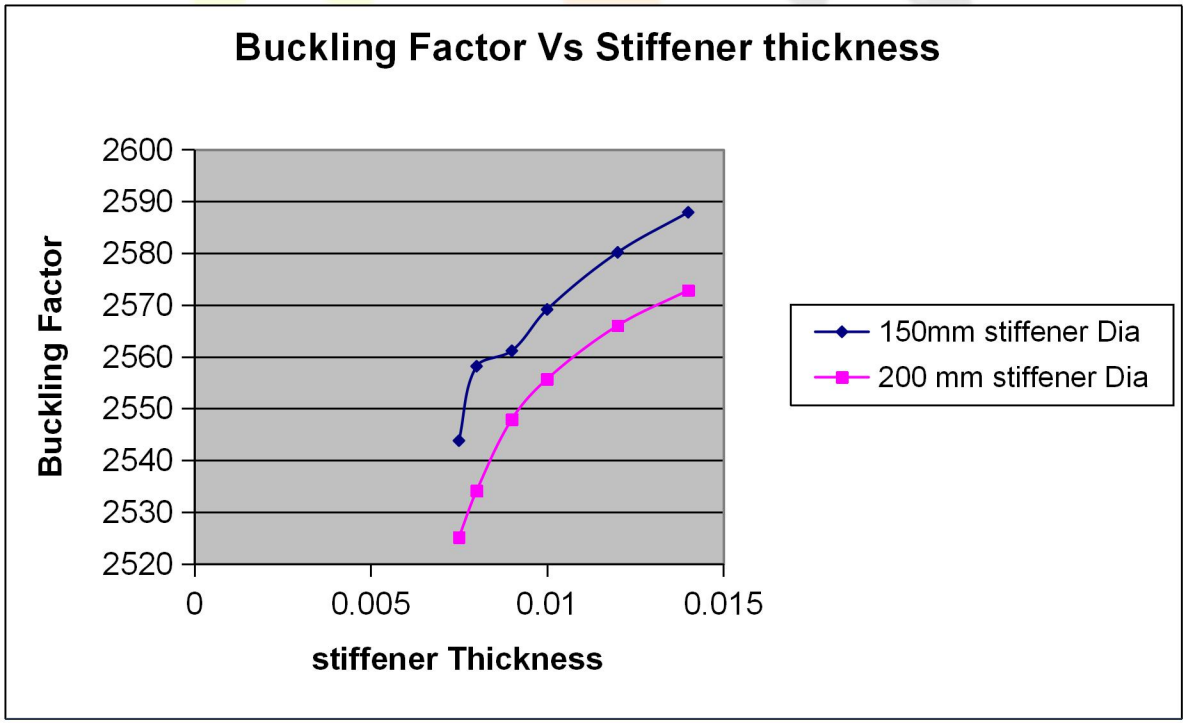


Figure18: Thickness of Stiffener Vs. Buckling factor

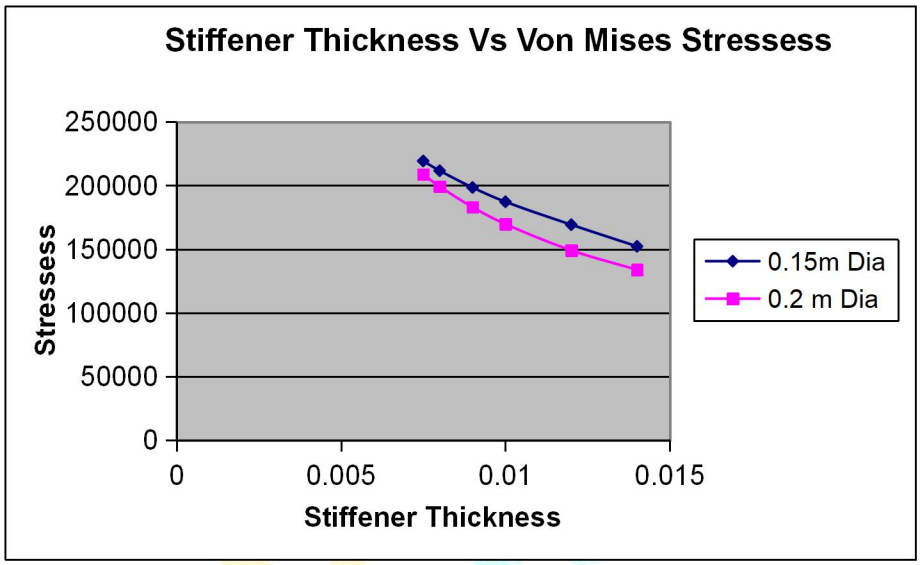


Figure19: Stiffener thickness Vs. Von mises Stresses

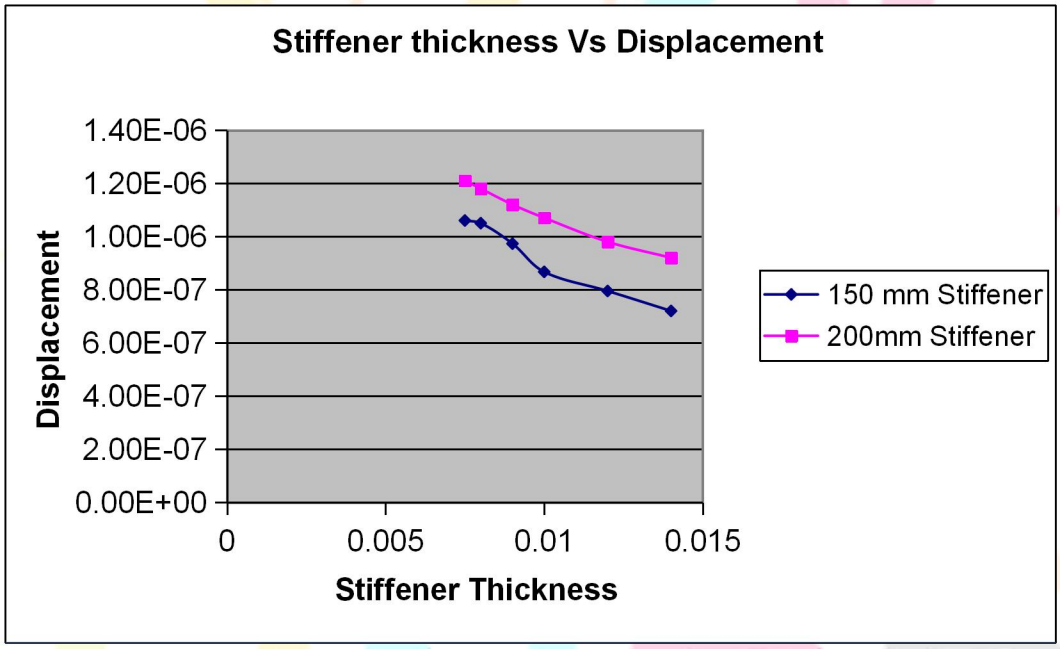


Figure20: Stiffener thickness Vs. Displacement

Buckling strength of the shell with two cutouts is calculated (i.e.2477.1) obtained from the table4 and is compared with shell without cutout (i.e.2499.51) obtained from the figure3 and observed the difference. This difference can be minimized or eliminated by adding the stiffener around the cutout. If we observe the table8, when the width of the stiffener is increased the buckling factor is also increases for the different thickness of the stiffener. Nodal solutions of the shell with cutout and stiffener are shown in figure21 and 21a.displacements with stiffeners are shown in figure22 and 22a.

Table8: Thickness of Stiffener Vs. Buckling factor

S.No	Thickness of the Stiffener (m)	Buckling Factor of shell with 150mm Stiffener for 100mm cut out size with Two Cutout	Buckling Factor of shell provided with 200mm Stiffener for 100mm cut out size with Two Cutout
1	0.0075	2525.12	2543.79
2	0.008	2534.10	2558.16
3	0.009	2547.92	2561.15
4	0.010	2555.70	2569.13
5	0.012	2566.02	2580.15
6	0.014	2572.83	2587.89

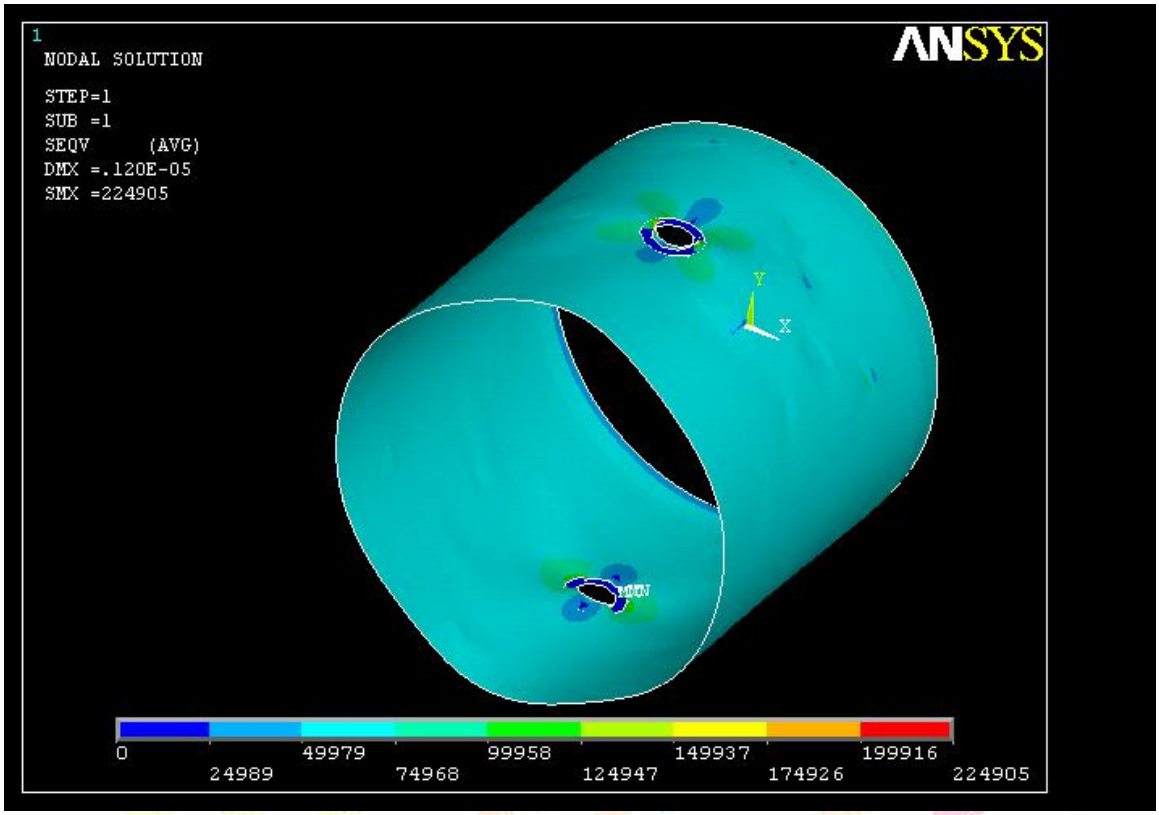


Figure21: Nodal solutions of two cutouts with stiffener

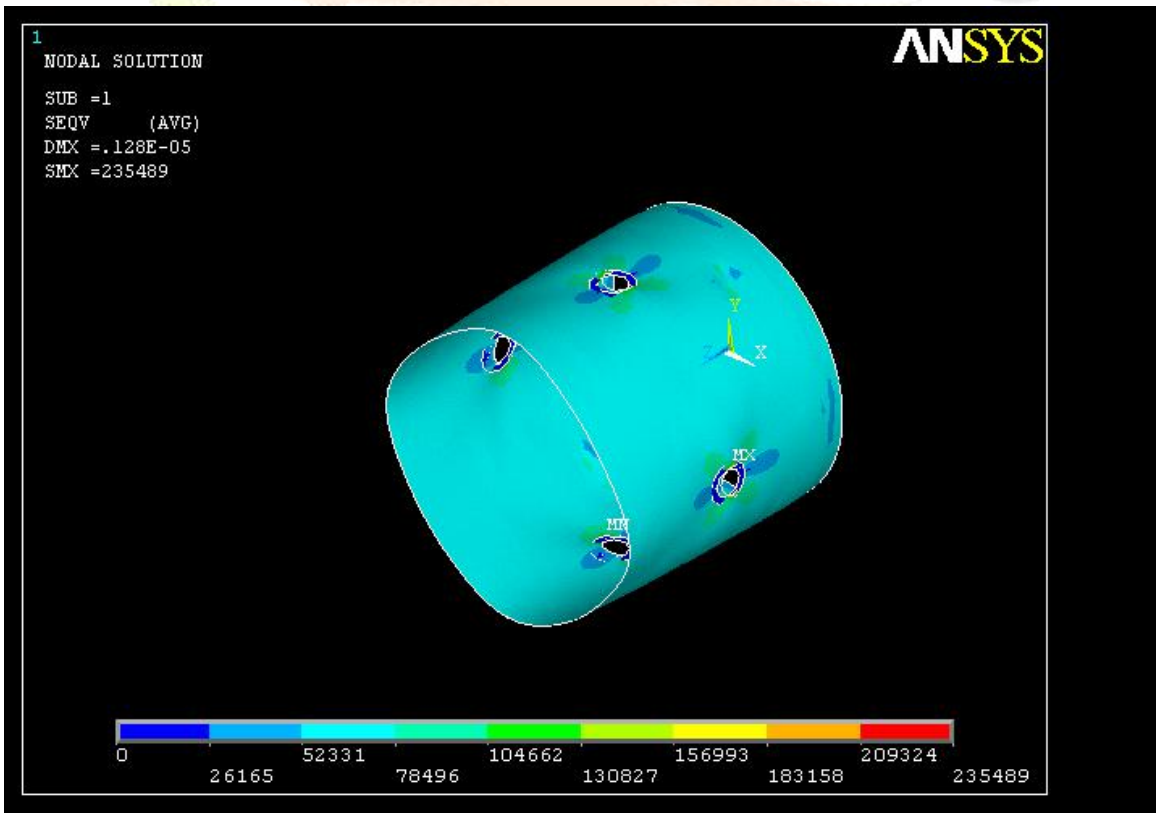


Figure21a: Nodal solutions of four cutouts with stiffener

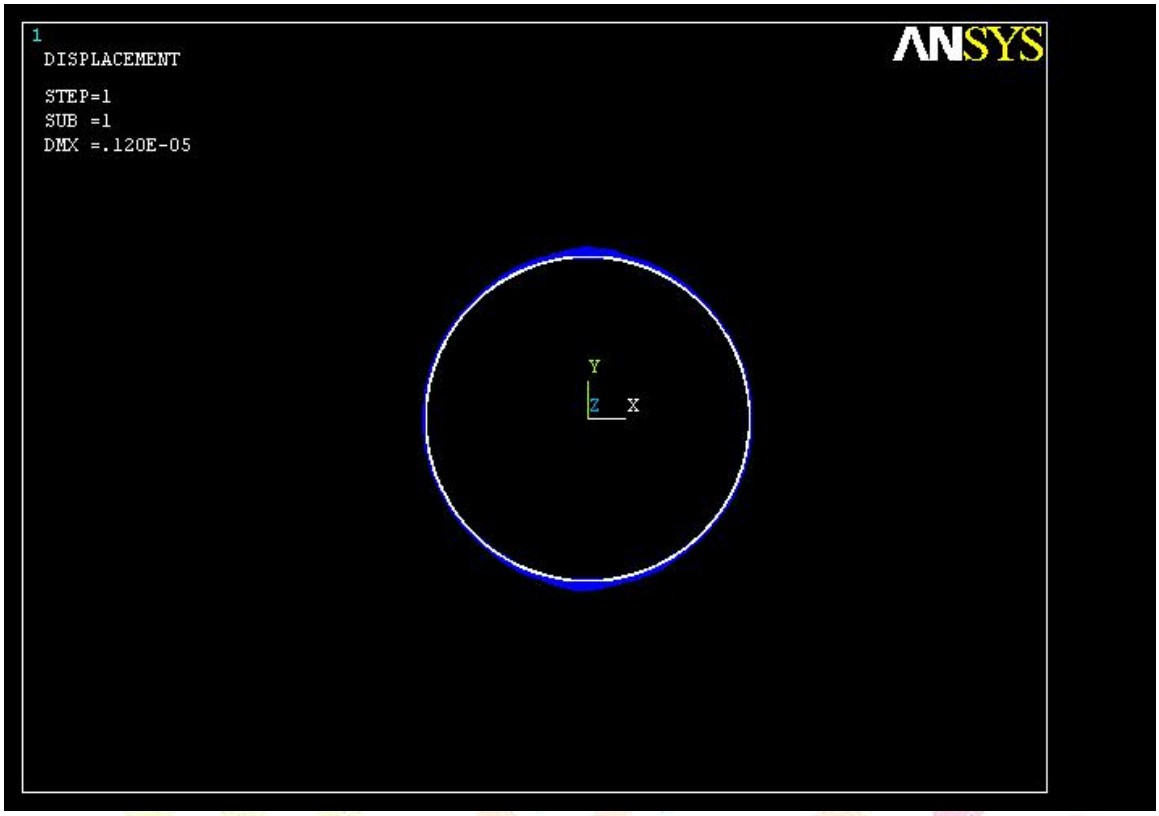


Figure22: Displacement of two cutouts with stiffener

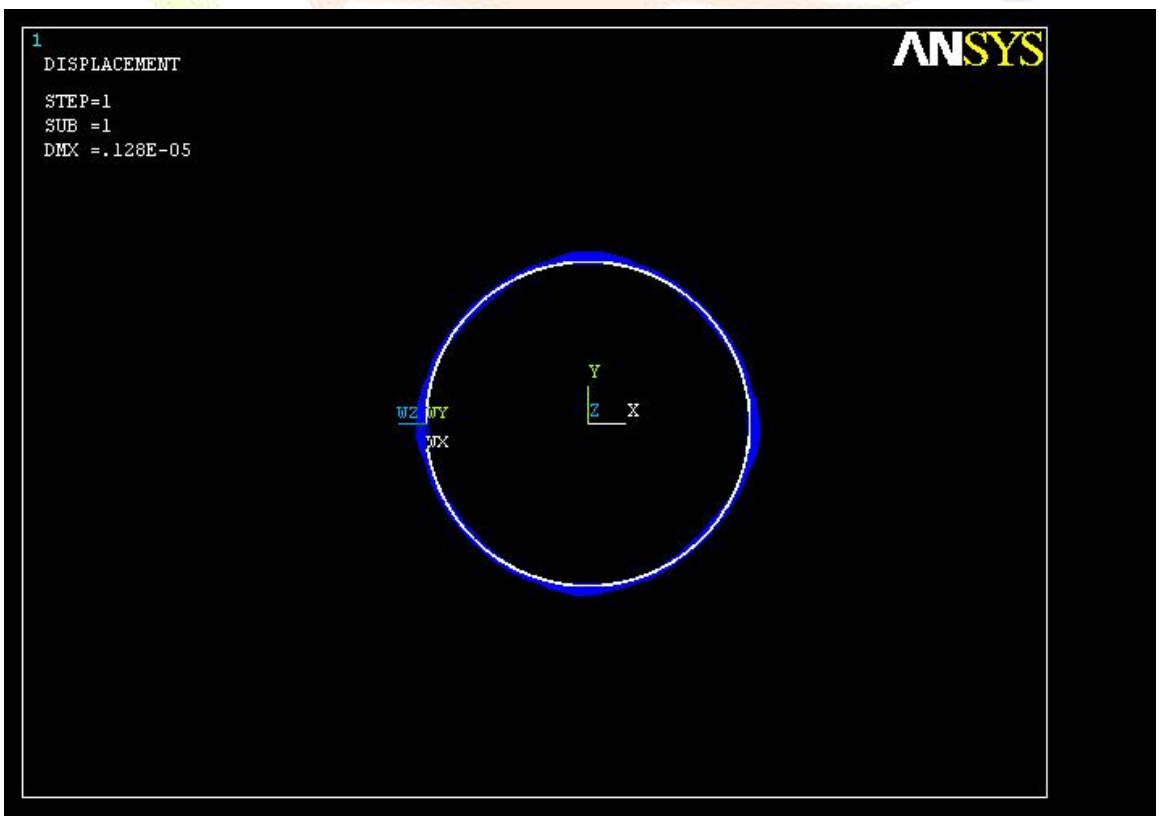


Figure22a: Displacement of four cutouts with stiffener

CONCLUSION

If a circular cutout in a circular cylinder loaded by axial compression is small enough, stress concentration at the cutout is not sufficient to cause buckling before the shell buckle due to some other initial imperfection. However larger cutouts can greatly reduce the buckling strength of

the cylinder. The amount of reduction of buckling load depends on parameter which is proportional to the cutout radius divided by the square root of the product of the shell radius and the shell thickness.

The character of the buckling of such shells can be described as a local buckling phenomenon, which leads to the general collapse of the shell.it appears that the stability of the local buckling model depends on whether or not the stress level in shell is high enough to make the shell sensitive to small disturbances.

Buckling factor depends also with the number of cutouts in shell. If the number of cutouts increased at small sizes there is no variation but in later sizes variations will be there because the total area of the shell will get reduced while increase the number of cutouts as well as the cutout. Hence the size of the cutout is limited to certain value depending upon the design considerations.

To reduce the stress concentration near the cutout, stiffener can be provided around the cutout.by increasing the stiffener thickness there is reduction in stress concentration similarly gradual increase in buckling strength. If the stiffener width around the cutout is increased the stiffener thickness can be reduced to get the same effect.

It can be stated that if stiffener area is 10% of the area of the shell, the thickness will around 20mm and if the stiffener is 20% of the area of the shell the thickness can be reduced to 15mm.the above results are compared with the buckling effect of shell without cutout and with cutout and stiffener.

The buckling load carrying capacity of shell is found to be dependent on the number of cutouts provided on the shell. The results shows that while analyzing for buckling loads there decreasing trend in the load carrying capacity in the shells from without cutout to four cutouts taken in the order.

The von mises stresses are in the increasing order with respect to the cutout sizes and number of cutouts. Adding stiffener in the form of ring at the cutouts improve all the three parameters as observed in the table8 for two cutouts.

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