

Buckling Analysis of Laminated Composite Cylindrical Shells

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ABSTRACT: Cutouts in cylindrical shell-type components are unavoidable in the construction of aerospace structures. This fact is significant because the structural failure of these components usually begins near the cutout because of high stress concentrations that initiate the formation of cracks. Hence, a cutout can trigger a local failure at a load level lower than the global failure load of a corresponding shell without a cutout. As a result, preliminary-design sizing of a cylindrical shell with a cutout is often based on the magnitude of the stress concentrations near the cutout. Therefore, an accurate assessment of the stress concentrations in a given shell subjected to various types of loading and support conditions is essential to the development of safe and reliable designs. In this paper, static and buckling analyses are performed for determining the stress and deformation fields in a thin laminated-composite cylindrical shell with and without an elliptical cutout is presented. The materials used are Steel, S2 Glass Epoxy and Kevlar. The analysis is carried out using FEA software and the modeling is in Pro/Engineer and analysis performed in Ansys..

KEYWORDS: Cylindrical shell, Cutouts, SCF, Deformation fields, Buckling factors, Stress criteria

I. INTRODUCTION

Circular cylindrical shells have been studied extensively due to its widespread application in modern engineering, such as in the aerospace industry, where fuselages and wings are mostly assemblies of shell structures. Fibre-reinforced composite materials offer more efficient structures for application due to their high specific modulus and strength compared to metallic material. Many theories of anisotropic shells have been developed, such as the classical theory (CST), first-order shear-deformation theory (FSDT) and high-order shear deformation (HSDT). Cutouts in cylindrical shell-type components are unavoidable in the construction of aerospace structures. In the paper by E. Oterkus, E. Madenci [1], a key finding of the results obtained with this analysis method is that the maximum tangential stress resultant concentration near a circular cutout in a tension-loaded, circular, quasi-isotropic shell increases by approximately 18% as the shell radius-to-length ratio decreases from 1.25 to 0.5. Likewise, increases in the maximum tangential stress-resultant concentration as large as 65% have been found to occur with a fivefold increase in cutout radius. In the work of S. K. Singh; A. Chakrabarti [2], the buckling behavior of laminated composite plates is studied using an efficient C^0 finite element model developed based on higher order zigzag theory. C.V. Srinivas, Y.J. Suresh, W.P. Prema Kumar [3], have studied the effects of the skew angle, aspect ratio, length-to-thickness-ratio, fibre orientation angle, and numbers of layers in the laminate and laminate sequence on the critical buckling load factor (K_{cr}) of antisymmetric composite laminates have also been presented. The critical buckling load factor (K_{cr}) is found to increase with the skew angle. When the number of layers in the laminate is large, the variation of critical buckling load factor (K_{cr}) with the number of layers is not appreciable. Dae Yong Park, Suk Yoon Chang, Sung Soon Yhim [4], have reported that the study for elastic buckling behavior of laminated composite plates with through-the-width delaminations are focused on various parameters, such as size of delamination, support condition, aspect ratio, width-to-thickness ratio, stacking sequences, and location of delamination and multiple delaminations. In another work by M Mohan Kumar [5], it was noted that variations in length-to-thickness ratio affects the critical buckling load. The buckling load decreases as the a/t ratio increases. The rate of decrease of buckling load is not uniform with the rate of increase of a/t ratio. As the aspect ratio increases, the critical buckling load of the plate decreases. When the aspect ratio changed from 1.0 to 1.7 then the rate of change of buckling load with a/b ratio is almost uniform. It was seen that the different fiber orientation angles affected the critical buckling load adversely. With the increase in the fiber angle, the buckling load decreased. The plate with [0]8 layup had the highest buckling load and the plate with [45]8 layup had the least. The reduction of the buckling load due to the presence of a cut-out is found to be significant. It is noted that the presence of cut-out lowers the buckling load and it varies with the cut-out shape. The plate with circular cut-out yielded

to the greatest critical buckling load while the rectangular cut-out failed for the lowest buckling load. Materials and their properties are given in Table1.

Table1: Material properties

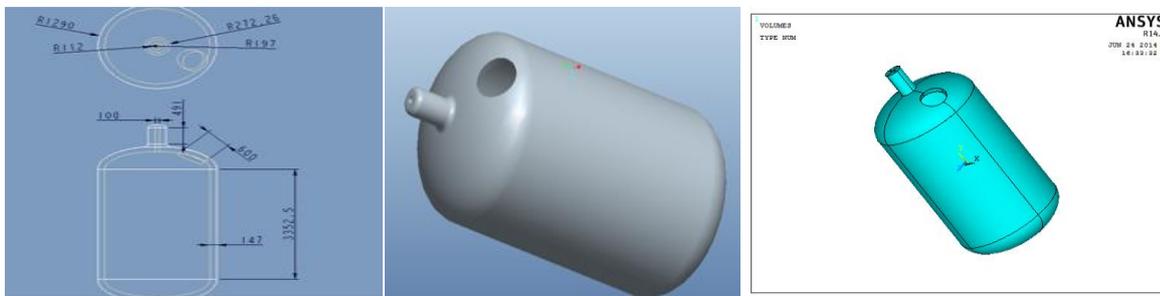
Physical properties	Glass	Kevlar
Density, g/cc	2.46	1.44
Ultimate Tensile Strength, MPa	489	362
Elongation at Break, %	5.70	2.40
Modulus of Elasticity, GPa	86.9	112
Poisson's Ratio	0.230	0.360
Shear Modulus, GPa	35.0	125

II. DESIGN OF ELLIPTICAL HEAD PARAMETER

The calculations are given here as per the relations among the design and geometry parameters. ASME design equation for elliptical head, $t = PDK / (2SE - 0.2P)$, Thickness of elliptical head t in mm, P = design pressure in N/mm^2 , D = outer diameter in mm, Design Pressure (P) = 21 N/mm^2 , Allowable Stress value (S) = 123 N/mm^2 , Joint Efficiency (E) = 1, Stress concentration factor (k) = $1/6 [2 + (\frac{a}{b})^2]$, Major axis length (a) = 2724 mm, Minor axis length (b) = 771.34 mm, $k = 1/6 [2 + (\frac{2724}{771.34})^2] = 0.62$, $t = (21 \times 2724) / (2 \times 123 \times 1 - 0.2 \times 21) = 146.67 \approx 147$, Hoop stress (σ_{t1}) = $(P \times D) / (2 \times t) = (21 \times 2724) / (2 \times 147) = 195$ MPa, Longitudinal stress (σ_{t2}) = $(P \times D) / (4 \times t) = (21 \times 2724) / (4 \times 147) = 97$ MPa.

Geometric Modeling and Meshing:

The cylindrical shell is modeled in Pro Engineer software with the dimensions given in Fig.1a and 3D model is shown in Fig.1b. This model is imported to ANSYS (FEA software) and Fig.2 shows the meshed model.

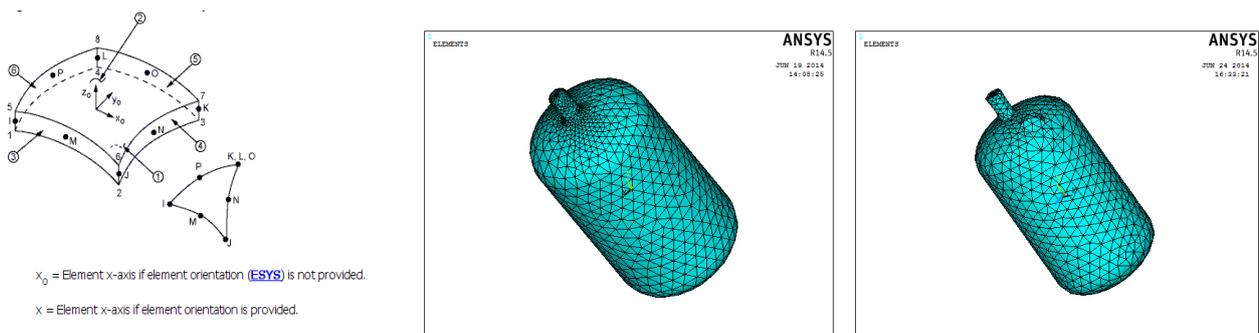


a. 2D model

b. 3D model

c. Ansys model

Fig.1: Pro-E model of pressure vessel with an elliptical cut out



a. Higher order shell element

b. Meshed model without and with hole

Fig 2 : Meshed Model of pressure vessel with and without hole

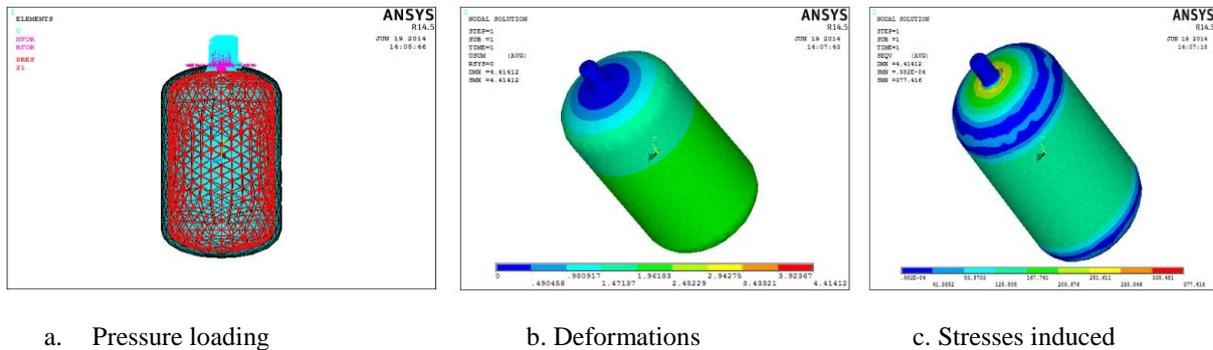


Fig. 3: Variation of displacement and stress induced in the cylinder made of steel without hole

Fig.3 shows the variation of displacement, stress induced in the cylinder when a pressure load of 21 MPa is applied on the mid surface. Table 2 gives the deformations, strains and stresses in the shell when pressure is acting on the cylinder made of steel. It is observed that the displacements of shell without hole are 4.414, 7.689 and 10.723 mm for the shell made of steel, Kevlar and S2 glass and stresses induced are 277.4 MPa, 349.4 MPa, and 399.108 MPa. On the other hand the shell with hole for the same materials the displacements are 1.256 mm, 2.674 mm, and 2.605 mm and stresses are respectively 166.156 MPa, 166.831 MPa and 165.583 MPa. By observing the Table 3, the buckling factor is more

Parameters	Material					
	Steel		Kevlar		S2 Glass	
	Without hole	With hole	Without hole	With hole	Without hole	With hole
Displacement, mm	4.414	1.256	7.689	2.674	10.723	2.605
Vonmises Stress, MPa	277.416	166.156	349.818	166.831	399.108	165.583
Strain	0.002	.001	0.003	.002	0.005	0.0014

for Steel, so buckling may occur first when Steel is used than S2 Glass and Kevlar.

Table 2: Structural Analysis of Shell made of Steel, Kevlar and S2 Glass materials with and without hole

III. BUCKLING LOAD CALCULATIONS

Bulking load for cylinder shell is $F = \frac{\pi^2 EI}{(KL)^2}$

F=maximum force, E=modulus of elasticity, I=moment of inertia, L=unsupported length of column, K=column effective length factor, K=1.0→ both side pinned, K=0.05→ both side fixed K=0.699→ one end fixed & one end pinned, K=2.0→one end fixed & one end free to move Length of the pressure vessel used, L=4686.72mm, $I = \frac{\pi}{64} d^3$, (d=100mm), $I = \frac{3.14}{64} (100)^3$ I=49062.5, A=251327mm²,

$$F = \frac{\pi^2 EI}{(KL)^2}$$

$$F = \frac{(3.14)(3.14)(209000)(49062.5)}{(5368.45)^2}$$

$$F = 3507.98 \text{ N}$$

$$\text{Pressure, } P = \frac{F}{A} = \frac{3507.98}{251327} = 0.0139 \text{ N/mm}^2$$

IV. BUCKLING ANALYSIS OF PRESSURE VESSEL

The buckling analysis is performed on pressure vessel with different materials using Ansys FEA software. The deformations and buckling factors estimated are shown in Table 3.

Table 3: Variation of displacement and BLF for shell made of Steel, S2Glass and Kevlar materials

Set	Displacement, mm			Buckling factor, kN		
	1	2	3	1	2	3
Steel	1.14	1.03	1.11	36	14	18.7
S2 glass	1.04	1.11	1.18	34	11	14.9
Kevlar	1.02	1.12	1.12	26	10.7	14.9

Structural Analysis:

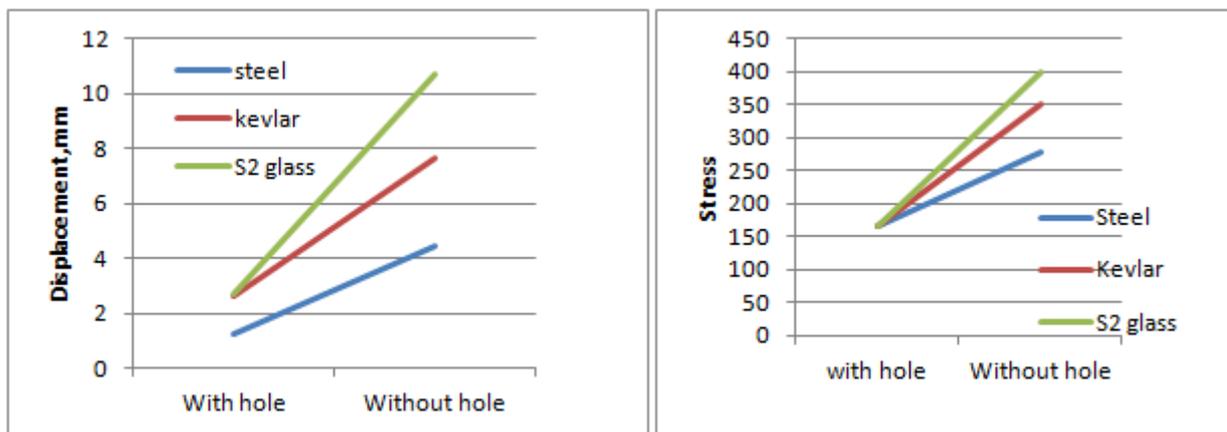


Fig 4: Displacements and stresses in Steel, Kevlar and S2 Glass for both with and without hole

By observing the above graph, the displacement is increasing for without hole and is more for S2 Glass, and the stress is increasing for without hole and is more for S2 Glass.

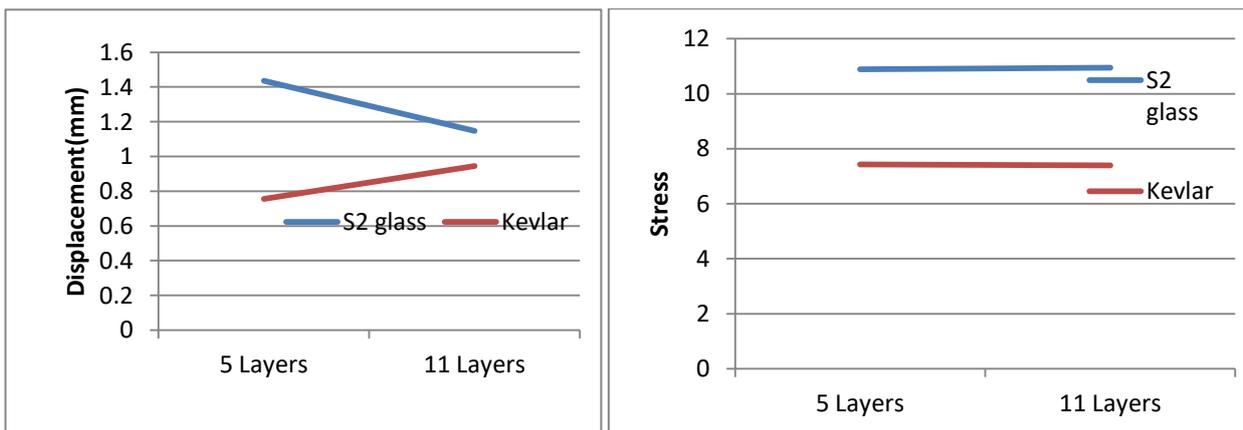


Fig 5: Displacement and Stress in Kevlar and S2 Glass shell elements with 5 and 11 layers

By observing the above graph, the displacement is increasing for 11 layers when Kevlar is used and decreasing for 11 layers for S2 Glass and the stress is almost same for 5 layers and 11 layers for both the materials.



Table 4: Displacement and Buckling factors in different material at various modes

Material	First Set		Second Set		Third Set	
	Displacement, mm	Buckling Factor	Displacement, mm	Buckling Factor	Displacement, mm	Buckling Factor
Steel	1.13636	36127.6	1.03838	14174.5	1.1126	18729.7
S2 glass	1.04198	33784.4	1.10944	10776.6	1.18223	14924.3
Kevlar	1.02738	26142.2	1.11873	10771.2	1.12145	14913.9

Table4 shows the buckling factors obtained in a cylindrical shell made of steel, S2 glass and Kevlar materials for first three modes. It is observed that the buckling factors are more for steel as compared to S2 glass and Kevlar.

Discussion: In this work, static and buckling analyses are performed for determining the stress and deformation fields in a thin laminated-composite cylindrical shell with an elliptical head without cutout and with cutout are presented. Modeling is done in Pro/Engineer. The materials used are Steel, S2 Glass Epoxy and Kevlar. By replacing the materials with composites, the weight of the vessel decreases. Analysis is performed in ANSYS. The analysis is carried using solid elements and shell elements with 5 and 11 layers are used as layer stacking for composite materials S2 Glass and Kevlar. By observing the structural analysis results, the stress values are decreased for vessel with hole than the vessel without hole and the stresses and displacements are within the range of permissible values for every material. . The stress values are almost reduced by 40% when steel is used, almost by 52.66% when Kevlar is used and almost by 58.19% when S2 Glass is used. Structural analysis is also done using shell element for composite materials for 5 layers and 11 layers. By observing the results, the stress values are less for Kevlar than S2 Glass Epoxy. The stresses are reduced by 31.75% when 5 layers are taken and by 32.45% when 11layers are taken.

Buckling analysis is also performed to calculate buckling load magnitudes that cause buckling and associated buckling modes. By observing the results, the deformation is more at the middle of the cylindrical shape so the failure starts at the area initially. By observing the Buckling Load Factors for the three materials, it is more for steel than the composite materials, so by using steel the failure chances are more than the composite materials. The factor is reduced by 20.31% for S2 Glass and by 20.37% for Kevlar. By comparing Kevlar and S2 Glass, the better performed material is Kevlar

V. CONCLUSION

As per the dimensions and the loads applied the cylindrical shell with and without cut out, the analysis of static and buckling is performed with three materials. The stress are reduced by 40% when steel is used, 52.66% when Kevlar is used and 58.19% when S2 Glass is used. By using shell elements of the same design of composite material for 5 layers and 11 layers it is found that the stresses are less for Kevlar than S2 Glass Epoxy. The buckling factor is reduced by 20.31% for S2 Glass and by 20.37% for Kevlar. By comparing Kevlar and S2 Glass, the better performed material is Kevlar

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