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Buckling behavior of composite cylindrical shells with cutout considering geometric imperfection

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Abstract. Creating different cutout shapes in order to make doors and windows, reduce the structural weight or implement various mechanisms increases the likelihood of buckling in thin-walled structures. In this study, the effect of cutout shape and geometric imperfection (GI) is simultaneously investigated on the critical buckling load and knock-down factor (KDF) of composite cylindrical shells. The GI is modeled using single perturbation load approach (SPLA). First, in order to assess the finite element model, the critical buckling load of a composite shell without cutout obtained by SPLA is compared with the experimental results available in the literature. Then, the effect of different shapes of cutout such as circular, elliptic and square, and perturbation load imperfection (PLI) is investigated on the buckling behavior of cylindrical shells. Results show that the critical buckling load of a shell without cutout decreases by increasing the PLI, whereas increasing the PLI does not have a great impact on the critical buckling load in the presence of cutout imperfection. Increasing the cutout area reduces the effect of the PLI, which results in an increase in the KDF.

Keywords: composite shell; buckling; cutout; single perturbation load approach; geometry imperfection; knock-down factor

1. Introduction

Composite structures are widely used in various industries, especially in the aerospace industry, due to their high specific strength and stiffness ratio. However, buckling and instability occur in the thin-walled cylindrical shells if the axial compressive load exceeds the design load. In the construction of aerospace structures the use of cylindrical shell-type components with cutout sections are unavoidable. Different shapes of cutouts are used in order to make doors or windows, decrease the structure weight or cross various mechanisms. Cutouts due to making high stress concentration affect the buckling behavior of the structure. Therefore, the design of thin-walled composite cylindrical shells with cutouts is of great important for engineers. The NASA empirical guideline (NASA-SP 8007) (Peterson *et al*. 1968) is mostly used for the buckling analysis and design of cylindrical structures. This guideline was first presented in 1968 and was then modified by Arbocz and Starnes (2002). According to this guideline, the changes in the knock-down factor (KDF or ρ: the ratio of the actual buckling load to the theoretical buckling load) are plotted, as seen in Fig. 1, versus the ratio of the cylinder radius to the thickness (R/t). Fig. 1 has been obtained from the buckling tests of metallic cylinders. The buckling load predicted for composite cylindrical shells from this diagram is conservative due to the increased precision of fabrication and loading using advanced construction and testing devices. Also, geometric parameters significantly affect the buckling load, but the NASA-SP 8007 diagram only considers the influence of cylinder thickness and radius. In other words, it is independent of cylinder length. Wagner *et al*. (2017a, b) proposed a robust design criterion for axially loaded cylindrical shells in order to improve NASA-SP 8007 with the aid of the single boundary perturbation approach. They concluded that the KDF values of short thin-walled shells (i.e., $L/R≈1\text{-}2$, $R/t > 200$) and long thick-walled shells (i.e., $L/R > 3$, $R/t < 250$) are significantly higher and lower, respectively than the values reported in NASA SP-8007. In addition to geometric effects, the impact of cutouts on the buckling load and KDF is not investigated comprehensively.

In addition to cutouts, a bundle of superficial distortions is created on thin shells known as geometric imperfections during the manufacturing process. In the finite element simulation, cutouts are physically modeled on the geometry, but superficial distortions are applied to the shell using one of the imperfection methods. There are several methods for modeling geometric imperfections in thin-walled composite cylinders: linear buckling mode shape imperfection, reduced energy method, single perturbation load approach (SPLA) and geometric dimple imperfection. Yamad *et al*. (2001) first used the linear buckling mode shape method for composite cylinders. They extracted linear buckling modes from the buckling solution of a cylinder. Then, knowing that one of these modes appears at the onset of buckling, it can be assumed that the fraction of one of these modes is effective on the cylinder. Thus, by applying a fraction of

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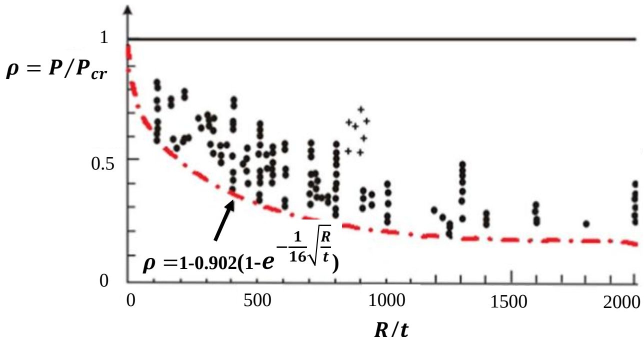


Fig. 1 Experimental data for an isotropic cylindrical shell under axial compressive load (modified by Arbocz and Starnes 2002)

a mode on the cylinder, the geometric imperfection can be modeled. Sosa *et al.* (2006) presented the reduced energy method. In this method, the linear buckling mode is first extracted, then the linear buckling mode is applied as the initial displacement to the cylinder and the critical buckling load is obtained by using the theory of strain energy. In SPLA, presented by Hühne *et al.* (2005, 2008), the effect of single load on the model surface is used to consider the worst geometric imperfection. The single perturbation load produces a fovea acting as an imperfection and increasing with the single load, which ends in buckling due to the reduction of the structural endurance. As a result, the structural sensitivity is eliminated from the single perturbation load and the load at that point can be considered as the buckling load of structure. Hühne *et al.* (2005, 2008) derived this concept from a study conducted by Esslinger (1969) who carried out some experiments using high speed cameras and realized that there was a local distortion before the occurrence of instability at a point. The geometric dimple imperfection method was proposed by Wullschleger and Meyer-Piening (2002). In this method, a dimple with one half-wave sinusoid in the circumferential direction and one in the axial direction is applied on the cylinder body before applying the axial load, and then the axial load increases until buckling occurs. Aktas and Balcioglu (2014) studied the effect of different parameters such as plate thickness, diameter of circular cutout, the distance between circular cutouts and rowing orientation angle on the buckling load of E-glass/vinylester pultruded composite beams with single and double circular cutouts. Khayat *et al.* (2016) investigated the buckling behavior of laminated composite cylindrical shells using semi-analytical finite strip method. Their results indicate that considering pressure stiffness causes buckling pressure reduction which in turn depends on various parameters such as geometry and lay-ups of the shell.

The SPLA is a new method that has been used by some researchers over the last few years to determine the critical buckling load of cylindrical shells. The SPLA is a promising deterministic procedure based on the mechanical considerations to determine reasonable design loads regarding cylindrical shells in axial compression. The EU DESICOS project (Degenhardt 2011) presented a new guideline on the design of composite structure imperfection that reduces the weight of the structure and facilitates its design. Orifici and Bisagni (2013) investigated the SPLA for cylinders with small and large square cutouts. They showed that in cylinders with cutouts, the place of single perturbation load was effective on the buckling load. Castro *et al.* (2013) used SPLA to develop semi-analytic models. Arbelo *et al.* (2014) reported several studies that estimated the knock-down factor by using the SPLA. Priyadarsini *et al.* (2012) carried out numerical and experimental studies on the buckling of composite cylindrical shells under axial load with and without defects. Ismail *et al.* (2014) inspected the NASA's empirical guideline to improve buckling in composite cylindrical shells using changes in the amplitude of imperfection. A good correlation between numerical and experimental data was found. Castro *et al.* (2014) studied the numerical solution of each geometric imperfection method and its effect on the composite cylindrical buckling and compared the results obtained from different methods. Ismail *et al.* (2015) investigated the response of cylindrical shells with different geometries and materials using both eigenvalue and perturbation load methods. Their goal was to develop a better approach for the design and evaluation of buckling. They concluded that the eigenvalue method was more conservative than the SPLA, and the SPLA showed better agreement with the experimental results. Wang *et al.* (2019a, b) obtained the KDF by using the SPLA and the worst multiple perturbation load approach (WMPLA) for cylindrical shell structures under axial compression. The WMPLA is performed to find the worst combination of dimple-shape geometric imperfections to predict the lower-bound buckling load. The results show that the SPLA-based methods produce higher KDFs than the test results and are sensitive to the distribution of measured imperfections. While the KDFs predicted by the WMPLA are very close to the experimental results.

Kriegesmann *et al.* (2016) investigated two main issues that should be fully understood to have a better insight on the potentials and limits of SPLA applications. The first issue is related to the number of perturbation loads applied concurrently. It was shown that the lower bound obtained by multiple perturbation load approach decreases with increasing the number of perturbation loads. It implies that the single perturbation load does not lead to the worst imperfection case. The SPLA method applied to some cylinders showed that certain cylinders seemed to be less sensitive to local imperfections than others. Buckling load of these cylinders is smaller than the design load because other effects such as load eccentricity and thickness variations become predominant.

Moniri-Bidgoli and Heidari-Rarani (2016) analytically and numerically analyzed the buckling behavior of metal-composite cylindrical shells under axial compressive load. They showed that metal layers are effective means of improving the buckling load of such structures. Taheri-Behrooz *et al.* (Taheri-Behrooz *et al.* 2017, Taheri-Behrooz and Omidi 2018) applied the numerical based linear buckling mode shape imperfection method and modified it using a stochastic method to assess the effect of geometrical imperfections on the buckling of cylindrical shells with and without circular cutout. To verify the accuracy and difference between various numerical and analytical results against real-world cases, composite cylinders made of glass/epoxy with the stacking sequence of [90/+23/-23/90] were tested. The experimental results were consistent with the nonlinear numerical results of cylinders without cutout. Khakimova *et al*. (2017) considered geometric imperfections including mid-surface imperfection, thickness imperfection and fiber volume fraction correction into the finite element model and compared the results of KDF with those of SPLA and NASA-SP 8007 method. Ma *et al*. (2018) tested a composite cylindrical shell with a central rectangular opening under axial compression and measured its critical buckling load. Also, a finite element model of the shell with Hashin failure criterion was established to analyze its buckling and post-buckling behaviors by nonlinear Newton-Raphson method.

From the above review, it can be found that a considerable number of studies have only focused on the inspection of SPLA in composite cylindrical shells as well as comparing this method with other imperfection approaches in shell structures. However, there are still some issues regarding buckling of composite shells such as effect of different shapes and areas of cutouts as well as simultaneous effect of geometric imperfection (GI) and cutout on the critical buckling load and KDF that they need more investigations. Therefore, the buckling behavior of composite cylindrical shells without and with cutout, and consideration of GI is investigated in this study. SLPA is used for modeling GI in shells. Moreover, the effect of simultaneous perturbation load imperfection (PLI) and Cutout imperfection (CI), as two separate imperfections, and their distinct effect on KDF are investigated.

Table 1 Geometric properties of composite cylindrical shells

	Bisagni (2000)	Eglitis *et al*. (2009)
Material	Carbon/epoxy	Glass/epoxy
Length, mm	520	560
Radius, mm	350	150
Thickness, mm	1.32	1.1
L/R	1.45	3.73
R/t	265	136
Ply arrangement	$[0/45/-45/0]_s$	$[0_4]_s$
t_{ply}, mm	0.33	0.275
Cutout area, mm^2	7850	7850

Table 2 Mechanical properties of composite cylindrical shell

	E_{11}, GPa	E_{22}, GPa	G_{12}, GPa	v_{12}	ρ, kg/m^3
Carbon/epoxy (Bisagni 2000)	52	52	2.35	0.302	1320
Glass/epoxy (Eglitis *et al*. 2009)	18.2	18.6	4.56	0.16	-

2. Finite element modeling

2.1 Geometric and mechanical properties

The geometric and mechanical properties of two composite cylindrical shells made of carbon roving tape wrapped/epoxy and E-glass fabric/polyester are extracted from Bisagni (2000) and Eglitis *et al*. (2009), respectively. Tables 1 and 2 show the related properties.

2.2 Modeling

In this section, the simulation process of buckling is described in the commercial finite element software, ABAQUS. A cylindrical shell is modeled according to Table 1 and mechanical properties and composite layup are assigned to the model according to Table 2. The model is meshed by four-node shell element with reduced integration (S4R). The boundary conditions are applied on the both edges of the shell (see Fig. 2). All six degrees of freedom are fixed at one edge and the five degrees of freedom except for displacement along the cylinder's length is constrained at another edge. First, a linear eigenvalue buckling analysis is performed on a glass/epoxy cylindrical shell to determine the appropriate element size and mesh refinement. Then, the size of elements is changed to reach an almost identical critical buckling load. Table 3 shows the results of the mesh refinement for a glass/epoxy cylinder without cutout. The suitable number of elements for carbon/epoxy and glass/epoxy cylindrical shells are obtained 7560 and 21056, respectively.

For the cylindrical shell with cutout, the elements around the cutout should be smaller than other parts owing to the stress concentration. After the mesh sensitivity

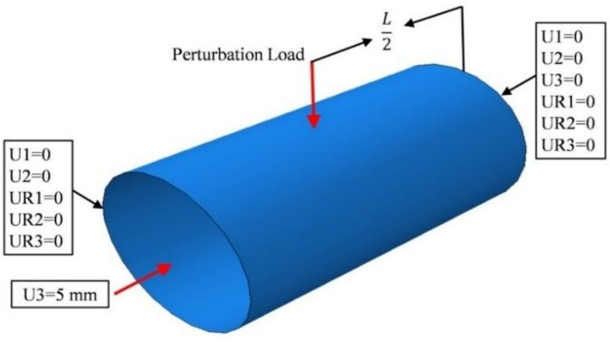


Fig. 2 Boundary conditions and location of applying single perturbation load

Table 3 Mesh refinement analysis in glass/epoxy shell without cutout

Element size, mm	0.013	0.0095	0.0066	0.0054	0.005
Number of elements	3139	5841	12155	18200	21056
Linear buckling load, kN	75.1	68.5	64.8	63.7	63.4

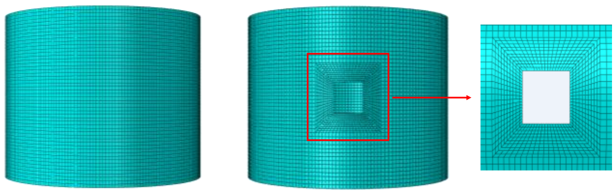


Fig. 3 Mesh density in finite element model of composite cylinders

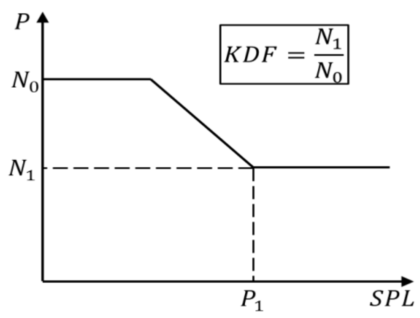


Fig. 4 Variation of buckling load according to SPL

analysis, the number of elements around the cutout in carbon/epoxy and glass/epoxy cylindrical shells is selected 960 and 1768, respectively. Thus, the total number of elements in cylindrical shell with and without cutout are 12000 and 17840, respectively. Fig. 3 shows the mesh density in cylindrical shells without and with cutout.

In this study, a linear analysis is first performed on all samples to obtain the local or global buckling load. In spite of high computational time of nonlinear buckling analysis rather than the linear buckling analysis, this method is preferred in structural design due to its accuracy. Therefore, an implicit nonlinear dynamic analysis is performed to obtain the load-end shortening curve. Nonlinear analysis is performed in two steps. In the first step, the single perturbation load is applied in the radial direction at the middle of the cylindrical shell (as shown in Fig. 2) while both edges of the cylinder are clamped. In the second step, the single perturbation load is kept constant and the axial displacement is applied to the shell's edge until the specimen buckles. These two steps are repeated and the single perturbation load is increased at each stage. As shown schematically in Fig. 4, if the buckling load from the linear analysis is considered as N_0, it will be constant without considering the single perturbation load (SPL). By increasing the SPL, the buckling load decreases continuously until it reaches to a constant load N_1 for a given value of the perturbation load, i.e., P_1. The load N_1 is introduced as the design buckling load. The N_1 (the corresponding buckling load for specific SPL, P_1) to N_0 (linear buckling load) ratio is called the knock-down factor (KDF). The goals of the first and second steps are to determine the worst GI and applying displacement on shell until the occurrence of buckling, respectively.

3. Results and discussion

3.1 Cylindrical shell without cutout

In order to demonstrate the potentials of SPLA, the finite element critical buckling load of the composite cylindrical shells without cutout are compared with the experimental data of Refs. (Bisagni 2000, Eglitis *et al*. 2009), as in Table 4. Also, the buckling load of the linear eigenvalue solution and NASA-SP 8007 guideline are presented.

Table 4 shows that linear eigenvalue solution and NASA-SP 8007 guideline respectively overestimates and underestimates the buckling load in comparison to experiments. In the case of eigenvalue solution, the overestimation can be attributed to ignoring the effect of imperfections on the structural model. The obtained buckling loads by SPLA are very close to the experimental ones. Thus, the finite element model is verified according to the Table 4 results.

Fig. 5 shows buckling load and corresponding mode shapes of initial 12 modes for a cylinder with circular cutout. As shown in Fig. 5, first ten mode shapes are

Table 4 Comparison of finite element and experimental critical buckling loads (kN) of composite cylindrical shells without cutout (values in parentheses show the percent of difference with respect to experiment)

Materials	Experiment	NASA-SP 8007	Finite element results-present study	
			SPLA	Linear eigenvalue
Carbon/ epoxy	163.46 (Bisagni 2000)	105.1 (-35.7 %)	159.1 (-2.7 %)	248.1 (51.8 %)
Glass/ epoxy	39.73 (Eglitis *et al*. 2009)	33.8 (-14.9 %)	40 (0.67 %)	63.4 (59.58 %)

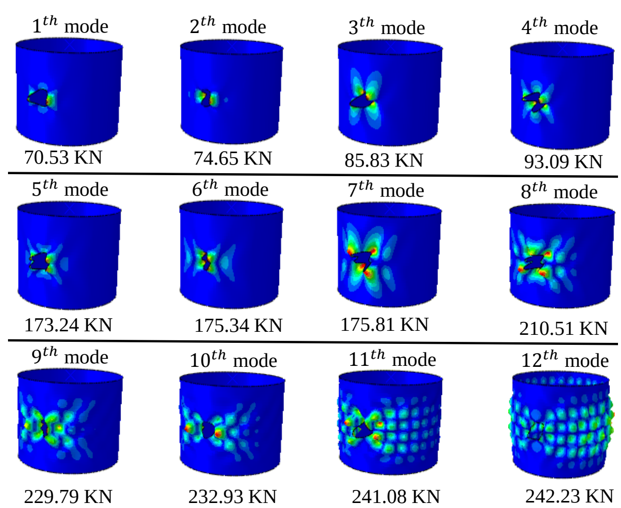


Fig. 5 Twelve mode shapes of carbon/epoxy cylindrical shell with circular cutout using linear buckling analysis

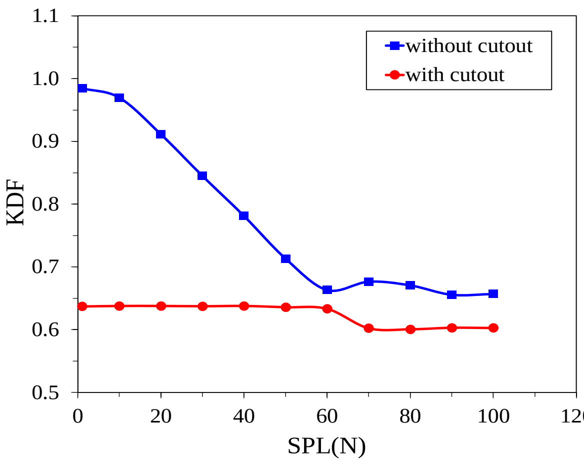
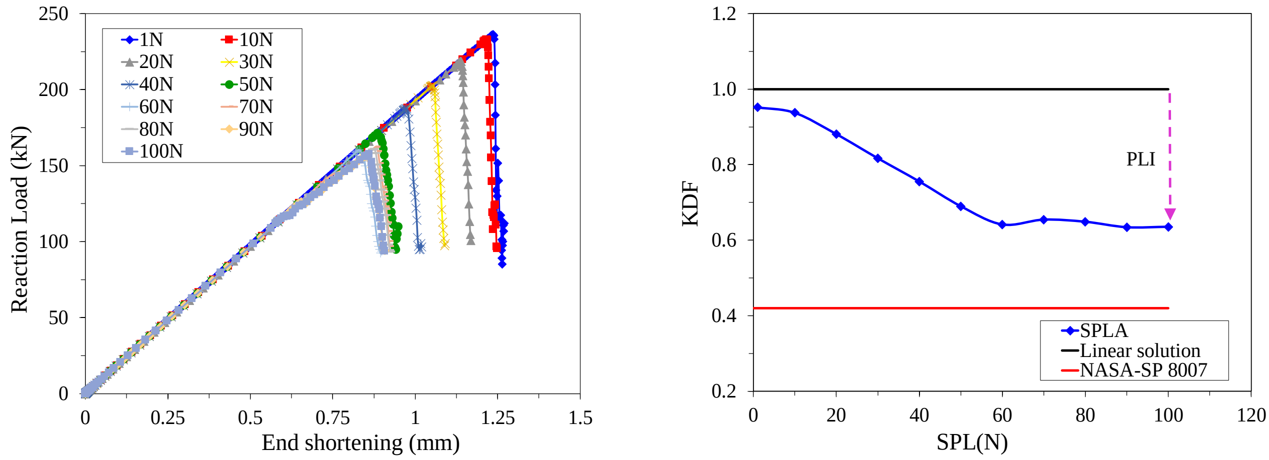


Fig. 6 Effect of cutout on KDF of a composite cylindrical shell

depicting local buckling in the cylinder while the mode shapes of 11 and 12 are associated with the cylinder global buckling.Fig. 6 shows the effect of CI on KDF of carbon/epoxy cylindrical shells without and with circular cutout. It can be seen that the presence of cutout reduces the effect of increasing the single perturbation load on the KDF.

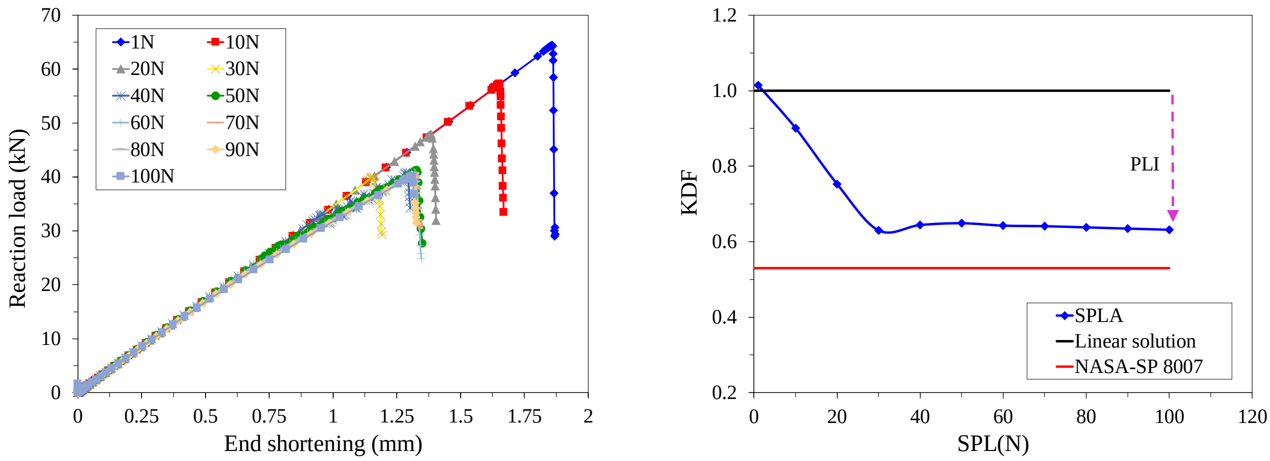
Figs. 7(a) and (b) respectively show the effect of single perturbation load on the load-end shortening and KDF of carbon/epoxy cylindrical shell. According to this figure, the maximum reaction load in the load-end shortening curves and KDF decrease when SPL increases up to 60 N. Consequently, it means that the buckling resistance of cylindrical shells decreases as GI increases. But the reaction load and KDF remain approximately constant for SPL of 60-100 N. Fig. 7(b) compares the KDF values of linear eigenvalue solution, NASA-SP 8007 and SPLA. The reaction load and KDF based on SPLA are 159.1 kN and 0.663, respectively. As stated before, the NASA guideline gives a very conservative KDF. Also, the KDF of SPLA deviates from the linear eigenvalue solution by increasing PLI. In Fig. 7(b), the KDF increases from SPL 60 to 70 and and decreases from SPL 70 to 80. These fluctuations in nonlinear finite element analysis are normal and the overall trend of KDF change is important.

Fig. 8 show the effect of perturbation load on the load-



(a) (b)

Fig. 7 Effect of increasing single perturbation load on load-end shortening and KDF of carbon/epoxy cylindrical shell



(a) (b)

Fig. 8 Effect of increasing single perturbation load on the load-end shortening and KDF of glass/epoxy cylindrical shell

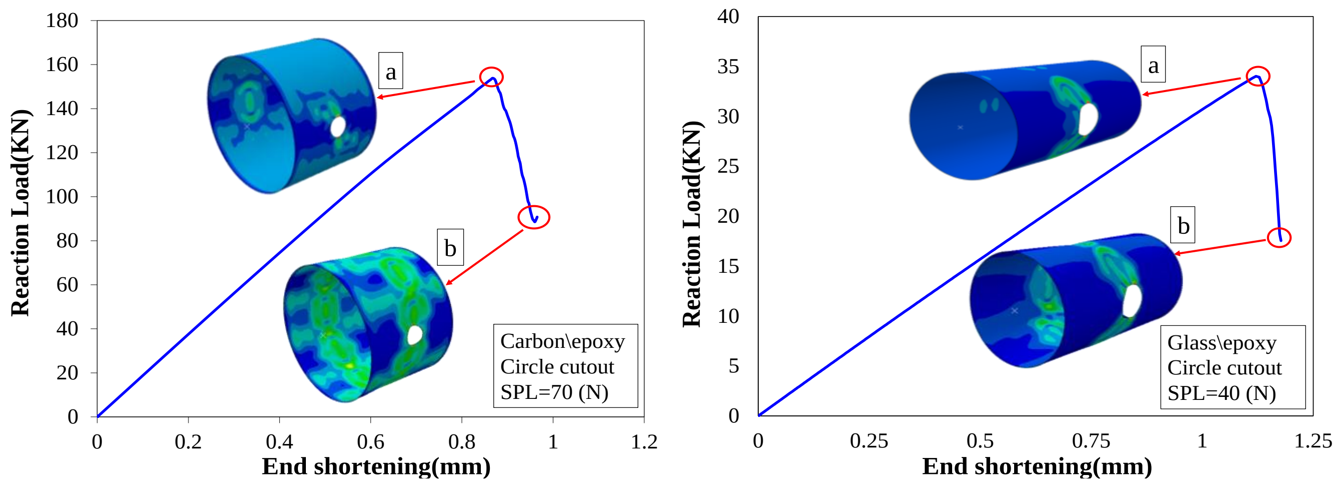


Fig. 9 Load-end shortening curves from nonlinear analysis with SPLA

end shortening and KDF of glass/epoxy cylindrical shell. The similar behavior is observed as the carbon/epoxy cylindrical shell. In this case, the maximum reaction load and KDF are 40 kN and 0.596, respectively. In Fig. 8(b), in spite of decreasing trend in KDF using SPLA, there is a local minimum at SPL = 30. Small fluctuations in results of nonlinear finite element analyses are evident. In numerical methods unlike the analytical methods, very small fluctuations happen due to cumulative errors.

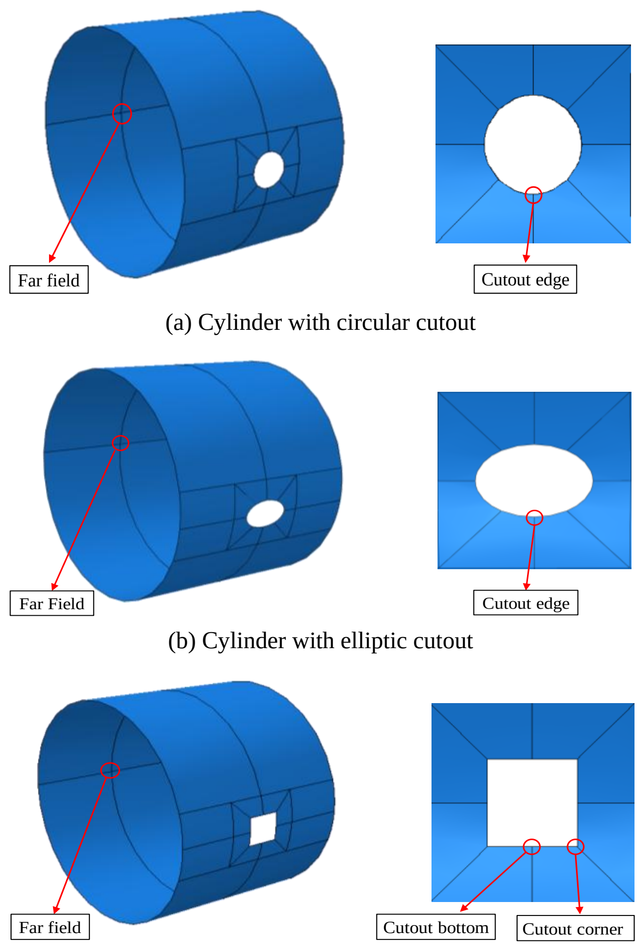
Fig. 9 show the load-end shortening of the carbon/epoxy and glass/epoxy cylindrical shell with circular cutout. As the compressive load increases, the local-neighbouring dimples are developed as shown in points (a). By more increasing of the compressive load, the axial and circumferential buckling modes are developed and post-buckling happens. Points (b) show this phenomenon.

3.2 Cylindrical shell with cutout

In the case of cylindrical shells with cutout, the cylinder loses its axisymmetric. Therefore, the single perturbation load should be applied to the critical areas surrounding the cutout as well as the regions far from it so that the lowest KDF obtains. Figs. 10(a) and (b) show the locations of applying perturbation load at the cutout edge and far field for circular and elliptic cutouts. For square cutouts as shown in Fig. 10(c), three locations, i.e., cutout edge, cutout corner and far field are considered for applying perturbation load.

3.2.1 Effect of single perturbation load on the critical buckling load

Fig. 11 shows the variations of KDF for a cylinder with different cutout shapes, materials and locations of perturbation load. From these results, it is found that the CI on the shell reduces the buckling resistance and thus the critical buckling load decreases. Moreover, the critical condition according to the location of the PLI happens when the perturbation load is applied far from the cutout. Due to the existence of cutout, the single perturbation load creates a GI in the cylinder. Accordingly, when these two imperfections are located far from each other, they make remarkable effect on the reduction of critical buckling loads.



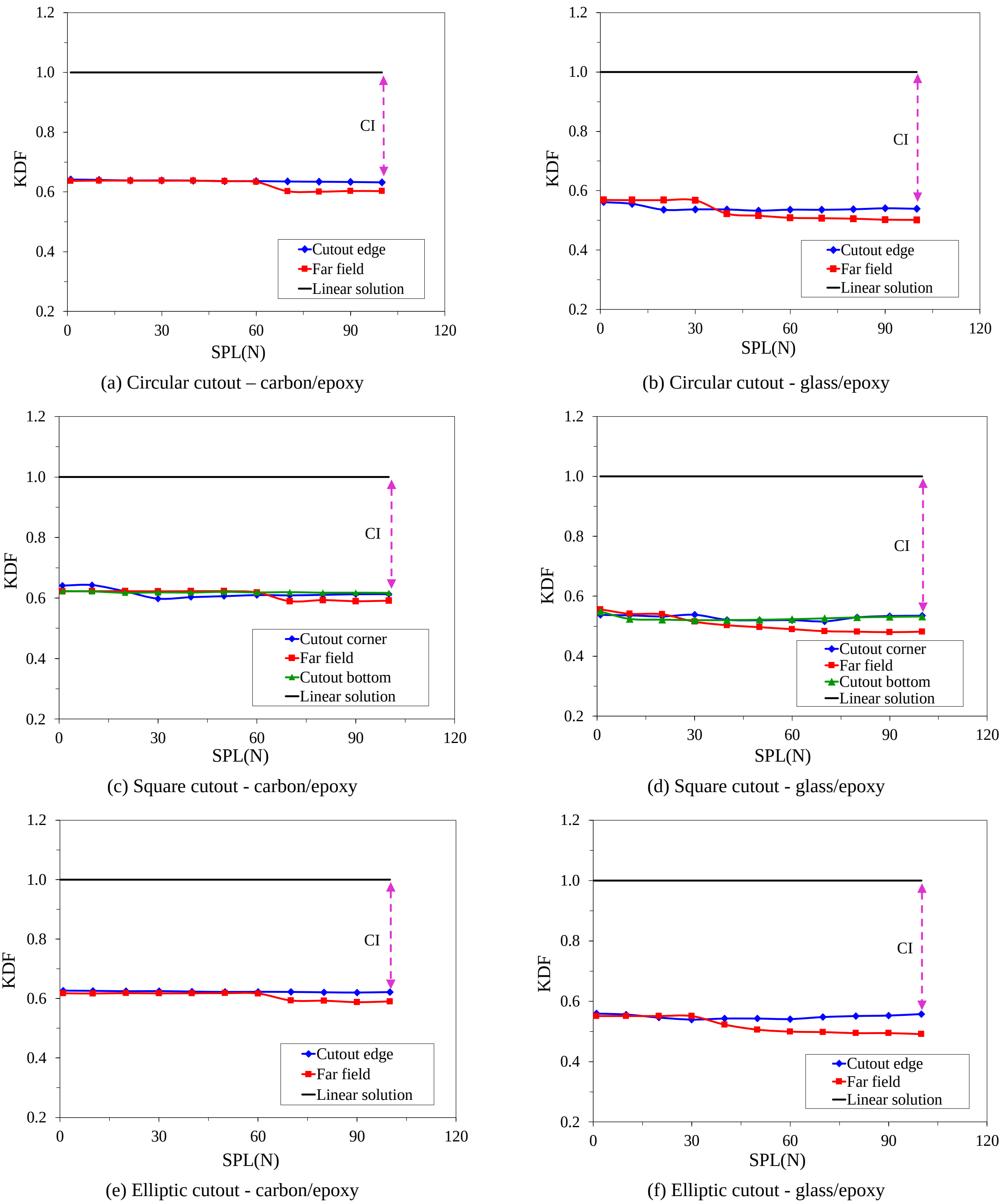
(a) Cylinder with circular cutout

(b) Cylinder with elliptic cutout

(c) Cylinder with square cutout

Fig. 10 Locations of single perturbation load in cylinders with cutout

Table 5 shows different numerical solutions for critical buckling load of composite shells with various cutouts. It can be seen that the local buckling is more conservative than other solutions for all three shapes of cutout due to the prediction of the first buckling mode that occurs locally



(a) Circular cutout – carbon/epoxy
(b) Circular cutout - glass/epoxy
(c) Square cutout - carbon/epoxy
(d) Square cutout - glass/epoxy
(e) Elliptic cutout - carbon/epoxy
(f) Elliptic cutout - glass/epoxy

Fig. 11 Changes in KDF for different cutout shapes and locations of single perturbation loads

around the cutouts. Moreover, the global buckling solution overestimates the buckling load in comparison to SPLA. Therefore, the linear buckling solution unrealistically predicts the critical buckling load. Also, the most reduction in the critical buckling load occurs in square cutout due to their sharp corners. These sharp corners result in stress concentration at these points and reduce the buckling resistance of cylindrical shells. In addition, owing to the larger area of circular cutouts, the circular cutout has less resistance against buckling than the elliptic cutout.

3.2.2 Effect of cutout area on KDF

In this section, the effect of square cutout area on KDF is studied. The areas are selected as 100, 400, 1600, 2500 and 7850 mm^2. As shown in Fig. 12, increasing the area of cutout reduces the effect of the single perturbation load,

Table 5 Critical buckling load (kN) of composite cylindrical shells with different shapes of cutouts

Cutout	Material	Linear solution, local buckling	Linear solution, global buckling	Nonlinear solution, without SPLA	Nonlinear solution, SPLA
Circular	Carbon/epoxy	70.53	242.2	155.5	144.44
	Glass/epoxy	14.19	59.9	33.66	30
Elliptic	Carbon/epoxy	60.98	246.7	154.7	144.67
	Glass/epoxy	12.5	60.07	33.69	30.1
Square	Carbon/epoxy	55.29	247.98	159.8	143.67
	Glass/epoxy	10.56	60.21	33.8	29

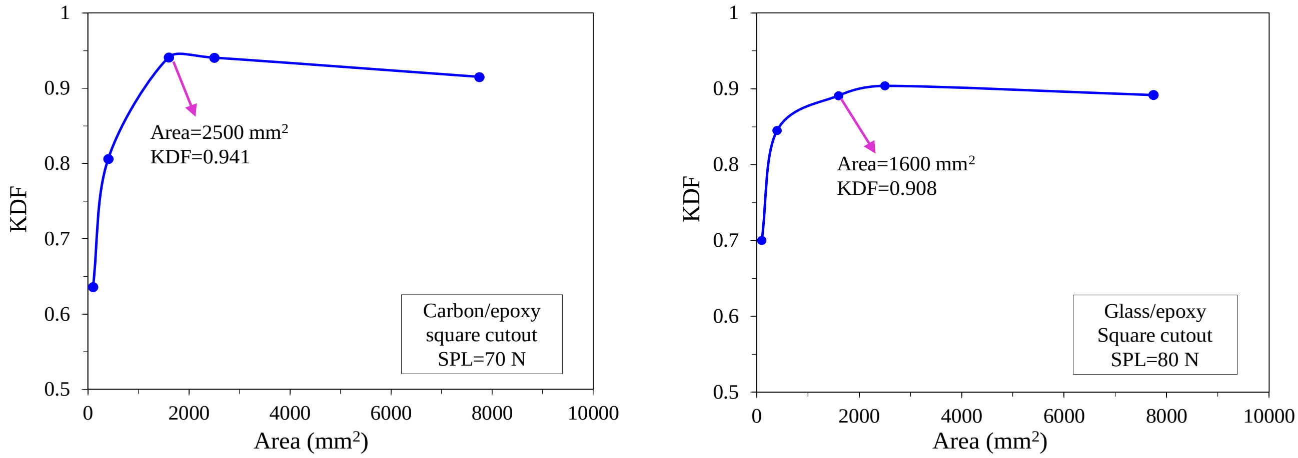


Fig. 12 Effect of cutout area on KDF

which leads to an increase in KDF. Furthermore, those cutouts with an area larger than 1600 mm^2 and 2500 mm^2 do not change KDF. By increasing the area of cutout, the effect of the cutout imperfection is more than the increasing of perturbation load imperfection. Hence, the effect of cutout on the buckling load is greater than the perturbation load. As a result, the KDF value, which is due to the perturbation load imperfection remains approximately constant.

4. Conclusions

In this study, the buckling behavior of carbon/epoxy and glass/epoxy composite cylindrical shells without and with cutouts is studied. The simultaneous effect of cutout and geometric imperfection is investigated on the prediction of critical buckling load. Three different shapes of cutout (i.e., circular, elliptic and square) are considered. Geometric imperfection is modeled using SPLA. Comparison of critical buckling load of composite cylindrical shells without cutout calculated by linear eigenvalue analysis, NASA-SP 8007 guideline and SPLA with experiments show that SPLA is more accurate than others. The cutout in a shell acts as an imperfection and the results show that a CI, regardless of its shape, reduces the critical buckling load. Hence, the effect of the single perturbation load is eliminated from the buckling load. In cylinders with cutout, the most critical condition based on the location of the single perturbation load happens when the single perturbation load is far from the cutout. Since the cutout and the single perturbation load both act as imperfections, the most noticeable effect occurs when these two factors are far from each other. The greatest decrease in buckling load arises in square cutouts due to sharp corners. Moreover, the circular cutout has less resistance against buckling than the elliptic cutout due to the larger area of the circular cutout in the circumferential direction. As the area of cutout increases, the critical buckling load decreases due to increasing imperfection. This trend, however, stops at a certain amount of cutout area and the critical buckling load remains constant.

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