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Numerical study on influence of dent parameters on critical buckling pressure of thin cylindrical shell subjected to uniform lateral pressure

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ABSTRACT

One of the common failure modes of thin cylindrical shells subjected to external pressure is buckling. The critical buckling pressures of these shell structures are mainly affected by the geometrical imperfections present in the cylindrical shell which are very difficult to alleviate during manufacturing process. Dent is one of the common geometrical imperfections present in thin shell structures which may be formed due to mechanical damage caused by accidental loading or impact. In this work, numerical parametric study is carried out to study the influence of dent parameters of centrally located dent (dent length, dent width, dent depth and angle of orientation of the dent), cylindrical shell parameters (L/R ratio and R/t ratio) and yield stress on the critical buckling pressure of externally pressurized thin dented cylindrical shells with simply supported boundary conditions at both top and bottom edges using non-linear static finite-element analysis of ANSYS.

1. Introduction

Thin cylindrical shells have wide applications in marine, nuclear, mechanical, civil and aerospace structures etc., as one of the important structural element because of their efficient load carrying capacity with weight economy. One method of greatly improving the buckling resistance of such long thin cylindrical shell is to stiffen them at their flanges. If the ring stiffeners are not strong enough general instability failure mode can occur in which the ring-shell combination collapse due to the application of external pressure. If the ring stiffeners are relatively strong enough, basic failure mode i.e., the bare shell in between the ring stiffeners will collapse which is also called as shell instability failure mode. Hence the design of thin cylindrical shells under external pressure should be based on buckling criteria. Buckling phenomena occurs when most of the strain energy which is stored as membrane energy can be converted to bending energy requiring large deformation resulting in catastrophic failure.

The thin cylindrical shell structures are prone to a large number of imperfections, owing to their manufacturing difficulties. These imperfections affect the load carrying capacity of these structures. The imperfections present in thin cylindrical shells are classified as geometrical imperfections, material imperfections and other imperfections. The imperfections such as circularity, cylindricity, local indentations, dents, cracks, swellings, non-uniform thicknesses, etc., fall under the category of geometrical imperfections whereas imperfections such as inhomogenity, vacancies, impurities, etc., are classified as material imperfections. The residual stresses and strains induced during manufacturing, etc. are grouped as other imperfections. Out of all these imperfections, the geometrical imperfections are more dominant in determining the load carrying capacity of thin cylindrical shells. Reliable prediction of collapse pressure of these structures is important because the buckling failure is catastrophic in nature.

2. Literature review

As stated by Guggenberger [1] the stability of cylindrical shells under external pressure has been in research continuously from the earlier work of von Mises (1914). The classical elastic buckling theory predicts the bifurcation buckling pressure of perfect thin cylindrical shell without stiffeners under uniform external pressure is given in Timoshenko and Gere [2] and analytical solution based on Donnell stability equation in uncoupled form is given in Brush and Almroth [3]. Most of the earlier experimental works discussed about buckling behavior of cylindrical shell under hydrostatic pressure condition. Only few authors discussed about the buckling behavior of cylindrical shell under lateral pressure (For example, Windernburg and Trilling [4], Seleim and Roorda [5], Park and Kyriakides [6], Lo Frano, Forasassi [7]. Bushnell [8] reported both analytical and numerical efforts and experimental investigations carried out before 1985. It was reported that both analytical results and experimental results are widely deviating. It was also reported that the cause for this deviation is

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due to inevitable differences called imperfections present in the real structure from the perfect structure. Analytical explanations for the same are overviewed in Teng [9] and the studies related to imperfections are overviewed in Schmidt [10]. A large number of theoretical and experimental works related to post-buckling behavior taking into account of initial imperfections are discussed in Yamaki [11]. In most of the earlier works, the imperfections were assumed as distributed geometrical imperfections which are distributed throughout the surface of the structure (For example Goncalves and Batista [12], Prabu et al., [13], Brar [14] and only few were discussed about the effect of local geometrical imperfections such as dent, dimple etc., Only studies related to effect of local geometrical imperfections on buckling behavior of thin cylindrical shell under external pressure are discussed here. Jurcke et al. (1983) (referred to in Guggenberger [1]) studied about effect of single circumferential long dent on post buckling behavior of dented cylindrical shell under external pressure. In related to this work, Guggenberger [1] studied about effect of a deep single longitudinal dent on load carrying behavior of externally pressurized plastic cylindrical shell and the numerical results obtained were compared with experimental results. Park and Kyriakides [6] in their work studied about the reduction of collapse pressure of long cylindrical shells (having L/R ratio approximately equal to 58) with local dents both numerically and experimentally in which dents were formed by spherical indentors on number of stainless tubes of different sizes. The geometrical imperfections details of each dented cylindrical shell were recorded using an imperfection scanning systems and these cylindrical shells were tested under external pressure. The denting and collapse pressure were simulated using non-linear FE analysis and the result obtained were compared with experimental values and they are found to have good agreement with each other. Schneider and Brede [15] numerically investigated about the equivalent geometrical imperfections that are to be applied in numerical analysis to determine equivalent experimental buckling resistances of cylindrical shell (having L/R ratio equal to 0.73) and it was suggested that single longitudinal sinus shaped concave dimple shaped (local) imperfections are better suited than eigen-affine initial imperfections. Rathinam and Prabu [16] discussed about numerical analysis of effect of dent parameters and dent orientation on collapse pressure of a thin cylindrical shell (shell thickness $t=1$ mm and L/R ratio$=2$) with a central dent. Rathinam and Prabu [17] numerically compared the effect of a dent located at half the height and one-third height of thin cylindrical shell on critical buckling pressure and it was concluded that the influence of a dent on critical bucking pressure of dented cylindrical shell with a dent located at half the height is more

Table 2
Maximum amplitude of imperfections vs Buckling Strength Ratio (BSR).

Sl: no	Maximum amplitude of imperfections (mm)	Buckling strength ratio (BSR)
1	0.001	0.99665
2	0.01	0.97901
3	0.1	0.89940
4	1	0.54756

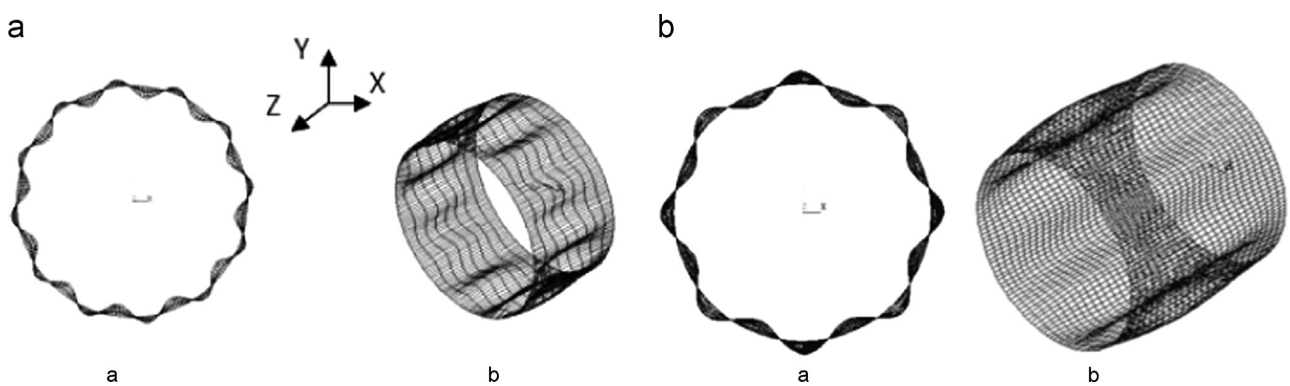

Fig. 1. (a) First eigen buckling mode shape in front view and isometric view of the perfect thin cylindrical shell C given in Combescure and Gusic [19]. (b) First eigen buckling mode shape in frontview and isometric view of model 53 given in Windernburg and Trilling [4].

Table 1a
Comparison with buckling pressure of the cylindrical shell given in Combescure and Gusic [19].

Cylindrical shell	Z	R (mm)	L (mm)	t (mm)	E ($\times 10^5$ N/mm^2)	Poisson's ratio (γ)	FE eigen buckling pressure (MPa)	
							From Reference	From present analysis
A	51	50	20	0.15	2	0.3	0.2936	0.2927
C	500	100	113.8	0.247	2	0.3	0.0624	0.0619
D	5000	100	508.5	0.247	2	0.3	0.0136	0.0135

Table 1b
Comparison with experimental buckling pressure given in Windernburg and Trilling [4].

Model number	Z	R (mm)	L (mm)	t (mm)	E ($\times 10^5$ N/mm^2)	Poisson's ratio (γ)	Critical buckling pressure (MPa)	
							From Reference	From present analysis
53	969	203.2	406.4	0.8	1.93	0.3	0.096 (8)	0.108 (8)
57	994	203.2	406.4	0.78	2.06	0.3	0.103 (8)	0.109 (8)

( ) indicates number of circumferential lobes, Z—Batdorf parameter $z=\sqrt{1-\nu^2}(L/R)^2(R/t)$.