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# Numerical and experimental investigations on buckling of steel cylindrical shells with elliptical cutout subject to axial compression

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ABSTRACT

The effect of cutouts on load-bearing capacity and buckling behavior of cylindrical shells is an essential consideration in their design.

In this paper, simulation and analysis of thin steel cylindrical shells of various lengths and diameters with elliptical cutouts have been studied using the finite element method and the effect of cutout position and the length-to-diameter ($L/D$) and diameter-to-thickness ($D/t$) ratios on the buckling and post-buckling behavior of cylindrical shells has been investigated. For several specimens, buckling test was performed using an INSTRON 8802 servo hydraulic machine and the results of experimental tests were compared to numerical results. A very good correlation was observed between numerical simulation and experimental results. Finally, based on the experimental and numerical results, formulas are presented for finding the buckling load of these structures.



## 1. Introduction

Cylindrical shells are frequently used in the manufacturing of aircrafts, missiles, boilers, pipelines, automobiles, and some submarine structures. These structures may experience axial compression loads in their longevity and yield to buckling. Furthermore, these structures usually have disruptions, such as cutouts, which may have adverse effects on their stability.

The problem of buckling in cylindrical shells has been a preoccupation of investigators for more than a century. At first, researchers focused on the determination of the buckling load in the linear elastic zone, but experimental studies [1,2] showed that the buckling capacity of thin cylindrical shells is much lower than the amount determined in the classic theories [3]. Based on the classic theories, the buckling load of thin cylindrical shells subject to uniform axial compression can be predicted using the formula

$$N_{cr} = \frac{E}{\sqrt{3(1-\nu^2)}}\left(\frac{t^2}{R}\right) \tag{1}$$

where $E$ is the Young modulus, $\nu$ is Poisson's ratio, $t$ is shell thickness, and $R$ is shell radius. It is noteworthy that this formula gives an appropriate result for thin shells without cutouts with $L/R \leqslant 5$ [4]. For shells with moderate thickness ($R/t < 50$), this formula often overestimates the buckling load, so that buckling occurs before reaching the specified load.

Van Dyke [5] determined the stress distribution around a hole in a cylindrical shell subject to axial, torsional, and internal pressure. Tennyson [6] performed an experimental study on the effect of circular cutouts on the buckling capacity of cylindrical shells with radius-to-thickness ratio of 162–331 subject to axial compression. He compared the measured buckling loads with analytical results of Van Dyke.

Brogan and Almorth [7] studied the effect of stiffened rectangular cutouts on the buckling load of cylindrical shells, and also compared the experimental results of shells with cutouts with and without stiffening with the results calculated by the STAGS finite element code. Jenkins [8] performed an experimental study on cylindrical shells in the range $75 \leqslant R/t \leqslant 150$ with two opposite circular cutouts. Additionally, Starnes [9] performed an experimental investigation on the buckling of cylindrical shells with circular cutout subject to axial compression. In this study, the radius-to-thickness ratio of shells was 400–960. Based on the findings from these experiments, he linearized the buckling problem and determined an upper bound for the buckling load using the Reyleigh–Ritz method.

Almorth and Holmes [10] investigated 11 thin-walled aluminum cylindrical shells with rectangular cutouts and various stiffeners which had been installed on seven test specimens. The buckling load of stiffened shells was compared to that of the non-stiffened specimens. Furthermore, the experimental results were compared with the results of the STAGS finite element code. Analysis results showed that the effects of stiffeners are negligible

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for thin cylindrical shells with small cutouts, unless for long shells. Almorth et al. [11] performed a complex nonlinear analysis for cylindrical shells with two opposite circular cutouts subject to axial compression. They showed that the calculated numerical results are comparable to the experimental results of Starnes. Starnes [12] performed another experimental and numerical study on the buckling effect of circular, square, and rectangular cutouts in cylindrical shells subject to axial compression. Toda [13], too, performed an experimental investigation on the cylindrical shells with circular holes subject to axial compression. Furthermore, he placed ring-shaped stiffeners around the cutout and studied the effect of stiffeners on the buckling of cylindrical shells with circular cutouts. The shells in his study were made of polyester with two opposite circular cutouts, and they had a radius-to-thickness ratio of 100 and 400. It was found that if the holes were small enough, they had no effect on the buckling resistance of cylindrical shells. Larger holes, however, caused considerable decrease in the buckling load. Jullien and Limam [2] studied the effect of square, rectangular, and circular cutouts on the buckling of cylindrical shells subject to axial compression, and developed a parametrical formula for the shape and dimensions of the cutouts. The influences of the position and number of cutouts were also studied. The software program used for the finite element method was CASTEM2000. At the same time, Yeh et al. [14] analytically and experimentally studied the bending and buckling of moderately thick-walled cylindrical shells with cutouts. The dimensions of their shells were diameter-to-thickness, $D/t = 50$ and length-to-diameter, $L/D = 7.9$. It was found that the limiting buckling moment would be higher if the cutout was on the tension side rather than on the compression side. They also performed parametric studies on the influences of shape, size, and location of a cutout on the buckling capacity. Hilburger et al. [15] analyzed the buckling behavior of thin composite cylindrical panels with central circular cutout. In this study, the effect of cutout dimensions, panel curvature, and initial geometric imperfections was investigated, and the numerical results were compared with experimental findings. The STAGS finite element code was used for numerical analysis in this study. It was found that the results of nonlinear analyses are much more accurate than the traditional linear analyses. Tafreshi [16] also numerically studied the buckling and post-buckling response of composite cylindrical shells subjected to internal pressure and axial compression loads using ABAQUS. She studied the influences of size and orientation of cutouts and found that an increase of internal pressure resulted in an increase in buckling capacity. Haipeng Han et al. [17] studied the effect of dimension and position of square-shaped cutouts in thin and moderately thick-walled cylindrical shells of various lengths by nonlinear numerical methods using the ANSYS software. They also compared their results with experimental studies on moderately thick-walled shells. Finally, they developed several parametric relationships based on the analytical and experimental results using the least squares regression method.

In this paper, linear and nonlinear analyses using the ABAQUS finite element software, were carried out in order to study the effect of the position of elliptical cutouts with identical dimensions on the buckling and post-buckling behavior of cylindrical shells. The shells with different diameters and lengths as follows, studied were: $(L/D_1) = 2.857,\ 6.5,\ 10$; $(D_1/t) = 53.846$; and $(L/D_2) = 2.495,\ 5.676,\ 8.732$; $(D_2/t) = 61.667$. Additionally, several buckling tests were performed using an INSTRON 8802 servo hydraulic machine, and the results were compared with the results of the finite element method. A very good correlation between experiments and numerical simulations was observed. Finally, based on the experimental and numerical results, formulas are presented for the computation of the buckling load in such structures.

## 2. Numerical analysis using the finite element method

The numerical simulations were carried out using the general finite element program ABAQUS 6.4-PR11.

### 2.1. Geometry and mechanical properties of the shells

For this study, thin-walled cylindrical shells with three different lengths ($L = 120,\ 273,\ 420$ mm), and two different diameters ($D = 42$, 48.1 mm) were analyzed. An elliptical geometry was selected for cutouts that were created in the specimens. Furthermore, the thickness of shells was $t = 0.78$ mm. Fig. 1 shows the geometry of the elliptical cutouts. According to this figure, parameter ($a$) shows the size of the cutout along the longitudinal axis of the cylinder, and parameter ($b$) shows the size of the cutout in circumferential direction of the cylinder. The distance between the center of the cutout and the lower edge of the shell is designated by $L_0$, as shown in Fig. 1.

Specimens were nominated as follows: *D*42-*L*120-$L_0$60-*a*-*b*. The numbers following *D* and *L* show the diameter and length of the specimen, respectively.

The cylindrical shells used for this study were made of mild steel alloy. The mechanical properties of this steel alloy were determined according to ASTM E8 standard [18], using the INSTRON 8802 servo hydraulic machine.

The stress–strain curves, stress–plastic strain curve and respective values are shown in Fig. 2. Based on the linear portion of stress–strain curve, the value of elasticity module was computed as $E = 187.737$ GPa and the value of yield stress was obtained as $\sigma_y = 212$ MPa. Furthermore, the value of Poisson ratio was assumed to be $\nu = 0.33$.

### 2.2. Boundary conditions

For applying boundary conditions on the edges of the cylindrical shells, two rigid plates were used that were attached to the ends of the cylinder.

In order to analyze the buckling subject to axial load similar to what was done in the experiments; a 10-mm displacement was applied centrally to the center of the upper plate, which resulted in a distributed, compressive load on both edges of the cylinder. Additionally, all degrees of freedom in the lower plate and all degrees of freedom in the upper plate, except in the direction of longitudinal axis, were constrained.

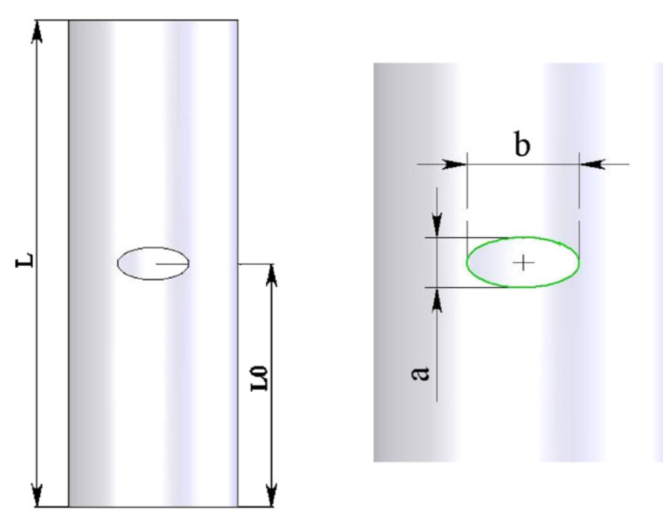


**Fig. 1.** Geometry of cutout.