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Recovery of capacity lost due to openings in cylindrical shells under compression

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ABSTRACT

Cylindrical shell structures such as silos and wind turbine towers often feature openings for internal access, and these may have a significant detrimental effect on the buckling load. This paper numerically investigates the buckling of cylindrical shells with cutouts near the cylinder end using nonlinear finite element analysis. The finite element (FE) model was validated against published experiments on the axial compression of moderately thin-walled cylinders with cutouts of various sizes and shapes. The results indicate excellent qualitative and quantitative agreements for loads, deflections, and buckled mode shapes. Using the validated FE model, it was found that the opening width is the most critical geometric parameter in terms of loss of carrying capacity. The model was then extended to explore a variety of stiffener configurations aimed at recovering the capacity lost due to the opening. These configurations included a frame ring, as well as the simpler option of straight ribs on either side of and above the hole. Thickness and protrusion were also considered in the analysis. It was found that the frame ring achieved full recovery of the lost capacity, while the straight stiffeners were effective where only up to 67% recovery was required. The results also indicated that the optimum placement of stiffeners was immediately adjacent to the cutout. This study demonstrates that the optimum stiffener is a fully enclosed ring immediately adjacent to the cutout edge, and that full load recovery can be achieved with an appropriate design.

1. Introduction

Cylindrical shells are important structural elements in silos, wind turbines, pressure vessels, and offshore structures. These commonly have geometric discontinuities that serve as doors, windows, or access ports. These shell structures experience compressive loads during their service life. The presence of cutouts in the shell structures under compressive loads influences stability and can significantly reduce the buckling load. Given the arbitrariness of cutout geometry and the ensuing complexity of the local membrane stress distribution around the cutout, there is limited extant research predicting the effects of a cutout on the buckling load of a cylindrical shell using analytical techniques.

The influence of unreinforced cutouts on the buckling behaviour of cylindrical shells was first investigated in the late 1940s by Lur'e [1]. Lekkerkerker [2], van Dyke [3] and Savin [4] extended Lur'e's analysis to present a parameter α that regulated the solution of the pre-buckling stress distribution and displacement of a cylindrical shell with a cutout. Toda [5] found that the buckling of these shells was nonlinear in nature due to the presence of pre-buckling bending deformation near a cutout region. Starnes [6] conducted experiments on the buckling behaviour of cylindrical shells with circular cutouts and suggested that there was a relative decrease in the buckling load of the shell as the size of the hole increased.

The earliest numerical results with respect to cylindrical shells with cutouts were calculated by Almroth and Holmes [7]; the cutouts considered were rectangular in shape. Based on the aforementioned results in combination with numerical and experimental studies on rectangular and circular cutouts, Eggwertz and Samuelson [8] proposed a simple design procedure for cylindrical shells under axial compression with circular and rectangular cutouts. Jullien and Limam [9] indicated that the buckling load was independent of the geometrical imperfections within the shell to a significant extent. However, the link between the initial geometrical imperfections and the openings has to be considered for small openings.

Han et al. [10] investigated experimentally and numerically the influence of cutouts on thin and moderately thick cylindrical shells (diameter/thickness $D/t = 45$ and 450) with various shell lengths subject to axial compression, stating that "The presence of a cutout in the moderately thick-walled shell induced global bending, leading to the collapse of the shells when the cutout was located near the mid-height of the shell, whereas when the cutout was located near the loaded end, local buckling occurred."

Shariati and Rokhi [11] also experimentally and numerically

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investigated buckling of cylindrical shells subjected to axial compression ($D/t = 54$ and 62), observing that an increment in the cutout opening height decreased the buckling load only minimally, but the buckling load in a cylindrical shell with a fixed cutout area increased with increases in the opening height-to-width ratio.

There have been many recent studies on buckling of cylinders with cutouts due to bending, include those of Poonaya et al. [12], Fam et al. [13] and Guo et al. [14]. Jay et al. [15] include a summary of sixteen flexural testing experiments involving D/t ratios of 21–800. Lee et al. [16] completed a comprehensive nonlinear finite element study of the effects of opening geometries typical of wind turbine towers, proposing empirical formulae for strength reduction in axial compression and pure bending. They found that the cutout location is much more significant than its shape.

As it is often necessary for tubes to contain openings, it is of interest to investigate methods of reinforcement to recover the capacity lost due to the cutout. Several previous studies have focused on this issue. Bennett et al. [17] performed experimental and analytical research on the use of the 'area replacement method' (ARM) to support the surroundings of a circular cutout within a cylindrical shell and its relation to the buckling strength of the shell under an axial compressive loading condition. Hilburger and Starnes Jr.[18] investigated common buckling and failure response behaviour and trends for a thin-walled circular cylindrical shell with a square cutout and several cutout reinforcement configurations. The results suggested that adjustments in the orthotropy, thickness, and dimensions of the cutout reinforcement in a compression-loaded shell could lead to an extensive increase in the buckling load of the shell and could reduce the local deformations, strains, and damage near the cutout as well.

More recently, Dimopoulos and Gantes [19] studied the buckling behaviour in bending of cantilever shells with stiffened openings that mirror the principal geometric characteristics of wind turbine towers. The results indicated that the existence of the opening caused a strength reduction of approximately 24%. The study also suggested that the selected stiffening proposal was adequate for recovering the strength loss. Dimopoulos and Gantes [20] compared these results with prediction methods proposed in the Eurocode EN1993-1.6 standard, concluding that GMNIA (Geometric and Material Nonlinear Analysis with Imperfections) is the most reliable analysis type. However the effect of imperfections is known to become less significant as the slenderness decreases [9,19].

Much of the above recent work has focused on bending of cylinders with cutouts but there has been relatively little work on axial loading, or on reinforcement of cutouts. One exception is the work of Ghanbari Ghazijahani et al. [21], who experimentally investigated the buckling behaviour of shells with different cutouts under compression and also examined stiffening by a rib each side of the opening. The findings revealed that the stiffener recovered nearly 33% of the lost capacity. The present paper presents a complementary numerical study of this work using nonlinear finite element analysis (FEA), significantly extending the investigation of stiffening options.

The main objectives of this paper are to (i) validate ANSYS nonlinear analyses against the experimental results of Ghanbari Ghazijahani et al. [21], (ii) investigate the sensitivity of the buckling load to several design parameters defining the cutout geometry, and (iii) examine a variety of stiffener configurations around the cutout to achieve as close as practically possible to full recovery of the lost capacity.

2. Finite element models and analysis methods

2.1. Finite element model

This study focuses on cylindrical shells with different cutouts as were investigated in the experimental study of Ghanbari Ghazijahani et al. [21]. Linear and nonlinear finite element models were developed in ANSYS, with both geometric and material nonlinearities considered.

The dimensions of the shells were the same as those studied by Ghanbari Ghazijahani et al. [21], with fixed cylinder dimensions as summarised in Table 1. Sixteen variants were modelled in the validation part: an intact cylinder, and the same cylinder with cutouts of five different shapes, each in three sizes. The cutout shapes are defined in Fig. 1, while the dimensions (height a and width b) are provided in Table 2. The shapes can be classified into three 'full' shapes, which had an aspect ratio (width/height) of 0.4, and two 'semi' shapes, which were half the height but the same width as their full counterparts. The distance between the bottom of the openings and the bottom end of the shell, denoted L_0, was 50 mm.

Table 1
Key geometric parameters of the baseline cylinder.

Length	Diameter	Thickness	End offset		
L (mm)	D (mm)	t (mm)	L_0 (mm)	D/t	L/t
400	76.2	1.6	50	47.6	250

All geometrical models were created in AUTODESK INVENTOR before importing into ANSYS as solids. Surface bodies were created from the imported solid body and a thickness of 1.6 mm was assigned to each surface body.

After determining suitable mesh characteristics (Section 2.2.4) the ANSYS model was validated against the experiments of Ghanbari Ghazijahani et al. [21] (Section 3). The validated ANSYS model was then used to investigate variations in the cutout geometry (Section 4). Then three types of stiffener were added and the effects of the stiffener dimensions were systematically investigated (Section 5).

Material properties, particularly the stress-strain curve, are critical for nonlinear analysis. The uniaxial stress-strain properties shown in Table 3 and Fig. 2 are those measured by Ghanbari Ghazijahani et al. [21], and represent the mild steel properties of the cylinders. These data were input as a multi-linear material model, however the linear analysis only recognised the initial modulus, which was 200 GPa.

2.2. Analytical process

2.2.1. Boundary conditions

In the experiments of Ghanbari Ghazijahani et al. [21] the specimens were compressed between two flat platens in a universal testing machine (see Fig. 6 (a) below). Vertical displacement was restrained at the bottom and controlled at the top, while lateral displacements were prevented at both ends by friction. It would be natural to assume that the ends were effectively simply supported, but close inspection of the inset figure in Fig. 6 (a) shows that the ends behaved as clamped, i.e. wall rotation was prevented. This comes about because the walls had finite thickness and were machined flat. Rotation therefore did not occur until the very advanced stages of buckling when the eccentricity of the local shell normal force exceeded half the wall thickness due to the increasing wall bending moment (or possibly later if this occurred only locally). In fact both simply supported and clamped boundary conditions were tried and, although the peak load differences were generally small, the clamped boundary condition gave significantly better agreement with experimental load-displacement curves than the simply supported boundary condition, particularly in the post-buckling behaviour. This will be discussed further in Section 3.

It could be of interest to perform experiments in future with simply supported ends (by finishing the edges with a rounded or pointed profile) or with elastic supports (as for example would be provided by a silo or tank end-cap that, due to its bending flexibility, only partially restrains the cylinder end rotations). However, the numerically computed peak loads differed by a maximum of 1.2% for the small cutout, 1.8% for the medium cutout, 4.8% for the large cutout (presumably higher because of its proximity to the ends) and 3.1% for the intact specimen. In a silo or tank the rotation of the ends of the cylinder would be expected to be bounded between those of the clamped case for a very rigid end and the simply supported case for a very flexible end. There is a wide variety of possible end-cap configurations, but given that the extreme cases differ by mostly much less than 5% in buckling load, the present results give good guidance to the