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Empirical equations to estimate non-linear collapse of medium-length cylindrical shells with circular cutouts

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ABSTRACT

This paper examines the load bearing capacity of medium-length steel cylindrical shells with a circular cutout under the action of axial compression. Numerical simulations were performed for a radius-to-thickness ratio (R/t) ranging from 100 to 500, and an imperfection parameter (α) of between 1 and 4. Structural steel and the behavior of perfect plastic material were considered within the scope of the study. The investigation includes the influence of the cutout size and number, radius-to-thickness ratio. The paper also contains a comparison between theoretical predictions, ABAQUS® FE results and experimental data for axially compressed cylinders. A reasonable correlation was obtained between numerical predictions and experimental tests. Details relating to the experimental procedure and FE model are provided. A parametric study was conducted to propose empirical equations, estimating the limit load of cylindrical shells as a function of geometry (R/t, α and cutout number) and material properties (E, ν and σ_y). Empirical formulas were produced, fitting a surface plot using the Least Square method, for both the perfect shell structure (reference shell load) and the limit load reduction factor (ϕ). All variables in the empirical formula were normalized. A stochastic error analysis was used as a tool to measure how the results of proposed equations approach the FE predictions. Finally, based on experimental and numerical results, formulas are presented to calculate the limit load of medium-length shells with and without circular cutouts having a discrepancy not exceeding ± 10% regarding error analysis.

1. Introduction

Thin-walled cylindrical shells are often used in wide range of engineering structures such as aircraft, storage tanks, pipelines, towers and pressure vessels. Due to their ease of manufacture and their endurance to axial compressive loading, thin-walled cylindrical shells are one of the most preferred structures in the industry. However, the stability of these structures may be seriously affected by discontinuities such as cutouts, stiffeners and geometric imperfections [1,2]. Structures often suffer from considerable stress concentrations caused by the above mentioned discontinuities. For this reason, understanding the influence of an imperfection on the buckling load of shell structures plays an important role prior to the manufacturing stage. Therefore, the buckling strength of thin-walled cylinders with a cutout under axial compressive load has been investigated in the literature a number of times. Initially, researchers focused on determination of the buckling load in the linear elastic region, but experimental studies [3–5] indicate that the load carrying capacity of cylindrical shells is considerably lower than in classic theories [6]. Tennyson [7] investigated the effects of circular cutouts on the buckling strength of cylindrical shells under axial compression. It was reported that large reductions in critical buckling loads result from the presence of relatively small holes in the structure. Similarly, Almroth and Brogan [8] present a numerical solution to the problem of the stability of elastic cylindrical shells with rectangular cutouts. They mainly focus on how to eliminate or decrease the negative effect of cutouts, introducing reinforcements around the cutouts. A more detailed work, to compare the cases for reinforced and unreinforced rectangular cutouts, was presented by Almroth and Holmes [9]. This study points out that for small to moderate size cutouts, reinforcement has no benefit unless the cylinder has a high geometrical quality. Starnes [10] also investigated the effect of a circular hole on the buckling of thin cylindrical shells under axial compression both analytically and numerically. Starnes shows that the buckling of a cylinder with a cutout related to the ratio of a^2/Rt rather than a/R which had formerly been suggested by Tennyson [7] where, a is the cutout radius, R is the shell radius and t is the shell thickness. The shell structures experience high stress concentrations through the presence of cutouts. To analyze the stress state around the cutouts of cylindrical shells, Pierce and Chou [11] performed experimental tests under tension and compression and verify with simplified analytical expressions

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[12]. It has been found that stress concentration factors around the cutouts may be much higher than the theoretical values since the shells are sensitive to imperfections and the presence of a local buckling near the cutouts.

The results of previous studies in the literature have led to the conclusion that the buckling loads of cylindrical shells with cutouts under axial compression are governed by the parameter $\alpha = a/\sqrt{Rt}$, where a is the characteristic cutout dimension, R is the shell radius, and t is the shell thickness. In this manner, Toda [13] presents an experimental study for shells with both reinforced and unreinforced cutouts at the middle of the axial length of the shell. He performed various tests on shells with R/t of 100 and 400, and α values ranging from 0 to 20. The experimental results show that increasing the cutout diameter has a significant effect on the buckling load, while relatively small holes have almost no significant effect. The influence of the location of the cutout attracted the attention of several researchers. For example, a study conducted by Jullien and Limam [14] examines the shapes, sizes, location and number of cutouts for steel shells subjected to uniaxial loading. They claim that the opening parameter $a/\sqrt{Rt}$ characterizes the effect of a cutout on the critical load of a cylindrical shell. The axial length and position of the cutout has little influence, which is supported by Ghazijahani et. al. [15]. However, analytical approaches are always not sufficient to predict the buckling behavior of the shell with a cutout. Therefore, it requires more complicated numerical simulations which account for nonlinearities and discontinuities caused by the presence of a cutout. A recent work by Han et. al. [16], evaluates the influence of square cutouts on the buckling responses of unstiffened aluminum cylindrical shells. Various cutout sizes, cutout locations and shell aspect ratios were investigated. The researchers observed a good match between numerical and experimental results. Moreover, several empirical equations were developed based on the results using the least square regression method. However, proposed empirical equations involve many coefficients because of the Lagrange polynomial and it is not applicable practically for a wide range of shell configurations. Using the same approach, Shariati and Mahdizadeh [17] also develop a series of empirical equations to calculate reduction in buckling load based on normalized shell parameters. To use these expressions, the buckling load for cylindrical shells without a cutout must be known. Furthermore, they report that changing the position of the cutout from the mid-height of the shell toward the edges increases the buckling load, and longer shells are more sensitive to a change in cutout position. Shariati and Mahdizadeh extend their previous study one step further by involving a new parameter (cutout angle) in the calculation of the buckling reduction factor [18]. Consequently, to examine the influence of the cutouts on the steel and composite structures, two current studies has been proposed by Dimopoulos et al. and Wang et al. which accounts material and geometry non-linearities [19,20].

Studies conducted to date emphasize the importance of understanding the buckling response of cylindrical shells comprising defects and cutouts. Until now, few researchers have focused on developing analytical expressions to predict the buckling load of imperfect shells, which can only be applied in a narrow shell parameter range. Additionally, many do not account for nonlinear material behavior and geometric stiffness. In this case, a proper FE model including nonlinear effects may be a suitable alternative in an evaluation of the buckling load of shells with a cutout, which is not required to realize extensive and numerous experimental tests.

This study aims to develop empirical equations for a wide range of medium length cylindrical shell configurations with circular cutouts, which enable the use of material properties as an input. The most significant and common type of loading and boundary conditions are used as suggested in [21]. In numerical analysis (ABAQUS®), a suitable meshing scheme, element formulation and perfect plastic material response, which give conservative results in comparison with a multi-linear material model, are considered [21,19]. A parametric study is conducted; where R/t is between 100 and 500, α ranges from 1 to 4 [13], σ_y varies from 200 to 600 MPa, regarding practical applications of medium-length cylindrical shell structures. α denotes the imperfection parameter. Accordingly, the circular cutout number (n) is limited to 4 and situated at the mid-length of the shell. Creating empirical equations will provide the possibility to predict the buckling load of such cylindrical shells with/-out circular cutout in a larger scale, which would fill a gap in the literature.

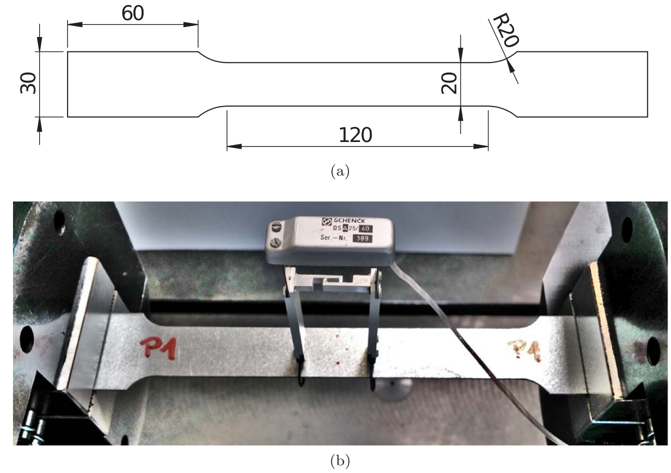

Fig. 1. (a)Dimensions of the dog bone test specimen [22], (b) Experimental setup for tensile test of S235JR steel with mechanical extensometer.

2. Experimental procedure

2.1. Mechanical properties of the shells

The cylindrical shells for experimental study are typically selected to be structural mild steel designated as S235JR due to their enormous application in thin walled shell applications. These kinds of structural steels exhibit ease of manufacture and application through their strength and stability. In order to determine mechanical properties, tensile test specimens machined from the walls of hot rolled pipe, were prepared according to the EN ISO 6892-1 [22] standard. The geometries of the tensile test specimen and experimental setup are presented in Fig. 1a and 1b, respectively. Tensile tests were performed in a fully computer controllable testing machine, an INSTRON®, with three repetitions at a crosshead speed of 5 mm/min. The samples were placed in jaws of the test machine, with a SCHENCK brand mechanical extensometer at a range of 25 mm from the midpoint as shown in Fig. 1b. Based on the experimental results, the engineering and true stress strain diagrams of the S235JR test samples are given in Fig. 2, considering the average values of three repetitions. The true stress strain diagram was constructed up to an instability point (Necking) using the equations given below [23].

$$\sigma_t = (1 + \varepsilon)\sigma \tag{1}$$

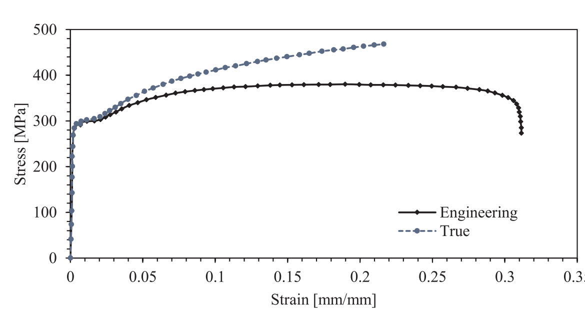

Fig. 2. Engineering and true stress-strain diagrams of S235JR steel.