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Buckling behavior of cylindrical steel tanks with concavity of vertical weld line imperfection

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ABSTRACT

Shell structures are built using a number of welded curved panel parts. Hence, some geometrical imperfections emerge. These imperfections have a direct impact on structural behavior of shells during the external compressive loading. In this research, a field study was accomplished on the implementation of the storage tanks in a refinery site and then, the resulted imperfections were identified and categorized. The survey of imperfections revealed that the imperfection in form of concavity of vertical weld line is the most prevalent type of imperfection seen in the steel tanks. This imperfection experimentally modeled and the buckling behavior of these tanks was evaluated under uniform external pressure. Comparing obtained results of estimation, ASME code and experimental research represented a considerable difference in the amount of buckling load. Results show that the imperfections due to concavity of vertical weld line are very important in buckling of the tanks under uniform external pressure. This imperfection decreases initial, full and post buckling capacity of the tanks under uniform external pressure, significantly. Findings of this research show that for design of steel tanks under uniform external pressure load, 65% of the buckling load obtained from the ASME Code should be used.

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1. Introduction

Shell structures have curved initial shapes with the thickness less than other dimensions. In some states, radius-to-thickness ratio reaches 3000. The force field generated in these structures includes membrane and bending forces, which can change depending on shell thickness. Steel tanks designed in cylindrical form are commonly used shell structures in industrial facilities. In geometrical terms, these tanks have a very small thickness compared to their other dimensions and are categorized as thin-walled structures for buckling failure caused by the influence of uniform external pressure loading on the tank wall [1].

The buckling of tanks under uniform external pressure is usually caused by operational problems during the discharge of the liquid contents in such a way that partial vacuum is produced. These events are usually classified as accidents, and occur in individual tanks in a tank farm, rather than affecting many tanks in the same event, such as those produced by a natural disaster.

The objective of this paper is identification and categorization of created imperfections in implementation of the storage tanks via a field study. By identifying the prevalent imperfection, buckling behavior and failure of these tanks is investigated under uniform external pressure loading.

2. Stability in shell structures

Evaluating the stability problems in thin-walled shells is important for two reasons. First, the ratio of thickness to other dimensions is very low in these structures, which highlights the instability problem. Second, shell structures are exposed to compressive stresses and forces; however, shell buckling is caused by low thickness. One of the main properties of shells is having much higher membrane stiffness than bending stiffness. Accordingly, a shell can absorb a large amount of membrane energy without experiencing a great deformation, while large deformations and rotation in the cross-section need to absorb this energy through bending deformations [1].

2.1. Buckling of shell structures

Buckling is considered as a nonlinear phenomenon, in which the structure cannot take further load with the same geometry and changes its shape in order to find alternative equilibrium configurations. Shell buckling occurs as the structure response to the membrane forces. Membrane forces act along the component axis and tangent to the middle surface of the shell and the buckling occurs when the structure

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converts membrane strain energy into bending strain energy without any change in the applied external load [1].

There have been a number of studies addressing buckling behavior and instability, see e.g. [2]. and references therein. Below we mention several contributions relying on experimental approach.

Influence of primary boundary condition on the buckling of shallow cylindrical shells was studied by Showkati and Ansourian experimentally [3]. Wang and Koizumi investigated the buckling of cylindrical shells with longitudinal joints under external pressure [4]. Buckling of cylindrical shells with stepwise variable wall thickness under uniform external pressure was considered by Chen et al. [5]. Aghajari et al. conducted an experimental study on the buckling of thin cylindrical shells with two-stepwise variable thickness under external pressure [6]. Experimental and numerical investigation of composite conical shells stability subjected to dynamic loading was investigated by Jalili et al. [7]. Ghazijahani et al. studied longitudinally stiffened corrugated cylindrical shells under uniform external pressure [8].

2.2. Geometric imperfections on shells buckling

Geometric imperfections include all the deviations in the form of a structural component compared with its ideal geometric composition. In construction of shell structures, due to the large dimensions, curved plates or panels can be used. The seam between various surfaces of the main source of deviation is in the real form, these deviations or imperfections can be generated as a result of welding or appropriate incompatibility of the plates with larger dimensions than other plates. In contrast to various structures, the buckling strength of shells with no imperfections is significantly different from the buckling strength of the same shell with imperfections. In some states, geometric imperfections may strengthen the structure and increase its capacity. This feature puts shells among the structures which are called sensitive to imperfection. In Fig. 1, the buckling behavior of columns, flat plates and cylindrical shells are schematically shown.

In these curves, black lines show the system with no geometric imperfections or perfect system, while red lines represent the corresponding behavior of the imperfect system. As can be seen, the flat plates and column elements are not sensitive to imperfection, while cylinders which are the examples of thin-walled structures are very sensitive to imperfections [9].

There have been a number of studies addressing the influence of imperfections on the buckling behavior of shell structures. Below we mention several contributions relying on experimental approach.

Calladine studied the causes of imperfection-sensitivity in the buckling of thin shells [10]. Teng et al. investigated the geometric imperfections in full-scale welded steel silos [11]. In other work, buckling behavior of large steel cylinders with patterned welds was considered

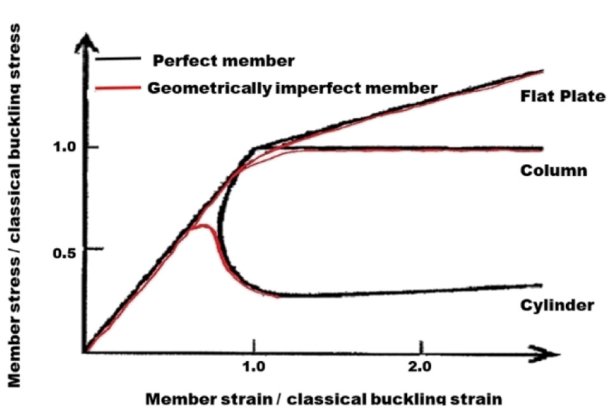

Fig. 1. Load-axial displacement graph of columns, flat plates and cylindrical shells in perfect and imperfect states [9].

by Hubner et al. [12]. Influence of imperfection on the buckling of thin cylindrical shell under uniform external pressure was studied experimentally by Lo Frano and Forasassi [13]. Maali et al. investigated the Buckling behavior of conical shells under weld-induced imperfections experimentally [14]. Yang et al. studied the buckling of cylindrical shells with general axisymmetric thickness imperfections under external pressure [15]. Fatemi et al. conducted experiments on imperfect cylindrical shells under uniform external pressure and observed that geometric imperfections have a more considerable impact on the behavior of shells [16].

In other work, inelastic stability of liners of cylindrical conduits with local imperfection under external pressure was studied by Khaled El-Sawy [17]. Ghazijahani et al. conducted experiments on dented cylindrical shells under peripheral pressure [18]. Thompson presented advances in shell buckling. He studied on the buckling of axially compressed cylindrical shells with arbitrary thickness imperfections theoretically and experimentally [19]. Cao et al. studied buckling of cylindrical shells with arbitrary thickness imperfections under axial compression analytically [20]. Lee et al. investigated the geometric role of precisely engineered imperfections on the critical buckling load of spherical elastic shells. Their investigation combined precision experiments, finite element modeling and numerical solutions of a reduced shell theory, all of which were found to be in excellent quantitative agreement [21].

Evkin et al. investigated the buckling of a spherical shell under external pressure and inward concentrated load [22]. In other work, Hutchinson and Thompson studied nonlinear buckling behavior of spherical shells subject to external pressure. They found that the nonlinear axisymmetric post-buckling behavior of perfect thin spherical shells and their asymmetric bifurcations are characterized providing results for a structure/loading combination with an exceptionally nonlinear buckling response [23].

Due to the mentioned studies about imperfections in shell structures, field study and vertical weld line imperfection has not been studied. In this research this imperfection has been introduced and buckling behavior of cylindrical steel tanks has been investigated under uniform external pressure.

3. Analytical equations for buckling of cylindrical shells

Donnell in 1933 obtained an equation for the buckling of cylindrical shells as follow [24]:

$$D\nabla^8 w + \frac{Et}{R^2}\frac{\partial^4 w}{\partial x^4} + \nabla^4\left(N_x\frac{\partial^2 w}{\partial x^2} + 2N_{xy}\frac{\partial^2 w}{\partial x\,\partial\theta} + N_\theta\frac{\partial^2 w}{\partial\theta^2}\right) = 0 \quad (1)$$

This equation can be used for various loading conditions. Let us take the simple case of applied lateral pressure, where $N_x = N_{xy} = 0$ and $N_\theta = Pr - \sigma_{cr}t$. Eq. (1) then becomes [24]:

$$D\nabla^8 w + \frac{Et}{R^2}\frac{\partial^4 w}{\partial x^4} + \nabla^4\left(N_\theta\frac{\partial^2 w}{\partial\theta^2}\right) = 0 \quad (2)$$

An expression for the deflection that satisfies the simply supported boundary condition of a cylinder can be expressed as:

$$w = w_{mn}\ \sin\frac{m\pi x}{L}\ \ \sin\frac{n\pi y}{\pi R} \quad (3)$$

Substituting this expression into Eq. (2) gives the nontrivial solution $\sigma_{cr} = \frac{\pi^2 KE}{12(1-\upsilon^2)}(t/L)^2$ where $K = \frac{(m^2+\beta^2)^2}{\beta^2} + \frac{12Z^2}{\pi^4\beta^2(1+\beta^2/m^2)^2}$, $\beta = \frac{nL}{\pi R}$, $Z = \frac{L^2}{Rt}\sqrt{1-\upsilon^2}$

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