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Comparative studies on buckling behaviors of T joint and pipe by varying geometric parameters and analysis methods

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ABSTRACT

In this study, buckling behaviors of T joint and pipe are comparatively investigated by varying geometric parameters and analysis methods. The effects of the wall thickness and ellipticity on buckling behavior are taken into account. According to the lowest potential energy principle, the equations of critical pressure and buckling wave number are established on the assumption of elastic buckling in the paper. However, in practice, if the structures deform largely, the T joint and pipe always experience elastic–plastic buckling, so the geometric and material nonlinearities are considered in buckling calculation. In achieving it, the finite element method (FEM) is adopted to explore the effects of those nonlinearities on buckling. Moreover, the effects of initial defect on the critical pressure are elucidated on the object of the T joint and pipe. Through rigorous FE numerical analysis, the buckling behaviors of the T joint and pipe are discussed in terms of deformation pattern, stress distribution, and critical pressure. Some interesting and useful conclusions are summarized in the paper.

1. Introduction

T joint and pipe are two typical structural elements in shell components, which are widely used in a great variety of applications from aerospace to marine industry. One of the most important issues attracting many researchers is their buckling behaviors because numerous factors can lead to different buckling behaviors, for instance, geometric parameter, material property, and combined loading. Accordingly, several approaches have been considered to tackle the buckling problems, among them are analytical approaches, numerical approaches, experimental approaches, and even some combination methods. As well known, instability problems with elastic feature under compressive loading have been extensively investigated theoretically and experimentally. A lot of analytical solutions on elastic structure can be found in many monographs [1,2]. However, for complex structure and nonlinear deformation in shell components, it is always impossible to develop analytical solutions for the plastic collapse strength, let alone buckling behaviors. FE analysis is therefore highly desirable and may be the only choice for someone lack of experimental budget.

Since an accurate evaluation on buckling of shell components is so crucial for their safety design, optimal and economic reasons, considerable studies have been carried out to quantitatively analyze the buckling behaviors by FEM. To author's knowledge, the code [3] issued by European Convention for Construction Steelwork (ECCS) is the first to be strongly oriented toward FE analysis on cylindrical shell buckling by incorporating geometric nonlinearity. The En1993-1-6 standard [4,5] offers new FE paradigm for thin shell stability design. In some standards [6], the buckling behaviors of imperfect elastic–plastic shell structure were evaluated by FEM, the linear bifurcation load as well as the plastic collapse load was determined [7,8]. In order to disclose post-buckling behavior, extensive plasticity was involved in large local strains, it was found that sometimes accurate determination of plastic collapse strength was extremely difficult [9]. In dealing with it, based on modification of some existing models, the plastic buckling was accurately evaluated, and remarkably close agreement was achieved between FE simulation and experimental test [10,11]. As the indication of the European standards for shell buckling, some topics have been the subject of the extended research for many years, but others are still open to new research [12]. Subsequently, the axially compressed cylindrical shell was chosen as an object illustrating the new research which needs to be conducted [13,14]. In order to validate their FE simulation, experimental methods were also adopted to study the buckling and post-buckling behavior of pipes that can be constrained in straight horizontal or curved way [15]. Regarding FE simulation, different stages of the buckling behavior and their relation to

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loadings, geometric parameters and boundary conditions had also been investigated, the results had shown that the buckling load is a strong function of geometric parameters and boundary condition [16], moreover, the influence of internal pressure on buckling behavior of pipes was insignificant. Through the review work on recent advances and trends in the area of thin shell buckling [17], one was able to find that the topics on initial defect influence was given emphasis as well as the use of FE buckling analysis in the stability design. In order to reduce the imperfection effect, some reinforcement measures were taken to reduce stress concentration around cutout or geometric discontinuity [18]. Through rigorous investigation on static strength of galvanized K-joint with different hole geometries and welding, different joint failures were exposed using nonlinear buckling analyses [19]. FEM and analytical method based on the elastic–plastic buckling theory were simultaneously employed to perform parametric studies on various buckling [20,21]. Consequently, the best reinforcement strategy was found, and the structural design recommendations were explained in detail to avoid metallic structural collapse [22]. Besides, buckling load optimization was maximized by using objective function [23], and the accuracy of results was improved by adding some terms to the equations of the critical pressure [24]. As for welded joint structures, it should be noted that the joint residual stresses from welding had a significant impact on their static strength [19]. Due to the coupling effect between geometric nonlinearity and material nonlinearity, the buckling behaviors were different from any results drawn from linear calculation or single buckling mechanism [25]. Some researchers [26] pointed out that the structure collapse was initiated by yielding, but interaction with geometrical deformation was meaningful because most nonlinear buckling reduced the load bearing capacity even further.

The objective of the paper is to comparatively investigate the buckling behaviors of the T joint and pipe by varying geometric parameters and analysis methods using FEM. In achieving the investigation, wall thickness and ellipticity were selected as geometric parameter, while the geometric nonlinearity and material nonlinearity were involved in the buckling analysis. Based on the minimum potential energy principle, the buckling equations on critical pressure and wave number were established in the paper. By imposing 10% initial defect on the joint and pipe, the imperfection effect was explored on the aspect of the critical pressure. Finally, the paper analyzed the difference on buckling behaviors between T joint and pipe in terms of stress concentration, critical pressure, and deformation pattern.

2. Theoretical buckling and FE modeling

2.1. Strain energy in shell cylinder

The general strain energy of a shell cylinder can be formulated as follows:

$$V=\frac{1}{2}\iiint\left[\sigma_x\varepsilon_x+\sigma_y\varepsilon_y+\sigma_z\varepsilon_z+\tau_{xy}\gamma_{xy}+\tau_{xz}\gamma_{xz}+\tau_{yz}\gamma_{yz}\right]dx\,dy\,dz \qquad (1)$$

where Eq. (1) is obtained on the assumption of thin shells and complying with Hooke's law. After some substitution handlings, the strain energy of the shell cylinder can be rewritten with the middle surface as shown in Fig. 1, assuming the pipe is isotropic thin shells. By integrating Eq. (1) with respect to Z from $-0.5\,t$ to $0.5\,t$ and rearranging the equation, Eq. (1) can be expressed as the summation of bending strain energy (V_{p1}) and extensional strain energy (V_{p2}):

$$V_p=V_{p1}+V_{p2}=\frac{Et^3}{24(1-\nu^2)}\iint\left(\chi_x^0+\chi_y^0\right)^2-2(1-\nu)\left(\chi_x^0\chi_y^0-\chi_{xy}^{02}\right)dx\,dy$$

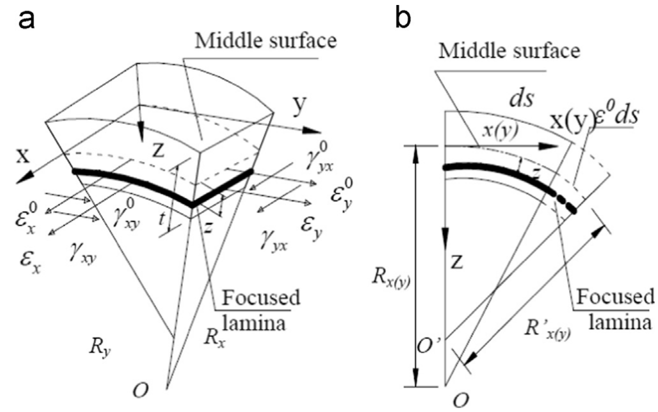

Fig. 1. Variations of an infinite small element and flexural deformation [27].

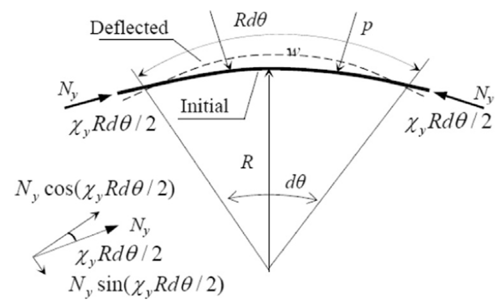

Fig. 2. Work mechanism of infinite small element during buckling [27].

$$+\frac{Et}{2(1-\nu^2)}\iint\left[\varepsilon_x^{02}+2\nu\varepsilon_x^0\varepsilon_y^0+\varepsilon_y^{02}+\frac{1}{2}(1-\nu)\gamma_{xy}^{02}\right]dx\,dy \qquad (2)$$

2.2. Work done by external loads

The work done by external pressure can be calculated from the bending deformation of infinite element shown in Fig. 2. Due to inextensional deformation of the shell cylinder, the work done by tangential force is so small that can be neglected, therefore the work done by perpendicular force dominates the buckling of the shell cylinder. Since the relative rotation angle θ is so small, $\sin(\chi_y R\,d\theta/2)$ is equal to $\chi_y R\,d\theta/2$, therefore the total works can be deduced as follows:

$$U=-\frac{R}{2}\iint N_y\chi_y^0 w\,d\theta\,dx\overset{N_y=pR}{=}-\frac{pR^2}{2}\iint\chi_y^0 w\,d\theta\,dx \qquad (3)$$

Where χ_y^0 is the curvature change of the middle surface shown in Fig. 2, p the external pressure, N_y the axial force.

2.3. Potential energy in shell cylinder

Due to the difficulty in handling the constant stiffness matrix, the buckling load is often approximately estimated using energy conservation. The energy method is an effective method for finding approximations to the lowest energy eigenstate or ground state in mechanics. Accordingly, the potential energy (Π) of the shell cylinder can be expressed as follows:

$$\Pi=U-V_p=\frac{\pi L}{4R}\frac{Et}{1-\nu^2}\left\{-A^2\left[m^2\alpha^2+\frac{1}{2}(1-\nu)n^2\right]+AC[(1-\nu)m\alpha]\right\}$$