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Buckling of bi-segment spherical shells under hydrostatic external pressure

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ABSTRACT

This paper focuses on bi-segment spherical shells under uniform external pressure. The numerical and experimental results of buckling of bi-segment spherical shells with rib-ring of different sizes were discussed. A total of six laboratory scale models with rib-ring of three different sizes were tested. The bi-segment pressure hull was assembly with two individual spherical shells and the rib-ring using the tungsten inert gas butt welding. Each segment was 150 mm diameter and about 0.8 mm wall thickness. Numerical collapse modes agreed well with those obtained from experiments. The predicted collapsed load is about 92–99% of the experimental one except the shell with no rib-ring.

1. Introduction

Deep submersibles are widely used in various military and commercial fields, including rescuing submarines and exploring oceanographic resources. The pressure hull is the most important component of a deep submersible. It has the capability of withstanding extreme pressure, but also it provides adequate interior space for pilots and scientists [1–3]. The pressure hull is traditionally single shell in submersibles. In the wake of developments in ocean research and exploitation, the multi-segment pressure hull, which is assembled with several segments of the same shape or the similar shape, could be used instead of the single shell. The buckling behavior of multi-segment pressure hull has attracted some recent attention [4].

Many studies have been undertaken on the multi-segment underwater pressure hull. Blachut et al. presented and tested two-barrel shell configuration of mild steel. The results revealed that numerical collapse pressures agreed well with experimental results obtained during the hydrostatic test [5]. Zhang et al. presented the new design of the double-segment egg-shaped pressure hulls and studied the effect of rib-ring length and rib-ring thickness on the collapse load [6]. Nevertheless, cylindrical and spherical pressure hulls are the most efficient and simple type for deep-sea pressure hull structures. Spherical pressure hull has the lowest buoyancy factor (weight-to-displacement ratio), but it hardly offers adequate internal space for staff and equipment than a cylinder with the same diameter. The multi-segment spherical shell has the advantages of both the cylindrical hull and the spherical hull. It has lower buoyancy factor and is convenient for interior arrangement. Liang et al. reported results of a numerical study into optimal design of multi-segment spherical hull. The paper contains details of sensitivity analysis of the design variables include the thickness of the rib-ring and the inner radius of the rib-ring, i.e. [7]. Gou et al. studied the structural optimization of the multiple sphere pressure hull, and presented two typical failure modes of bi-segment spherical pressure shell and three new multiple intersecting forms [8]. Hall et al. [9] proposed that the bi-spherical pressure shell was manufactured using graphite/epoxy material, and the reinforcing ring of titanium alloy material are adopted. This approach achieved a weight saving of 46% of that of steel intersecting spheres with the same diameter [9]. Leon et al. tested the bi-spherical pressure shell of titanium alloy. The experimental results showed that the payload of the pressure hull is increased by 10% because the titanium-reinforced ring is replaced by the light weight and high strength ceramic material(Al_2O_3) [10]. Furthermore, Leon's [10] work and Hall's [9] work are concerned only with the effects of the different materials to reduce the weight of the pressure hulls. It appears that few experimental results are available for bi-segment spherical shells based on the design variable of rib-ring. Further study on the buckling of multi-segment spherical shell is still necessary numerically and experimentally.

The buckling behavior of bi-segment spherical shell was studied, firstly numerically and then experimentally. The buckling of bi-sphere shell with rib-ring of different size was analyzed carefully. The corresponding models with rib-ring of three sizes were manufactured. A total of six laboratory-scale bi-segment spherical shells were measured and tested. These shells were collapsed subjected to incremental pressure.

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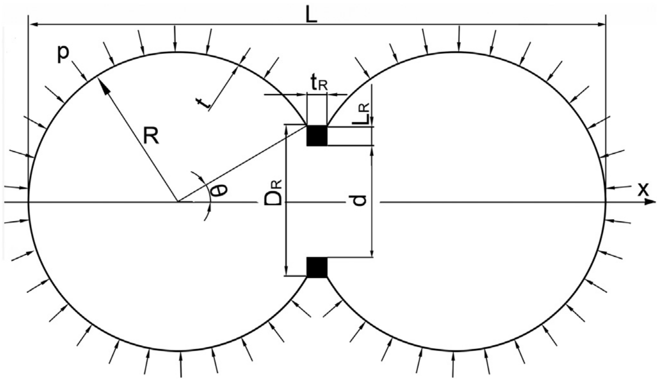

Fig. 1. Geometry of a bi-segment spherical shell subjected to external pressure.

The paper aims to analyse the buckling behavior of bi-segments pressure shell subjected to external pressure.

2. Background

The bi-segment spherical shell under hydrostatic external pressure, p, is investigated in this paper, as sketched in Fig. 1. Consider middle surface radius of spherical shell (R), wall thickness of spherical shell (t), length of the rib-ring (L_R), thickness of the rib-ring (t_R), outside diameter of the rib-ring (D_R), inside diameter of the rib-ring (d), angle of intersection of the spherical shell (θ) and total length of the bi-segment spherical shell (L).

The buckling behavior of spherical shell is mainly characterized by R/t, and the effect of rib-ring thickness and rib-ring length is significant for bi-segment spherical shell. For the individual spherical shell, the buckling pressure drops shraply due to the lost part of spherical shell. The reinforcing ring strengthens the stability of the spherical shell. When the strength of the rib-ring can completely replace the lost part, the bi-segment spherical shell reaches the same strength as the complete spherical shell [8]. Therefore, when the radial deformation of spherical shell and the rib-ring is consistent, the bi-segment spherical shell will be optimally structured. The radial direction is the direction of perpendicular to the rotating axis. It was called deformation compatibility principle and related formulas are as follows.

Firstly, the radial displacement of spherical shell under hydrostatic pressure is evaluated from the following formula,

$$\delta = \varepsilon \mathrm{R} \sin\theta, \tag{1}$$

Where, the strain of spherical shell can be written as

$$\varepsilon = \frac{\mathrm{pR}}{2\mathrm{Et}}(1-\nu). \tag{2}$$

And then, the radial displacement of rib-ring is obtained from

$$\delta_{\mathrm{R}} = \frac{\mathrm{p_R} D_R}{2E}\left(\frac{D_{\mathrm{R}}{}^2 + \mathrm{d}^2}{D_{\mathrm{R}}{}^2 - \mathrm{d}^2} - \nu\right), \tag{3}$$

Where, the pressure on the rib-ring ($\mathrm{P_R}$) is given by the following relations:

$$\mathrm{p_R} = \frac{\mathrm{pR}\cos\theta}{2t_R} + \frac{pR\cos\theta}{2t_R} + p = p\left(\frac{R\cos\theta}{t_R} + 1\right). \tag{4}$$

Finally, in order to coordinate the deformation between the spherical shell and the rib-ring, then

$$\delta = \delta_{\mathrm{R}}, \tag{5}$$

taking into account the expressions (1)- (5) the following values are determined:

$$\mathrm{t_R} = \frac{D_R^3}{2\mathrm{R}(1-\nu)\tan\theta} \times \frac{1}{L_R(1-L_R)} - \frac{D_R(1+\nu)}{R(1-\nu)\tan\theta} \tag{6}$$

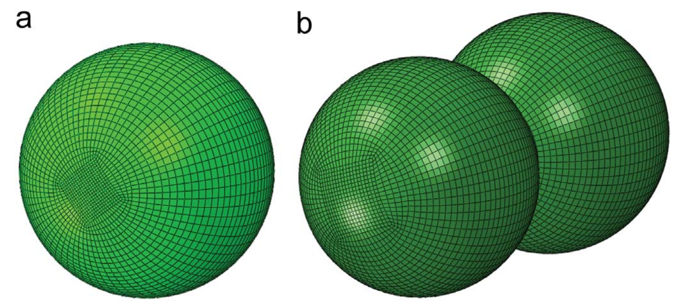

Fig. 2. Finite elements of a single sphere(a) and a bi-segment spherical shell(b).

Where, E is Young's modulus, ν is Poisson's ratio. The relationship between the length of the rib-ring and the thickness of the rib-ring were determined through above-mentioned equations.

3. Numerical analysis

Numerical analyses were carried out to analyse the buckling behavior of bi-segment spherical shells. The detailed parameters of geometry are as follows: R is 75 mm, t is 0.8 mm, D_R is 75 mm, θ is 30°, t_R are 0–18 mm and L_R are 0–8 mm. A single spherical shell with the similar geometry (R = 75 mm, t = 0.8 mm) was also analyzed, which was discussed in previous studies [11]. The nonlinear numerical analysis was implemented using the modified Riks method in ABAQUS according to CCS 2013 [12] and EN 1993-1-6 [13]. It is assumed that the shell was subjected to uniform external pressure, and a p_O = 1 MPa external pressure was applied. Boundary conditions at three random spatial points were respectively constrained in the three orthogonal directions, in order to avoid rigid body motion. A fully integrated S4 shell was selected to avoid hourglassing, and element number of all numerical models were about 7100 for the bi-segment spherical and that was 4538 for a single spherical shell as shown in Fig. 2. The number of elements was determined through mesh density convergence analysis, as reported in Ref. [14]. Material was assumed to be elastic-perfectly plastic with 304 stainless steel, its yield stress, σ_y, is 300 MPa, Young modulus, E, is 195 GPa and Poisson's ratio, ν, is 0.3. The elastic-perfectly plastic material model has been widely used by Blachut and Ifayefunmi and a good agreement between experiment and theory was obtained [15,16]. The effect of rib-ring length, L_R, and rib-ring thickness, t_R, on the collapse load of the bi-segment spherical shell is depicted in Table 1 and Fig. 3.

As can be seen, the pressure increases nearly linearly with the increase of rib-ring thickness and/or rib-ring length, and after the initial increase of collapse load there is a plateau where the magnitude of collapse load does not increase along with an increase of rib-ring length and/or rib-ring thickness. There are 2 lines descried in the figure, named line1 and line2 respectively. Line1 shows that for $t_R \geq 10$ mm

Table 1
The collapse loads (in MPa) for bi-segment spherical shells with rib-ring of different sizes.

t_R (mm)	L_R (mm)							
	1	2	3	4	5	6	7	8
2	1.44	1.69	1.90	2.10	2.30	2.50	2.70	2.90
4	1.57	2.05	2.45	2.98	3.22	3.61	4.00	4.38
6	1.75	2.40	2.98	3.55	4.11	4.67	5.22	5.77
8	1.93	2.74	3.48	4.21	4.94	5.66	6.34	6.40
10	2.10	3.04	3.94	4.83	5.71	6.40	6.38	6.40
12	2.23	3.26	4.31	5.35	6.34	6.41	6.41	6.42
14	2.31	3.44	4.62	5.80	6.40	6.40	6.40	6.40
16	2.34	3.55	4.86	6.15	6.40	6.40	6.40	6.40
18	2.40	3.58	5.10	6.36	6.40	6.40	6.40	6.40

The collapse load of bi-segment spherical shell with no rib-ring (t_R = 0) is 1.18 MPa.