Ocean Engineering 127 (2016) 226–235

Contents lists available at ScienceDirect

Ocean Engineering

journal homepage: www.elsevier.com/locate/oceaneng

Dynamic modeling and vibration characteristics analysis of submerged stiffened combined shells

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ARTICLE INFO

Keywords:
Transfer matrix method
Combined Shells
Wave superposition method
Vibration
Stiffeners
Conical-cylindrical-spherical shell

ABSTRACT

A semi-analytical method is present to analysis the vibration response of submerged stiffened combined shells. In general, the submarine hull can be modeled as submerged stiffened combined conical-cylindrical-spherical shells. The precise integration method is imported to develop a precise transfer matrix method (PTMM). The dynamic model is established to solve the dynamic responses of the combined shell in vacuo. The fluid load is described by wave superposition method (WSM). Then the structural responses of a submerged stiffened submarine hull can be obtained by coupled PTMM and WSM method. The effectiveness of the present method has been verified by comparing the frequency parameters of the combined shells and the structural responses of the submerged spherical shell with existing results. Furthermore, the effects of the model truncation, stiffness, damping and fluid load on the structural responses of the combined shells are studied.

1. Introduction

Nowadays combined shells are widely used in main ocean engineering, such as submarine, underwater vehicle, torpedo, underwater robot, and pipelines. The dynamic response of underwater elastic structure can be solved by many numerical methods including finite element method (FEM) (Everstine, 1997; Damatty et al., 2005), boundary element method (BEM) (Seybert et al., 1990; Ventsel et al., 2010) and so on (Galletly and Mistry, 1974; Liu and Chen, 2009; Ming et al., 2013). Based on the discrete element meshes and constructing the low order shape functions to solve the fluid-structure interaction problem, all these numerical methods above are limited by the calculated frequency and cannot explain the physical mechanism. Analytical methods are only applicable to the vibration analysis of a few simple underwater structures such as spherical shells (Chen, 2003) and cylindrical shells (Laulagnet, 1990). Up to now, many scholars focus on the vibration and acoustic radiation of the cylindrical shell. The cone vertex of conical shell results in tension-bending coupling term existing in the constitutive equation. It certainly increases the mathematical complexity. Then semi-analytical and semi-numerical methods are being welcome (Sivadas and Ganesan, 1993). Irie et al. (1984) presented the transfer matrix method to study the free vibration of joined conical-cylindrical and annular plate-cylindrical shells. Efraim and Eisenberger (2006) proposed a dynamic stiffness matrix method for analyzing the dynamic behaviors of a coupled conical-cylindrical shell. Caresta and Kessissoglou (2010a, 2010b) analyzed the vibro-acoustic responses of a submarine under harmonic force excitation. A wave superposition method which has better computation efficiency and calculation accuracy than the boundary element was proposed based on Helmholtz boundary integral equation (Koopmann, 1989). Based on the two-dimensional fast Fourier transform (2D FFT) algorithm, a wave superposition spectral method with complex radius vector efficiently analyzed the acoustic radiation of an axisymmetric body (Lu, 2011). Qu et al. (2013) investigated free vibration characteristics of conical-cylindrical-spherical shell combinations with ring stiffeners by a modified vibrational method. An accurate modified Fourier series solution which is used to analyze free vibration of truncated conical shells with general elastic boundary conditions was developed (G. Jin et al., 2013a, 2013b, 2014a, 2014b). Su et al.. (2014) employed a unified solution method for free vibration analysis of functionally graded cylindrical, conical shells and annular plates with general boundary conditions. Chen et al. (2015) analyze free and forced vibration characteristics of ring-stiffened combined conical-cylindrical shells with arbitrary boundary conditions.

In this paper, the idea of precise integration method (PIM) (Zhong, 2004) is borrowed to improve the traditional transfer matrix method (TMM). The combined shell is divided into N sections. Based on precise integration method, the field transfer matrix of each section can be solved precisely by using a power series solution. After assembling the whole transfer matrixes of theses sections into an entire matrix and

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http://dx.doi.org/10.1016/j.oceaneng.2016.10.008
Received 6 November 2015; Received in revised form 31 August 2016; Accepted 3 October 2016
Available online 12 October 2016

considering the boundary conditions, the dynamic response in vacuo can be solved easily. Based on the above calculation process, the method called precise transfer matrix method (PTMM) is developed. The authors propose a PTMM for solving structural vibration of combined shells in vacuo. A wave superposition method (WSM) is employed to analyze the effect of the fluid load. Then the authors present a coupled PTMM and WSM for determining the structural responses of stiffened combined shells in fluid. This method is particularly suitable for evaluating the vibration response of distributive systems regardless of the frequency and structural dimension. Then the dynamic response of a complex combined shell can be obtained finally. A combined shell and typical spherical shell taken as an example is calculated by the presented method and compared with the literature data. The comparison exhibit the present method is of high accuracy and efficiency. The convergence analysis for the present method is also analyzed. At last the structural parameters and fluid load are taken into account besides the effects of model truncation.

2. Dynamic model of the combined shell

The submarine hull can usually be modeled as a finite ring-stiffened conical-cylindrical- spherical shell. All the shell components of combined shell accord with the form of shells of revolution, as illustrated in Fig. 1. The combined shell is described with a o-s,θ,r coordinate system, in which s is measured along the meridian, r is the radial coordinate, and θ is the circumferential coordinate. It is assumed that all the shell components and ring stiffeners are made of homogeneous and isotropic materials..

2.1. The combined shell

The time dependent harmonic term $e^{j\omega t}$ is suppressed in the formulation for simplicity of the analysis. Based on the Flügge shell theory (Flügge, 1973), the equations of shells of revolution are written in a matrix differential equation as follows

$$\frac{d\boldsymbol{Z}(\xi)}{d\xi} = \boldsymbol{U}(\xi)\boldsymbol{Z}(\xi) + \boldsymbol{F}(\xi) - \boldsymbol{p}(\xi) \tag{1}$$

where $\boldsymbol{Z}(\xi) = \{\tilde{u}\ \ \tilde{v}\ \ \tilde{w}\ \ \tilde{\varphi}\ \ \widetilde{M_s}\ \ \widetilde{V_s}\ \ \widetilde{N_s}\ \ \widetilde{S}_{s\theta}\}^T$is state vector, the quantities $\tilde{u}$,$\tilde{v}$,... marked with an over score are the respective dimensionless variables. u, v and w are the displacement components in the axial, circumferential and radial directions, respectively. The coefficient matrix $\boldsymbol{U}(\xi)$ is related toξ. $\xi = \frac{s}{R}$ is the dimensionless variable. The nonzero elements in $\boldsymbol{U}(\xi)$ are given in Appendix A. $\boldsymbol{F}(\xi)$ and $\boldsymbol{p}(\xi)$ denote the external driving forces and fluid load, respectively. These dimensionless state vectors are

$$(u, w, \varphi, N_s, V_s, M_s) = \sum_{\alpha=0}^{1}\sum_{n}\left(h\tilde{u}, h\tilde{w}, \frac{h\tilde{\varphi}}{R}, \frac{K\widetilde{N_s}}{R^2}, \frac{K\widetilde{V_s}}{R^2}, \frac{K\widetilde{M_s}}{R}\right)\sin\left(n\theta + \frac{\alpha\pi}{2}\right); \tag{2a}$$

$$(v, S_{s\theta}) = \sum_{\alpha=0}^{1}\sum_{n}\left(h\tilde{v}, \frac{K\widetilde{S}_{s\theta}}{R^2}\right)\cos\left(n\theta + \frac{\alpha\pi}{2}\right); \tag{2b}$$

where R, h is the radius and thickness of the shell. K is bending rigidity $Eh^3/12(1-\mu^2)$.$\alpha = 0, 1$ denotes symmetric mode and anti-symmetric mode.

2.2. The external load

The external load result from the propulsion electric machine, lubricant pump, and sea water pump and the fluctuating force due to the rotation of the propeller. So the conical shell component of the combined shell is excited by the fluctuating forces in both the axial and radial directions. The mechanical excitations acting on the shell can be simplified as concentrated forces f_i (i=1, 2...N). Then the ith concentrated force can be expressed as

$$f(x_i, \theta_i) = f_{0i}\delta(s - s_{0i})\delta(\theta - \theta_{0i})/R \tag{3}$$

where f_{0i}denotes the amplitude of the force, s_{0i}denotes the position of the force in the generatrix direction, θ_{0i}denotes the position of the force in the circumferential angle.

For a given circumferential wave number n, the external force acting on the combined shell can be expressed as

$$\boldsymbol{F} = RK^{-1}[0\ \ 0\ \ 0\ \ 0\ \ 0\ \ f_n^r\ \ f_n^c\ \ f_n^a]^T, \tag{4}$$

where f_n^r,$f_n^c$and f_n^a represent the radial force, circumferential force and axial force respectively.

2.3. Junction of the shell components

The relation between the displacement of the cylindrical shell component u, v, w, φ and the displacement of conical shell component u_c, v_c, w_c, φ_c should keep the continuous. Then the displacement compatibility conditions in the junction can be derived as (Ma et al., 2014)

$$u = u_c\cos\alpha - w_c\sin\alpha,\ v = v_c,\ w = u_c\sin\alpha + w_c\cos\alpha,\ \varphi = \varphi_c \tag{5}$$

Similarly, the relation between force of the cylindrical shell component N^s, $S^{s\varphi}$, V^s, M^s and force of conical shell component N_c^s, $S_c^{s\varphi}$, V_c^s, M_c^s should maintain the balance. The force compatibility condition in the junction can be derived as

$$N^s = N_c^s\cos\alpha + V_c^s\sin\alpha,\ S^{s\varphi} = S_c^{s\varphi},\ V^s = -N_c^s\sin\alpha + V_c^s\cos\alpha,\ M^s = M_c^s \tag{6}$$

According to the force and displacement compatibility condition, the state vectors of the conical shell component and the cylindrical shell component both located in the junction satisfy

$$\boldsymbol{Z}(s = L_{cyl}^L) = \boldsymbol{T}_P\boldsymbol{Z}(s = L_{con}^R), \tag{7}$$

where $\boldsymbol{Z}(s = L_{cyl}^L)$ denote the state vectors of right section of the junction. $\boldsymbol{Z}(s = L_{con}^R)$ denote the state vectors of left section of the junction. $\boldsymbol{T}_P$is a point transfer matrix of order 8×8. In the same way, the transfer matrix in the other junction can be obtained according to the force and displacement compatibility condition.

2.4. The ring-stiffener

If the ring-stiffener is laid on the cylindrical shell component, the stiffener is perpendicular to the shell's axis. According to the displacement compatibility condition in the junction of the cylindrical shell and the stiffener, the relation between centroidal displacement of the stiffener components u, v, w, φ and neutral surface displacement of cylindrical shell component $\overline{u}$, $\overline{v}$, $\overline{w}$, $\overline{\varphi}$can be derived as

$$u = \overline{u} - e\frac{\partial\overline{w}}{\partial x},\ v = \frac{R-e}{R}\overline{v} - \frac{e}{R}\frac{\partial\overline{w}}{\partial\theta},\ w = \overline{w},\ \varphi = \overline{\varphi} \tag{8}$$

where e denotes linear eccentricity. Similarly, the force compatibility condition in the junction can be derived as

$$F_w = -\overline{F}_w,\ F_v = -\overline{F}_v,\ F_u = -\overline{F}_u,\ M_\phi = -\overline{M}_\phi + e\overline{F}_u \tag{9}$$

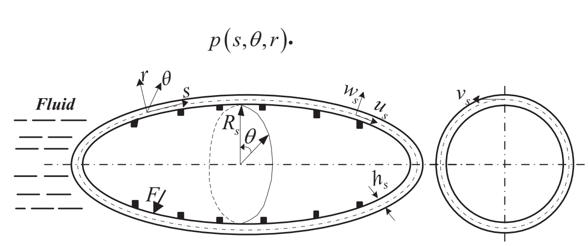

Fig. 1. Schematic diagram of a stiffened shell of revolution.