The Stability and Vibration Characteristic Optimization of the Pressure Shell of a Buoyancy Regulator of an Underwater Vehicle

Yonghui Cao, Chiye Yang, Jing Liu, Yu Xie, Shumin Ma, and Yong Cao

Abstract Underwater vehicles (UV) with a deeper operation ability are the important research field in the marine industry. In order to obtain a better and safer operation performance, the strength, stability and vibration characteristics of the pressure shell of UV should be analyzed. In this paper, a finite element model of the pressure shell of buoyancy regulator is developed. The influences of the stiffener thickness, width, position, and shell thickness on the strength, deformation, and load factor of the pressure shell are studied. In addition, a lighter and safer shell structure is obtained by using the response surface optimization method. The simulation results show that the above factors have great influence on the shell characteristics, such as strength, stability and modal parameters. Moreover, a lighter pressure shell used in the UV can be helpful for providing a better possibility to carry more equipment.

Keywords Underwater vehicle · Pressure shell · Stability · Vibration characteristic

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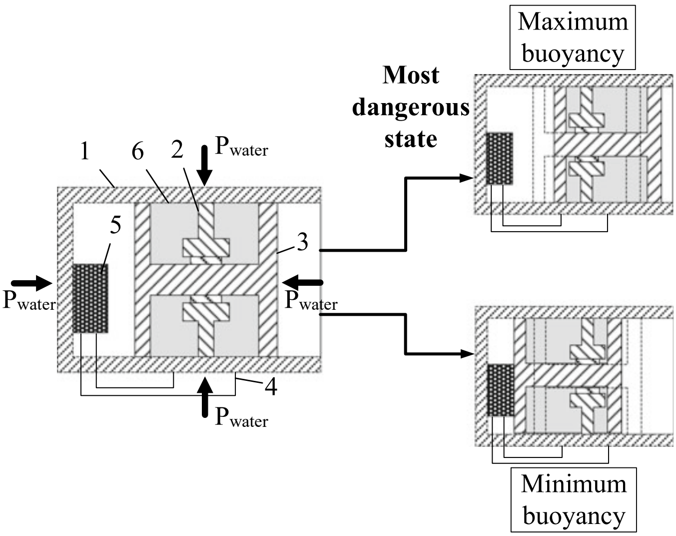
1 Introduction

The piston-type buoyancy regulator is generally used in large-scale UVs such as gliders, which is the key device for adjusting buoyancy. As shown in Fig. 1, when the piston moves outwards, the drainage volume of the buoyancy regulator increases, and the buoyancy provided increases accordingly; on the contrary, the buoyancy provided decreases. When the piston moves to the outermost side, the compartment where the piston cylinder and the hydraulic block are located; and it will generate a larger cavity. Due to the effect of water pressure, this is the weakest status of the whole device. Since the inside is a cavity and the outside are under the water pressure, this section of the cavity can be regarded as a pressure shell.

In 2014, the pressure cabin of the “Nereus” submarine developed by the Woods Hole Institute of Oceanography in the United States [1] ruptured due to the insufficient strength, which caused the submarine to be lost in the Kermadec Trench in northeastern New Zealand. Therefore, it is necessary to calculate the strength of the pressure shell. In addition, it can be known from the critical length formula that this pressure-resistant shell is a thin wall short cylinder at the external pressure, and buckling may occur before the strength yields [2]. Therefore, it is also necessary to conduct the stability analysis. Finally, the moving parts such as the motors inside the device may cause the casing to vibrate during operation. In order to ensure the normal operation of the device, a better dynamic performance is also a required



1.Cylinder (pressure shell), 2. Division plate, 3. Piston, 4. Oil tube

5.Hydraulic Manifold Block, 6. Hydraulic oil

Fig. 1 Working principle of piston-type buoyancy regulator feature. Vasilikis Daniel and Karamanos Spyros A examined the mechanical response of thin-walled cylinders surrounded by a rigid or deformable medium, subjected to uniform external pressure [3]. Rouzbeh Hashemian and Magdi Mohareb developed a general eigenvalue buckling solution for the buckling analysis of sandwich pipes with thick cores subjected to internal and external hydrostatic pressure [4]. Razakamiadana and Zidi studied buckling and postbuckling of concentric cylindrical tubes under external pressure by finite element method [5]. Isvandzibaei et al. presented the energy method for the vibration of thin-walled homogeneous isotropic and manifold layered isotropic cylindrical shells under uniform external lateral pressure [6]. Liu et al. presented an analytical procedure and closed-form vibration solutions with analytically determined coefficients for orthotropic circular cylindrical shells having classical boundary conditions [7]. This paper presents a finite element analysis to simulate the shell to study the strength, stability and vibration characteristics of the shell.

The organization structure of this article is as follows: Sect. 2 introduces the finite element model. Section 3 introduces the simulation results. The lightweight optimization of the shell is completed by the response surface optimization method. Finally, the modal analysis of the shell before and after the optimization verifies that the dynamic performance of the optimized shell is also better than that before the optimization. A conclusion and suggestions for future work will be presented in Sect. 4.

2 Finite Element Model

Aiming at the most dangerous state of the buoyancy adjusting device as the calculation condition, the external pressure shell composed of the inner side of the piston cylinder and the compartment where the hydraulic block is located is taken as the calculation object, and the work is simulated at a water depth of 2000 m. The original model can be simplified to a cylinder with a wall thickness of 12.5 mm as shown in Fig. 2. The wall thickness of the piston cylinder is H; the number of stiffeners is n; the height of the stiffener is h; and the width of the stiffener is w.

For the cylindrical model, Shell181 has higher calculation efficiency and accuracy. Therefore, the calculations in this paper use Shell181 to construct the conceptual model. A quadrilateral dominant mesh is used, the mesh size is 3.0 mm; and the number of mesh is 57,749. Since the rigidity of the piston and the end cap is much greater than that of the cylinder; two ends can be regarded as the fixed constraints; and a pressure of 20 Mpa is applied to the outer wall of the cylinder to simulate a water depth of 2000 m. The mesh division and loading conditions are shown with six stiffeners as shown in Fig. 3.

The response surface optimization method is used to optimize the pressure shell. The variable range is as follows: The number of stiffeners is set to $2 \leq n \leq 6$, where n is an integer; the thickness of the stiffener is $5 \leq h \leq 25$; the width of the stiffener is $5 \leq w \leq 15$; the wall thickness of the piston cylinder is $5 \leq H \leq 30$, and h, w,

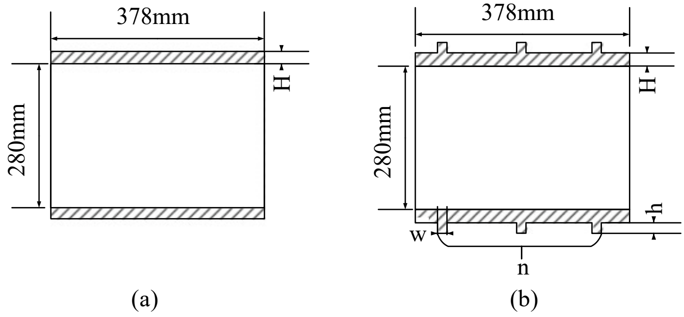


Fig. 2 Geometric model

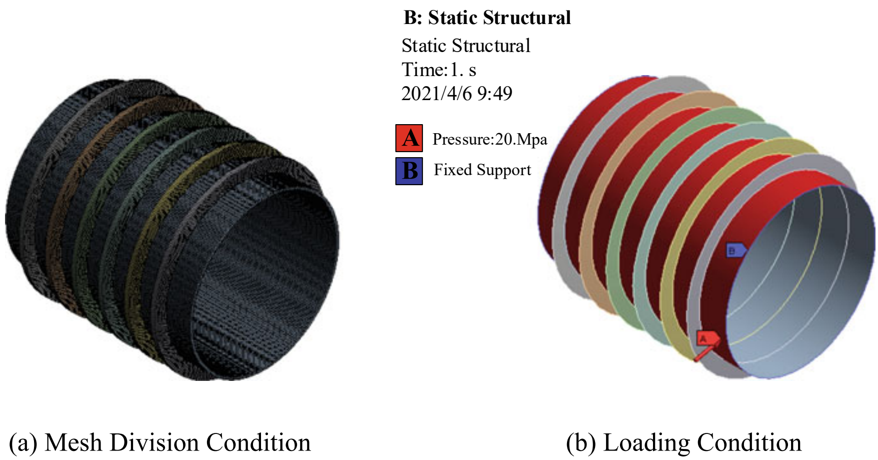


Fig. 3 Mesh division and loading conditions

and *H* are continuous values. The central composite design method (CCD) is used to produce experimental points for calculation. The genetic aggregation method is used to generate the response surface for the calculated experimental points.

3 Simulation Results

The sensitivity of each variable to the impact of the shell's strength, stiffness and total mass, and the results are shown in Fig. 4. The impact of the shell strength can be reflected by the maximum stress. The impact on the stiffness can be reflected by the load factor. It can be seen from the results that the stiffener has little effect on the strength of the shell, far less than the impact of the thickness of the shell on the strength; with the increase of the stiffeners, the influence of the thickness of stiffeners

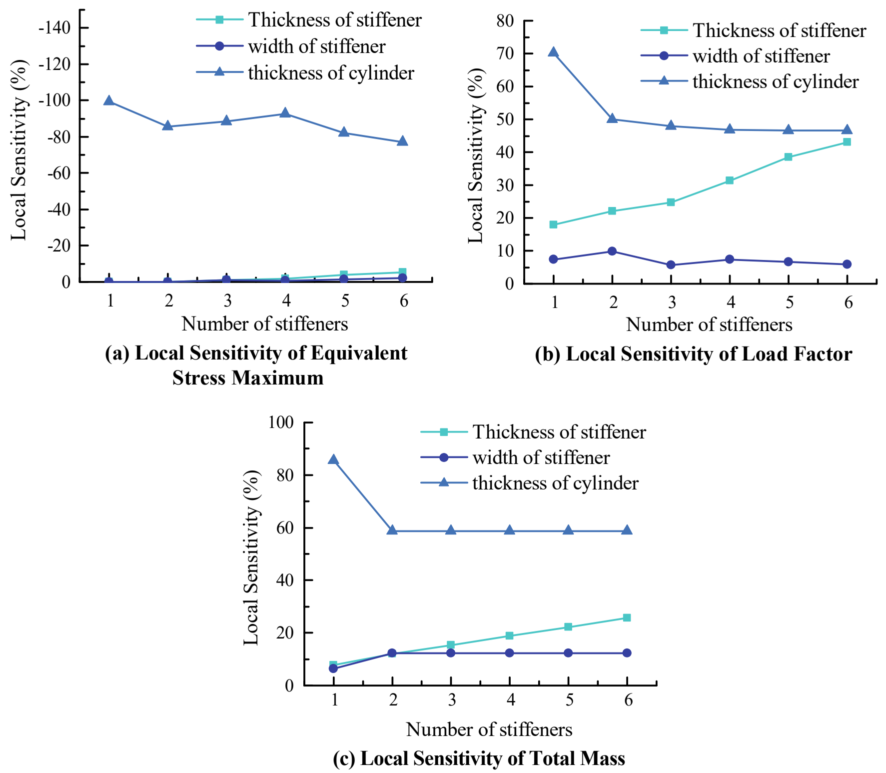


Fig. 4 Local sensitivity curve with the number of stiffeners

on the stiffness of the shell and mass of the shell increases gradually. The width of stiffeners has a small influence on the stiffness. However, the effect of stiffeners on the stiffness of shell is less than that of the wall thickness of shell. The quickest and most convenient way to improve the strength and stiffness of the shell is to increase its wall thickness. The mass of the whole shell will also be greatly increased. The number and thickness of the stiffeners can be increased appropriately in order to improve the stiffness of the body without causing a substantial increase in the mass of the shell.

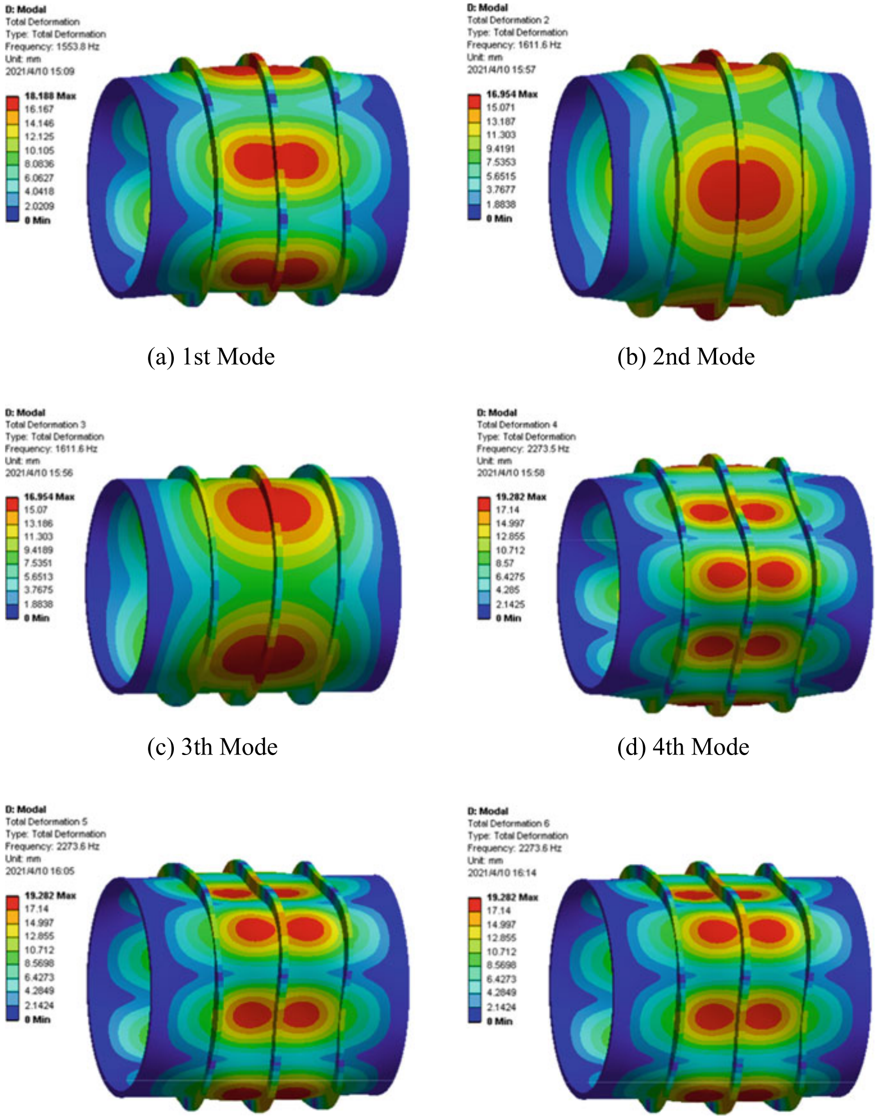
Then, the parameters of the shell are optimized. The optimization method is MOCA method, and the boundary conditions and optimization objectives are: minimum mass, minimum total shape variable, and load factor greater than 3. The optimization results are: $n = 3$, $H = 9.2975$ mm, $h = 23.134$ mm, $w = 7.218$ mm. Keep one decimal place: $n = 3$, $H = 9.3$ mm, $h = 23.1$ mm, $w = 7.2$ mm. At this time, the maximum stress of the shell is 478.39 Mpa, which is less than the yield stress, and the strength is qualified; Since the calculated critical load is often 3 to 5 times larger than the actual critical load [8], the safety factor should not be less than 3, that is, the load factor ≥ 3. In the result, the shell load factor is $3.0946 > 3$, and the stability is qualified; the weight is 10.9 kg, which is 12.5% lighter than before the optimization, achieving the goal of lightening.

The modal analysis of the shell before and after the optimization is carried out. The natural frequency and mode shape results of the first six orders are as shown in Fig. 5 and Table 1. It can be seen from the calculation results that the natural frequency of the optimized shell has been improved, and its dynamic performance has also been optimized.

4 Conclusions

In this paper, the main influence factors on the strength and stiffness of the pressure shell are analyzed by using a finite element analysis method. It seems that the thickness of shell has the greatest influence on the strength, stiffness and total mass of the shell. With the increase of the number of stiffeners, the influence of the thickness of stiffeners on the stiffness of the shell increases greatly. Therefore, in order to lighten the shell, the number and thickness of stiffeners can be increased appropriately. In addition, the existing shell is optimized, and the optimized shell is analyzed too. The results show that the weight of the optimized shell is reduced by 12.5% on the basis of meeting the requirements of strength and stability. Finally, the modal analysis of the optimized shell shows that the natural frequencies of the optimized shell have been improved.

This paper only analyzes the uniformly distributed stiffeners, and obtains better optimization results, which proves that this analysis method is feasible and can be used in practical engineering in the future. In addition, it can also analyze the non-uniform distribution of stiffeners and stiffeners with different cross-section shapes, and find out the best form of stiffeners, and apply it to practical engineering.



(a) 1st Mode (b) 2nd Mode

(c) 3th Mode (d) 4th Mode

(e) 5th Mode (f) 6th Mode

Fig. 5 The front six modes

Table 1 Comparison of the first six natural frequencies before and after optimization

Mode	Frequency after optimization (Hz)	Frequency before optimization (Hz)
1	1553.8	1513.1
2	1553.8	1513.1
3	1611.6	1768.7
4	1611.6	1768.7
5	2273.5	2044.4
6	2273.6	2044.4

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