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On the Design of Pressure-Resistant Torpedo Casing Based on the Buckling Mode

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Abstract

In order to safeguard national security and marine resources, torpedoes are increasingly being developed for deeper depths, which poses higher demands on working conditions. This, in turn, imposes stricter performance requirements on the torpedo casing. Therefore, this study focuses on the design and research of stability indicators for a specific model of torpedo casing. Traditional empirical formula design methods, parameter optimization design methods, and a novel design method based on buckling modes are compared to evaluate their advantages and disadvantages. The aim is to demonstrate the superiority of the new design method in the context of buckling mode design.

Keywords: torpedo, buckling analysis, design method, finite elements, thin-walled structure, reinforced structure.

1 Introduction

Modern torpedoes are underwater weapons characterized by their good stealth, strong anti-interference capabilities, high hit rate, and significant destructive power. They can automatically pursue targets according to preset plans and received target information [1,2].Nowadays, torpedoes are increasingly trending towards the development of greater diving depths [3].

Greater diving depths bring more severe service environments, posing new challenges to the design of torpedo casing structures. Simply increasing the wall thickness to enhance the torpedo's pressure resistance can lead to an increase in the torpedo's mass, which in turn can cause issues with the flexibility and propulsion of the torpedo's normal navigation and tracking. Additionally, increasing the wall thickness without changing the outer surface shape can lead to a mismatch between the center of gravity and the buoyancy center [4], as well as a reduction in the effective internal volume of the torpedo. Therefore, the study of torpedo casing design methods is of great significance.

Stiffened shell structures have become a prevalent architectural choice across various industries, including machinery, aerospace, and shipbuilding [5, 6]. These structures are renowned for their exceptional performance, delivering high rigidity, strength, and efficiency [7]. The design of stiffened shell structures integrates reinforcing stiffeners with shell elements, enhancing their capabilities. When compared to shells of equivalent mass, stiffened shell structures exhibit superior bending rigidity and strength. Moreover, they are characterized by their excellent dynamic and static mechanical properties, making them an ideal solution for applications demanding robust structural integrity.

2 Methodology

In this section, we provide a brief introduction to three methods of torpedo casing design.

2.1 Traditional design method based on empirical formula

Referring to the current design methods for torpedo casings [8], the stresses at various points of the torpedo casing and the critical pressure should meet the following standards:

1. Casing Strength

The most significant stress in the casing is the transverse average stress in the middle of the casing between the stiffeners. Generally, this stress value is relatively high, and the transverse stress over a longer section of the casing between the stiffeners is close to this value. Secondly, the longitudinal stress at the stiffeners is also substantial, but it is of a local nature and decreases rapidly away from the stiffeners, making it relatively less important. Under the action of the calculated pressure p, the stresses at these two locations should meet the following requirements: the transverse average stress in the middle of the stiffened casing should not exceed 0.85σ, as given by

$$\sigma_2^0 = K_2^0 p_j R/t \leq 0.85\sigma_s \tag{1}$$

The longitudinal equivalent stress at the stiffeners should not exceed the yield limit, as given by

$$\sigma_{\text{leg}}|_{x=l/2} = (0.91K_1 - 0.3K_r)p_j R/t \leq \sigma_s \tag{2}$$

2. Stiffener Strength

Under the action of the calculated pressure p, the stiffener stress should not exceed 55% of the material's yield limit, as given by

$$\sigma_r = K_r p_j R/t \leq 0.55\sigma_s \tag{3}$$

3. Casing Stability

The actual critical pressure $p_c r$ of the casing should not be lower than the calculated pressure p_j, as given by

$$p_{cr} \geq p_j \tag{4}$$

4. Overall Stability

The critical pressure for overall instability is set higher than that for the casing to ensure that even if local instability occurs in the casing, the stiffeners still have sufficient stiffness to confine the instability to a local area, thus providing a certain level of safety for the casing. Therefore, the actual critical pressure for overall instability $(p_{cr})_g$ should be not less than 1.1 to 1.3 times the calculated pressure p_j, as given by

$$(p_{cr})_g \geq (1.1 \sim 1.3)p_j \tag{5}$$

Based on the aforementioned strength criteria, a variety of empirical formulas have been derived, and the design of the casing structure parameters is carried out from these.

2.2 Parameter design method via optimization

Parameter design method via optimization is commonly used in structural optimization design, aiming to achieve the optimal design of a structure's performance under specific loading conditions by extracting structural characteristic parameters and employing optimization algorithms. This approach is generally applied to structures with determined configurations and clear geometric features, where the configuration does not change during the optimization process.

During the parameter optimization process, the calculation of the objective function value is typically derived from post-processing results of finite element analysis. Since this process cannot be explicitly represented, the finite difference method is often employed to compute the sensitivity of the objective function with respect to the design variables. Consequently, the number of finite element analyses required for each iteration is directly related to the number of design variables. When the number of design variables is large, the time required for each iteration can become very lengthy. Therefore, it is generally not feasible to use solid models for optimization; instead, simplified models are utilized to enhance the efficiency of the optimization process. This approach is predicated on the simplified model passing an equivalence validation, meaning that the sensitivity of the simplified model to different design variables is consistent with that of the original model.

The selection of optimization algorithms has a significant impact on the outcomes of the optimization process. In this context, the Globally Convergent Method of Moving Asymptotes (GCMMA) is employed [9]. Generally, this algorithm can to some extent avoid the optimization results from becoming trapped in local optimal solutions. Moreover, the direction of inner loop iterations is informed by sensitivities calculated from the outer loop, which reduces the frequency of sensitivity calculations and greatly improves the efficiency of the optimization process. Additionally, this algorithm has been widely used within our research team and has been proven effective through practical application.

For the conduct of parameter optimization, the development of an automated modeling program is indispensable, serving as a critical link between data and geometry, as well as between optimization algorithms and finite element models. Such a program automatically reads optimization parameters and generates structural finite element models based on these parameters. Subsequent simulation analysis can yield the desired structural performance, facilitating the next steps in optimization.

2.3 Fast design method guided by buckling mode

Fig.1 presents the routine procedure of the mode-guided design approach. Firstly, analyse the unreinforced structure for linear buckling and obtain the corresponding eigenmodes. Secondly, perform a geometric analysis of the obtained buckling eigenmodes and the unreinforced model to calculate the deflection angles of the normal vectors at each node. Then, the characteristic '*Max*' and '*Min*' points are extracted. In the third step, the stiffening stiffeners are generated by connecting the extracted characteristic ('*Max*' and '*Min*') points. During the fourth step, the dimensions of the stiffeners are established according to the predefined design criteria and performance objectives. If the structure requires increased stability or if the design specifications require a multi-tiered approach to stiffening, it is important to repeat the previous steps to ensure that the final design meets these higher criteria.

The conventional design of thin-walled reinforced structures is based on the solution of the buckling eigenvalue problem, which serves as a benchmark for structural design. However, the modal reinforcement method departs from this convention by adopting the buckling mode information as the key basis for stability design, a concept that has been largely overlooked in previous studies. Eigenmode analysis reveals the mechanical response characteristics of thin-walled structures under the influence of specific external loads. It essentially describes the relative proportionality of the

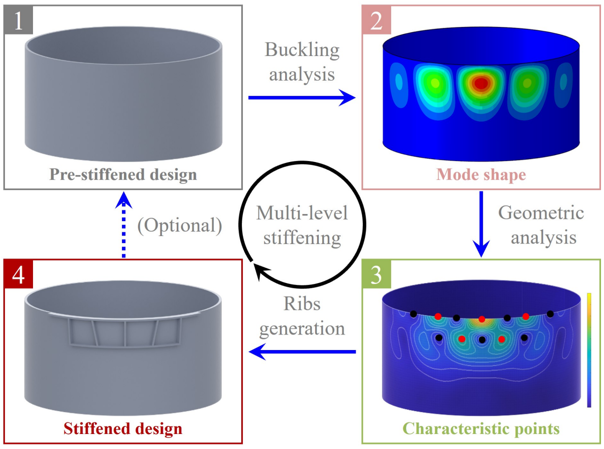

Figure 1: Procedures of buckling mode-guided stiffening casing design.

structure's displacement after buckling deformation. Compared with the conventional optimisation methods that rely only on the buckling eigenvalues, this method provides an innovative idea and solution for the buckling reinforcement design of thin-walled structures by making full use of the high-dimensional information of the modes. In terms of design methodology, the buckling mode-guided design approach proposed in this study has clear advantages. In comparison to conventional design methods, it does not necessitate complex parametric modelling or multiple iterations of optimisation processes. Instead, the designer is able to complete the entire design process through linear buckling analysis. This method significantly reduces the technical threshold for design, enabling engineering designers to design buckling reinforcement of thin-walled structures in a more convenient and efficient pattern.

3 Numerical Examples & Discussion

In this section, we utilize three different torpedo casing design methods to carry out the design of a specific segment of the torpedo casing, in order to compare the advantages and disadvantages of the methods.

3.1 Problem statement

A certain type of torpedo with an outer diameter of $533.4mm$ and a total length of $2100mm$ is assumed. Whose resistance to buckling is expected to be greater when operating underwater. As shown in Fig.2, the work pressure of this torpedo casing is $P_0 = 1.275MPa$. With a load safety factor of 1.2, the design pressure $P_j = 1.53MPa$.

Since traditional design methods take into account both structural load-bearing and stability, while the other two methods focus solely on structural stability, only the traditional design method can provide structural wall thickness parameters. To ensure a fair comparison of the stability design effects without being influenced by the load-bearing design, the other two methods adopt the calculation results from the traditional design method.

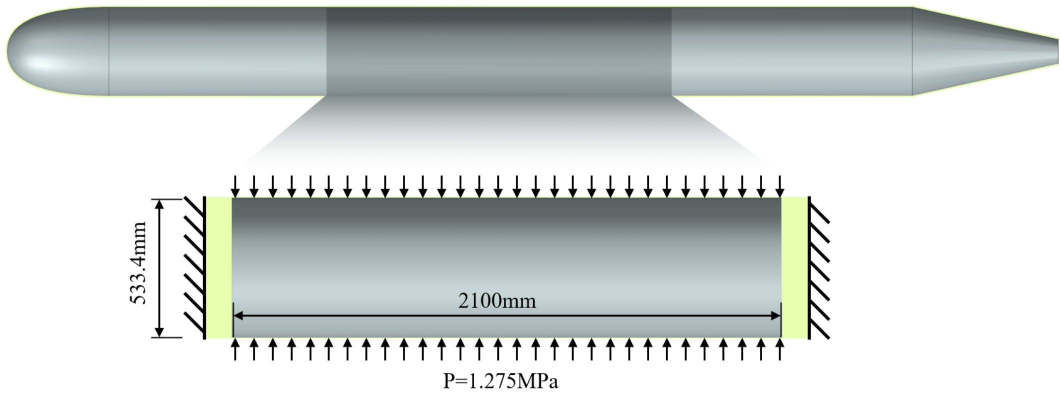

Figure 2: Size and load of stiffening casing.

For the verification of design results, finite element analysis (FEA) tool Abaqus is utilized uniformly. The boundary conditions and load settings are as shown in Fig.2. Steel grade 20A is chosen, with a density of $7.8 \times 10^3 kg/m^3$, a Young's modulus of $196GPa$, and a Poisson's ratio of 0.3. To ensure consistent boundary conditions, two additional cylindrical blocks were created as fixed parts on both sides of the casing, which are fixed in 3 degrees of freedom for displacement, and a uniform pressure of $1.275MPa$ is applied to outer surfaces of the design part.

3.2 Design of torpedo casing based on empirical formula

According to the torpedo design manual [8], the calculated pressure is $p_j = 1.53MPa$, and aluminum alloy is chosen as the material for both the casing and the stiffeners, with a yield stress $\sigma_s = 215.75MPa$. The thickness t of the casing should satisfy the Eq.(6):

$$t \geq \frac{K_2^0 p_j R}{0.85\sigma_s} = 2.394mm \tag{6}$$

where R is the outer radius of the torpedo casing. Given that the thickness $t = 2.4mm$, this value can be applied to the Eq.(7) to calculate l:

$$l \leq \frac{1.029}{p_j}(\frac{100t}{R})^{3/2}100t + 0.62\sqrt{Rt} = 149.8mm \tag{7}$$

Select rectangular as stiffener cross-sectional shape, with a height of $24mm$ and a width of $3mm$. Calculate the moment of inertia I of the combined section of the stiffener and the attached casing:

$$I = \sum(A_i Z_i^2 + I_i) - \frac{(\sum A_i Z_i)^2}{\sum A_i} = 1.41 \times 10^4 mm^4 \tag{8}$$

in which A_i represents the area of each section, Z_i represents the distance from the centroid of each section to the o-o axis, and I_i represents the moment of inertia of each section itself. The section includes both the casing section and the stiffener section. Eq.(8) must satisfy Eq.(9), which is derived from the local stability critical:

$$I = \frac{p_j}{\eta_1 \eta_2} \frac{R^3 l}{3E} = 1.30 \times 10^4 mm^4 \tag{9}$$

The theoretical critical pressure for the casing to buckle according to globe stability critical is:

$$(p'_{cr})_g = \frac{E}{1 + \frac{\alpha_1^2}{2(n^2-1)}} [\frac{t}{R} \frac{\alpha_1^4}{(n^2 + \alpha_1^2)(n^2 - 1)} + (n^2 - 1) \frac{I}{R^3 l}] = 3.84 MPa > 1.53 MPa \tag{10}$$

where n=2, represents the total number of circumferential waves associated with overall instability, and $\alpha_1 = \frac{\pi R}{L} = 0.42$, represents a structural parameter. Since $(p'_{cr})_g > 1.2 p_j$, this parameter of stiffener section is suitable.Construct a thin-walled stiffened model corresponding to the parameters, and the structural buckling analysis result of the torpedo casing design based on empirical formula is shown in Fig.3, whose first eigenvalue is 2.7.

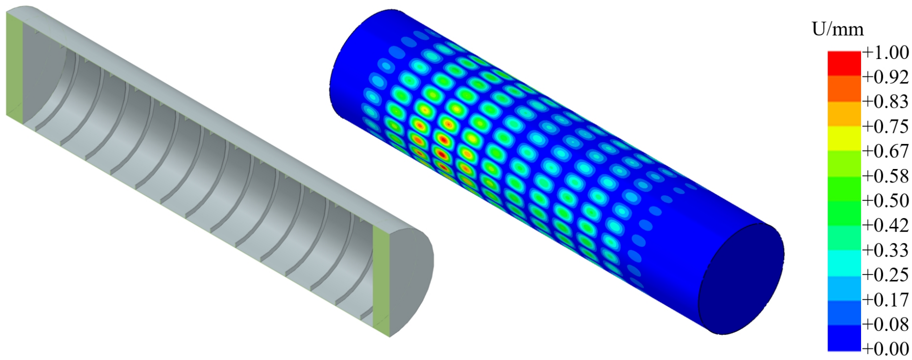

Figure 3: Buckling analysis results of the structure based on empirical formula, whose $\lambda_1 = 2.7$.

It should be noted that in the actual design process of empirical formula method, the stiffener parameters have undergone several iterations and various formulas, ultimately achieving correct results. Since the article only demonstrates this method, the result of design parameters were directly used, and a large number of design formulas were omitted.

The structure obtained through traditional design method based on empirical formula is uniformly stiffened cylindrical casing, with a 1st-order buckling eigenvalue of 2.7. The design efficiency of this method depends on the designer's engineering experience. Designers with more experience can choose reasonable initial parameters, which greatly reduces the number of iterations. Due to lack of engineering experience, the author spent about $480min$ in the design process. This method involves almost full human participation throughout the design process, resulting in high labor and time costs.

3.3 Design of torpedo casing via optimization

The structural configuration of a torpedo casing can be approximately modeled as an outer skin with several internal circumferential stiffeners, which is simple in geometry and facilitates parametric design. For the structural characteristics of the torpedo casing, the thickness $t = 2.4mm$ is determined by traditional design method. Regarding the stiffeners of the torpedo, the characteristic parameters for the i-th stiffener are identified as the positional parameter x_i and the thickness parameter t_i. A parametric schematic diagram of the torpedo casing structure is presented as Fig.4.

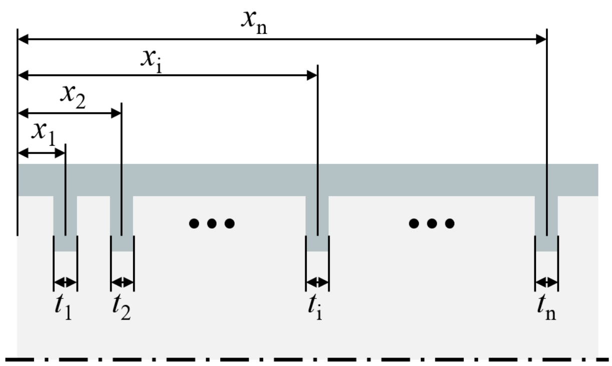

Figure 4: Parametrization of torpedo casing model.

In the process of parameter optimization, the calculation of the objective function value is facilitated by post-processing the results of finite element analysis. Due to the infeasibility of explicit representation in this process, the finite difference method is employed to compute the sensitivity of the objective function with respect to the design variables. Consequently, the number of finite element analyses required in each iteration is correlated with the number of design variables. When the number of stiffeners increases, the efficiency of the parameter optimization can be significantly reduced. Taking into account both design efficiency and performance, the design is conducted with 14 stiffeners, and an equivalent analysis of the shell model is also carried out, as Fig.5, with the expectation of further enhancing the optimization efficiency.

It can be observed that the shell model and the solid model maintain consistent modal deformation up to the fifth order, and the error in the buckling eigenvalues for each order does not exceed 5%. Therefore, it is considered feasible to use the shell model to replace the solid model for optimization design.

Taking the first-order buckling eigenvalue as the optimization objective and the volume as the constraint, the rib parameters are optimized. The constructed optimization model is as follows:

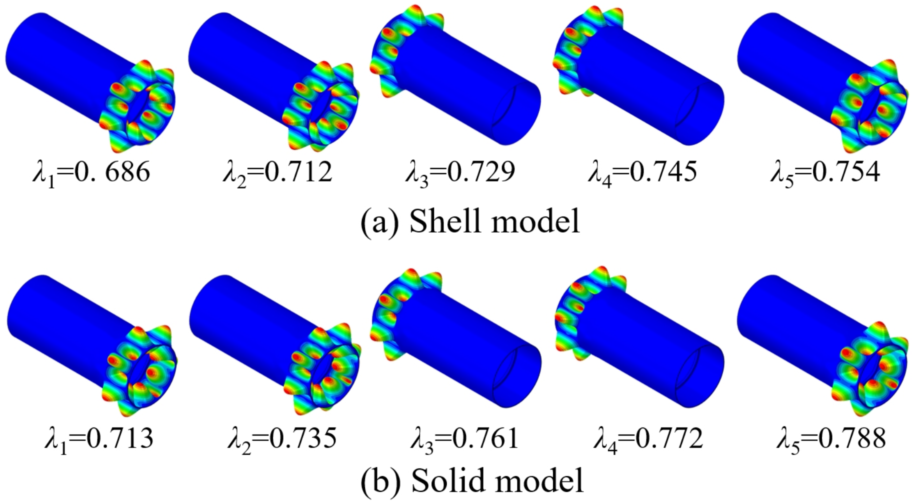

Figure 5: Verification of the equivalence of shell model substitution.

$$\begin{array}{ll} \text{Find} & (\boldsymbol{x}, \boldsymbol{t}) = (x_1, \cdots, x_n, t_1, \cdots, t_n) \\ \text{Max} & \lambda_1 = f(\boldsymbol{x}, \boldsymbol{t}) \\ s.t. & \left\{ \begin{array}{l} \boldsymbol{KU} = \boldsymbol{F} \\ (\boldsymbol{K} + \lambda_i \boldsymbol{K}_{\mathrm{G}})\boldsymbol{\varphi}_i = 0 \\ V = g(\boldsymbol{t}) \leq \bar{V} \end{array} \right. \end{array} \tag{11}$$

Subsequently, the automatic modeling program is developed for the beam-shell model. Using Python language for secondary development of Abaqus, an automatic modeling program is created that can read parameters, generate finite element models, and output buckling characteristic values automatically. This program is designed to prepare for the execution of the parameter optimization algorithm.From Fig.6, it can be observed that after 8 iterations, the objective function and the constraint functions have converged.The buckling analysis result of the casing structure designed by Parameter design method via optimization is shown in Fig.7.

The structure obtained through parameter design method via optimization is approximately a uniformly ribbed cylindrical casing, with a 1st-order buckling eigenvalue of 2.7. The design efficiency of this method depends on the efficiency of the optimization algorithm and the settings of the optimization parameters. The optimization model proposed in this article underwent 8 external iterations and 20 internal iterations. Each external iteration required 30 times finite element analyses, while internal iteration required 1 time. The average time for a single finite element analysis of the shell model is $8.7min$, totaling $2262min$. It should be noted that the long design time results in a low tolerance for errors, which is due to incorrect parameter settings potentially causing the optimization model to start iterating from the beginning.

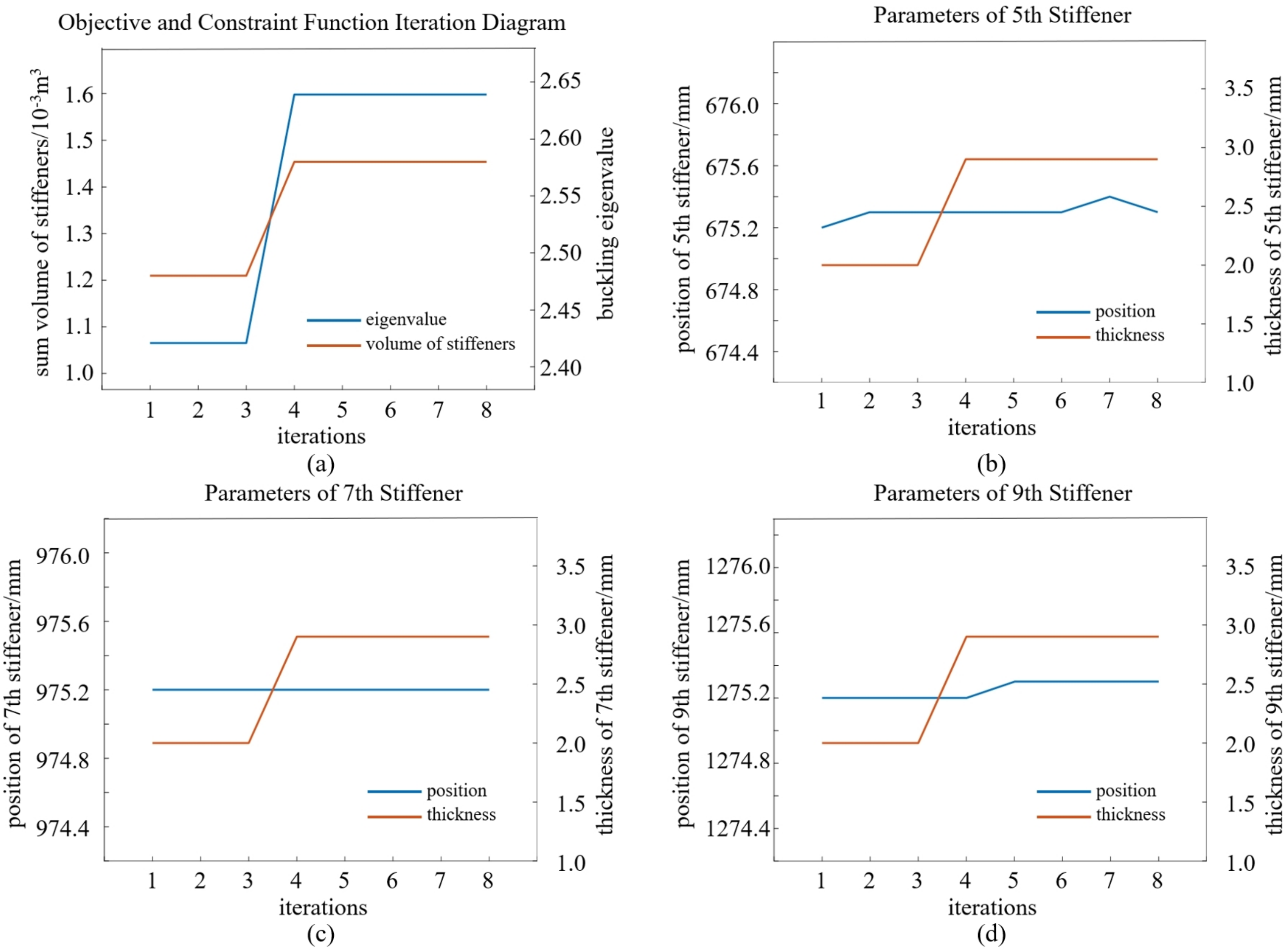

Figure 6: Iteration process of optimization: (a)The objective and constraint functions, (b)The parameters of 5th stiffener, (c)The parameters of 7th stiffener, (d)The parameters of 9th stiffener.

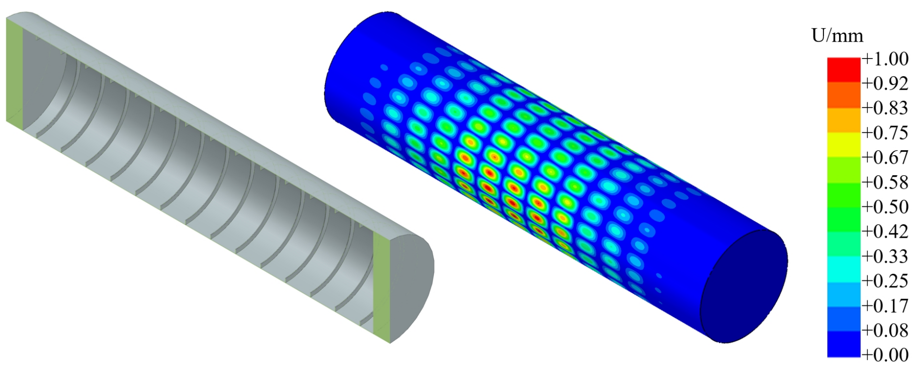

Figure 7: Buckling analysis result of the structure based on optimization method, whose $\lambda_1 = 2.7$.

3.4 Design of torpedo casing guided by buckling mode

Based on the design results from the traditional design method, the thickness of the torpedo casing is taken as $2.4mm$. A finite element model of an unstiffened thin-walled cylindrical structure is established and buckling analysis is conducted, with the buckling deformation information shown in Fig.8. Due to the circumferential symmetry of the cylindrical structure and the load, the buckling mode of the cylinder also exhibits circumferential periodicity. According to the conclusions of Section 2.3, the modal deformation that we focus on, with the maximum and minimum points of the deflection angle connected, just forms a circle around the circumference. This is consistent with the form of circumferential stiffeners in the traditional design method.

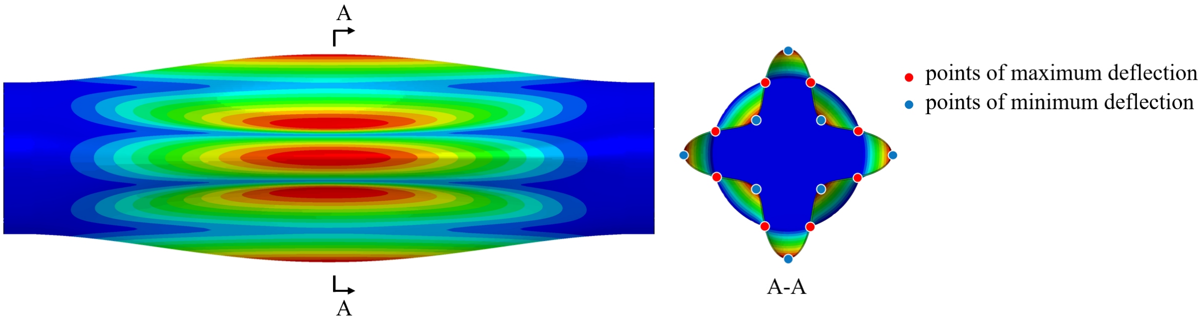

Figure 8: Buckling analysis results of the unstiffened structure.

The circle connecting the maximum and minimum points of the deflection angle located at the center of the buckling deformation. In other words, the design of pressure-resistant torpedo casing based on the buckling mode can be simplified to finding the position of the buckling deformation center and applying circumferential stiffeners. This greatly simplifies the difficulty of the pressure-resistant torpedo casing reinforcement design. Finally, after 7 iterations, we applied 15 stiffeners, and the buckling analysis result of the casing structure obtained is shown in Fig.9.

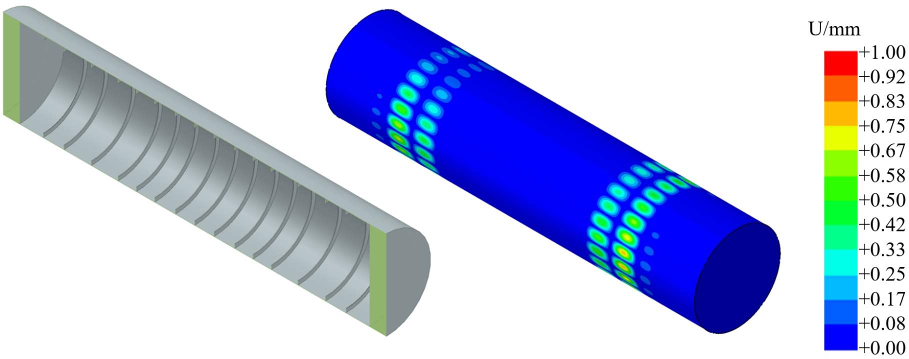

Figure 9: Buckling analysis results of the structure based on the buckling mode, whose $\lambda_1 = 2.6$.

The structure obtained through fast design method guided by buckling mode has stiffeners that are denser in the middle and sparser on the ends, with a 1st-order buckling eigenvalue of 2.6. This method requires less engineering experience from the designer and exhibits a "reinforce where it's weak" pattern in specific pressure-resistant thin-walled cylindrical shell reinforcements. This method underwent a total of 7 iterations of solid model, each of which requires $49.8min$, totally $348.6min$.

4 Conclusions

This article introduces three methods of torpedo casing design, which are traditional design method based on empirical formula, parameter design method via optimization, fast design method guided by buckling mode, respectively. The three methods are applied to the proposed numerical examples to compare the design results and design efficiency.

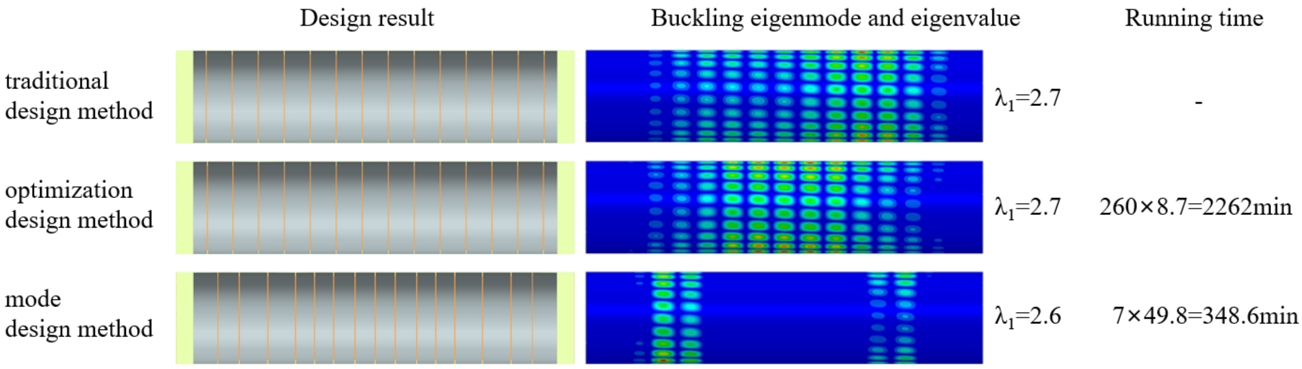

Figure 10: Comparison of Design Results from Three Methods

The 1st-order eigenvalues obtained by the three methods are 2.7, 2.7 and 2.6 respectively, whose error does not exceed 5%. This confirms the effectiveness of the three design methods. However, the design efficiency of the three methods varies significantly. The traditional design method based on empirical formula has a high labor cost, while the parameter design method via optimization has a high time cost. In contrast, fast design method guided by buckling mode can greatly reduce the labor and time costs associated with the design of pressure-resistant ribbed cylindrical shells.

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