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STRUCTURE DESIGN OF THE SHIP PEDESTAL BASED ON TOPOLOGY OPTIMIZATION

CHUNHUI ZHANG, ZECUI ZENG, CHEN JI

Naval Research Academy, Beijing100000, China
e-mail: 502773429@qq.com

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Abstract. The optimization study of the ship pedestal structure is of great significance to the lightweight and the anti-shock performance of the ship. Therefore, the TOSCA software is used to design the ship pedestal in topology optimization. of a ship's pedestal. By setting the load, determining the objective function, selecting the constraints, and selecting the optimization region, the topology-optimized pedestal structure is obtained. Then, the structure was redesigned to determine the final structure of the pedestal. Finally, compared with the traditional pedestal for modal and anti-shock performance, it is verified that the designed pedestal in this paper has improved performance over the traditional one.

1 INTRODUCTION

The ship pedestal is a kind of structure specially designed for installing the equipment on the hull. Generally, the ship equipment must be connected to the hull structure through the pedestal to prevent the equipment from damage caused by shocking and vibration.

When engineering technicians actually design the pedestal, they often rely on existing experience to design the structure and size of the pedestal, making the actual pedestal used to be excessively conservative and bulky. In order to reduce the weight of the pedestal and improve the performance of it, scholars in various countries generally consider both material and structure. In order to improve the performance of the ship pedestal, various materials are used in it. Zhang Xiangwen[1] used the good energy absorption characteristics of Woven bee materials to design two kinds of honeycomb pedestals with macroscopic negative and positive Poisson ratio effects, and compared their stiffness, strength, vibration isolation and shocking isolation performance. The results show that the honeycomb pedestal has excellent anti-vibration and anti-shocking performance. Luo Zhong and Mao Liang[2-3] used the sandwich composite materials to propose a sandwich pedestal. Research has shown that the sandwich pedestal has a lighter weight and impedance damping designing.

It can be seen that the pedestal made of new materials is light in weight and excellent in performance, but most of them are still in the theoretical research stage. Moreover, the threshold of new materials is relatively high, and the preparation and welding processes are not mature enough to be widely used. Therefore, many people start with structural aspects and adopt the new structures to improve the performance of the pedestal. Cheng Huanbo[4] carried out topological optimization analysis on the pedestal of the concrete conveying arm, and designed a new pedestal according to the optimization results. The results show that when the new pedestal meets the strength and natural frequency, the base weight is reduced by 18.3kg. Sun Yumei[5] carried out sensitivity analysis and size optimization for a naval gun pedestal, which reduced the weight of the pedestal by 13.2%, achieving the goal of lightweight. Huang Haiyan [6]took the weight of the host pedestal structure as the objective function, and used stability, allowable stress, and fatigue strength as constraint conditions. Optimized the structure of the host pedestal using the annealing algorithm. San Xiaogang[7] topologically optimized the design of a large-scale theodolite pedestal. According to the relative density cloud map of the material obtained, the new pedestal was rebuilt with the hollow square tube. Under the requirements of strength and stiffness, the weight of the pedestal was reduced by 27.8%.

The optimization of the pedestal structure is of great significance for ship lightweight and anti-shock performance[8]. Therefore, with the help of TOSCA software, the topology optimization design of a ship pedestal will be carried out. In the period of conceptual design, we should jump out of the original thinking formula for the pedestal design to find a structure style with lighter weight and better performance.

2 TOPOLOGY OPTIMIZATION METHOD BASED ON MINIMUM FLEXIBILITY

2.1 Model Establishment

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The variable density method based on minimum flexibility is the foundation of other topological optimization methods with the global volume as a constraint. In this paper, the material difference model based on SIMP is used to solve the problem. The mathematical model is as follows:

$$\begin{cases} Min: C = F^T U \\ s.t.: V < \varepsilon V_0 \\ F = KU \\ 0 < \rho_{\min} \le \rho_i \le 1, \left(i = 1, 2, 3 \ldots n\right) \end{cases} \tag{1}$$

Where C is the structural flexibility (the deformation energy generated by the structure under external force, the smaller the deformation, the smaller the flexibility and the greater the stiffness); F is the external force matrix of the structure; U is the total displacement matrix of the structure; K is the total stiffness matrix of the structure.

2.2 Structure Discretization

For the sake of narrative convenience, the density *ρ* is represented by the variable *x*, then in the discrete finite element structure

$$V = \sum_{i=1}^{n} x_i v_i \tag{2}$$

Where *n* is the number of elements; v_i is the volume of element *i*.

At the same time, assuming that the element stiffness and element elastic modulus before and after optimization are the same, they are also an exponential relationship with the density. That is:

$$k_i = \left(x_i\right)^p k_0 \tag{3}$$

Where k_i is stiffness of the *i-th* element after optimization; k_0 is stiffness of the *i-th* element before optimization.

Since the total stiffness of the structure

$$K = \sum_{i=1}^{n} k_i \tag{4}$$

The total flexibility of the structure

$$C = F^T U = U^T K U = \sum_{i=1}^{n} u_i^T k_i u_i = \sum_{i=1}^{n} \left(x_i\right)^p u_i^T k_0 u_i \tag{5}$$

Where U_i is the displacement of the *i-th* element.

Therefore, from Equation (3) to Equation (8), we can known that under the constraint of volume fraction, with the maximum stiffness as the objective function, the mathematical model of the variable density method based on the SIMP method can be written as

$$\begin{cases} Min: \quad C = \sum_{i=1}^{n} (x_i)^p u_i^T k_0 u_i \\ s.t.: \quad \sum_{i=1}^{n} x_i v_i - \varepsilon V_0 \le 0 \\ \quad ku = f \\ 0 < x_{\min} \le x_i \le 1, (i = 1,2,3...n) \end{cases} \tag{6}$$

Where x is the design variable (unit relative density) and x_i (i=1,2,3...n) is the unit design variable. To avoid the singularity phenomenon in the stiffness matrix when calculating the finite element, x_{min} is usually taken as 0.001and X_i is between x_{min} and 1.

2.3 Sensitivity analysis

In order to get the optimization direction of the design variables, the structural response needs to be partial derivative of the element relative density, that is the sensitivity analysis. Then the relative density of the element is calculated by the displacement u to obtain the finite element equilibrium equation:

$$ku = f \tag{7}$$

Derivatives for x_i on both sides:

$$\frac{\partial k}{\partial x_i} u + k \frac{\partial u}{\partial x_i} = \frac{\partial f}{\partial x_i} \tag{8}$$

The load f is an external force and is independent of the element relative density.

$$\frac{\partial f}{\partial x_i} = 0 \tag{9}$$

Bring equation (9) into equation (8), and it will get

$$\frac{\partial k}{\partial x_i} u = -k \frac{\partial u}{\partial x_i} \tag{10}$$

Find the partial derivative of the relative density for the volume V to the element. Because of:

$$V = \sum_{i=1}^{n} x_i v_i \tag{11}$$

So that:

$$\frac{\partial V}{\partial x_i} = v_i \tag{12}$$

The flexibility C finds partial derivatives of the element relative densities. The expression of element flexibility is:

$$\frac{\partial C}{\partial x_i} = \frac{\partial U^T}{\partial x_i} KU + U^T K \frac{\partial U}{\partial x_i} + U^T \frac{\partial K}{\partial x_i} U \tag{13}$$

By the balance equation $KU = F$, we get:

$$\frac{\partial K}{\partial x_i} U + K \frac{\partial U}{\partial x_i} = 0 \tag{14}$$

With the simultaneous expressions (13) and (14), the sensitivity of the objective function can be obtained:

$$\frac{\partial C}{\partial x_i} = -U^T \frac{\partial K}{\partial x_i} U = -\sum_{i=1}^{n} u_i^T \frac{\partial k_i}{\partial x_i} u_i \tag{15}$$

Equations (6) and (15) are the mathematical model and sensitivity of the minimum flexibility optimization problem respectively. It can be seen that the sensitivity of volume and flexibility is a local variable and only relates to the element. Theoretically, a mathematical model similar to the above equation can be given with any objective function and constraints, and the corresponding sensitivity can be obtained. In fact, some responses are difficult to define, and sensitivity is difficult to deduce. The type of optimization that can be used in engineering practice is very limited. Most of the problems are focused on the optimization of structures subjected to static loads and the optimization of improving the first-order natural frequencies of structures[9-10].

3 OPTIMIZATION DESIGN OF A SHIP PEDESTAL

The topology optimization method based on the minimum flexibility is described above. Based on the above theoretical methods, the TOSCA software is used to optimized design the topology of a ship pedestal with the maximum stiffness as the objective function and the volume fraction as the constraint condition. Based on the optimization results, the influence of structural parameters such as support form and panel shape on the performance of the pedestal is discussed, and the pedestal is redesigned. Finally, Abaqus software was used to compare the modal and anti-shock performance of the two pedestals. The optimized design flow chart of the base is shown in Fig. 1.

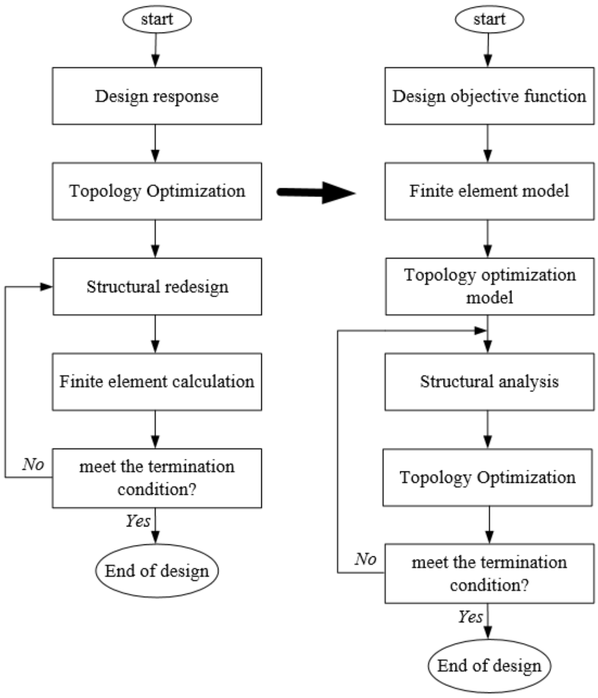

Figure 1: Flow chart of pedestal topology optimization design

3.1 Traditional pedestal model

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Select a device pedestal on the inner bottom of a frigate for topology optimization. The pedestal is a typical "box shaped" pedestal that is welded directly to the inner bottom. The device is rigidly connected to the pedestal plate by bolts. The base is 800mm in length, 670mm in width, 150mm in height, 10mm in a panel thickness, 8mm in thickness of abdominal plate and elbow plate, and 46.8kg in weight, as shown in Fig. 2 in detail. The pedestal material is 907A steel with a density of 7.8e-9t/mm^3, an elastic modulus of 2.06e5MPa, a Poisson Ratio of 0.3, and a yield limit of 380MPa.

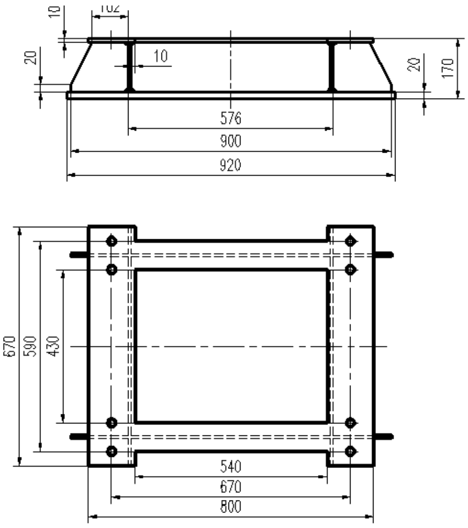


Figure 2: An engineering drawing of a pedestal

3.2 The optimal setting of the pedestal

1) Shocking load setting of the pedestal

When the pedestal is subjected to a three-phase shocking, the vertical shocking is the most dangerous situation. When undergoing topology optimization, the vertical shocking is the input load. The topology optimization of the pedestal in this paper is rigidly installed on the inner bottom, which belongs to the hull installation, and the equipment it carries is Class A equipment. The input value of shocking load is designed according to the standard GJB1060.1-91[11]. The pedestal weighs 46.8kg and the equipment weighs 240kg. When calculating, 80% of the total mass is taken as the effective mass. It was calculated that: the equal acceleration spectrum is As=2203m/s^2; the equal speed spectrum is Vs=2.92m/s; and the equal displacement spectrum is Ds=0.045m.

2) Objective Function

The pedestal is rigidly connected to the device. When subjected to shocking loads, the pedestal is not allowed to deform in order to ensure the accuracy of the equipment it carries. The objective function for optimization is the maximum stiffness of the pedestal, which is set to the minimum in the TOSCA software for structural flexibility (ie strain energy).

3) Constraints

Spatial aspects: Ship equipment not only has high weight requirements, but also has strict requirements on the volume. To ensure that the equipment environment does not require much adjustment, the size of the space occupied by the structure must not change much. That is, the variable area does not exceed the cube area formed by the original structure of length, width and height.

In terms of connection: The pedestal serves as a "bridge" between the equipment and the hull, and is connected to the equipment through bolts. In order to ensure the reliability and convenience of connection with the equipment before and after optimization, the position and size of the screw hole can not be changed.

In terms of process: the geometric structure of the pedestal and the load it bears are all symmetrical about the horizontal and vertical planes passing through the gravity center of the pedestal, so there is a symmetry constraint in the topology optimization. In order to eliminate the small transmission path in the optimization result and make the topology optimization structure more regular, the size of the smallest member of the structure is required to be constrained. The optimization result requires that the smallest member size of the structure is ≥10mm. The pedestal is welded through the plate. If the plate is too thin, it will burn easily, or it will cause large deformation due to uneven heat. Therefore, the thickness of the plate should be ≥5mm.

4) Setting the Optimization Zone

On the one hand, the pedestal limits the space occupied. On the other hand, the number, position and specification of the bolts remain unchanged. Therefore, an optimization model is designed to constrain different volume fractions and to research the effect of different structural parameters on the topology results. A cubic block is constructed based on the length, width and height of the original pedestal, and the bolt holes are excavated in the middle. The remaining area is used as the optimization area, as shown in Fig. 3.

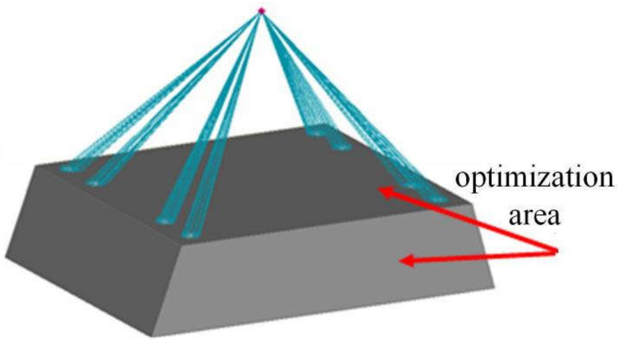


Figure 3: Optimization model

The general process of topology optimization is designing-optimization-redesigning. This optimization aims to find the best force transmission path through topological optimization under constraint conditions, and retain the most efficient materials. Through the analysis of the topology optimization results, valuable conclusions are obtained, and used for the guidance of anti-shock designing of the pedestal, so as to design a new pedestal and compare the modal and anti-shocking performance of the old and new pedestals.

3.3 Analysis of Optimization Results

With TOSCA software, topology optimization was performed on two pedestal structure models under different volume constraints. After many iterations, the pedestal quality and the stiffness gradually decreases as shown in Fig. 4. Both the constraints and the objective function converge very quickly. The objective function tends to stabilize after the 5th iterations, and the volume fraction reaches the setting value 0.4 after the 15th iterations.

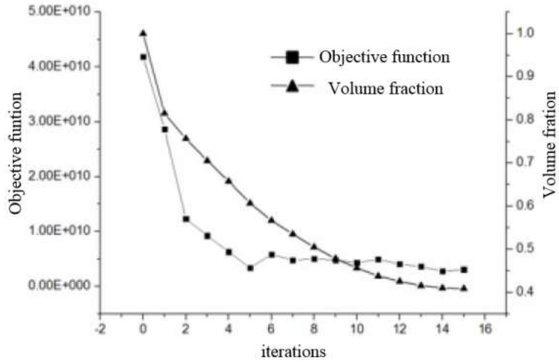


Figure 4: Constraint conditions and objective function iteration curves with volume fraction of 0.4.

Fig. 5 shows the pedestal structure after the topology optimization of Model 2 at a volume fraction of 0.4. It can be seen that many elements have been removed. Through the processing of the Smooth module in the TOSCA software, a relatively smooth structure is obtained, and several formats commonly used in CAD software can be output to facilitate the redesign of the pedestal.

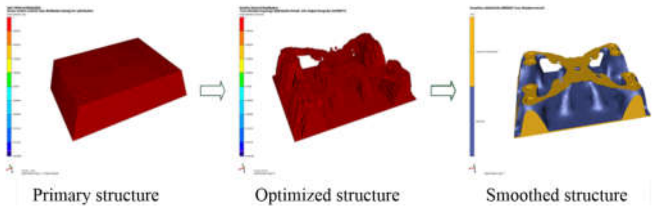


Figure 5: Optimization results when the volume fraction is 0.4

3.4 The parameters discussion of the pedestal structure

Although the structural topology optimization results can indicate the optimal material allocation of the structure, the optimization results tend to produce some ambiguous structures locally. Extract the concept of wireframe and panel for the optimization result. Restore the structure concept as realistically as possible, and try to make the similar endpoints reach one point.

1) Support Form

From the previous analysis, it can be found that the pillar below the bolt is the most important force transmission structure. It exists when the volume fraction and retention quality are low, so the support structure of the panel is the most important structural parameter.

Ensure that the pedestal quality is almost constant. All the panel thickness of the pedestal is 10mm, and the truss sections are all 12 × 12 rectangles. The material is 907A steel. By comparing the shocking responses of different supporting structures such as trusses, plates, '工' shaped beams and 'box' shaped beams, the changing regular that the anti-shocking performance of the pedestal changes with the structure is studied.

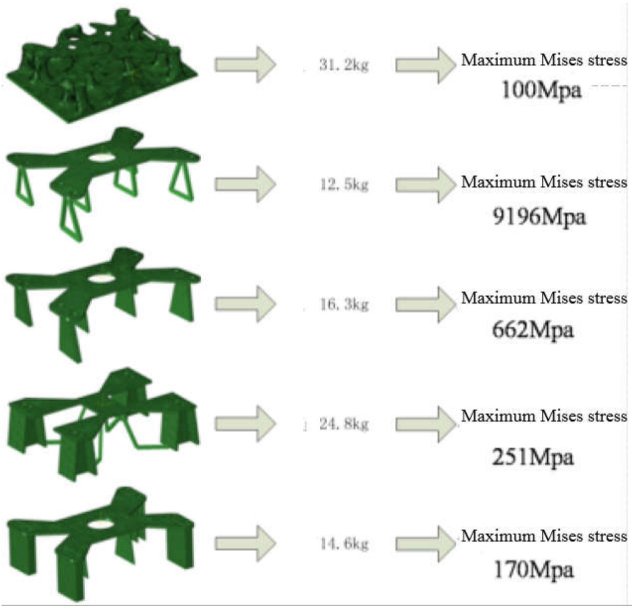


Figure 6: The shocking response of the pedestal in different supporting forms

Fig. 6 shows the maximum Mises stress corresponding to the pedestal under different cross-sectional shapes. It can be seen that using the box beam supports the least stress and best results. The "box" shaped beam supporting structure was used as the research object for force analysis, and the panel was considered as a fixed constraint. The other end of the "box" shaped beam is subjected to bending moment *M* and pressure *F*. The force diagram is shown in Fig. 7.

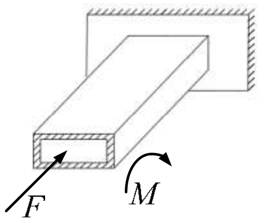


Figure 7: Force diagram of "box" beam support structure

From the knowledge of material mechanics, it can be known that for cantilever beams, the maximum stress is at the fixed support , ie at the panel. The stress size is

$$\sigma=\frac{F}{A}+\frac{M}{W} \tag{16}$$

Where, F and M are the axial pressure and bending moment of the cantilever beams respectively; A and W are the sectional area and the section modulus of the cross-section respectively.

Therefore, increasing the contact area between the support structure and the panel can reduce the stress of the pedestal. When the Sectional area does not change significantly, the higher the section modulus, the smaller the stress of the pedestal. Under different supporting forms, the maximum stress of the pedestal is ranked from large to small as truss > plate > '工' shaped beams > 'box' shaped beam.

2) Plate Shape

In the process of topology optimization, the shape of the plate is also changing. According to the optimization results of the pedestal under different volume fraction constraints, the plate shape can be divided into four cases as shown in Fig. 8 .

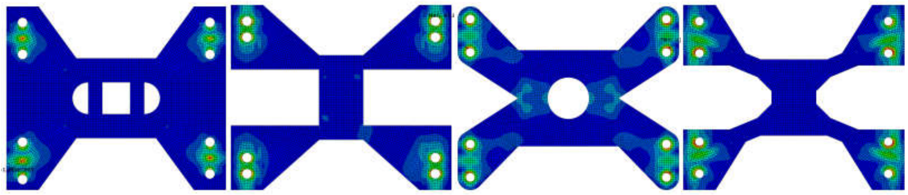


Figure 8: Different shapes of the pedestal plate

Ensure that the support of the pedestal is unchanged and compare the maximum Mises stress in the structure. This type of support is all supported by '工' shaped beams, and the pedestal have approximately the same mass and are subject to the same shocking load.

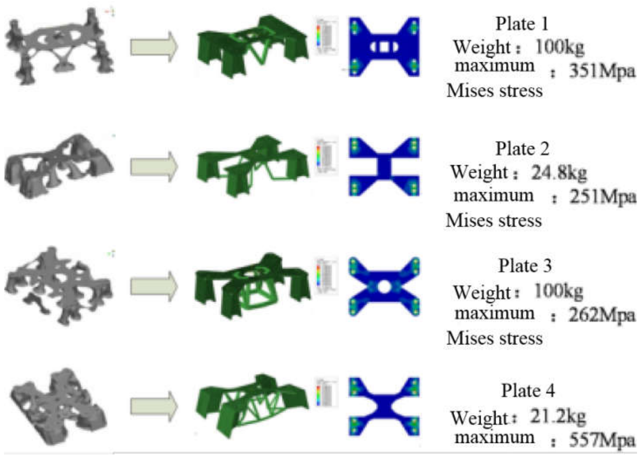


Figure 9: Shocking response of the pedestal with different plate shapes

Fig. 9 shows the maximum Mises stress in the pedestals of different plates under the same shocking load. It can be seen that these four plates shape do not have much influence on the shocking response; the difference in stress is largely determined by the form of support and the form of the structure. In view of the process and the structural durability, the plates corresponding to cases 2 and 4 are thinner at the four corners and the middle connection, and they are not as safe as the plates corresponding to cases 1 and 3.

3) Redesign of the Pedestal

Based on the optimized results and parameter analysis of the pedestal, it can be seen that it is more suitable to use a “box” shaped beam and the shape of the panel corresponding to Case 1 or Case 3. However, during the actual manufacturing process, it was found that the screw used to secure the equipment bolts on the panel would penetrate deep into the “box” beam, resulting in the inability to install the nuts. For this purpose, change the section shape to "E" which is as shown in Fig. 10. At the same time, in order to avoid structural damage caused by stress concentration, the structural boundary should be as smooth as possible. For this reason, a slight improvement is made on the basis of the plate 3. The length, width and height of the final optimized designing pedestal are basically the same as those of the original pedestal, and the positions of the bolts do not change. The plate thickness is 10mm which is supported by four brackets. The cross-section of the bracket is in an "E" shape and the thickness of it is 6mm. The pedestal is supported by equilateral angle steel truss in the middle and its weight is 36.2kg (22.6% of weight has been reduced). The finite element model of the optimized pedestal is shown in Figure 11, and its engineering drawings are shown in Fig. 12.

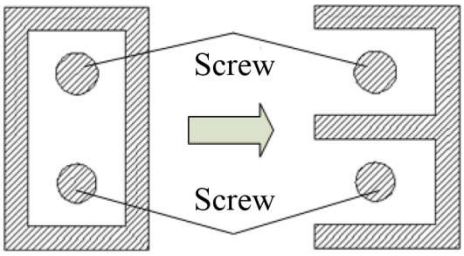


Figure 10: Ideal and actual sections

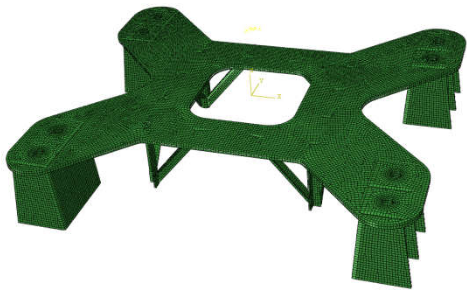


Figure 11: The final optimized pedestal finite element model

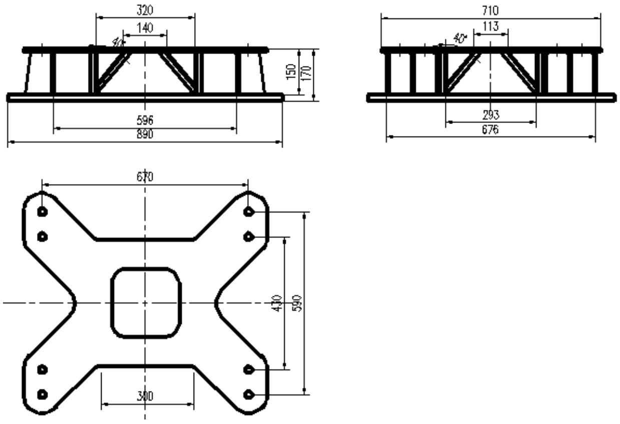


Figure 12: Engineering drawings for the optimized pedestal

4 COMPARISON OF PEDESTAL PERFORMANCE

4.1 Modal comparison

The main headings should be written left aligned, in 12pt, boldface and all capital Roman letters. There should be a 12pt space before and 6pt after the main headings.

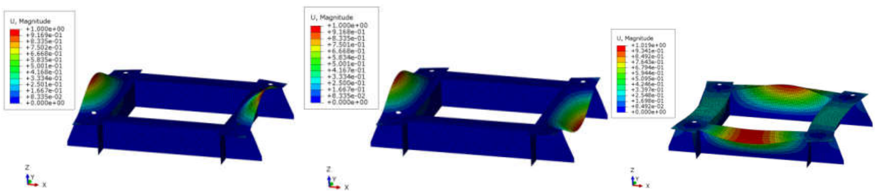
This paper considered the actual working environment of the pedestal and separately constrained the six degrees freedom of the original pedestal and the optimized pedestal in contact with the insole. In the Abaqus software, the six modes were calculated separately. The vertical participation quality of each mode was shown in Tab. 1.

Table 1: The first 6 modes of the pedestal

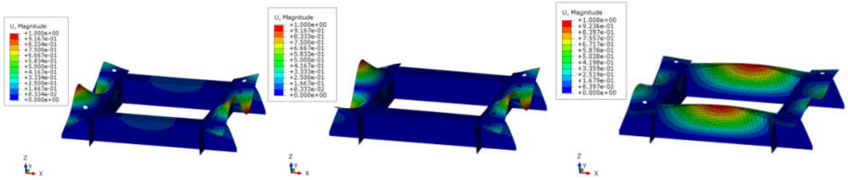
Modal order	Traditional pedestal		Optimized pedestal	
	f_1 (Hz)	p_1 (t)	f_2(Hz)	p_2(t)
1	306.82	3.1E-03	430.82	2.9E-04
2	331.15	2.6E-08	764.35	1.1E-10
3	448.70	2.2E-14	825.59	3.4E-03
4	448.76	1.3E-12	885.15	2.5E-10
5	554.66	5.8E-07	943.05	1.9E-14
6	592.96	6.1E-16	999.27	4.1E-10

In this table, f_1 is traditional pedestal frequency, p_1 is traditional pedestal participation quality, f_2 is optimized pedestal frequency, p_1 is optimized pedestal participation quality.

The first six vibration modes of the two pedestals are shown in Figs. 13 and 14, and it can be seen that the first vibration modes of the two pedestals are mainly vertical vibrations.

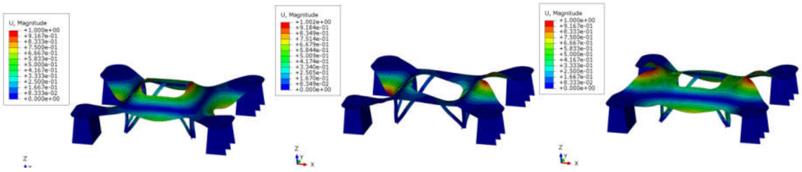


a. The 1st order mod b.The 2nd order mode c. The 3rd order mode

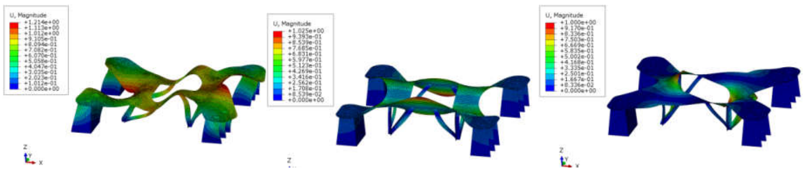


d. The 4th order mode e. The 5th order mode f. The 6th order mode

Figure 13: The first six-order modes of the traditional pedestal



a. The 1st order mode b. The 2nd order mode c. The 3rd order mode



d. The 4th order mode e. The 5th order mode f. The 6th order mode

Figure 14: The first six-order modes of the optimized pedestal

It can be seen that the vibration modes of the first two stages of the two pedestals are similar. The vertical minimum resonant frequency of the traditional pedestal is 306.82 Hz; the vertical minimum resonant frequency of the optimized pedestal is 430.82 Hz. After the pedestal is optimized, its vertical stiffness is increased.

4.2 Shocking Comparison

Secondary headings should be written left aligned, 12 pt, boldface Roman, with an initial capital for first word only. There should be a 12pt space before and 6pt after the secondary headings.

According to the method specified in GJB 1060.1-91, the shocking load of the traditional pedestal and the optimized pedestal is designed, and it is converted to a dual-triangular acceleration time-domain curve according to the German specification BV430/85 as the input load for shocking calculation[12]. The vertical, lateral and vertical directions of the two pedestals were checked for anti-shocking. According to formula (15), the input value of the shocking load can be obtained, as shown in Tab. 2. It can be seen that the spectral value calculated according to the specification is related to the weight. When the mass of the optimized pedestal is small, the input load obtained will be slightly larger.

Table 2: Input loads in different shocking directions

Direction	Value	Model pedestal	Topology pedestal
Vertical	A_s (m/s^2)	2203	2213
	V_s (m/s)	2.92	2.93
	D_s (m)	0.045	0.045
Lateral	A_s (m/s^2)	881	885
	V_s (m/s)	1.17	1.17
	D_s (m)	0.045	0.045
Longitu-dinal	A_s (m/s^2)	441	443
	V_s (m/s)	0.58	0.59
	D_s (m)	0.045	0.045

In this table, A_s = Acceleration spectrum, V_s= Speed spectrum, D_s = Displacement spectrum.

1) Response curve of the device acceleration

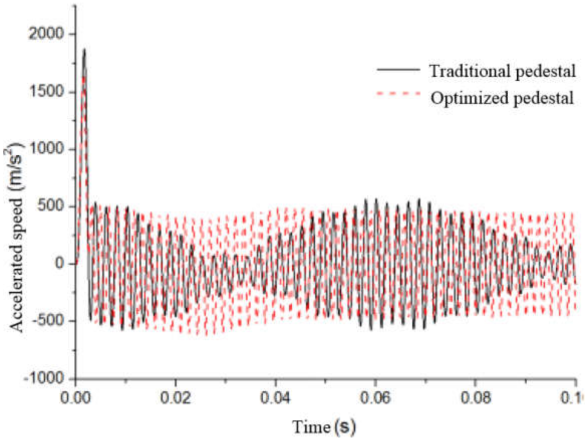


Figure 15: Vertical acceleration response of the device

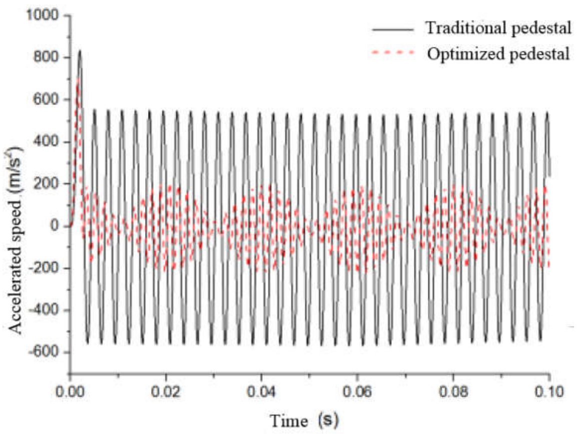


Figure 16: Lateral acceleration response of the device

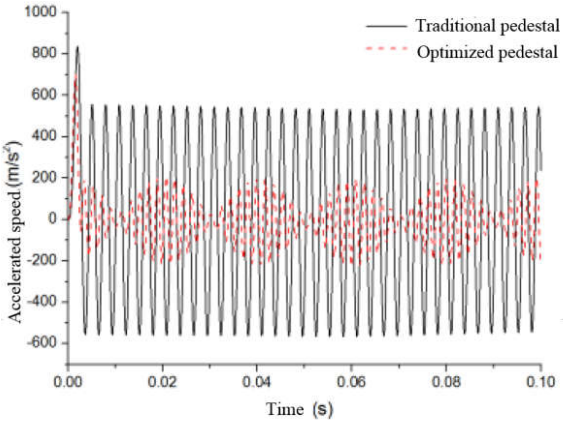


Figure 17: Longitudinal acceleration response of device

Figs. 15 to 17 are the vertical, lateral and vertical acceleration curves of the traditional spectral and the optimized spectral. It can be seen that the device acceleration curve shapes of the two bases in the three directions are very similar. Under the action of the shocking, the acceleration value increases rapidly and basically reaches the maximum value at the same moment, and then decays rapidly. Since no damping is set, the acceleration of the equipment will continue to oscillate after the shocking load ends. It can also be seen that despite the rigid connections, the isolation performance is limited. However, in both the vertical and lateral directions where the shocking loads are larger, the input value of the optimized pedestal is slightly larger than that of the traditional pedestal, but the maximum acceleration at the gravity center of the device is smaller than that of the traditional pedestal. From this point of view, the anti-shocking performance of the optimized pedestal is slightly better than that of the traditional pedestal.

2）Response of pedestal stress

Figs. 18 to 20 are Mises stress cloud figures for the maximum stress in vertical, lateral, and longitudinal directions of the traditional pedestal and the optimized pedestal. The bolt connection is simulated using the MPC-Beam method, and the calculated stress at the nodes near the bolts will be larger. The three level units near the bolt have been hidden in the figure. It can be seen that when the pedestal is subjected to a vertical shocking, the stress value is the highest, which is the most dangerous condition of the pedestal. The shapes of stress cloud figures for the two bases are similar. The maximum stress appears near the plate bolts, and the maximum stress value of the traditional pedestal is 616 MPa, which is beyond the requirements of the Chinese military standard GJB1060.1-91 (The stress is not exceeded the material's static yield strength of 390 MPa). The maximum Mises stress of the optimized pedestal is only 272 MPa, which is reduced by 55.8% and meets the requirements of GJB 1060.1-91. Under the lateral and vertical shocking loads, the maximum Mises stress of the traditional pedestal is low. They are 97.6 MPa and 71.2 MPa respectively, which are far less than the static yield limit of the material. The maximum Mises stress at the optimized pedestal was increased compared to the traditional pedestal. They are 237 MPa and 116 MPa respectively, but they are still smaller than the static yield limit of the material and the pedestal was in a safe state

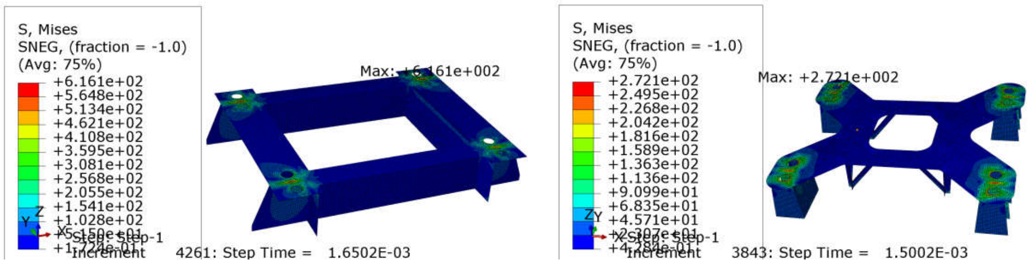


Figure 18: Pedestal stress cloud figure at vertical shocking

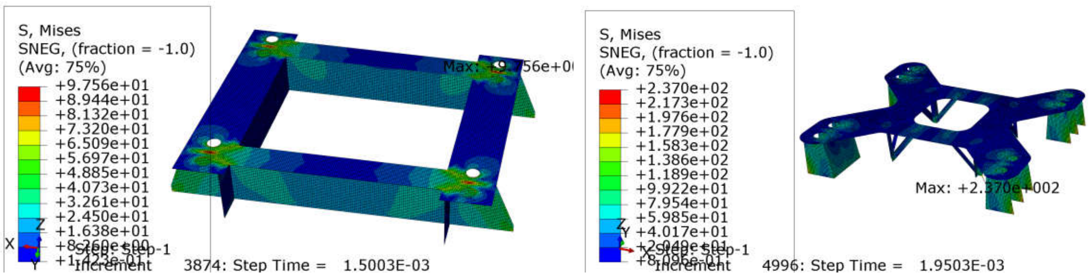


Figure 19: Pedestal stress cloud figure at lateral shocking

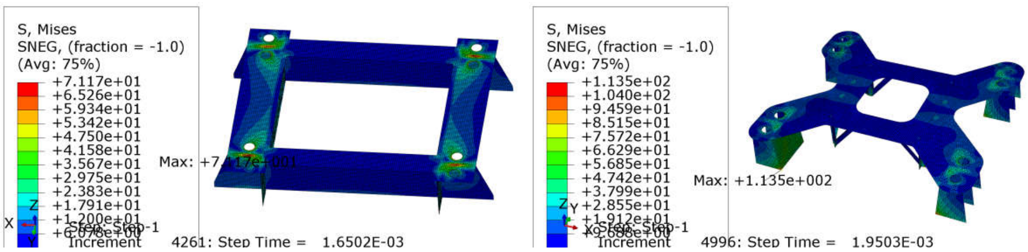


Figure 20: Pedestal stress cloud figure at longitudinal shocking

5 CONCLUSION

This paper mainly used TOSCA software to optimize the designing of a ship pedestal, and designed a new structure of the pedestal. Firstly, established a finite element model of a pedestal, and set the shocking load of the pedestal according to the national military standard of GJB1060.1-91. Then, the topology optimization of the two pedestals is designed. The maximum stiffness is used as the objective function, and the volume fraction is the constraint condition. A series of pedestals with different structural forms are set up based on the optimization results, and the effecting of the support form and the plate on the pedestal anti-shocking performance is studied. Finally, considering the stress concentration effect, the constraints of the actual processing technology, and the convenience of equipment installation, the shape of the support frame was changed from the most ideal "box" shaped to the "E" shaped and the edges and corners of the panel were smoothed. The pedestal designed in this paper was finally obtained. Compared with the traditional pedestal, the weight was reduced by 22.6%, the vertical minimum resonance frequency is increased from 306.82Hz to 430.82Hz, the vertical and horizontal anti-shocking performance are slightly improved, and the anti-shocking performance is greatly improved. When the shocking strength of the pedestal was checked in the form of “box” support, the maximum Mises stress of the pedestal was reduced from 616 MPa to 272 MPa, which was a drop of 55.8% under the vertical shocking load.

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