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Ship mounting structure damping material optimization distribution and experimental study

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Abstract: [Objectives] The mounting structure of a ship is an important piece of equipment for vibration reduction. In order to improve the vibration reduction effect of the mounting structure, damping material is often pasted onto its surface. **[Methods]** The vibration level difference of the mounting structure's acceleration parameters is defined as the evaluation index. Based on the Solid Isotropic Material with Penalization (SIMP) model, a topological optimization model of free damping material distribution is established. In the optimization formulation, the constraints ensure the optimal distribution of damping material on the surface of the structure while the total volume of damping material is certain. Finally, based on the Finite Element Model (FEM) of the structure, the optimal damping materials for the laying scheme are ascertained. The results of topological optimization are tested and verified by the model test. **[Results]** The optimal free damping material distribution of a mounting structure is obtained. **[Conclusions]** The research results have value as a reference for the design of the mounting structures of ships and the application of composite materials.

Key words: free damping; topological optimization; vibration level difference; experimental verification

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0 Introduction

With the improvement of people's requirements for the comfortableness of ships and acoustic stealth of naval ships, designers have carried out systematic research on the structural vibration reduction problem of the mounting structure, which connects the power equipment and the hull, under the action of harmonic excitation[1-2]. To improve the vibration reduction effect on the mounting structure, various new materials including composites have been widely applied. Jiang et al.[3] carried out research on the optimization of material selection for marine damping materials based on the modal strain energy method and the structural dynamics optimization theory. Lv et al.[4] verified that the vibration isolation effect of steel-composite mounting structure was better than that of steel mounting structure based on the power flow level difference parameter. The evaluation for the vibration reduction effect of mounting structure mainly includes the parameters such as power flow, acceleration, and their differences[5], where the vibration level difference is the ratio of the vibration responses at the input and output terminals of the elastic support (mounting structure) of the system under the condition of elastic mounting, and the vibration level difference parameter has been widely applied because it is easy to be measured[6].

Damping material is a new kind of composite. It uses the viscoelasticity of macromolecular materials to convert the mechanical energy of vibration into thermal energy for consumption, so as to realize the target of vibration and noise reduction. Damping material can be divided into two kinds, i.e., constrained damping material and free damping material. The laying thickness, location, and laying method of the

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material directly affect its vibration reduction effect. Zhu et al.[7] pasted the isobutylene isoprene rubber damping material on the mounting structure, and compared the effects of the laying position and thickness on the vibration insertion loss. Shi et al.[8] analyzed the vibration characteristics of the cylindrical shell structure, part of which was pasted with the constrained damping material, and discussed the effect of the damping material distribution on the vibration characteristics. Saravanan et al. [9] studied the vibro–acoustic characteristics of the stiffened cylindrical shell pasted with damping material, and discussed the effect law of the damping material configuration on the vibration response.

The problem of damping material laying is a topological optimization problem. Its basic principle is to convert the problem of searching for the optimal topological structure into the problem of searching for the optimal material distribution in the given design area. In 1988, Bendsøe et al.[10] carried out research on the structural topological optimization based on the homogenization theory, and opened up a new field of topological optimization of continuous structure. Zheng et al.[11] studied the layout optimization of damping materials, aiming at the minimization of the structural vibration energy. Kang et al.[12] studied the damping material distribution with the topological optimization theory, taking the vibration characteristics of the specified position as references. Takezawa et al.[13] used the complex mode superposition method under the state space to carry out topological optimization on the damping materials, taking the minimization of the damping materials as the constraint and the maximization of the first–order modal frequency of the structure as the optimization target. Zheng et al.[14] optimized the beam structure, part of which was pasted with constrained damping materials, taking the minimization of the vibration energy as the target. Yun et al.[15] carried out topological optimization on the structure with viscoelastic damping materials under the action of variable excitation force, and realized the minimization of the structural vibration parameters under the condition that the amount of the damping materials is certain.

Although people have made some achievements in the topological optimization of damping materials, most of the research achievements are restricted to theoretical calculation and numerical analysis. There are some limitations in solving practical engineering problems. Based on this purpose, in this paper, we will take the laying problem of damping materials on the mounting structure of a ship as the research object to carry out topological optimization research on the free damping materials, and used the model test method to verify the optimization results, so as to support the results of numerical calculation.

1 Vibration level difference of the structural dynamic response

1.1 Structural dynamic response equation

It is supposed that the connection of the damping material and the elastic mounting structure is ideal. When the external harmonic excitation force is applied on the structure, the dynamic response differential equation within the structural domain Ω^{S} can be written as follows:

$$\boldsymbol{M}\ddot{\boldsymbol{u}} + \boldsymbol{C}\dot{\boldsymbol{u}} + \boldsymbol{K}\boldsymbol{u} = \boldsymbol{f}(t) \tag{1}$$

where $\boldsymbol{u}$ is the displacement vector of node; $\boldsymbol{M}$ is the mass matrix, and $\boldsymbol{M} = \boldsymbol{M}_{\mathrm{s}} + \boldsymbol{M}_{\mathrm{d}}$; $\boldsymbol{K}$ is the structural stiffness matrix, and $\boldsymbol{K} = \boldsymbol{K}_{\mathrm{s}} + \boldsymbol{K}_{\mathrm{d}}$; $\boldsymbol{C}$ is the viscous damping matrix, and $\boldsymbol{C} = \boldsymbol{C}_{\mathrm{s}} + \boldsymbol{C}_{\mathrm{d}}$; and $\boldsymbol{f}(t)$ is the external excitation force. Among them, $\boldsymbol{M}_{\mathrm{s}}$, $\boldsymbol{K}_{\mathrm{s}}$, and $\boldsymbol{C}_{\mathrm{s}}$ are the mass matrix, stiffness matrix, and damping matrix of the elastic mounting structure, respectively, while $\boldsymbol{M}_{\mathrm{d}}$, $\boldsymbol{K}_{\mathrm{d}}$ and $\boldsymbol{C}_{\mathrm{d}}$ are the mass matrix, stiffness matrix, and damping matrix of the damping material layer, respectively. If $\boldsymbol{M}_{\mathrm{d}}$, $\boldsymbol{K}_{\mathrm{d}}$ and $\boldsymbol{C}_{\mathrm{d}}$ are 0, there is no damping material on the elements of elastic mounting structure. Since the loss factor of the damping material is much larger than that of the elastic mounting structure, at this moment, the damping matrix $\boldsymbol{C}$ in Eq. (1) can be written as $\boldsymbol{C} = \boldsymbol{C}_{\mathrm{d}}$.

1.2 Vibration level difference

The vibration isolation effect of the mounting structure can be estimated by the calculation of the acceleration level difference of the reference points at the input/output terminal of the mounting structure. The acceleration level difference (unit: dB) is defined as follows:

$$L_{\mathrm{r}} = 10\lg\frac{a_{\mathrm{up}}}{a_{\mathrm{down}}} \tag{2}$$

where a_{up} and a_{down} are the acceleration values at the reference points of the input terminal and the output terminal of the mounting structure within the calculated frequency range.

2 Topological optimization equation of damping materials

2.1 Damping material overview

When being applied for structural vibration reduction, the damping materials with a certain thickness are usually pasted onto the surface of the elastic mounting structure. Fig. 1 shows the laying diagram of damping materials on the elastic mounting structure.

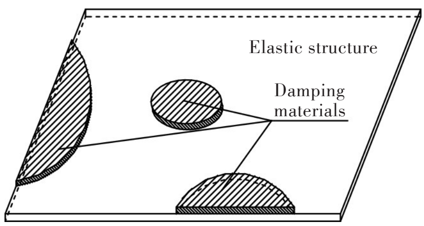

Fig.1 Sketch of damping material distribution

In the topological optimization of damping materials, the elastic mounting structure is defined as the non-design area, and the damping material layer is the design area. Since the stiffness of the elastic mounting structure is much greater than that of the damping materials, when the structural stiffness is calculated, the eccentric effect caused by the laying of damping material layer can be neglected, i.e., it is assumed that the neutral surface of the structure after the damping materials are pasted does not change.

2.2 Topological optimization of damping materials based on the vibration level difference

As shown in Fig. 1, the problem of the topological optimization of damping materials is an integer optimization problem for the 0-1 discrete variables. If damping material is pasted in the design domain, it is expressed as 1; if there is no damping material in the design domain, it is 0. The common method for the topological optimization of damping materials includes the homogenization method, the variable density method, the topological function description method, the variable thickness method, etc. In the variable density method, the application of the Solid Isotropic Material with Penalization (SIMP) method is very extensive. In this paper, we establish the mathematical model of topological optimization for damping material laying based on the SIMP method. By introducing the penalty factor, we establish a nonlinear relationship between the elastic modulus of materials and the relative density of the elements. When the density value of the design variable is the median between 0 and 1, it will be punished by the penalty factor to make it close to the two ends, so that the model can better approach to the optimization model of 0-1 discrete variables.

According to the design goal of the mounting structure, we can establish the topological optimization equation aiming at the optimal target of the vibration level difference at the reference points. We define the relative density of the damping materials in the design area as the design variable. Under the condition that the total amount of damping materials is constant, we can obtain the maximum acceleration level difference, i.e., $\mathrm{Max}(L_r) = 10\lg\dfrac{a_{\mathrm{up}}}{a_{\mathrm{down}}}$.

3 Example analysis

We take the mounting structure of the main engine of a ship for example to carry out the numerical analysis and test verification for the topological optimization of the damping material distribution.

3.1 Model description

The design parameters of the mounting structure of the main engine of the ship are as follows: the length is 0.68 m, the width is 0.64 m, and the height is 0.53 m. The mounting structure is composed of panel, transverse ribbed plate, longitudinal ribbed plate, bottom transverse ribbed plate, etc. The finite element method is applied for the finite element modelling of the mounting structure. The finite element model of the mounting structure includes 3 340 plate elements and 3 502 nodes (see Fig. 2).

In the mounting structure, the material of the mounting structure is steel. In order to improve the vibration reduction effect of the mounting structure,

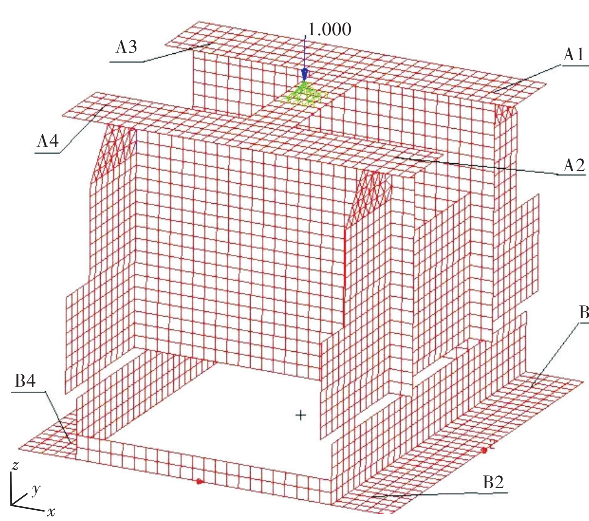

Fig.2 Finite element model of the mounting structure

SA-3 damping material with the thickness of 1.0 mm is pasted on the outer surface of the transverse ribbed plate and longitudinal ribbed plate. The mechanical properties of the steel and damping materials are shown in Table 1.

Table 1 The mechanical properties of materials

Material	Elastic modulus E / MPa	Poisson's ratio	Loss factor η	Density /(kg·m^{-3})
Steel	210 000	0.30	0.001	7 800
Damping material	75	0.49	0.750	1 500

3.2 Analysis of the structural dynamic response

Before calculating the dynamic response of the mounting structure, we first use the measured data of low order modes of the mounting structure and their corresponding frequencies to correct the model of the mounting structure based on parametric model modification so as to improve the precision of the finite element model of the mounting structure. The numerical calculation of the dynamic response of the mounting structure is carried out by applying harmonic excitation force on the top of the mounting structure. Combined with the experimental scheme of the mounting structure model, we separately define the reference point group A (points A1, A2, A3, and A4) and the reference point group B (points B1, B2, B3, and B4) as the input terminal and output terminal (see Fig. 2).

We adopt the MSC.Patran/Nastran software to carry out dynamic response analysis for the mounting structure. The calculation frequency ranges from 1 Hz to 500 Hz, and the calculation step is 2 Hz. We define the acceleration level difference of the reference point groups at the input and out terminals as the evaluation index of vibration reduction. The acceleration value of the reference point group A at the input terminal is $a_A = \sqrt{\dfrac{(a_{A1})^2 + (a_{A2})^2 + (a_{A3})^2 + (a_{A4})^2}{4}}$, and the acceleration value of the reference point group B at the output terminal is $a_B = \sqrt{\dfrac{(a_{B1})^2 + (a_{B2})^2 + (a_{B3})^2 + (a_{B4})^2}{4}}$.

Fig. 3 shows the acceleration values of the reference point groups at the input terminal and the output terminal. Fig. 4 shows the acceleration level difference between the reference point groups at the input terminal and the output terminal.

From Fig. 3, we know that, in the high frequency band, the acceleration value at the input terminal is far larger than that at the output terminal; in the low frequency band, the difference of the acceleration values at the input and output terminals is small. From Fig. 4, we know that, in the high frequency band, the vibration acceleration amplitude at the input terminal is large, and the vibration level differences are all large, which is more than 10 dB in most of the frequency bands and even more than 15 dB in some frequency bands. In the low frequency band, the vibration level differences of some frequency bands are negative, which indicates that, in this frequency band, the acceleration value of the output terminal is larger than that of the input terminal. Fig. 4 shows that, in the low frequency band, the vibration reduction effect of damping materials needs to be improved.

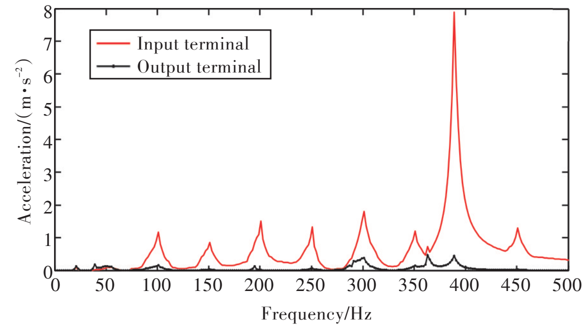

Fig.3 Acceleration values of the evaluation points

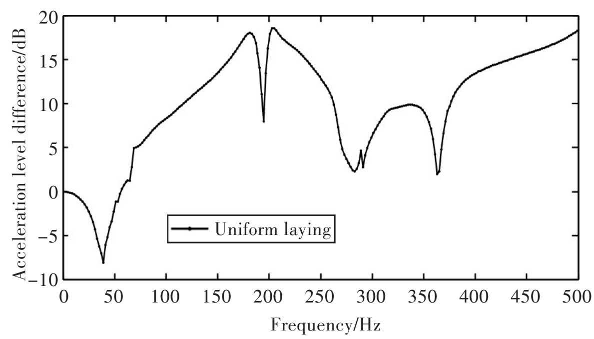

Fig.4 Acceleration level difference of the evaluation points

4 Damping material optimization of the mounting structure

4.1 Optimization equation of the vibration level difference of the mounting structure

From Figs. 3 and 4, we know that, to improve the vibration reduction effect of the mounting structure, the vibration level difference of the typical reference points at the frequencies ranging from 1 Hz to 500 Hz needs to be optimized so as to improve the maximum vibration level difference and improve the vibration reduction effect of the mounting structure.

In engineering practice, damping materials are often pasted on the surfaces of the transverse and longitudinal ribbed plates. We define these two regions as the design area of topological optimization for damping material, and there are 2 240 elements in total. Combining the vibration level difference equation, we define the relative density of the damping materials as the design variable and the damping material amount as the volume constraint condition. Then, the topological optimization equation for the damping materials of the mounting structure can be written as follows:

$$\begin{aligned}&\text{Find} \quad x=\{x_1, x_2, \cdots, x_i\}^{\mathrm{T}}\ ;\ i=1, 2, \cdots, 2\,240\\ &\text{Max} \quad L_{\mathrm{r}}=10\lg\frac{a_{\mathrm{up}}}{a_{\mathrm{down}}}\\ &\text{s.t.} \quad \boldsymbol{M}\ddot{\boldsymbol{u}}+\boldsymbol{C}\dot{\boldsymbol{u}}+\boldsymbol{K}\boldsymbol{u}=\boldsymbol{f}(t)\\ &\qquad \sum_{i=1}^{2\,240} x_i V_i - V_0 \leqslant 0,\ V_0=\zeta\sum_{i=0}^{2\,240} V_i\\ &\qquad 0<x_{\min}\leqslant x_i\leqslant 1\\ &\qquad g(x_i, v_i)\leqslant 0 \end{aligned} \tag{3}$$

where x_i is the relative density of the i^{th} element in the damping layer, i.e., the design variable; $x_{\min}$ and 1 are the upper and lower constraints of the design variable x_i. In order to prevent the stiffness matrix singularity in the penalty process, we set $x_{\min}$ = 0.001; L_{r} is the acceleration level difference of the reference point groups at the input/output terminal of the mounting system, i.e., the objective function; V_0 is the volume of damping material before topological optimization; V_i is the volume of the i^{th} element of the damping layer; ζ is the volume percent; $g(x_i, v_i)$ is the other constraint conditions, e.g., the parameters such as the natural frequency of the structure, the structural deformation, and the structural dynamic stress. Combined with the mechanical parameters of the SA–3 material, in the initial design, the mass of damping material is about 1.5 kg.

To solve Eq. (3), we establish an artificial damping penalty model based on the SIMP model. The penalty model can be used to obtain the intermediate density of the damping material so as to obtain a more explicit topological structure for the damping material. The elastic modulus matrix, mass matrix, and stiffness matrix of the elements in the damping material layer after penalty are

$$\boldsymbol{E}_i=x_i^p\boldsymbol{E}_i^*\ ,\ \boldsymbol{M}_i=x_i^p\boldsymbol{M}_i^*\ ,\ \boldsymbol{K}_i=x_i^p\boldsymbol{K}_i^* \tag{4}$$

where $\boldsymbol{E}_i^*$, $\boldsymbol{M}_i^*$ and $\boldsymbol{K}_i^*$ are the elastic modulus matrix, mass matrix, and stiffness matrix of the i^{th} element in the damping material layer; p is the penalty factor, and $p\geqslant 0$. In this paper, $p=3$.

Meanwhile, after the introduction of the penalty factor, the intermediate density of the damping materials can be punished so that the relative density of the damping materials can be as close to 0 and 1 as possible. Then, the total damping matrix of the structure can be written as follows:

$$\boldsymbol{C}_d=\sum_{i=1}^{2\,240}(\gamma(x_i)^{p_1}\boldsymbol{M}_i^*+\eta(x_i)^{p_2}\boldsymbol{K}_i^*) \tag{5}$$

where γ and η are the Rayleigh damping coefficients when the relative densities of the damping materials are all 1, and the penalty factors satisfy $p_1\geqslant 0$, $p_2\geqslant 0$. In this paper, $p_1=p_2=3$.

Combining Eqs. (3)–(5), we can carry out the topological optimization for the damping material distribution of the mounting structure so as to obtain the optimal distribution of the damping materials on the surface of the mounting structure. In the optimization process, we carry out the secondary development on the iSIGHT integrated optimization platform. We adopt the genetic algorithm, which satisfies the discrete variable problem and has good global search ability for optimal solution, to carry out the topological distribution optimization for damping materials, so as to achieve the vibration reduction target of the mounting structure.

4.2 Optimization of damping materials

In the topological optimization process for damping materials of the mounting structure, to simplify the calculation, we assume that the laying thickness of the optimized damping materials is equal, i.e., optimization of equal thickness. Through topological optimization, we can obtain the distribution of the damping materials on the mounting structure. The optimized mounting structure model is shown in Fig. 5. In Fig. 5, the red area is the laying area of damping materials, the distribution area of the optimized damping materials is half of the area before optimiza-

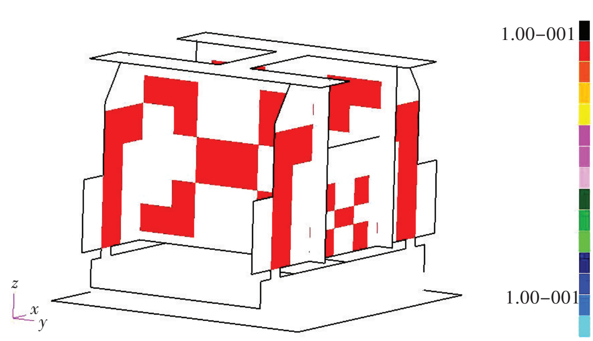

Fig.5 Distribution of damping materials on the mounting structure

tion, and the thickness of the optimized damping materials is 2.0 mm.

Fig. 6 shows the acceleration values of the reference point groups at the input terminal and the output terminal after optimization. Fig. 7 shows the acceleration level difference between the reference point groups at the input terminal and the output terminal after optimization.

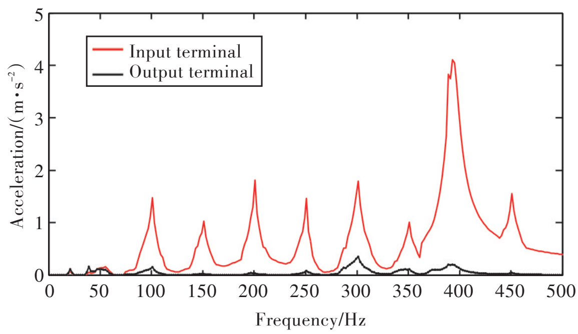

Fig.6 Acceleration value of evaluation points after optimization

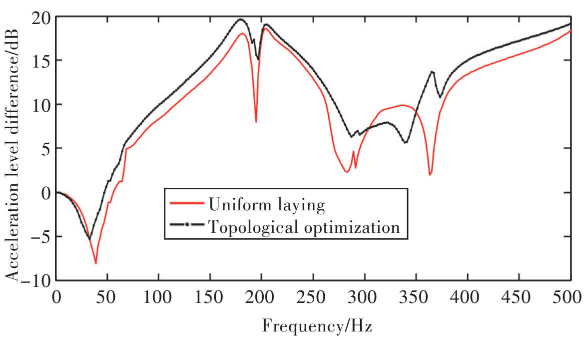

Fig.7 Acceleration level difference of evaluation points after optimization

Comparing Figs. 3 with 6, we know that, through the topological optimization of the damping materials, the output of the acceleration value at the output terminal of the mounting structure obviously reduces. Meanwhile, through the comparison of Figs. 4 and 7, we know that, whether in the high frequency band or in the low frequency band, the values of the vibration level differences all have a certain degree of increase after the topological optimization of damping materials. This shows that the vibration reduction effect of the mounting structure after topological optimization obviously increases, and the optimization is effective.

5 Model test

5.1 Experimental model

In order to verify the topological optimization results of the damping materials of the mounting structure, we carry out model test, where the laying position of the damping materials is based on the topological optimization results. The experimental model is made of welded all-steel materials (see Fig. 8).

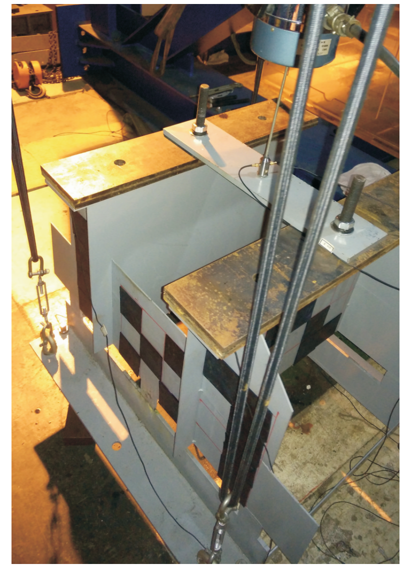

Fig.8 Damping materials distribution experiment

The experimental equipment and instruments include MTS structural dynamic loading system, SCADSIII dynamic data analysis and acquisition system, multi-channel dynamic strain meter, PCB force sensor, etc. By measuring the vibration acceleration of typical reference points, we obtain the vibration reduction effect of the mounting structure model in the typical frequency domain. In the model experiment, the position for the finite element calculation is referred as the position of the measuring points.

5.2 Comparison of the experimental results

Fig. 9 shows the acceleration level difference between the reference points at the input terminal and the output terminal after the topological optimization of damping materials and the model experiment are implemented.

From Fig. 9, we know that, the numerical calculation results of the topological optimization of damping materials agree well with the values of the vibration level difference of the model experiment. This

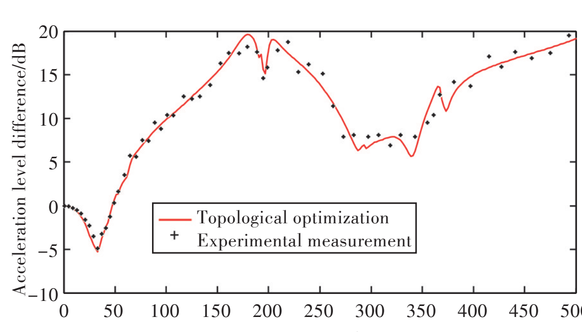

Fig.9 Comparison of acceleration level difference

verifies the validity of the numerical calculation.

6 Conclusions

Aiming at the improvement of the vibration reduction effect of the mounting structure, in this paper, we establish the topological distribution optimization equation for damping materials, carry out numerical analysis and experimental verification, and obtain the following main conclusions:

1) By carrying out the topological distribution optimization for damping materials, the utilization efficiency of damping materials is improved.

2) By model test, the validity of numerical calculation and the feasibility of optimization are verified.

3) In practical engineering applications, if the topological optimization of damping materials is integrated with the thickness optimization, a better vibration reduction effect will be achieved.

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船舶基座阻尼材料敷设优化及实验研究

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摘　要：[目的]船舶基座结构是船舶减振的重要设备，为提高基座结构的减振效果，在结构表面敷设阻尼材料是常用手段。[方法]以基座结构的加速度振级落差为评价指标，应用各向正交惩罚材料密度法(SIMP)，建立自由阻尼材料的拓扑优化数学模型。在优化模型中，其约束条件是确保在阻尼材料总使用量一定的情况下实现阻尼材料在基座结构表面的最优分布。最后，以某型船的主机基座为例，在建立的有限元模型的基础上，开展基座结构阻尼材料拓扑优化的数值计算，并利用基座模型实验的方法对拓扑优化数值计算结果进行实验验证。[结果]经验证，获得了阻尼材料的最优敷设方案。[结论]所得成果对船舶基座结构设计和复合材料的应用具有一定的参考价值。

关键词：自由阻尼；拓扑优化；振级落差；实验验证