Multi-objective optimization design of lens barrel for the carbon dioxide detector

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Abstract: To meet lens barrel′s requirements for high stiffness, lightweight and good thermal environment adaptability, multi-objective optimization that combines the compliance and first-order natural frequency under multiple load cases was studied. Firstly, based on the SIMP (Solid Isotropic Materials with Penalization) method, weight sum method and target-vector approach were adopted to define objective function of multi-objective topology optimization. Then lens barrel was optimized. The optimization procedure consisted of two main steps of topology optimization and thickness optimization. During topology optimization, with stiffness and maximum first-order natural frequency as the objective function, a creative frame layout was obtained. After approximating the result of topology optimization as a thin-walled structure, a new lens barrel 3-D model was established. The approximated thickness of lens barrel were optimized to minimize its mass. The stiffness and natural frequency obtained by topology optimization were set as design constraints for this thickness optimization. The optimization results show that the mass of final lens barrel decreases to 12.46 kg from 18.63 kg, the maxim deformation subjected to gravity load decrease to 0.001 9 mm from 0.013 mm, the maximum deformation caused by thermal load decreases to 0.062 mm from 0.098 mm, and the first-order natural frequency increases to 189.83 Hz from 65.6 Hz. With this optimization method, the design period is effectively shortened, and the performance of lens barrel is enhanced, which shows that optimized lens barrel can meet the system design requirements.

Key words: optimization; multi-objective; topology; thickness optimization; lens barrel;
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二氧化碳探测仪镜筒结构的多目标优化设计

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摘 要: 为满足镜筒结构高强度、轻量化及良好的热环境适应性的要求,研究了多工况下同时考虑结

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构刚度和一阶固有频率的多目标优化问题。首先,基于变密度连续体结构拓扑优化方法,采用加权和法和靶向量法定义了多工况下多目标优化函数;然后进行了镜筒的优化设计。整个优化设计包括拓扑优化和厚度优化两部分。在拓扑优化中,以刚度和频率最大为目标函数,得到了满足要求的镜筒结构材料的最佳分布,并以此为依据,完成了镜筒结构的三维建模;以拓扑优化所得结构的柔度值和一阶频率为约束,质量最小为目标函数,对镜筒结构各厚度进行了优化。优化设计结果表明,镜筒质量由 18.63 kg 减少至 12.46 kg;重力作用下最大变形由 0.013 mm 减少至 0.001 9 mm;热载荷作用下最大变形由 0.098 mm 减少至 0.062 mm;一阶固有频率从 65.6 Hz 提高至 189.83 Hz。该优化设计方法有效缩短了设计周期,提高了镜筒结构的性能,满足了系统设计要求。

关键词:优化; 多目标; 拓扑; 厚度优化; 镜筒

0 Introduction

Carbon dioxide detector is used to monitor carbon dioxide globally from space, to ensure that its performance and structure have enough strength and stiffness to bear the complicated dynamics load and impacting load. Because of the special working environments and stringent requirements, development of structure components with features of high stiffness, lightweight and dimensional stability in the wide temperature range has become a significant task for space applications [1]. Lens barrel is the supporter of various optical components, which determines their axial and radial position. In the traditional lens barrel design, its structural form is largely based upon a benchmark design or on previous designs, then it′s checked and modified. One of the biggest disadvantages of this design is that when changes occur in the testing phase, it′s very hard and costly to modify the design model. What can do is just local adjustments [2]. The root causes of this problem are in conceptual design stage. The material layout of structure is mainly based on designer′s experience. Besides, lens barrel is a thermal-structure coupling system, because of the multi-physical coupling, the optimal topology for lens barrel is very far different from the experience and intuition. Experienced designer can estimate approximately the topology of optimal structure under single physical field, but in multi-physical field, the experiences always do not work. Continuum topology optimization can solve the deficiencies of traditional design method, and provide designer with a better structure layout at the conceptual design stage. Since topology optimization is usually invoked in the conceptual design stage, it becomes a rational method for innovative structural designs.

Continuum structural topology optimization nowadays is mainly concentrated on the single objective optimization [3-6]. Some papers about lens barrel design and optimization are on its limitations[7-8]. Firstly, single objective optimization can´t always make the structure optimal in practical engineering applications. Secondly, in the previously design, the thermal analysis is mostly made in the later stages of design to check whether it satisfies the requirements, the thermal deformation and stress is seldom considered in the early design stage and taken as a design variables, consequently, not only it wastes time, but also a optimal structure may not be got. Based on the above problems, in this paper, multi-objective topology optimization that combines total compliance and first-order natural frequency of lens barrel for carbon dioxide detector subjected to thermal load and gravity load is discussed and analyzed by using the Altair OptiStruct software. During the topology optimization, a creative lens barrel layout with stiffness and first-order natural frequency maximization is obtained, then the thickness optimization is found to have a minimum lens barrel mass without affecting the performances obtained from topology optimization.

1 Fundamental concepts about topology optimization

Topology optimization is a mathematical technique that produces an optimized shape and material distribution for a structure within a given package space, with a view to get optimum performance [9] with the least amount of material. The basic method of continuum structure is to discretize the domain into a finite element mesh and calculate material properties for each element. Then, based on certain algorithm, parts area of the design domain are deleted. The numerical analysis based topology optimization methods for continuum can be broadly classified into two categories: macroscopic approaches and microscopic approaches. The former one includes bubble method and recently developed level set method. The latter one includes varied thickness method, homogenization method, evolutionary structural optimization method, and SIMP method. The SIMP method is adopted in the paper. In the SIMP formulation, the material density for each element is taken as a design variable, varying continuously between 0 and 1, which represents the state of void and solid respectively, and the stiffness of the material is assumed to be linearly dependent on the density. In general, the optimal solution of problems, using both formulations mentioned above, involves large gray areas of intermediate densities in the structural domain. Such solutions are not meaningful when the topology of a given material is looked for. Therefore, techniques are needed to be introduced to penalize intermediate densities and to force the final design to be represented by densities of 0 or 1 for each element. The used penalization technique is "power law representation of elasticity properties", which can be expressed for any solid 3-D or 2-D element as follows[10]:

$$\boldsymbol{k}_e=(x_e)^p\boldsymbol{k}_0 \tag{1}$$

where $p\geqslant 2$ is the penalty factor, $\boldsymbol{k}_e$ and $\boldsymbol{k}_0$ represents penalized and the real stiffness matrix of an element, x_e is the element density. Compared with other approaches, SIMP method is more general and requires less design variables, so it improves the calculation process and solution efficiency.

2 Mathematical model of topology optimization

2.1 Formation of topology optimization for stiffness

In the structure design, conventionally, the design goal is to increase the stiffness. The relationship between stiffness and compliance can be expressed as[11]

$$\boldsymbol{C}=\boldsymbol{U}^{\mathrm{T}}\boldsymbol{F}=\boldsymbol{U}^{\mathrm{T}}\boldsymbol{K}\boldsymbol{U} \tag{2}$$

where $\boldsymbol{C}$ is the compliance of the structure, $\boldsymbol{K}$ is the global stiffness matrix, $\boldsymbol{U}$ is vector of nodal displacement. With the compliance minimum as the objective function, volume as constraint, it can be expressed as

Find:

$$\boldsymbol{X}=\{x_1,x_2,x_3,\cdots,x_e\},\quad e=1,2,\cdots,N$$

Min:

$$C(\boldsymbol{X})=\boldsymbol{U}^{\mathrm{T}}\boldsymbol{K}\boldsymbol{U}=\sum_{e=1}^{N}x_e\boldsymbol{u}_e^{\mathrm{T}}\boldsymbol{k}_e^{0}\boldsymbol{u}_e \tag{3}$$

Subjected to: $V(\boldsymbol{X})=f_vV_0$

$$0\leqslant x_{\min}\leqslant x_e\leqslant x_{\max}\leqslant 1$$

where $C(\boldsymbol{X})$ represents the total compliance, x_e is the density of e-th element, $\boldsymbol{u}_e$ is the elemental displacement vector, $\boldsymbol{k}_e^{0}$ is the stiffness matrix of the e-th element when it is made of the solid material, N is the total number of elements, V is the material volume, V_0 is the design domain volume, f_v is the prescribed volume fraction. In order to avoid numerical difficulties caused by non-positive definite stiffness matrix, a lower-bound limit $x_{\min}>0$ is imposed on the material density.

2.2 Formulation of topology optimization for first-order natural frequency

When structure is in vibration, the amplitude of higher modes is smaller compared with lower modes, so lower modes indicate the dynamic characteristics of structure[12-13]. Taking the first order natural frequency maximum as the objective function, volume as constraint, it can be expressed as

Find: $\boldsymbol{X}=\{x_1,x_2,x_3,\cdots,x_e\}$, $e=1,2,\cdots,N$

Max:

$$W(\boldsymbol{X})^2=\frac{\boldsymbol{\theta}^{\mathrm{T}}\boldsymbol{k\theta}}{\boldsymbol{\theta}^{\mathrm{T}}\boldsymbol{M\theta}} \tag{4}$$

Subjected to: $(\boldsymbol{K}-W^2\boldsymbol{M})\boldsymbol{\theta}=0$

$$\boldsymbol{M}=\sum_{e=1}^{N}\boldsymbol{m}_e$$

$$0\leqslant x_{\min}\leqslant x_e\leqslant x_{\max}\leqslant 1$$

where $\boldsymbol{M}$ is global mass matrix, W is the first mode frequency, $\boldsymbol{\theta}$ is the eigenvector, $\boldsymbol{m}_e$ is the element matrix.

2.3 Definition and formulation of multi-objective topology optimization

Multi-objective optimization is also called multi-criteria optimization or vector optimization, the basic mathematical model is as follows[14]

$$F(\boldsymbol{X})=\min(F_1(\boldsymbol{X}),F_2(\boldsymbol{X}),F_3(\boldsymbol{X})\cdots) \tag{5}$$

Subjected to: $\boldsymbol{X}=\{x_1,x_2,x_3,\cdots,x_e\}$, $e=1,2,\cdots,N$

$$a_i\leqslant x_i\leqslant b_i$$

$$g_i(\boldsymbol{X})\leqslant 0$$

$$H_k(\boldsymbol{X})=0$$

There are many methods to solve the multi-objective optimization, like weight sum method, vector evaluated genetic algorithm, lexicographic ordering, target-vector approach and so on. Weight sum approach is often used. Its essence is to construct a function, known as merit function, which can turn n number of sub-objective functions into a single objective function, and the merit function will be optimized as follows:

$$\min F(\boldsymbol{X})=a_1f_1(\boldsymbol{X})+a_2f_2(\boldsymbol{X})+\cdots+a_nf_n(\boldsymbol{X}) \tag{6}$$

where $F(\boldsymbol{X})$ denotes the merit function, $f_n(\boldsymbol{X})$ is the sub-objective function, a_n is the weighted index, which presents the relative importance of different sub-objectives. Another method is target-vector approach. It sets each sub-objective an expected value, then make the deviation from the desired value as small as possible, the merit function can be represented as

$$F(\boldsymbol{X})=\sum_{e=1}^{N}\left(\frac{f_e(\boldsymbol{X})-f_e^{\Delta}}{f_e^{\Delta}}\right)^2 \tag{7}$$

where $f_e(\boldsymbol{X})$ denotes the sub-objection function, f_e^{Δ} is the ideal value of corresponding sub-objective. The merit function is ideally zero when $f_e(\boldsymbol{X})$ matches the ideal value.

In engineering practice, for some structures, when its natural frequency is maximum, the stiffness may not or when the stiffness is maximum, the natural frequency may not. In order to solve this problem, for a structure which needs stiffness and first-order natural frequency as maximum as possible under all working conditions, a multi-objective function as a merit function based on weight sum method and target-vector approach can be expressed as[15-16]

$$\min f(\boldsymbol{X})=$$

$$\left\{\left[\sum_{k=1}^{N}w_k\frac{C_k(\boldsymbol{X})-C_k^{\min}}{C_k^{\max}-C_k^{\min}}\right]^2+\left[w_{\Lambda}\frac{A_{\max}(\boldsymbol{X})-A(\boldsymbol{X})}{A_{\max}-A_{\min}}\right]^2\right\}^{1/2} \tag{8}$$

where $C_k^{\max}$ and $A_{\min}$ are the initial value of compliance and first natural frequency, $C_k^{\min}$ and $A_{\max}$ are the ideal value of compliance and first-order natural frequency, k denotes the number of working conditions, w_k and w_{Λ} are the weight index. Apparently, when $f(\boldsymbol{X})$ is zero, the stiffness and first-order frequency are maximization.

3 New lens layout design by topology optimization

Carbon dioxide detector is used to measure carbon dioxide in the three visible and near-infrared spectral ranges. It consists of a telescope and three spectrometer optical systems. One of the spectrometer optical systems is shown in Fig.1. Based on the characteristic of the optical system, the original model of lens barrel is designed as in Fig.2. As the original model is rational to some extent, topology optimization based on this model would be more intuitive and reasonable considering the manufacturing, assembling and so on. Gravity load and thermal load are the main loads for lens barrel on earth and in space

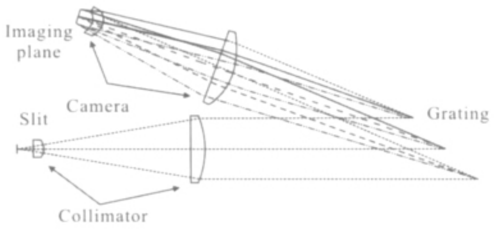


Fig.1 Schematic diagram of the spectrometer optical system

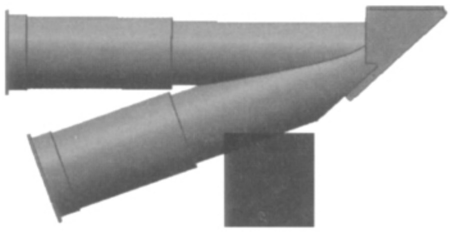


Fig.2 Original model of the lens barrel

environment respectively. In order to make the deformation minimum and dynamic characteristic better, the stiffness and first natural frequency should be as maximum as possible. The original model is analyzed by using the FE methods, the results show that the maximum deformation is 0.013 2 mm subjected to gravity load; the first mode frequency is 66.02 Hz, as shown in Fig.3; the maximum location defection between the collimator and grating derived from the 10 ℃ thermal load is 0.064 mm. To assure detector´s performance, the maximum tilt between the optical elements is required to be less than 10″, which is equivalent to the defection difference

$$\frac{10''\times 890}{57.32\times 3\ 600}\leqslant 0.043\ \text{mm} \tag{9}$$

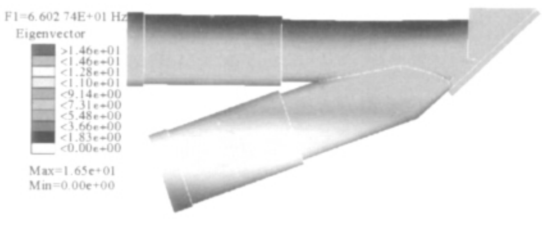


Fig.3 First-order natural frequency mode of lens barrel

The first-order natural frequency is required to be above 150 Hz. Apparently the original model can´t meet these specifications, so it must be optimized. In this paper, lens barrel optimization consists of two main steps: topology optimization and thickness optimization. For topology optimization, the objective is to increase the stiffness and first-order natural frequency by identifying the locations for ribs in the design domain. After approximating the result of topology optimization as a thin-walled structure, the approximated thicknesses of lens barrel are optimized to minimize its mass. Figure 4 illustrates the proposed procedure of the optimizations.

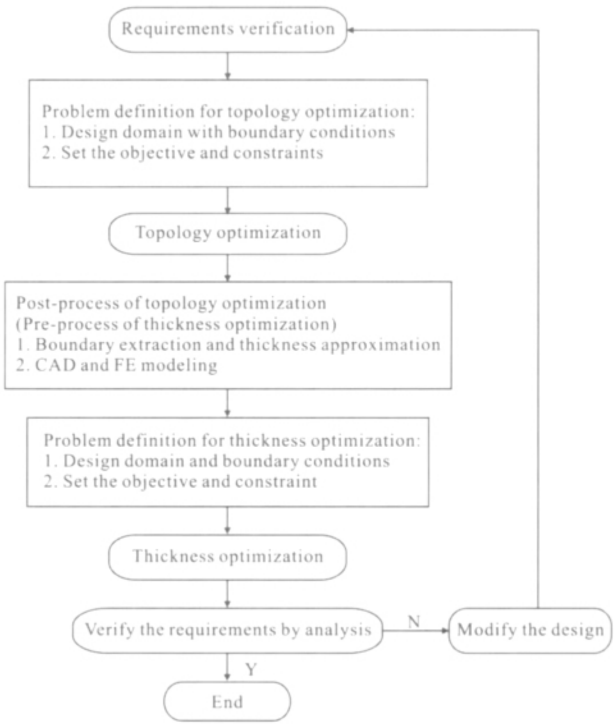
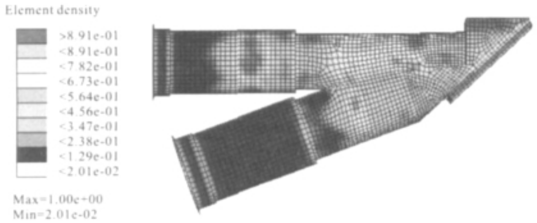


Fig.4 A design procedure for lens barrel optimization

As discussed before, for a structure whose stiffness is maximum, its first-order natural frequency may not. To verify this, Fig.5(a) and (b) shows the optimized topology when only the stiffness and the first-order natural frequency is considered as the objective function respectively. Regions that need reinforcement tend towards a density of 1, areas that don′t need reinforcement tend towards a density of 0. From the

(a) Plot of element density only considering the first-order natural frequency



(b) Plot of element density only considering the stiffness subjected to multiple loads

Fig.5 Topology optimization results with different objective function

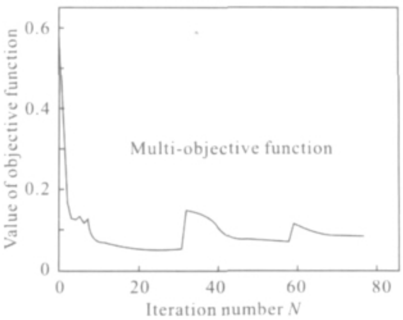


Fig.7 Iteration history of multi-objective function

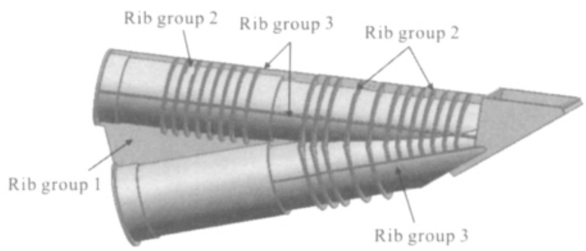


Fig.8 Post-processed lens barrel from Fig.6

two figures, the lo cations where the ribs should be added are very different. It indicates that for a structure when there are many sub-objectives, it´s very necessary to construct a multi-objective function. Based on this, using the Eq.(8), both the stiffness and the first natural frequency are considered f or the design objective. The result is shown in Fig.6. In comparison with the stiffness maximized results in Fig.5(b), the stiffness and frequency is greatly increased mainly by reinforcing the elements in the constraint and connection parts. Figure 7 shows the iteration history of multi-objective optimization. Based on the optimization result in Fig.6, the location where should be added ribs can be got. Considering the manufacturing and aesthetic property, a post-processed lens barrel is shown in Fig.8. There are some problems. Topology optimization only gives where the ribs should be added, but the ribs′ form, height and space between them are not known. It′s unavoidable that some extra ribs would be added to the lens barrel. To avoid this and make the structure more optimized, a second topology optimization is necessary. The design domain are rib group 1, rib group 2 and rib group 3 as shown in Fig.8, the objective function still uses the Eq.(8), with volume as constraint. Figure 9 is the optimization result with density threshold above 0.25. The transparent parts in Fig.9 denote the material would be removed. Compared with Fig.8, some ribs are deleted. This proves the necessity of the second topology optimization. Figure 10 shows the iteration history.

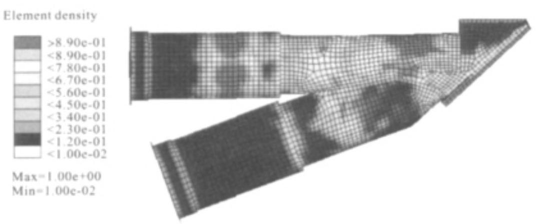


Fig.6 Topology optimization of lens barrel when considering both the stiffness and first-order natural frequency

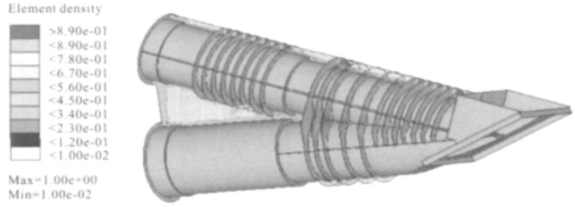


Fig.9 Result of the second topology optimization

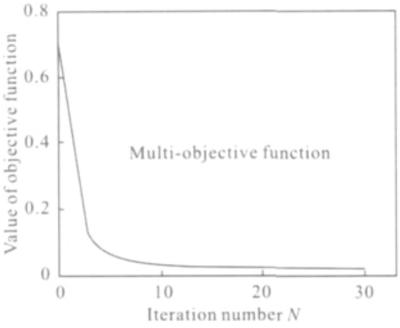


Fig.10 Iteration history of the second topology optimization

4 Thickness optimization

In Fig.1, 1, 2, 3, 4, 5 and 6 denote the design variables. The post-processed lens barrel in Figure 11 is reconstructed by using 6 shell members, each of

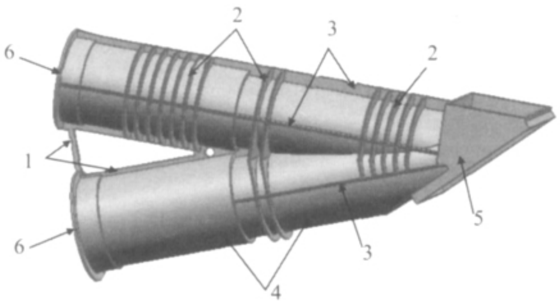


Fig.11 Post-processed lens barrel from Fig.9

which has its corresponding thickness. The formulation for present thickness optimization is given as

Find: $\boldsymbol{T}=\{t_1,t_2,t_3,t_4,t_5,t_6\}$

Min:

$$f(\boldsymbol{T})=\sum_{k=1}^{6}\rho A_k t_k \tag{10}$$

Subjected to: $\sigma_{\max}\leqslant\sigma_e$

$$g_1(\boldsymbol{T})=\frac{C}{C_0}-1\leqslant 0$$

$$g_2(\boldsymbol{T})=\frac{f}{f_0}-1\geqslant 0$$

Where t_k is the thickness of k-th shell of lens barrel, denoting the area, $f(\boldsymbol{T})$ is the total mass of lens barrel, σ_e is the maximum stress on lens barrel, $\sigma_{\max}$ is the allowable stress, C_0 and f_0 represent total compliance and first-order natural frequency of lens barrel in Fig.11 respectively, so the stiffness and first mode frequency is at least equal to thos e post-processed lens barrel from topology optimization in Fig.11. Thus the goal of thickness optimization is to reduce the mass of lens barrel as much as possible without deteriorating the performance of topology optimization result. Figure 12 is the iteration history of design variables. Figure 13 is the iteration history of mass. Table 1 is the result of optimized thickness of lens barrel after the thickness optimization. Considering the manufacture and cost, optimized thickness should be adjusted, so the thickness $\boldsymbol{T}=\{t_1,t_2,t_3,t_4,t_5,t_6\}=\{3,3,3,3,4,4\}$.

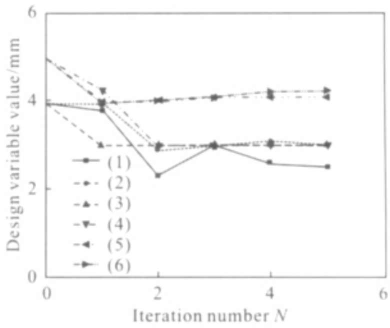


Fig.12 Iteration history of design variables

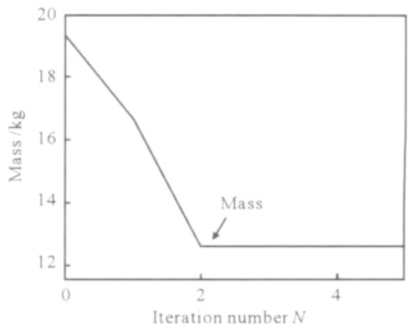


Fig.13 Iteration history of mass

In order to verify the performance of the final model in Fig.9 after topology and thickness optimization, it has been analyzed. The results show that the maximum deformation is 0.001 9 mm

Tab.1 Optimized thickness of lens barrel (mm)

	t_1	t_2	t_3	t_4	t_5	t_6
Initial thickness	4	4	4	5	5	5
Optimized thickness	2.5	3.1	3	3	4.1	4.2

subjected to the gravity load; the first-order natural frequency is 189.83 Hz. The maximum defection difference is 0.032 mm caused by the thermal load, all of these are less than specified value. Fig.14–16 are the corresponding plots. Table 2 is a comparison of the performances of the original model and the final model.

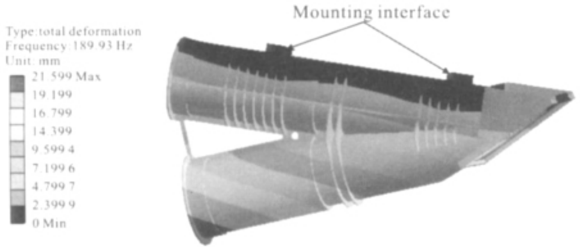


Fig.14 Model analysis of the final model

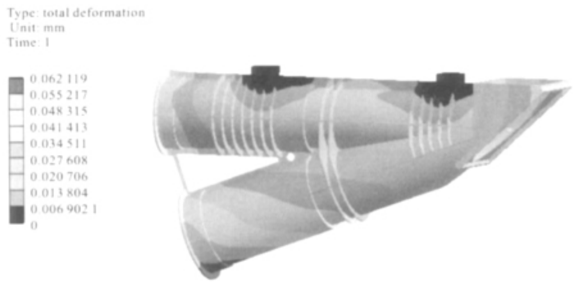


Fig.15 Displacement of the final model with gravity load

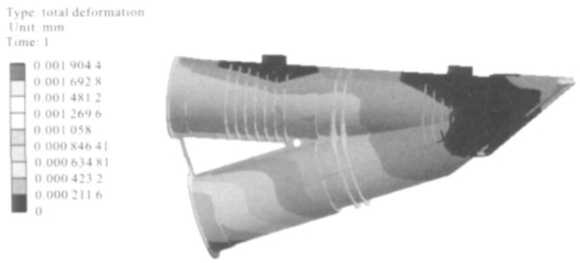


Fig.16 Deformation of the final model with 10 ℃ thermal load

Tab.2 Comparison of the performances of the original model and the final model

	Maximum deformation with gravity load/mm	Maximum deformation with thermal load /mm	First natural frequency /Hz	Mass /kg
Original model	0.013	0.098	65.6	18.63
Final model	0.001 9	0.062	189.83	12.46

5 Conclusion

A new lens barrel with increased stiffness and frequency was designed by applying topology and thickness optimization. Through multi-objective topology optimization, a creative layout was obtained as an optimum design to have maximized stiffness and first-order natural frequency. By applying thickness optimization, the approximated plate thicknesses from the result of topology optimization were adjusted to minimize total mass of the lens barrel without losing increased stiffness from the topology optimization. Compared with the original model, the mass of final lens barrel can decrease to 12.46 kg from 18.63 kg, the maximum deformation subjected to gravity load can decrease to 0.001 9 mm from 0.013 mm, the maximum deformation caused by thermal load can decrease to 0.062 mm from 0.098 mm, and the first-order natural frequency can increase to 189.83 Hz from 65.6 Hz. The numerical results indicate that after adopting the multi-objective topology and size optimization design method, not only the design period is shortened, but also the lens barrel weight is enhanced, which shows that lens barrel can meet the system design requirements.

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下期预览

激光距离选通成像及其非视域成像应用

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摘　要：非视域成像是国外近几年出现的一种新的成像模式，能够绕过拐角对难以直接观察的场景进行成像。介绍了基于激光距离选通成像技术的非视域成像模式，给出了国外的几个典型非视域成像实验及其结果。搭建了基于 532 nm 激光器和 ICCD 探测器的距离选通成像系统，以窗户玻璃和墙面瓷砖作为中介反射面，分别获得了 50 m 和 20 m 处目标的非视域图像。对实验结果进行分析认为，非视域成像的效果与中介反射面的反射特性有关，许多具有一定镜面反射特性的建筑材料均可作为中介反射面，用于非视域成像；非视域成像模式在城市巷战、公安侦察以及抗灾救援等领域展现出潜在的应用前景，是一种具有发展前景的新型光电成像模式。