



ARTICLE

Topology Optimization Design of a Hub Central Component Considering Fatigue Performance

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ABSTRACT: To address the design challenges of helicopter hub central components under high-performance requirements, this paper conducts safe-life topology optimization design research considering fatigue performance for rotor hub central components under multi-load conditions, combined with helicopter fatigue strength engineering design theory. For dealing with the issues of derivative calculation difficulties when directly considering fatigue constraints in existing topology optimization methods, this study establishes a mathematical formulation suitable for structural topology optimization of hub central components by combining modified structural safety fatigue limits based on iso-life curves. Then the sensitivity analysis of design variables is derived, and an optimization design model for typical main rotor hub central components is constructed. By controlling the safe-life equivalent stress of the hub central structure, the goal of managing structural fatigue life is achieved, providing new insights for long-life, high-reliability hub central component design. The paper presents a topology optimization case study of a typical five-arm hub central component, completes optimized structure reconstruction and fatigue strength analysis, which validates the effectiveness of the proposed methodology.

KEYWORDS: Hub central component; equivalent stress; fatigue performance; safe-life design; topology optimization

1 Introduction

Helicopters, renowned for their vertical takeoff/landing and hovering capabilities, demonstrate exceptional low-altitude maneuverability, making them indispensable in both military operations and civilian applications. As a critical dynamic component of the rotor system, the helicopter hub central component is subjected to complex multiaxial loading conditions during operational missions [1,2]. With future helicopters striving to overcome speed limitations inherent to conventional rotor aerodynamics and achieve high-speed flight, hub central components will face unprecedented demands for enhanced structural performance. Concurrently, stringent requirements for fatigue resistance and operational reliability must be rigorously addressed. Consequently, the efficient design of hub central components that simultaneously fulfill these competing performance criteria remains a formidable challenge for engineering practitioner.

Traditional hub central component design methodologies predominantly follow an empirical paradigm. Engineers initiate the process by developing preliminary configurations based on legacy design practices, constrained by interface compatibility requirements. This process then enters lengthy development cycles requiring repeated structural and strength analyses. However, two critical flaws persist: heavy dependence on



historical experience and time-consuming validation processes. With the rapid advancement of science and technology, particularly in computational methods, structural optimization design has become a vital tool for innovative structural design [3–5]. Among these methods, topology optimization has garnered significant attention in engineering due to its independence from empirical knowledge and its ability to generate novel configurations [6–8].

Since Bendsøe and Kikuchi [9] established the theoretical foundations of structural topology optimization, the methodology has undergone continuous refinement and has found broad application in modern aerospace structures [10–13]. Most practical implementations focus on enhancing structural stiffness and reducing mass. However, helicopter rotor components are subjected to sustained high-frequency dynamic loading during flight [14–16], where repetitive mechanical stress accumulation becomes the primary cause of failure [17–19]. Consequently, fatigue resistance emerges as the foremost design requirement for rotor hub assemblies. Yet, current topology optimization approaches generally fail to incorporate fatigue performance adequately, limiting their engineering applicability to rotor systems.

In the field of fatigue-constrained topology optimization, several representative methodologies have been proposed. Holmberg et al. [20] developed a decoupled framework that separates fatigue life evaluation from the optimization process, enabling lightweight design while maintaining independent fatigue life assessment. Jeong et al. [21] integrated dynamic fatigue and static failure criteria by transforming fatigue performance constraints into equivalent stress constraints, thereby satisfying both fatigue and static strength requirements within the optimization. Oest et al. [22] proposed a modeling approach for high-cycle fatigue damage under quasi-static loading, providing a practical numerical framework for implementing fatigue constraints in topology optimization. More recently, Suresh and Lindstrom [23] introduced an adjoint sensitivity analysis-based framework for high-cycle fatigue-constrained topology optimization, offering significant advantages in computational efficiency and scalability, and enabling direct integration into general-purpose finite element solvers.

Building upon these foundational approaches, researchers have further explored fatigue-oriented topology optimization strategies with enhanced engineering applicability. Zhang et al. [24] developed a method for topology optimization under non-proportional cyclic loading, calculating fatigue damage from stresses and applying the same strategies as in stress-constrained topology optimization to overcome singularity, localization, and high nonlinearity issues. Chen et al. [25] proposed a penalization formula to compute the cumulative fatigue damage of finite elements, using the principle of linear superposition and quasi-static finite element analysis to evaluate structural responses. Their results demonstrated that increasing the penalty factor produces smoother geometric transitions to reduce stress concentrations, decreases volume fraction, and yields more uniform damage distribution, thereby enabling more effective material allocation to prevent fatigue failure. Collet et al. [26] used a set of stress-based constraints to equivalently represent fatigue constraints, simplifying the minimum-volume design problem under both fatigue and compliance limits. Trudel et al. [27] modified the fatigue damage calculation by altering local stress, yield strength, or the intercept of the $S-N$ curve, and compared solutions to the fatigue minimization problem using gradient-based methods such as the Method of Moving Asymptotes (MMA) and Sequential Quadratic Programming (SQP).

Based on helicopter-specific safe-life design principles, this study proposes an equivalent dynamic stress criterion specifically tailored for the topology optimization of rotor hub components. Unlike conventional approaches that rely on P -norm stress aggregation or cumulative damage models, the proposed method directly incorporates a safety-fatigue limit corrected by iso-life curves into the optimization constraints. This enables explicit fatigue life control during the design process, improving both the interpretability and practical applicability of the optimization results. The framework offers a robust, engineering-oriented

approach for optimizing critical hub structures under realistic service loading conditions, ensuring compliance with widely adopted fatigue design standards in helicopter engineering while achieving significant weight reduction.

2 Optimization Problems

2.1 Geometry Model

The typical five-arm hub central component in helicopter rotor systems features dual mechanical interfaces: five mounting holes for blade connections and five lug fittings for damper attachments. This structural configuration transmits complex periodic loads from rotating blades through its load-bearing architecture. Fig. 1 illustrates the baseline hub central component configuration prior to topology optimization implementation.

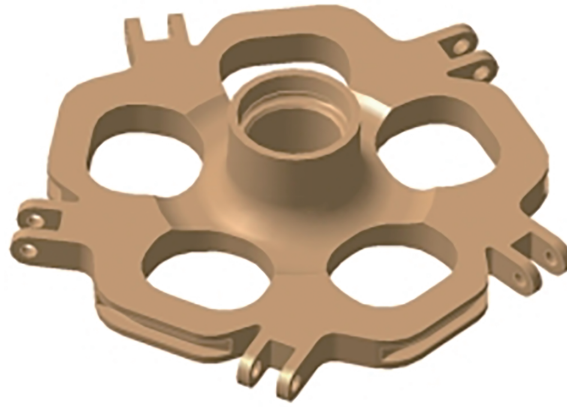


Figure 1: Structural model diagram of the baseline hub central component

To facilitate topology optimization of the hub central component, an initial design model (Fig. 2) was developed while preserving structural interface integrity. Based on the component's connection requirements, the baseline configuration was partitioned into design-enabled and design-restricted zones. Design-enabled zones (green regions in Fig. 2) permit structural topology modifications to optimize load transfer paths and material distribution. Design-restricted zones (blue regions in Fig. 2) maintain fixed geometries throughout the optimization process, primarily comprising mechanical interfaces with adjacent subsystems.

2.2 Loading Spectrum

To ensure both computational accuracy and geometric adaptability of the central component, the finite element model was meshed primarily with hexahedral elements supplemented by tetrahedral elements. The mesh size was 3.5 mm, resulting in a total of 7.28×10^6 elements. The helicopter hub central component serves as the primary load-bearing structure for blade-induced centrifugal load F_{xi} , flapping load F_{yi} , and lead-lag load F_{zi} , in addition to damper link loads F_{di} from vibration attenuation systems, as shown in Fig. 3.

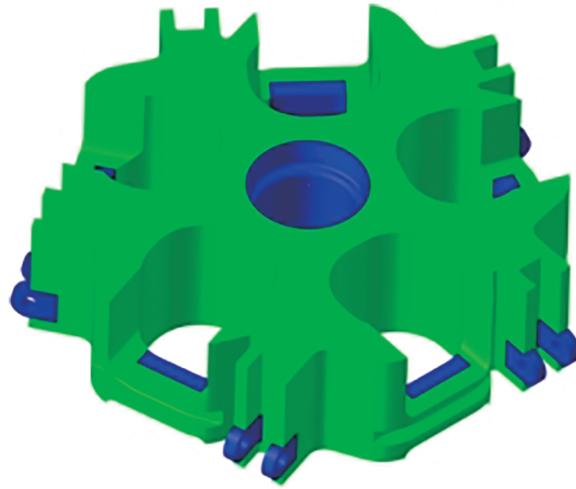


Figure 2: Design domain illustration for topology optimization

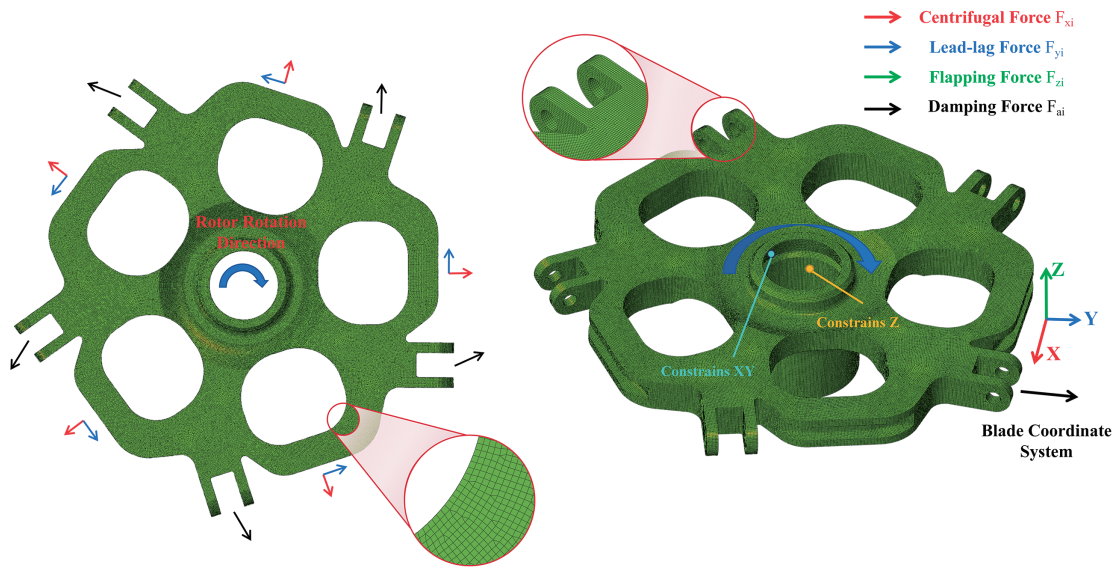


Figure 3: Schematic of the finite element model

The operational loads are classified into static and dynamic components. For the static loads, two cases are considered, as specified in [Table 1](#) of this study. Static load analysis excludes phase relationship considerations, while dynamic load components incorporate phase relationships as follows:

$$F_{xi} = F_{xd} \times \cos\left(\omega t + \pi + \frac{2\pi}{5}i\right) \quad (1)$$

$$F_{yi} = F_{yd} \times \cos\left(\omega t + \pi + \frac{2\pi}{5}i\right) \quad (2)$$

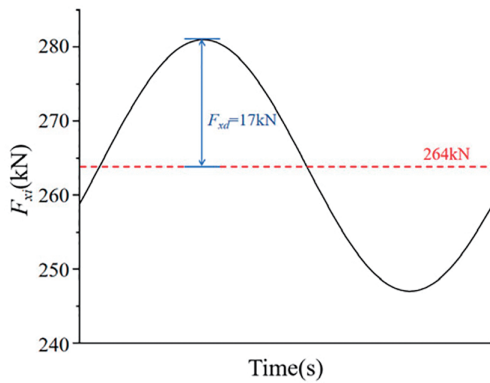
$$F_{zi} = F_{zd} \times \cos\left(\omega t + \frac{2\pi}{5}i\right) \quad (3)$$

$$F_{ai} = F_{ad} \times \cos\left(\omega t + \pi + \frac{2\pi}{5}i\right) \quad (4)$$

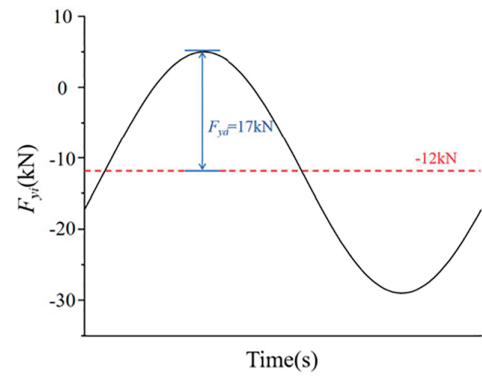
The baseline hub central component underwent load application process in strict accordance with the load case matrix specified in Table 1 and Force Time History Curve are shown in Fig. 4.

Table 1: Load spectrum of hub central component

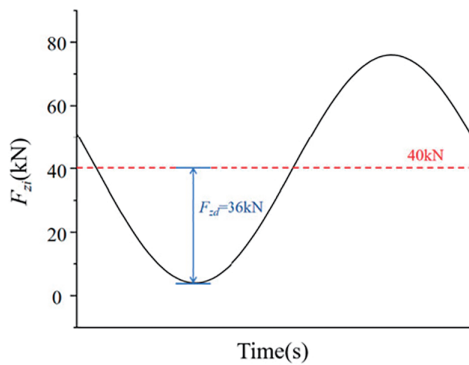
Case	Centrifugal load F_{xi}/kN	Flapping load F_{yi}/kN	Lead-lag load F_{zi}/kN	Damper link loads F_{ai}/kN
Static load case 1	430	40	25	15
Static load case 2	281	5	77	19
Dynamic load	Mean	-12	40	0.5
	Deviation	17	36	19



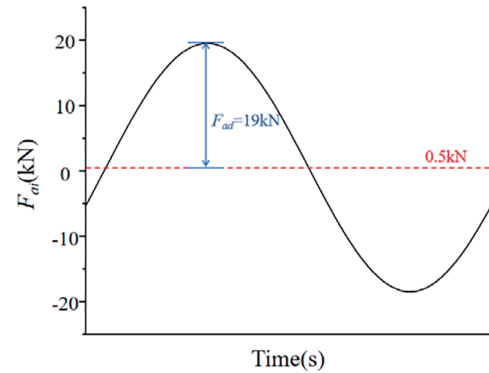
(a) Centrifugal Force Time History Curve



(b) Oscillatory Force Time History Curve



(c) Flapping Force Time History Curve



(d) Damping Force Time History Curve

Figure 4: Load spectrum

2.3 Fatigue Theory for Helicopter Structure

Fatigue failure constitutes the predominant failure mode for helicopter dynamic components. In engineering practice, these components experience variable-amplitude loading where operational stress levels, though below material ultimate strength, still induce fatigue damage. For dynamic structures primarily subjected to high-cycle fatigue (HCF) loads, fatigue life validation through S-N characteristic testing is mandatory. Engineering protocols require both load spectrum development for critical components and determination of safety life curves, followed by cumulative damage theory-based damage calculation across operational conditions to derive fatigue life. The S-N curve equation for metallic dynamic components under dominant HCF loading is expressed as:

$$\frac{S_a}{S_\infty} = 1 + \frac{A}{N^\alpha} \quad (5)$$

where N is fatigue life, S_a denotes the equivalent dynamic stress and S_∞ represents the safe fatigue limit ($N \rightarrow \infty$). A and α are material dependent parameters, and for the titanium alloy employed in this study. The values $A = 0.205$ and $\alpha = 0.49$ are adopted from the design manual [28].

In helicopter structural applications, the mean load and load amplitude of fatigue loads during flight operations are complex and variable. However, in structural fatigue characteristic tests for curve determination, loads are typically applied as fixed symmetric cyclic loads. This necessitates accounting for the influence of mean loads on alternating load-induced damage when calculating safe fatigue life. Correction of alternating loads via iso-life curves is required to simulate real operational loading conditions. Engineering practices commonly employ iso-life curves such as the Goodman line, Soderberg line, and Gerber parabola. The safety fatigue limit correction formula incorporating the Haigh coefficient is expressed as:

$$S_\infty = \left(1 - \frac{S_m}{S_{0.2}}\right) \times S_{-1} \quad (6)$$

where S_{-1} is the safe fatigue limit under symmetric cyclic loading, and $S_{0.2}$ denotes the material yield limit. For titanium alloy used in this study, $S_{-1} = 350$ MPa and $S_{0.2} = 830$ MPa. S_m corresponds to the structural stress induced by the mean load.

By substituting Eq. (6) into Eq. (5), the revised structural dynamic stress calculation formula is derived:

$$S_a = \left(1 + \frac{A}{N^\alpha}\right) \cdot \left(1 - \frac{S_m}{S_{0.2}}\right) \times S_{-1} \quad (7)$$

Through the above formulas and cumulative damage theory, the maximum dynamic stress level S_a in the structure corresponding to any design life N can be determined. By controlling the dynamic stress levels in each operating condition $\sigma_d \leq S_a$, dynamic components of the hub assembly with a service life reaching or exceeding the design life can be achieved.

3 Topology Optimization Formulations

3.1 Formulations

In the Solid Isotropic Material with Penalization (SIMP) topology optimization method, the relative density of each finite element is used as the design variable. The structure is discretized into a finite element mesh, and each element is assigned a relative density ρ_e ($0 \leq \rho_e \leq 1$) representing the topology design variable, where $\rho_e = 0$ indicates that no material is allocated to the element, and $\rho_e = 1$ indicates that material is fully allocated. The equivalent dynamic stress and structural weight are defined as constraints, while

the objective is to maximize the global stiffness of the structure (minimize compliance). Based on these considerations, the topology optimization problem for the hub central component is formulated, and the mathematical statement of the optimization problem is expressed as follows. The overall workflow of the proposed methodology, including the initialization of design variables, finite element analysis, sensitivity analysis, and MMA-based optimization, is illustrated in Fig. 5.

$$\begin{aligned}
 & \text{find } \rho_e (e = 1, \dots, Num) \\
 & \min obj = \sum_{r=1}^{nu} C_r \\
 & \text{s.t. } \mathbf{K}_r(\rho_e) \mathbf{U}_r = \mathbf{F}_r \\
 & \quad \sigma_{d,r} \leq S_{a,r} \\
 & \quad M_{\max} \leq \overline{M}_{\max}
 \end{aligned} \tag{8}$$

Here, Num denotes the number of structural elements, and nu denotes the number of loading cases. The specific expression for the structural stiffness (compliance) C_r under the r -th loading case is given as follows:

$$C_r = \mathbf{U}_r^T \mathbf{K} \mathbf{U}_r = \sum_{e=1}^N \mathbf{U}_{r,e}^T \mathbf{K}_e \mathbf{U}_{r,e} \tag{9}$$

Here, $\mathbf{U}_{r,e}$ denotes the displacement vector of the e -th element under the r -th load case, and $\mathbf{K}_e$ represents the stiffness matrix of the e -th element.

$$\mathbf{K}_e = \int_{\Omega_e} \mathbf{B}^T \mathbf{D}_e \mathbf{B} d\Omega_e \tag{10}$$

In the equation, Ω_e denotes the integration domain of the element, $\mathbf{B}$ denotes the strain–displacement matrix of the element, and $\mathbf{D}_e$ is the elasticity matrix of the e -th element. The elasticity matrix is interpolated using the SIMP scheme:

$$\mathbf{D}_e = (\rho_{\min} + (1 - \rho_{\min}) \rho_e^p) \cdot \mathbf{D}_0 \tag{11}$$

In the equation, ρ_e denotes the relative density of element e , and $\mathbf{D}_0$ is the elasticity matrix of the metallic material, corresponding to the elasticity matrix when the element density equals 1. To avoid singularity of the stiffness matrix when the element density is 0, a minimum element density $\rho_{\min}$ is introduced. The parameter p is the penalty exponent; under its effect, intermediate density values tend to approach either 0 or 1. In this study, p is set to 3.

3.2 Sensitivity Analysis for Optimal Design

The sensitivity of the objective function with respect to the design variables ρ_e can be obtained using the adjoint method:

$$\begin{aligned}
 \frac{\partial c}{\partial \rho_e} &= \sum_{r=1}^n \frac{\partial C_r}{\partial \rho_e} \\
 &= - \sum_{r=1}^n \sum_{e=1}^N \mathbf{U}_{r,e}^T \frac{\partial \mathbf{K}_e}{\partial \rho_e} \mathbf{U}_{r,e} \\
 &= - \sum_{r=1}^n \sum_{e=1}^N \mathbf{U}_{r,e}^T \left(\int_{\Omega_e} \mathbf{B}^T \frac{\partial \mathbf{D}_e}{\partial \rho_e} \mathbf{B} d\Omega_e \right) \mathbf{U}_{r,e}
 \end{aligned} \tag{12}$$

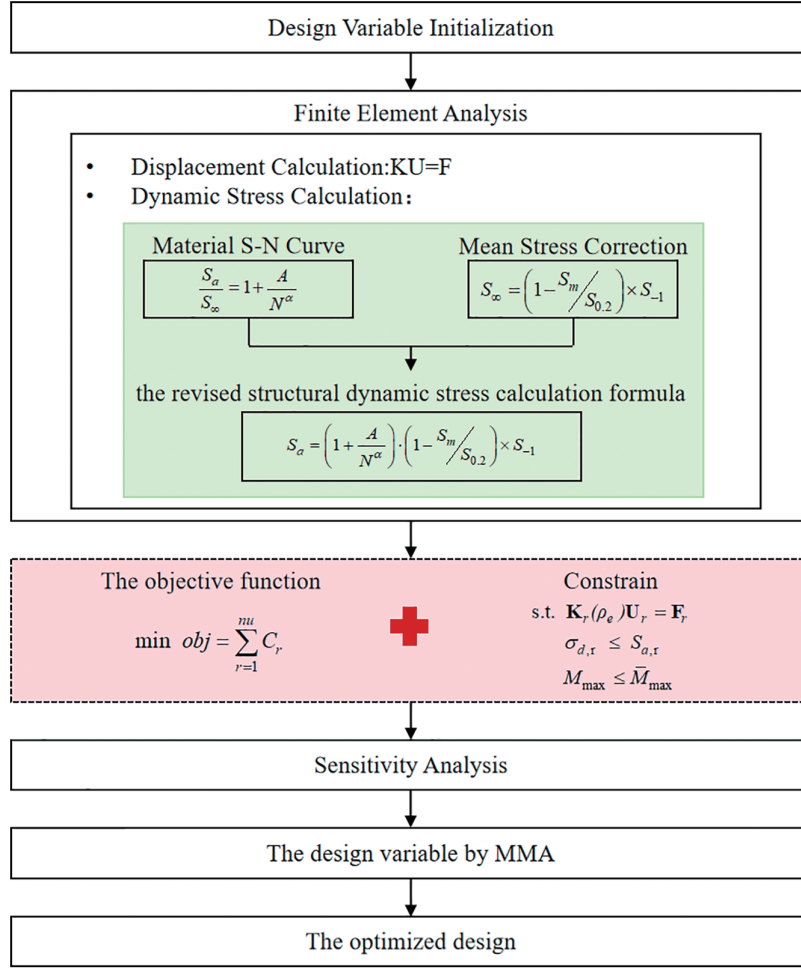


Figure 5: Flowchart of the proposed method

In the equation, the partial derivative of the objective function with respect to the design variables is ultimately reduced, via the chain rule, to the partial derivative of the elasticity matrix with respect to the design variables, which is calculated as follows:

$$\begin{aligned} \frac{\partial \mathbf{D}_e}{\partial \rho_e} &= \frac{\partial \rho_e^p}{\partial \rho_e} \cdot \mathbf{D}_0 \\ &= p \cdot (1 - \rho_{\min}) \cdot \rho_e^{p-1} \cdot \mathbf{D}_0 \end{aligned} \quad (13)$$

By substituting Eq. (13) into Eq. (12), the final sensitivity expression of the objective function with respect to the design variables can be obtained:

$$\frac{\partial c}{\partial \rho_e} = - \sum_{i=1}^n \sum_{e=1}^N \mathbf{U}_{i,e}^T \left(p \cdot (1 - \rho_{\min}) \cdot \rho_e^{p-1} \cdot \int_{\Omega_e} \mathbf{B}^T \mathbf{D}_0 \mathbf{B} d\Omega_e \right) \mathbf{U}_{i,e} \quad (14)$$

The sensitivity expression of the mass constraint with respect to the design variables is given as follows:

$$\frac{dM}{d\rho_e} = 1 \quad (15)$$

According to the helicopter fatigue stress theory adopted in this study, stress is used as a surrogate for fatigue life as the fatigue constraint. The stress condition that ensures the structure meets the fatigue strength requirement can be described as follows:

$$\sigma_d \leq S_a \quad (16)$$

Here, σ_d denotes the dynamic stress of the structure, and S_a is the allowable stress that satisfies the design life requirement. The calculation formula for the dynamic stress constraint of the structure is given as follows:

$$\sigma_d \leq \left(1 + \frac{A}{N^\alpha}\right) \cdot \left(1 - \frac{S_m}{S_{0.2}}\right) \times S_{-1} \quad (17)$$

Since stress is a local physical quantity, for a large-scale structure such as the hub central component, imposing stress constraints on each individual element would result in an enormous number of constraints, leading to a significant computational burden. To address this issue, the P-norm is used to aggregate the stress constraints of all elements into a single global stress constraint, as follows:

$$\sigma_p = \left(\sum_{e=1}^N \rho_e (\sigma_d)^p \right)^{\frac{1}{p}} \quad (18)$$

The sensitivity of the stress constraint σ_d with respect to the design variables can be approximated by the sensitivity of the P-norm stress σ_p , which is calculated as follows:

$$\frac{\partial \sigma_p}{\partial \rho_e} = \frac{\partial \sigma_p}{\partial \sigma_d} \cdot \frac{\partial \sigma_d}{\partial \rho_e} \quad (19)$$

In the equation, the calculation formulas for each physical quantity are given as follows, and the sensitivity of the dynamic stress σ_d is taken as the sensitivity of the first principal stress σ_{11} :

$$\frac{\partial \sigma_p}{\partial \sigma_d} = \left(\sum_{e=1}^N \rho_e (\sigma_{11})^p \right)^{\frac{1}{p}-1} \cdot \rho_e (\sigma_{11})^{p-1} \quad (20)$$

$$\frac{\partial \sigma_d}{\partial \rho_e} = p \cdot \rho_e^{p-1} \cdot \sigma_{11}^e + \rho_e^p \cdot \frac{\partial \sigma_{11}^e}{\partial \sigma_e} \cdot \frac{\partial \sigma_e}{\partial \rho_e} \quad (21)$$

The calculation expression for the element stress σ_e vector is given as follows:

$$\sigma_e = \mathbf{DB} \mathbf{u}_e \quad (22)$$

The sensitivity of the element stress vector with respect to the design variables is calculated as follows:

$$\frac{\partial \sigma_e}{\partial \rho_e} = -\mathbf{DBK}^{-1} \frac{\partial \mathbf{K}}{\partial \rho_e} \mathbf{u}_e \quad (23)$$

By substituting Eqs. (20), (21), and (23) into Eq. (19), the sensitivity of the stress constraint with respect to the design variables is obtained as follows:

$$\frac{\partial \sigma_p}{\partial \rho_e} = \left(\sum_{e=1}^N \rho_e (\sigma_d)^p \right)^{\frac{1}{p}-1} \cdot \rho_e (\sigma_d)^{p-1} \cdot \left(p \cdot \rho_e^{p-1} \cdot \sigma_{11}^e - \rho_e^p \cdot \frac{\partial \sigma_{11}^e}{\partial \sigma_e} \cdot \mathbf{DBK}^{-1} \frac{\partial \mathbf{K}}{\partial \rho_e} \mathbf{u}_e \right) \quad (24)$$

4 Optimization Results

In the topology optimization, a penalty factor of $p = 3$ was employed to ensure a clear 0–1 material distribution. A density filtering technique with a filter radius of $R_{min} = 7$ mm was applied to eliminate checkerboard patterns. The convergence criterion was defined by the change rate of the objective function, and convergence was considered achieved when $Objtol < 0.001$. The topology optimization of the typical hub central component was completed in 78 iterations, executed on 16 threads, requiring a wall-clock time of 66 h and a total CPU time of 113.5 h. The convergence curves of the objective function and the constraint functions over the entire iterative process are shown as follows (in Figs. 6 and 7):

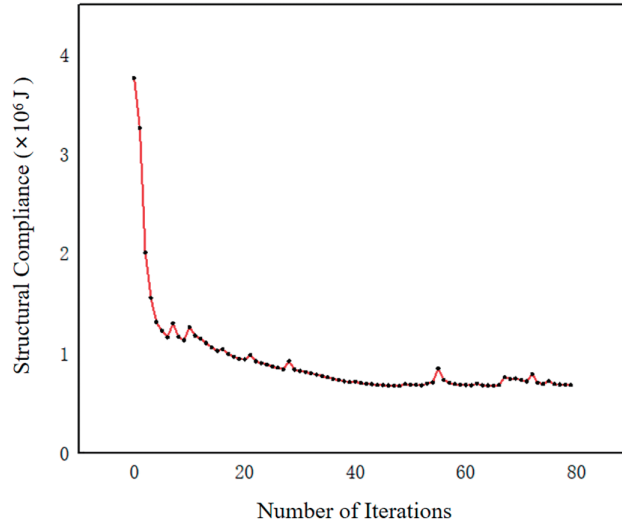


Figure 6: Iteration curve of the objective function

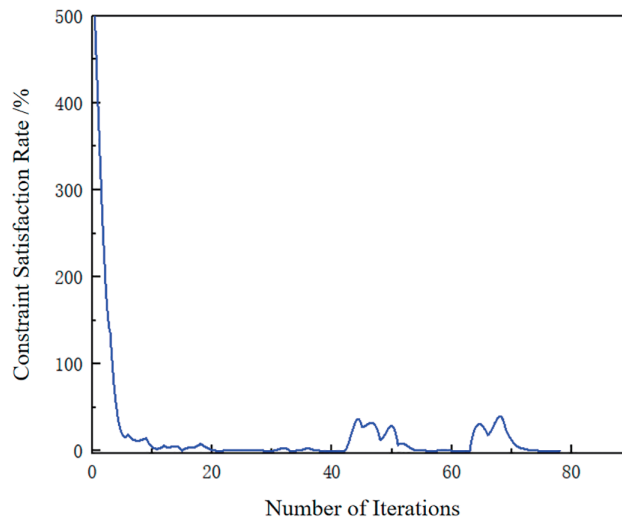


Figure 7: Iteration curve of the constraint function

After the 78th iteration of the topology optimization converged, the optimized topology of the hub central component was extracted, as shown in Fig. 8:

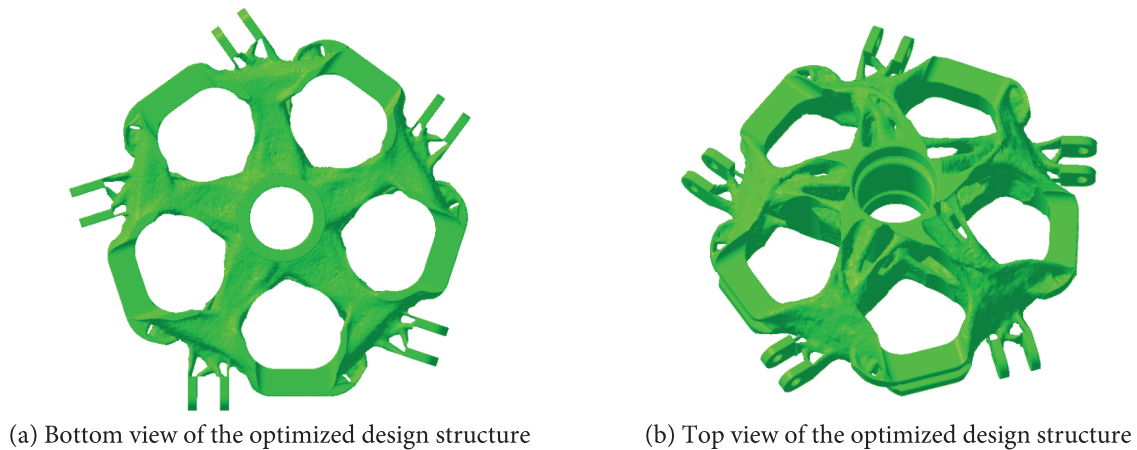


Figure 8: Topology optimization result of the hub center

Using the topology-optimized structure as a reference for conceptual design, detailed modeling of the component was carried out in digital modeling software. First, in the CAD software, an envelope that fully encloses the entire topology-optimized structure was created, as shown in [Fig. 9](#):

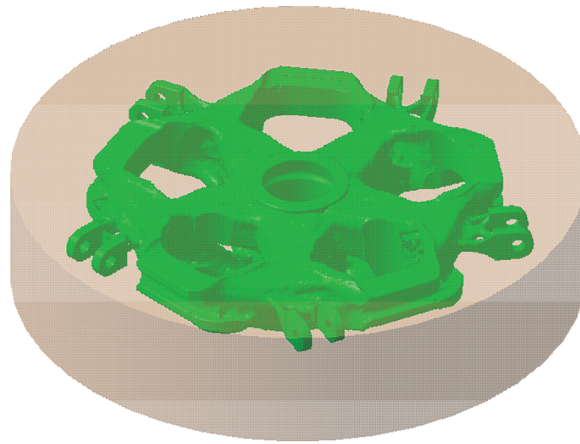


Figure 9: The central part reconstructs the structural envelope

Using the topology-optimized structure as a reference, the typical hub central component was reconstructed. The envelope was manually modeled using common geometric modeling operations for central hub design. The main process is illustrated in [Fig. 10](#):

During the detailed design process, material was arranged along the structural paths obtained from the topology optimization. Using this as the conceptual configuration and considering manufacturability, combined with the hub central component modeling methodology, the final design of the typical hub central component was completed. The final weight of the topology-optimized hub central component is 87.97 kg, satisfying the design constraint of 88 kg.

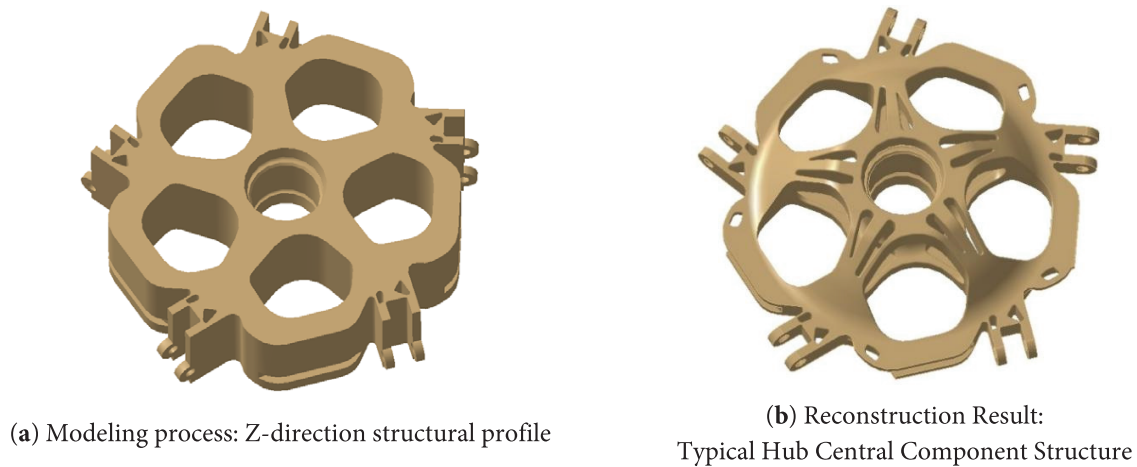


Figure 10: Structure modeling of hub central component

Finite element models of both the unoptimized and topology-optimized hub central components were established, and the loads and boundary conditions used in the previous topology optimization were applied. The specific loading conditions are listed in Table 2. Finite element calculations were completed for both models, and the displacement and the stress contours of the hub central component before and after optimization are shown below (in Figs. 11–14):

Table 2: The analysis results for the design results

Condition	Design before optimization	The topology optimization design
Dynamic displacement/mm	2.067	0.3733
Static displacement/mm	4.019	0.5122
Compliance	3.86×10^5	1.92×10^5
Dynamic stress/Mpa	212.7	120.9
Corrected static stress/Mpa	426.1	259
Number of cycles	5,095,909	∞
Fatigue life/H	202.2	∞

The verification results of both the unoptimized and topology-optimized hub central components were substituted into the fatigue performance constraint function (7). The finite element analysis results before and after optimization are summarized below:

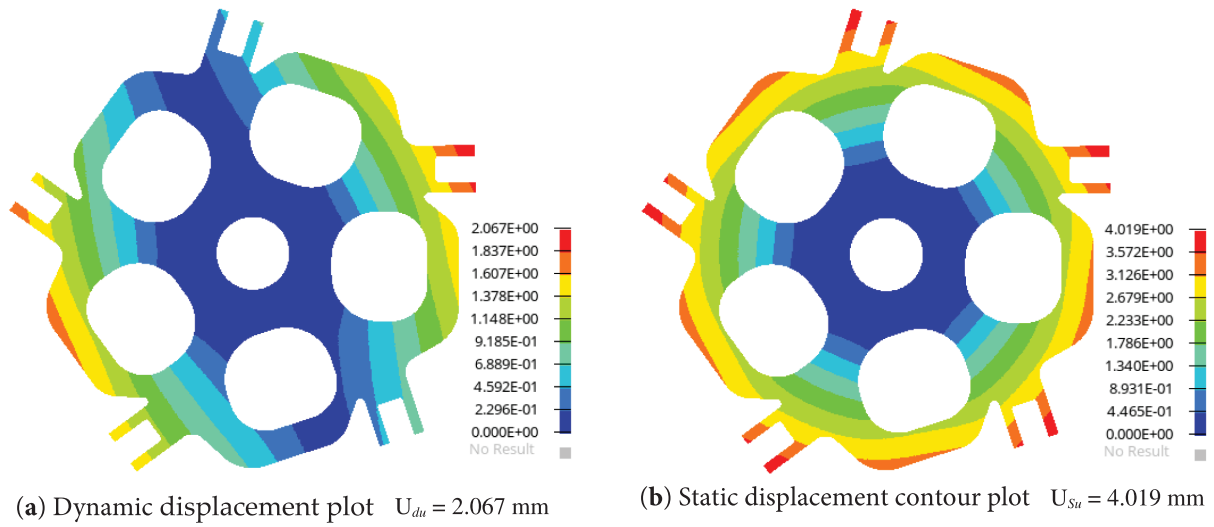


Figure 11: The displacement contour plots for the design before optimization

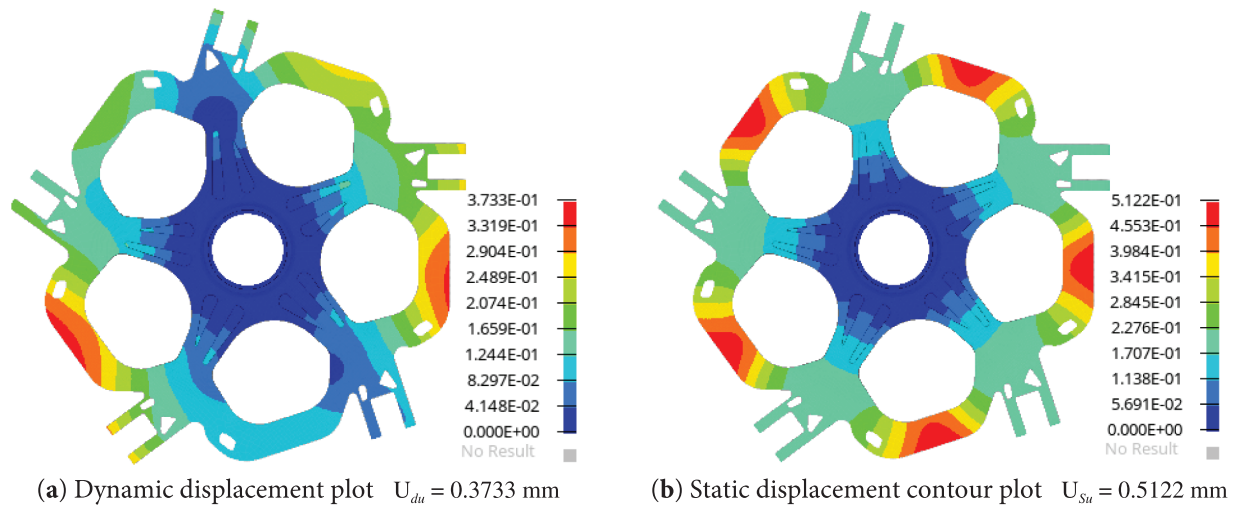


Figure 12: The displacement contour plots for the optimized design

Fatigue life verification indicates that the topology-optimized hub central component, with fatigue performance considered, achieves an infinite-life design, outperforming the unoptimized component.

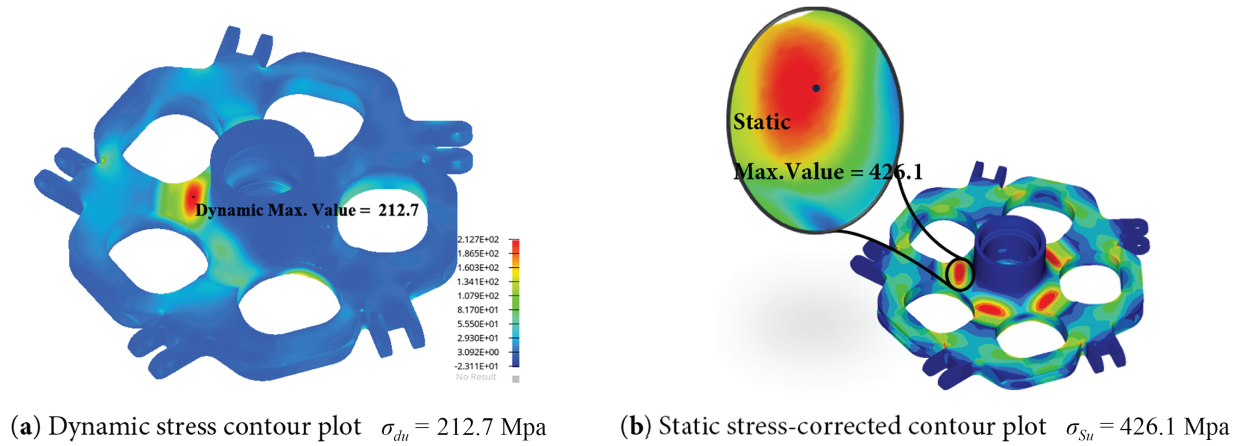


Figure 13: The stress contour plots for the design before optimization

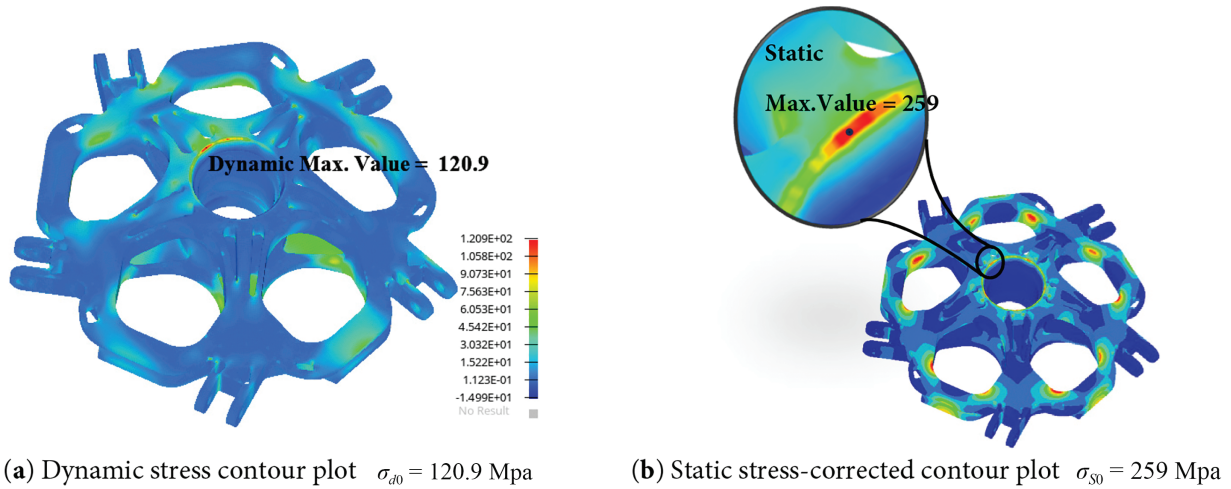


Figure 14: The stress contour plots for the optimized design

5 Conclusions

The central component of the helicopter rotor hub, due to its unique structural characteristics, endures complex cyclic fatigue loads during flight. To address the critical technical challenges in fatigue-driven forward optimization design of the hub central component, this study proposes a topology optimization method incorporating fatigue performance metrics, specifically tailored for multi-load conditions and cyclic failure mechanisms. Validation through a representative hub central component case confirms the effectiveness of the proposed method, with the following key conclusions:

- (1) Based on the S-N curve applicable to helicopter hub dynamic components, cumulative damage theory, and modified structural safety fatigue limits derived from iso-life curves, this study establishes an equivalent stress constraint that quantifies fatigue performance. This constraint can be effectively integrated into topology optimization formulations.
- (2) A topology optimization framework for helicopter hub central components is developed, achieving fatigue life control through regulation of safety-life equivalent stress. This forms a forward

design methodology for hub central component topology optimization, extendable to other rotor system structures.

- (3) Applying the proposed method to a typical five-arm hub central component, the optimized design achieves infinite fatigue life, significantly outperforming the baseline configuration. This research provides a robust foundation for high-reliability, fatigue-resistant hub central component design, offering an efficient design strategy and establishing critical technical groundwork for fatigue-conscious helicopter hub development.

The methodology effectively bridges fatigue performance requirements with structural optimization, advancing the design of helicopter dynamic systems under complex cyclic loading. However, it should be noted that, this study only focuses exclusively on high-cycle fatigue, and the validation is restricted to a numerical case of a single five-arm hub central component, without experimental evidence or verification involving multiple components. In future work, we plan to extend the methodology to other rotor hub components, incorporate low-cycle fatigue models, and carry out experimental validation to enhance the reliability and applicability of the proposed approach.

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Author Contributions: The authors confirm their contribution to the paper as follows: study concept, writing, and interpretation of results: Rui Xu, Quhao Li; data collection: Chaogan Gao; analysis and design: Guorui Yu; draft manuscript preparation: Jiale Shi. All authors reviewed the results and approved the final version of the manuscript.

Availability of Data and Materials: Details on the numerical implementation for the replication of the results have been provided in [Section 4](#), with the pseudocode and the optimization parameters. The design problems, mesh size, and boundary conditions are given in [Section 2](#). If the information provided in the paper is not enough, we sincerely welcome scientists or interested parties to contact us for further explanation.

Ethics Approval: Not applicable.

Conflicts of Interest: The authors declare no conflicts of interest to report regarding the present study.

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