Structure Design and Optimization of a gas turbine blade

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Abstract: Aiming at the fatigue life failure of a gas turbine blade, an optimization design of the blade was performed based on the test phenomenon and results of fatigue life analysis. As a result, the stress of the blade is decreased, and the fatigue life of the blade is increased.

1. Introduction

The single-stage gas turbine rotor (cooling turbine blades, turbine disk) is the core component of the turboshaft engine, and the blades are connected to the gas turbine disk through fir-trees[1]. Due to the high radius of the structure and the large mass of the blade, the gas turbine blades are subjected to huge mechanical loads, high temperature loads, aerodynamic loads, etc., and it is difficult to design the structural strength and life. Domestic scholars[2-10] have carried out certain research work on gas turbine blade structure design and life analyzer technology.

Gas turbine blades in aero-engines experience complex loadings, depending on the location of the structure, design method and service. Therefore, the failure of the blade can be result from low cycle fatigue[2-3], creep-fatigue, thermal fatigue[9] and thermal mechanical fatigue[10]. According to the temperature and stress/strain history at the detailed location of the blade[4-5], low cycle fatigue may be the main failure mode under the platform. Therefore, one purpose in the literatures to optimize the turbine blade structure was stress minimization[1].

The complexity of turbine blade structure brings great difficulties to parametric optimization. However, the optimization of turbine blades is extremely complicated[11-12]. Many studies are based on local optimization, such as fir-trees[1], film holes[13], blade profile, leading edge[14], blade Tip[15]. Lee et al. [13] used the multi-objective optimization strategy to analyze the gas film hole shape. The shape of the cooling was optimized to improve the cooling efficiency. In order to improve aerothermal performance of a turbofan turbine blade, multi-objective design optimization of turbine blade leading edge was performed[14]. The leading edge is parameterized by Be’izer curves. The Latin hypercube sampling plan is used to sample the design space. Therefore, local optimization is a feasible path considering the difficulty of parameterization caused by blade structure and the complexity of thermal mechanical load.

According to the experimental failure phenomenon, this paper analyzes the main reasons for the failure of gas turbine blades. Several improvement measures is studied and structure optimization as well as simulation is carries out. Finally, an optimization plan for blade structure is achieved. The
optimization results show that the structural stress can be effectively reduced and the low cycle fatigue life of the blade can be improved.

2. Failure phenomenon and analysis

2.1. Failure phenomenon
A certain type of engine was tested and stopped for a certain period of time when a failure occurred. The decomposition inspection found that a gas turbine blade was broken, and the rest of the blades were damaged and chipped to varying degrees at the blade tip. The broken gas turbine blade is shown in Figure 1.

![Fig.1 Broken gas turbine blade](image)

2.2. Fracture Analysis
Fluorescence inspection was performed on other leaves, and it was found that three leaves had linear fluorescence display at the same position of the leaf break. The inspection results are shown in Table 1.

| blade number | Fluorescence display length |
|--------------|----------------------------|
| 21#          | About 5mm                  |
| 23#          | About 4mm                  |
| 43#          | About 5mm                  |

![Fig.2 21# blade fluorescence display morphology](image)

21# blade with cracks on the outer surface was selected, and the inner cavity surface was observed after longitudinal sectioning on the exhaust side. It was found that there was a crack on the fractured part of the two blades (ie, the inner cavity surface of the first cooling cavity on the exhaust side). The crack is bent and opened in the surface, as shown in Figure 3.
The source area of the fatigue zone on the exhaust side end face presents the characteristics of a large line source, and there is no metallurgical defect in the source area. An obvious fatigue arc can be seen on the cross-section. The characteristics of the fatigue strips are clear and fine, and the strip spacing is less than 1 μm. There is a certain height difference between the cross-sections on both sides of the arc, which causes the arc to be enlarged and appear inclined, as shown in Figure 4.

Measure the bending and torsion dimension of the faulty part of the root extension of the inner cavity, as shown in Figure 6. The dimensions in the X direction and the Y direction are 6.7mm and 5.2mm respectively, and the dimension in the Z direction is only 3mm. Since the dimensions in the X direction and the Y direction are the two parts in the Z direction, the fault part of the root extension of the inner cavity is severely bent and twisted, which is easy to cause a large stress concentration.
3. Structural Analysis and Simulation Calculation
The three-dimensional finite element method is used to analyze the linear elastic stress of the blade. In the design state, the maximum linear elastic equivalent stress of the gas turbine working blade is 2037MPa, low cycle fatigue life less than 400 times. The local maximum stress is high, far exceeding the normal level. The stress distribution is shown in Figures 7 to 9.
From Figure 7, it can be seen that the maximum equivalent stress of the gas turbine working blade is located at the position where the extension root of the blade cavity is severely bent and twisted, which is basically consistent with the crack initiation position of the blade fault.

4. Optimization and improvement measures

4.1. Optimization direction

In view of the large stress area at the trailing edge of the root extension section of the gas turbine working blade, the original fault part is filled to reduce the stress. The inner cavity of the part, the boss under the edge plate and the ventilation groove structure at the bottom of the tenon are optimized. The optimized design variables and optimization constraints are shown in Table 2 and Table 2. The structure comparison before and after optimization is shown in Figure 10 and Figure 11.

| serial | variable | Attributes | lower | upper |
|--------|----------|------------|-------|-------|
| 1      | A1       | Inner cavity bending angle(°) | 30    | 45    |
| 2      | A2       | Clapboard root rounded(mm)    | 0.2   | 0.8   |
| 3      | A3       | Clapboard root rounded(mm)    | 0.2   | 0.8   |
| 4      | A4       | width(mm)              | 2.5   | 4.5   |
| 5      | A5       | rib width(mm)           | 0     | 2     |
| 6      | A6       | Vent groove depth(mm)    | 0.5   | 3     |
| 7      | A7       | Boss width(mm)           | 0.5   | 3     |
| 8      | A8       | Weight reduction groove depth(mm) | 0.2 | 0.8 |

Table 3 Optimized constraints

| serial | variable | Attributes     | lower | upper |
|--------|----------|----------------|-------|-------|
| 1      | CON1     | blade quality(g) | -     | 31    |
| 2      | CON2     | frequency margin(%) | 8    | -     |

Fig. 10 Schematic diagram before and after cavity optimization

Fig.11 Schematic diagram before and after shape optimization
4.2. Optimization Results
The three-dimensional linear elastic finite element method is used to analyze the stress. After the optimization, the stress and strain of the gas turbine working blade have a large level of decrease. The design variables and constraint values before and after optimization are shown in Table 4. The stress and strain comparison are shown in Table 5, The location diagram is shown in Figure 12.

Table 4 design variables and constraint values after optimization

| serial | variable | original | optimization |
|--------|----------|----------|--------------|
| 1      | A1       | 42       | 37           |
| 2      | A2       | 0.34     | 0.6          |
| 3      | A3       | 0.36     | 0.7          |
| 4      | A4       | 3.0      | 4.1          |
| 5      | A5       | 0        | 1.5          |
| 6      | A6       | 3.0      | 1.5          |
| 7      | A7       | 0.2      | 0            |
| 8      | A8       | 0.3      | 0.05         |
| 9      | CON1     | 30.1     | 30.4         |
| 10     | CON2     | 8.3      | 8.5          |

Table 5 Comparison of stress and strain before and after optimization

| location | original | optimization |
|----------|----------|--------------|
|          | equivalent stress (MPa) | total strain (%) |
|          | equivalent stress (MPa) | total strain (%) |
| 1        | 2037     | 1.222        | 1045 | 0.697       |
| 2        | 1952     | 1.289        | 859  | 0.608       |
| 3        | 1545     | 0.944        | 1066 | 0.680       |

After optimization, the low cycle fatigue life of the gas turbine working blades can reach 4807 times, which is more than 10 times higher than that before optimization.

5. Conclusion
The stress level of a certain type of engine gas turbine blade is relatively high, and the analysis and test results show that the design requirements are not met, and an optimized design needs to be carried out. In view of the problem of excessive local stress in the inner cavity of gas turbine blades, a series of structural optimization schemes are proposed, which greatly reduces the maximum stress of the blade and increases the low-cycle fatigue life of the blade by more than 10 times. The research results of this paper have great engineering value for the design and improvement of turbine blades.

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