Vibration Traits and Mechanism Analysis of an Engine Blades Fracture Failure

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Abstract: For the failure of twice centrifugal wheel blade fracture of an engine, the paper obtained the vibration traits of the fracture failure by analyzing the gross trend, time domain wave type, axial trajectory map and frequency spectrum diagram of the vibration signals at the failure time. The failure mechanism was analysed systematically based on the failure analysis of the fracture vane, the vibration characteristic analysis and stress distributing research of the vane. Finally, the paper verified the failure mechanism through the vane dynamic stress measurement test.

1 Preface

With the continuous improvement of the aero engine performance, the working state of the aero engine has gradually developed to high temperature, high pressure and high speed. The quality of the rotor blades directly impacts on many performance parameters of the engine. Aero engine’s blades are one kind of vulnerable parts of the engine [1]. They are complex in design, high requirements in processing and poor in working environment. According to incomplete statistics, 70%~80% of the engine failure are caused by blades failure. The references [2-9] are the analysis of blades failure, such as vibration, cracks, chipping and so on. Therefore, it has great significance to research and analyze the engine blades failure.

During the development of the aero engine, saltation phenomenon of the vibration signals occurred in the two tests. It was found that all the faults were blades tip fracture of the centrifugal impeller splitter blade. And the falling blocks were in the same position and had similar size. Through the analysis of the vibration signals and the fault blades, the vibration characteristics and the failure mechanism of the blocks were explored in this paper.

2 Failure of Blade Block

2.1 Anomalous Phenomena of Vibration

During the two lasting tests, the vibration amplitude of all vibration points of the engine changed at a certain time suddenly when the engine’s speed was constant and the working state was unchanged.

When the vibration was abnormal at first time, the vibration amplitude of all points reduced abruptly, which the accumulative working time of the engine was about 10 hours, as shown in Figure 1.

![Figure 1. Total vibration curve of the first vibration saltation (Normalized Amplitude).](image1)

When the second time abnormal vibration happened, the vibration amplitude of all points increased abruptly, and the accumulative working time of the engine was about 61 hours, as shown in Figure 2.

![Figure 2. Total vibration curve of the second vibration saltation (Normalized Amplitude).](image2)
2.2 Situation of Overhaul

The centrifugal impeller of the engine had chipping on a piece of splitter blade, after the first time of the vibration amplitude saltation. The chipping located at the tip of the blade, and the concave of the blade bent to convex side. The length of the missing block was about 14mm in tip blade, and about 18mm in leading edge. The chipping centrifugal impeller was shown in Figure 3.

Figure 3. First time dislocated part of centrifugal impeller splitter blade.

After the breakdown of the second vibration saltation occurred, it was found that the chipping generated similarly at the centrifugal impeller. And the concave of the blade bent to convex side at the tip of the blade. The length of the missing block was about 14mm in tip blade, and about 16mm in leading edge. The situation of the chipping of the centrifugal impeller splitter blade was shown in Figure 4.

Figure 4. Second time dislocated part of centrifugal impeller splitter blade.

The fracture failure of the centrifugal impeller shunting blade had occurred in the two tests, which seriously affected the test progress and the safety of the engine. Moreover, the two faults were at same location, and the size of the block was similar. These showed that there were some common problems in the two faults. Therefore, it is very important to analyze the fault signals and explore the fault mechanism as soon as possible for locating the cause of the failure.

3 Vibration Signals Analysis

The vibration signals of the vibration amplitude saltation were analyzed, when the fracture of centrifugal impeller splitter vane occurred.

3.1 Times-Domain Signals Analysis

The vibration signals of all measurement points were analyzed in time domain. The variation of the time-domain wave pattern of the vibration signals when the chipping dropped is shown in Figure 5. It can be seen that the time-domain wave of 1# and 2# test points located in the front of the engine had not obvious change. But the 3# and 4# turbine casing points near the centrifugal compressor outlet showed the impact signal obviously that should be caused by the chipping impacted.

Figure 5. Time-domain wave pattern of vibration velocity saltation.

The axis orbit map of 3# and 4# located at the vertical angle of the same section was shown in Figure 6. It can be seen that the trajectory of the axle center deviated seriously at the moment of the saltation. This was caused by the collision of the splitter blade’s chipping.

Figure 6. Comparison of axis trajectory.

3.2 Frequency Domain Signal Analysis

The vibration signals of the twice chipping faults were analyzed by spectrum. The frequency components and the vibration amplitude were compared before and after the chipping, as shown in Figure 7 and 8. The Figure 9 shows the time-frequency analysis spectrum of the Tz-point for the first time fracture failure.
By comparing the vibration spectrum before and after the saltation, it can be found that the vibration frequency components were consisted by the engine main-rotor working frequency (fundamental frequency) $f_1$ and its double frequency $2f_1$. The saltation manifested as the $f_1$ vibration amplitude mainly, while the vibration frequency was unchanged, and the abnormal frequency didn’t appear.

Table 1 shows the change that the vibration amplitude of the $f_1$ before and after the saltation.

<table>
<thead>
<tr>
<th>Measuring Code</th>
<th>Compare Project</th>
<th>1#</th>
<th>2#</th>
<th>3#</th>
<th>4#</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Before Saltation</td>
<td>11.3</td>
<td>17.8</td>
<td>17.3</td>
<td>13.6</td>
</tr>
<tr>
<td>First Time</td>
<td>After Saltation</td>
<td>3.5</td>
<td>7.4</td>
<td>1.9</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td>Saltation Value</td>
<td>-7.8</td>
<td>-10.4</td>
<td>-15.4</td>
<td>-10.3</td>
</tr>
<tr>
<td></td>
<td>Before Saltation</td>
<td>5.8</td>
<td>20.6</td>
<td>14.6</td>
<td>8.8</td>
</tr>
<tr>
<td>Second Time</td>
<td>After Saltation</td>
<td>13.3</td>
<td>26.8</td>
<td>24.2</td>
<td>20.7</td>
</tr>
<tr>
<td></td>
<td>Saltation Value</td>
<td>7.5</td>
<td>6.2</td>
<td>9.6</td>
<td>11.9</td>
</tr>
</tbody>
</table>

Compared the amplitude variation of the fundamental frequency in table 1:

In the first time of the vibration saltation, the vibration amplitude of all vibration points decreased. The 3# point amplitude reduced the largest value 15.4mm/s. It decreased by 89% than that before the saltation.

In the second time of the vibration saltation, the vibration amplitude of all vibration points increased. The 4# point amplitude increased the largest value 11.9mm/s. It augmented by 135.2% than that before the saltation.

So the weight distribution and the balance of the rotor were changed by blade chipping. Therefore, the vibration amplitude of the rotor fundamental frequency suddenly changed, and the total vibration was changed accordingly.

3.3 Vibration Signals Traits

Synthesizing the analysis of the vibration signals, the vibration traits of the splitter blade fracture fault are as follows.

1) The total vibration of all vibration points changed suddenly. The spectrum of vibration was mainly represented by the rotor fundamental frequency and its double frequency vibration. And there was no abnormal frequency appeared.

2) The time domain signals of the 3# and 4# points nearly by the outlet of the centrifugal compressor shown the impact signal obviously.

3) The axis trajectory which drawn according to the measurement points of turbine casing was obviously deviated at the moment of the vibration saltation.

4 Analysis of Fault Mechanism

4.1 Blade Failure Analysis

The blades of two centrifugal impellers were analyzed by metallographic analysis [5-7]. The material of the fault blade was same as the normal blade. The chipping blades had the common characteristics as follows.

1) Partial thickness of fault blade was thin, and the leading edge diameter of the blade was smaller than the design value. These reduced the anti-fatigue property, and affected the blade natural frequency.

2) The fracture of the fault blade leading edge was flat. Then the fracture became coarse gradually, and there was obvious bending deformation near the tip blade. The fractures were transient overload, and their properties...
were high cycle fatigue. The fractures began at the leading edge and then extended to the tip blade.

3) The blade surface roughness at the origin of the fracture fault exceeded the required range of technical conditions. Moreover, there were obvious machining marks on the side surface of the fracture source area. The marks on the origin of the crack were more serious.

![Figure 10. Machining marks comparison of normal with defective.](image)

The material used in centrifugal impeller was very sensitive to surface integrity. The thin blade thickness, the thick machining marks and the exceeding tolerance surface roughness could promote the fatigue cracks.

### 4.2 Vibration Traits of Blades

The centrifugal impeller has 11 splitter blades. In order to contrast the difference of the blades’ frequencies, the natural frequencies of the shunt blades were measured. The results were shown in Table 2. And the natural frequencies distribution comparison was shown in the Figure 11.

![Figure 11. Natural frequencies contrast diagram of centrifugal impeller shunt blades.](image)

It was shown that the frequencies of the fault blades were greater than the normal blades significantly. Moreover, the natural frequencies of the first failure blades were much larger than the design value.

The Campbell diagram can check whether the frequency of the various excitation forces produced by the rotating parts is consistent with the natural frequency. It can check the possibility of resonance.

We calculated the average values of the natural frequencies of the three impellers 11 blades respectively. The rotation frequency and its multiple frequencies of the rotor are used as the excitation force frequency. So we draw the Campbell chart with these calculated data, as shown in Figure 12.

![Figure 12. Campbell of splitter blades first-order vibration.](image)

It can be seen that the first-order natural frequency of the measured blades was close to the fourfold-frequency of the rotor. The possibility of the resonance was large. The rotation frequency of the rotor at the 100% speed is 753.7Hz, so the fourfold-frequency is 3014.8Hz. The comparison is made by using the concept of Resonance Margin, which is used to measure the probability of two frequencies. In general, the resonance margin is more than 10% [10]. The formula for calculating the resonance margin is as follows:

\[
\text{Resonance Margin} = \frac{\text{Max Value} - \text{Min Value}}{\text{Min Value}} \times 100\%
\]
The resonance margin of the splitter blades natural frequency with rotor quadruple frequency was shown in table 3.

Table 3. Resonance margin of the splitter blades natural frequency with rotor quadruple frequency (%).

<table>
<thead>
<tr>
<th>Blade Number</th>
<th>Normal Impeller</th>
<th>First Fault Impeller</th>
<th>Second Fault Impeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12.41</td>
<td>3.23</td>
<td>6.01</td>
</tr>
<tr>
<td>2</td>
<td>11.18</td>
<td>1.80</td>
<td>6.17</td>
</tr>
<tr>
<td>3</td>
<td>11.81</td>
<td>4.09</td>
<td>5.91</td>
</tr>
<tr>
<td>4</td>
<td>11.81</td>
<td>2.32</td>
<td>7.73</td>
</tr>
<tr>
<td>5</td>
<td>12.01</td>
<td>2.04</td>
<td>8.03</td>
</tr>
<tr>
<td>6</td>
<td>11.98</td>
<td>3.44</td>
<td>7.57</td>
</tr>
<tr>
<td>7</td>
<td>11.88</td>
<td>3.93</td>
<td>8.16</td>
</tr>
<tr>
<td>8</td>
<td>12.74</td>
<td>2.32</td>
<td>8.56</td>
</tr>
<tr>
<td>9</td>
<td>12.54</td>
<td>2.94</td>
<td>9.36</td>
</tr>
<tr>
<td>10</td>
<td>12.31</td>
<td>3.26</td>
<td>4.98</td>
</tr>
<tr>
<td>11</td>
<td>12.54</td>
<td>4.09</td>
<td>5.64</td>
</tr>
</tbody>
</table>

| Result | Yes | No | No |

So the resonance margin of the normal splitter blades meets the requirements, but the resonance margin of the failure splitter blades is less than 10%. And the resonance margin of the first time failure blades is smaller. It shows that the blades are easily affected by the fourfold-frequency of rotor. That confirms that the engine ran less time in the test of the first fracture failure.

4.3 Analysis of Blade Stress

The results of vibration analysis of splitter blades indicate that the possibility of first order resonance is large, when the splitter blades are excited by fourfold frequency. In order to verify the analysis results, the vibration stress analysis of the splitter blades is carried out while excited by the first natural frequency. The result is shown in Figure 13.

Figure 13. First order vibration stress distribution of blade back.

The maximum values of the vibration stress distribute on both sides of the tip blade whose distance to tip was 13.3mm and 16.1mm. This size is similar to the size of the blade chipping.

In addition, the stress distribution of the centrifugal impeller blade is analyzed under the condition of the standard working condition. It is found that the maximum radial stress is basically consistent with the origin of the fracture. The result is shown in Figure 14. However, the maximum stress is far less than the yield strength of the blade material.

Figure 14. Radial stress distribution of splitter blade (Unit:MPa)

The first order vibration stress analysis and the radial stress analysis of the splitter blade show that the resonance margin is small when excited by the fourfold frequency of the rotor. The maximum values of the vibration stress distribute is similar to the size of the blade chipping. The maximum radial stress is basically consistent with the origin of the fracture.

5 Testing Verification

In order to verify the results of the failure mechanism analysis, the dynamic strain test of the centrifugal impeller splitter blade was carried out.

Strain test system consists of strain gauges, strain lead, slip ring system, strain conditioner and computer. The speed pulse signal outputted by the speed measuring instrument was connected to strain conditioner. So the speed signal was synchronously collected with the strain signal. The composition of the strain test system is shown in Figure 15.

Figure 15. Composition of the strain test system.

The test points are selected according to the stress distribution results, as shown in Figure 13. And the positions of the strain gauges are shown in Figure 16.
Figure 16. Strain gauges’ position of splitter blade.

The strain spectrums of 1# and 2# were obtained by the dynamic strain test, as shown in Figures 17 and 18.

Figure 17. Strain spectrum of 1# measuring point (100% speed)

Figure 18. Strain spectrum of 2# measuring point (100% speed)

It can be seen that the strain spectrums are based on the first order resonance which mainly caused by the fourfold frequency of the rotor. The first order resonance does exist. At the same time, the measured stress amplitude indicates that the dynamic stress amplitude can meet the design requirements. So that the vibration stress can not cause the blade fracture simply.

Summary

The fault blades have the obvious machining knife marks, the surface roughness exceeds the technical requirement and the thickness of the blade is thin. All the above problems can easily lead to the decrease of blades fatigue resistance. At the same time, the machining error changes the vibration characteristics of the splitter blades. The first order resonance margin of the splitter blades is small when excited by the fourfold frequency of the rotor.

The first order vibration stress analysis, the blade radial stress analysis and the blade dynamic stress test show that the vibration strain distribution of the splitter blade is concentrated, and the blade vibration frequency is mainly caused by the first order resonance which equal the fourfold frequency of the rotor.

In the process of long time test, the defective blades had high cycle fatigues gradually, because of the influence of high frequency vibration. That led to the fracture failure eventually. The chipping of blades changed the mass distribution and balance of the rotor. So the axis trajectories diverged instantly, the vibration amplitude of fundamental frequency was changed. Finally, the total vibration changed correspondingly.

References