Aeroengine turbine blade containment tests using high-speed rotor spin testing facility

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Abstract

The blade containment test is regarded as an essential assessment of aeroengine safety. This paper presents the results of a series of blade containment tests where a double edge notched blade was released at certain rotating speed which subsequently impacted the inner wall of the containment ring. These tests were conducted over a range of blade lengths (113–123 mm) and releasing speeds (6800–15 000 rpm) using the high-speed rotor spin testing facility in the laboratory. It is shown that great attention should be paid to the failure of the containment rings caused by the second impact. Numerical simulations are carried out using nonlinear finite element method to study the impact process. The simulation results agree with the experimental results. Current experimental and numerical methods will be extended to actual aeroengine cases involving more complex blades and containment rings.

Keywords: Gas turbine aeroengine; Blade containment test; High-speed spin testing facility; Nonlinear finite element method

1. Introduction

A major risk in modern aviation is the failure of aeroengine turbine blades at very high rotating speed. The blade is the component that is most likely to be failed due to high cycle fatigue (HCF) [1]. HCF-induced failure can not be avoided in modern aircraft gas turbine engines [2]. High energy blade fragments may penetrate the wall of the containment ring and damage fluid lines, control cables, oil tanks and airframes, which may seriously affect the flying performance and threaten the safety of airplane and passengers. In order to contain the blade fragments, a containment ring around the gas turbine rotor is required. Federal Aviation Administration (FAA) in the United States requires all commercial aeroengines to have a containment ring which will not be perforated in the event of a blade failure during engine operation [3,4]. FAA further requires engine manufacturers to demonstrate through a certification test that the most critical blade must be contained when it is released at the maximum rotating speed of the engine. Similar requirements are also shown in British defense standard 00-971 [5].

In the design of a modern engine containment ring, a careful selection of material, geometry and wall thickness is needed to reduce the weight of the ring and offer sufficient perforation resistance. For this objective, the blade containment remains an active research area for aircraft industry. Such kind of tests is very costly and requires special testing facilities and instrumentations. Only limited research results can be found in open publications. A literature review on the design and analysis of turbine rotor fragment containment has been presented in [6]. With increasing computational power and improving dynamic nonlinear finite element methods, a number of 3D simulations on the blade containment design have been conducted, as shown in [7–9].

The purpose of this paper is to present a series of blade containment tests and their simulations using nonlinear finite element method. This article consists of four sections. Following this brief introduction, Section 2 describes the experimental devices, procedures and results. Section 3 presents a detailed case study using the finite element simulation, which is followed by conclusions.

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2. Experiments and results

A blade containment experiment using high-speed rotor spin testing facility is an appropriate method that can provide details of the impact effects. A complete description of the tester and the instruments for data acquisition is beyond the scope of this paper, but a brief summary will be given.

2.1. High-speed rotor spin testing facility

The tests had been conducted using the high-speed rotor spin testing facility at the High-Speed Rotating Machinery Laboratory in the Institute of Chemical Process Machinery in Zhejiang University. The facility has an upward vertical configuration and employs a 45 kW, 0–3000 rpm speed variable DC motor. The motor is connected to the drive shaft of a speed increasing gear box (1:10) to give an output speed range of up to 30 000 rpm. A flexible shaft is connected to the driven shaft of the gear box with a tapered mating surface so that the flexible shaft and the testing bladed disk can be installed and removed easily. The testing chamber is armored with two layers of lead bricks and a steel ring to protect and enhance the safety of operational personnel and prevent the damage of the outer casing of the facility. A cylindrical copper bearing is placed close to the overhung bladed disk as a contact protection bearing. After a blade is released from a disk, large vibration caused by mass unbalance is limited by rolling surface contact between the flexible shaft and the copper bearing, giving time for the operator to slow and stop the rotor without damage the testing facilities.

2.2. Blade containment tests

In these tests, the responses and deformations of both the containment ring and the released blade were studies within the preset speed range. The test rig was designed to satisfy several requirements, i.e. (a) the blade must be released at a preset speed, (b) the rotor speed at the instant of blade release must be recorded by speed meter, (c) the blade and the containment ring should be easily replaced for further tests, and (d) the test must be operated in vacuum so that aerodynamic effects can be eliminated. A plate blade and its balance mass on the opposite sides were connected to the rim of the disk with round pins, as shown in Figs. 1 and 2. Two edges of the blade were notched to the same depth. With the different lengths of the slots, the blades were predicted to release at different rotating speeds. Initial rotating speed and kinetic energy of the released blade can be calculated by a method presented in Section 2.3. The containment ring was mounted on the testing chamber floor. There was a radial clearance between the blade tip and the inner surface of the ring. The offset between the rotational axis of the disk and the geometric center axis of the ring was maintained within an acceptable tolerance by adjusting the position of the ring. A series of tests were performed using different containment rings and different blades mounted to the same disk. The material of the containment ring is Chinese standard #20 steel. Different wall thicknesses of the containment rings, various materials, and dimensions of the blades which were used in the tests, are as listed in Table 1.

![Pre-test photo in testing chamber](image)

![Sketch of the experimental set-up](image)

![Table 1 Parameters of the rings and the blades](image)

Although great efforts were made to achieve a preset blade releasing speed, it is practically difficult to release the blade at the preset speed. This might be caused by the variations of the tensile strength of the tested blades and/or by the cutting accuracy of the notches. Thus, one of the major requirements in the tests was to catch the instant rotating speed of the rotor at the time when the blade was released. An enamel insulated circuit wire with 0.4 mm in diameter was affixed on inner wall surface of the containment ring, which was connected in series with the...
signal line of the speed pickup transducer. When the released blade impacts the inner wall surface of the containment ring, the wire is cut off to stop the speed counting meter. The last number was taken as the instant rotating speed of the rotor at the releasing time of the blade.

Two pumps were used to vacuum the air to a pressure less than 1330 Pa in the testing chamber in order to prevent the unsteady aerodynamic force and impacting-induced fire. After the chamber was vacuumed, the bladed disk was accelerated slowly. As soon as the blade impacted the containment ring, the rotor was forced to stop quickly. Post-test examination was carried out after the cover of the tester was opened. The residual pieces of the containment ring and the released blade were collected for failure analysis.

2.3. Calculation of the releasing speed and the initial kinetic energy

The blade releasing speed and the initial kinetic energy are two important parameters that need to be considered in the design of testing components. Geometrical dimensions of the blade are presented in Fig. 2. The root of the blade has a notched cross-section in which a largest tensile stress was produced by the centrifugal force due to the rotation. The stress on this cross-section increases along with the rotating speed. When the stress is greater than the ultimate tensile strength of the material, the failure occurs in the notched section and the blade is released with certain kinetic energy. The stress on the notched cross-section due to the centrifugal force can be calculated as

\[
\sigma = \frac{1}{2} \frac{\rho b L (r_b^1 + r_b^2) \omega^2}{b - 2a},
\]

(1)

where \( \rho \) is the density of the blade material; \( b \) and \( L \) are the width and length of the released blade; \( r_b^1 \) and \( r_b^2 \) are the inner and outer radii of the released blade; \( \omega \) is the rotating speed; \( a \) is the length of the slots.

Substituting \( \sigma = \sigma_u \) and \( \omega = \omega_{\text{off}} \) into Eq. (1), it can be rewritten as

\[
\omega_{\text{off}} = \sqrt{\frac{2(b - 2a)\sigma_u}{\rho b L (r_b^1 + r_b^2)}},
\]

(2)

where \( \omega_{\text{off}} \) is the blade releasing speed and \( \sigma_u \) is the ultimate tensile strength of the material.

The initial kinetic energy of the released blade is given by

\[
E_k = \int dE = \frac{1}{6} \rho b \delta_b \omega_{\text{off}}^2 (r_{b2}^3 - r_{b1}^3),
\]

(3)

where \( \delta_b \) is the thickness of the blade.

2.4. Experimental results

Table 2 lists a comparison between the predicted and the experimental data of the blade releasing speed for each case. Good agreement is obtained within a difference of less than 7.3%. The initial kinetic energy of each released blade is also listed in this table.

The blade stem and the balance mass connected to the disk are not involved in impacts in the tests. The tested containment rings and the deformed blades are presented in a series of photos in Fig. 3. All released blades are deformed into “U” shape except in case 4, in which the blade was released at a relatively low speed with small kinetic energy and it was bent slightly at the tip of the blade. The released blade perforated the wall of the containment ring in cases 1 and 6, and was contained in other five cases, as list in Table 3.

There are two impacted regions on the ring in cases 2, 3, 5 and 7, and three in case 1, 4 and 6. The penetration and/or perforation did not happen in the first impacted region, but in the second or third impacted regions. An outline of impact deformations and failures on the containment ring for case 1 is shown in Fig. 4, which is also representative for other cases. It is shown in Fig. 4 that three plastically deformed regions are presented in the rotating direction with three different penetration depths in radial direction \( h_1 \), \( h_2 \) and \( h_3 \), and two circumferential distances between the adjacent penetrations, \( C_1 \) and \( C_2 \), as listed in Table 3. According to the examination of the failure rings, the second impact in cases 1 and 6 leads to tearing failure of the wall in circumferential direction by the cutting of the two side edges of the high speed moving blade. Tearing and splitting of the ring at about 45° from vertical is observed in and after the second impact region impacted by the curled blade. Furthermore, the blade runs away from the torn hole on the wall of the ring and impacts on the outer protection structure. From the examination of the blades and the rings in other contained cases, it is also found that where only one side of the blade surface contacts with the inner wall surface of the ring, which leads to a noticeable scratch between the impacted regions and after the last impacted region. It is estimated that the curved blade slides on the inner wall surface of the ring to decelerate until it drops finally on the testing chamber floor in the blade contained cases. This implies that the released blade could impact the next blade in a full bladed disk test [7] and more blades may be involved in the impact and failure triggered by single blade failure.

In cases 2, 3 and 5, only the lengths of the notch of the released blades are different. It is shown in Fig. 5 that the maximum radial penetration depth in the first impacted region is only about 30% of that in the second impacted region. The kinetic energy of the released blade decreases gradually as a consequence of impacts, and the kinetic energy loss of the released
Fig. 3. Deformation and failure of the rings and the blades.
Table 3
Results of the blade containment tests

<table>
<thead>
<tr>
<th>No.</th>
<th>h₁ (mm)</th>
<th>h₂ (mm)</th>
<th>h₃ (mm)</th>
<th>C₁ (mm)</th>
<th>C₂ (mm)</th>
<th>Contain results</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>9.5</td>
<td>13.5</td>
<td></td>
<td>120</td>
<td>150</td>
<td>Penetrated</td>
</tr>
<tr>
<td>2a</td>
<td>3.5</td>
<td>14.5</td>
<td></td>
<td>160</td>
<td></td>
<td>Contained</td>
</tr>
<tr>
<td>3a</td>
<td>4.5</td>
<td>15.0</td>
<td>120</td>
<td></td>
<td></td>
<td>Contained</td>
</tr>
<tr>
<td>4</td>
<td>6.0</td>
<td>3.0</td>
<td>180</td>
<td>100</td>
<td></td>
<td>Contained</td>
</tr>
<tr>
<td>5a</td>
<td>6.0</td>
<td>17.0</td>
<td></td>
<td></td>
<td></td>
<td>Contained</td>
</tr>
<tr>
<td>6</td>
<td>9.5</td>
<td>14</td>
<td>100</td>
<td>175</td>
<td></td>
<td>Penetrated</td>
</tr>
<tr>
<td>7</td>
<td>5.54</td>
<td>4.64</td>
<td>270</td>
<td></td>
<td></td>
<td>Contained</td>
</tr>
</tbody>
</table>

a Wall thickness of containment ring is 3 mm.

Fig. 4. Schematic of impacted deformation and failure on wall of the ring.

Fig. 5. Maximum radial penetration depth on wall of the rings versus initial kinetic energy of the blades in cases 2, 3 and 5.

This deformation mode disappears when the initial impact kinetic energy becomes low, which is replaced by the bending near the tip of the blade with more impacted regions on the wall of the ring (case 4).

The blade containment tests demonstrate that the initial kinetic energy of the released blade is partially dissipated by the plastic curling of the released blade and the plastic deformation of the impacted regions on the wall of the ring. The tests also demonstrate that a part of the kinetic energy is dissipated by the sliding friction between the surface of the released blade and the inner wall surface of the ring.

3. Numerical simulation

Numerical modeling may offer great details in a blade containment process. For example, some important phenomena, e.g. the transfer and dissipation of the initial kinetic energy of the released blade, which are very difficult to be observed in the testing process, can be studied by numerical simulations. Limited containment tests can be used to validate the methodology of nonlinear finite element analysis, and the validated numerical model may be used as a useful tool for the design of the containment rings.

3.1. Finite element model

The structure model of the blade containment tests consists of a single released blade with initial rotating velocity distribution and a containment ring with a constrained lower edge. It is assumed that the disk and the balance mass are not involved in the impact event, and therefore, they are not included in the model. Results of nonlinear numerical simulation are highly sensitive to the mesh of finite element and the following meshing guidelines [10] for the dynamic analysis are used:

1. Element aspect ratio should be near 1:1:1 for 3D impact numerical simulations.
2. Very fine meshes (i.e. small elements) are required to resolve localized events such as material failure.

According to above guidelines, the ring is modeled by 50400 four noded shell elements with a single element layer in the thickness direction while the blade is modeled by 12096...
eight noded brick elements with three element layers in the thickness direction. All possible contact surface pairs between the blade and the ring are defined for the impact simulation. Self-contact pairs in the blade are also defined as the large bending of the blade was observed in the tests.

When the blade is released at a high rotating speed, the linear speed of the blade tip will reach 300–500 m/s. Large plastic deformation and high strain rate are expected to be reached in both the blade and the containment ring. Therefore, strain rate effects on the material constitutive equation have been included in the analysis model. The Cowper–Symonds relationship [11] is used for the consideration of strain-rate effects for both the blade and the containment ring, in which the dynamic yield stress is scaled with strain rate according to the following relationship:

\[ \sigma = \sigma_0 \left[ 1 + \left( \frac{\dot{\varepsilon}}{D} \right)^{1/p} \right], \]  

(4)

where \( \sigma \) is the dynamic yield stress; \( \dot{\varepsilon} \) is the strain rate; \( \sigma_0 \) is the quasi-static yield stress; \( D \) and \( p \) are the strain rate sensitivity constants.

Furthermore, friction effects are included in the finite element model using the following function based on Coulomb formation

\[ F_t = \mu |F_n|, \]  

(5)

where \( F_t \) is the friction force; \( F_n \) is the normal compressive force; \( \mu \) is the dynamic friction coefficient. \( \mu = 0.15 \) is used in the numerical simulation.

3.2. Case study

The details of a case study are presented here to illustrate the application of the finite element method described previously to investigation of blade containment. For test case 7, the geometrical and material properties of the blade and the ring are given in Tables 4 and 5. The blade is given an initial angular speed of 1464 rad s\(^{-1}\). The computation is conducted by nonlinear dynamical analysis software, MSC.Dytran, using a Windows computer with 2.4 GHz Intel P4 processor and 1 GB RAM. The total simulation time is 0.877 milliseconds in 9000 steps. As the calculation is proceeding, time steps are controlled automatically to assure the stability. The output results include strain, stress, displacement, kinetic and plastic deformation energy, etc. Equivalent plastic strain on the wall of the ring and the deformed shape of the blade are sequentially plotted in Fig. 7 at three intervals. It should be noted that two impacted regions on the ring are clearly separated by a circumferential distance and the maximum equivalent plastic strain occurs in the second region. As discussed in Section 2.4, the impact damage in the second impacted region is larger than the impact damage in the first impacted region. After first impact, the blade begins to curl from the tip side and continues to keep moving forward and sliding on the inner wall surface of the ring. The deformed blade impacts the ring again when the blade is folded. The blade curls into “U” shape at the end of the second impact as seen in Fig. 7. Although the maximum plastic strain on the ring shown in Fig. 7 is far below the effective failure plastic strain of the ring material, large plastic deformations actually occur locally at two impacted regions. The ring with wall thickness 6 mm is able to contain the high speed released blade. These simulation results show good agreement with the test results. The simulation gives good predictions for the overall deformed shapes of the blade and the ring in case 7. The numerical model may be further improved to predict the dynamic response in an actual blade containment event and optimize the design of a containment ring.

In addition, it is easy to integrate the kinetic and plastic deformation from elements to elements during a finite element computational process. The kinetic and plastic deformation energy of the blade and the ring are plotted against time in Fig. 8. As the clearance is 6 mm between the blade tip and the inner wall surface of the ring, the kinetic and plastic deformation energy remain constant or zero until the blade comes into contact with the ring. After the blade impacted on the ring, the kinetic energy is decreased and the plastic deformation energy is increased. Both kinetic and plastic deformation energies of the ring increase during the impact process. Because the wall thickness of the ring is relatively thick, the deformations of the ring are relatively small in comparison with the deformation of the blade. The energy dissipated in the blade deformation is almost twice as much as the plastic deformation energy of the ring. The multiple impact impacts lead to the increase of the kinetic energy in the ring, which is exhibited by instantaneous vibration of the ring that are dissipated gradually by internal damping. Two impact phases are clearly separated in the variation histories of the kinetic energy and the energy rate of the blade, as shown in Fig. 9. Between two impacts, the blade slides on the inner wall surface of the ring, causes only small deformation of

<table>
<thead>
<tr>
<th>Table 4</th>
<th>Geometric properties of the blade and the ring</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade inner radius</td>
<td>213 mm</td>
</tr>
<tr>
<td>Blade outer radius</td>
<td>328 mm</td>
</tr>
<tr>
<td>Blade axial length</td>
<td>50 mm</td>
</tr>
<tr>
<td>Blade thickness</td>
<td>3.6 mm</td>
</tr>
<tr>
<td>Blade mass</td>
<td>162 g</td>
</tr>
<tr>
<td>Blade initial angular velocity</td>
<td>1464 rad s(^{-1})</td>
</tr>
<tr>
<td>Containment ring inner diameter</td>
<td>1464 rad s(^{-1})</td>
</tr>
<tr>
<td>Containment ring thickness</td>
<td>6 mm</td>
</tr>
<tr>
<td>Containment ring axial length</td>
<td>150 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 5</th>
<th>Material properties of the blade and the ring</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material no. (China)</td>
<td>45</td>
</tr>
<tr>
<td>Density (kg m(^{-3}))</td>
<td>7.81 E3</td>
</tr>
<tr>
<td>Elastic modulus (N m(^{-2}))</td>
<td>2.0 E11</td>
</tr>
<tr>
<td>Shear modulus (N m(^{-2}))</td>
<td>7.69 E10</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Yield stress (N m(^{-2}))</td>
<td>7.50 E8</td>
</tr>
<tr>
<td>Failure plastic strain</td>
<td>0.2</td>
</tr>
<tr>
<td>Cowper–Symonds ( D ) factor</td>
<td>100</td>
</tr>
<tr>
<td>Cowper–Symonds ( p ) factor</td>
<td>10</td>
</tr>
</tbody>
</table>

For test case 7, the numerical model may be further improved to predict the dynamic response in an actual blade containment event and optimize the design of a containment ring.
peaks are presented in the kinetic energy rate curve of the blade. A and C correspond to the impacts between the blade and the ring, B is associated with the bending and the deformation of the blade.

4. Concluding remarks

The blade containment tests were carried out in nearly vacuum condition using the high-speed rotor spin testing facility, which, together with nonlinear finite element analyses, lead to following conclusions:

- Deformation and failure characteristics of the released blades and the containment rings vary significantly for different initial rotating speed of the blade and the wall thickness of the containment ring. Large plastic deformation area and/or penetration and perforation failure may appear in the second impacted region on the containment ring when the releasing speed of the blade is high. With the increase of the wall thickness, the containment ring can absorb much more impact energy to prevent failure.
- Maximum radial penetration depth in second impacted region is larger than that in other impacted regions. Maximum radial penetration depths and the circumferential distance between two impact regions have linear relationship with the initial kinetic energy of the released blade.
- The released blade is bent as it impacts the containment ring. For a high initial rotating speed of the blade, it is curled into a “U” shape. For a low rotating speed, the impact bending occurs only near the tip of the blade.
The results reveal that the energy transfer and dissipation in a blade containment process involve a variety of events including multiple impacts, plastic deformation, surface sliding friction, penetration and perforation.

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