An investigation on fatigue failure of turbine blades of aircraft engines by high cycles fatigue test

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Abstract: Thermal stress, wear and material damage produce effects of high-cycle fatigue failures in aircraft engines. The loading configuration on turbine blades of aircraft engines consists of an axial load. The axial load is the centrifugal force combined with the tensile and compressive loads, caused by the natural vibrations of the blades themselves. Low-cycle fatigue and high-cycle fatigue loading tests simulate these flight loads experienced by engine components. The objective of this study is to illustrate the most important features of high-cycle fatigue to pass the final controls for the assurance of quality of turbine blades in aircraft engines. A finite element model, useful to predict the results of experimental tests, is also presented.

Keywords: aircraft engines; fatigue tests; natural vibrations; finite elements model.

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1 Introduction

Blade failures in gas turbine engines often lead to loss of all downstream stages and can have a dramatic effect on the availability of the turbine engines. Failure investigation is essential for the effective management of engine airworthiness. In this paper blade fatigue failures are investigated by mechanical analyses and by examination of failed blades.

Hou et al. (2002) proposed a series of mechanical analyses to identify the possible causes of the failures by examining anomalies in the mechanical behaviour of the turbine blade. A non-linear finite element method was utilised to determine the steady-state stresses and dynamic characteristics of the turbine blade. The steady-state stresses and dynamic characteristics of the blade were evaluated and synthesised in order to identify the causes of blade failures.

An experimental procedure has been developed for the investigation of fatigue, crack growth resistance of materials and real compressor blades and for predicting the life of blades with cracks by Troshchenko and Prokopenko (2000). Investigations have been performed on the influence of manufacturing residual stresses and surface defects in the form of simulators of dents, corrosion pits, and non-metallic inclusions on fatigue strength of steels and a titanium alloy. The characteristics of the material crack growth resistance have been studied by considering the effect of the medium and stress ratio in a cycle.

Damage to materials used in rotating components of gas turbine engines can be presented in the form of manufacturing defects, or more often can appear during service operation. Low Cycle Fatigue (LCF), Foreign Object Damage (FOD), and fretting are the major sources of in-service damage. These sources can alter the HCF resistance individually or all together. Methodologies for treating such damage in establishing allowable materials are considered. Some recent results on the effects of damage on the Haigh (Goodman) diagram and a discussion of the life management aspects of HCF are presented by Nicholas (1999).

The theory of fatige strength is described widely by Fuchs and Stephens (1980), Collins (1981), Gurney (1979), Rolfe and Barsom (1987), Hertzberg (1989), Winterstein (1988) and Wirsching et al. (1995). There are lots of models of fatigue tests. In the field of high cycle fatigue, the Authors propose a staircase method to evaluate the quality of turbine blades of aircraft engines. We determine the temperature distribution and the correlation between tension and deformation of turbine blades, with one end fixed and the other one free, forced by centrifugal load. This method has the following parts:

- preliminary cold test to find the first flexion frequency
- high-cycle fatigue test to find the correlation between tension and deformation and to check the stress distribution of turbine blades, rotating at the rate of natural period per revolution.

2 Theory of high-cycle fatigue test

The turbine blade, as a pine or a dovetail, has been placed in a block featured by a screw on its lower side which simulates the centrifugal load (Figure 1). The turbine blade has been inspected by fluorescent piercing liquids at the end of every fatigue tests. If there are causes of the failures or fretting marks, the background will not be used in the next phase.

During the fatigue tests, we have acquired peak-to-peak stress values of the gundrill of the exit edge of the turbine blade, forced by electrodynamic or electromagnetic shaker at the first lateral frequency. The mean stress value S of turbine blades has been determined by the following equation:

$$S = 2.7 \times 10^{8} K_1 f A \tag{1}$$

where

S(N/m^2): the stress peak

f(Hz): the frequency

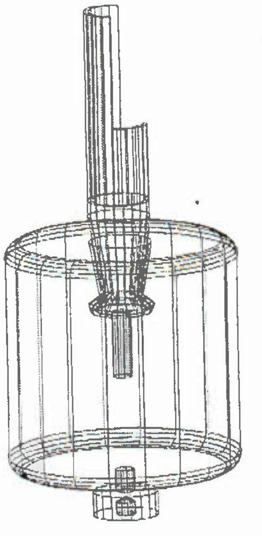
A(m): the peak-to-peak value of the top of turbine blade.

K_1: the constant obtained by the following equation:

$$K_1 = \frac{S_0}{2.7 \times 10^{8} f_0 A_0} \tag{2}$$

where the index 0 is the nominal values. We calculate the value of the factor K_1 by the strain gauges positioned on a turbine blade, when turbine blade overcomes 10^7 cycles of the fatigue test without reaching any break. The strain gauges have to be applied to the convex side on the top of the turbine blade, where it is C-shaped and have to be oriented to 90° with reference to the background.

Figure 1 Assembling of the test blade



The first sample of fatigue tests should be tested to the stress value obtained by the relation (1). The second sample should be tested to higher or lower stress value than the previous result. So we obtain the following cases:

- if the sample breaks, the second one will be tested to a lower stress level than the first one
- if the sample does not break, the second will be tested to higher stress level than the first one
- if the first one of two contiguous samples registers a crack and the second one does not registers a crack, the next sample of turbine blades will be tested to lower stress value than the previous stress value.

The increment or decrement of the stress value is about $d = 10\%$ of the previous stress level. If the turbine blade breaks after m cycles, the next turbine blade will be tested to amplitude S_1, decreased by a value 'd' compared with the previous:

$$S_1 = S_0 - d. \tag{3}$$

On the other hand, if the turbine blade does not break after m cycles, the next turbine blade will be tested to amplitude S_1, increased by a value 'd' compared with the previous:

$$S_1 = S_0 + d. \tag{4}$$

The same approach will be applied to the next turbine blades to have the stress value S_2:

$$S_2 = S_1 - d. \tag{5}$$

Every sample of turbine blades should be examined at first lateral frequency with a tolerance of 2%. If frequency value is constant and the sample of turbine blades does not break, the intact sample will be examined carefully by fluorescent piercing paints to have marks of defects. If n is the number of intact samples of turbine blades, forced at stress value level S_i, constant values A and B of mean fatigue resistance can be calculated by the following equations:

$$A = \sum i n_i \quad \text{and} \quad B = \sum i^2 n_i \tag{6}$$

where

$i = 0$: index of lower stress level S_0

$i = 1$: index of stress level S_1

$i = 2$: index of stress level S_2.

The value of mean fatigue resistance is obtained by the following formula:

$$\bar{X} = S_0 + d\left(\frac{A}{\sum n_i} \pm \frac{1}{2}\right) \tag{7}$$

where values $\pm 1/2$ depend on efficient turbine blades. When there are few efficient turbine blades we will have $+1/2$; when there are a lot of efficient turbine blades we will have $-1/2$. Therefore, the value of deviation is:

$$S_X = 1.62d\left[\frac{B\sum n_i - A^2}{\left(\sum n_i\right)^2} + 0.029\right]. \tag{8}$$

The acceptance criterion is based on the statistic prevision with reference to the stress level $S_{\min}$. The acceptance criterion requires to overpass the minimum level $S_{\min}$ obtained by the following equation:

$$S_{\min} = \bar{X} - K_2 S_X \tag{9}$$

where $\bar{X}$ and $\bar{S}_x$ are calculated by the application of equations (7) and (8) and the factor K_2 is function of the principal dimensions of the sample. $S_{\min}$ value from equation (9) is an average value calculated on 15 samples.

3 Experimental test

A HCF cell operates in the field 0–20 kHz. The HCF cell is attached to a servo-hydraulic fatigue machine. Figure 2 shows a scheme of the experimental set-up. An electrical sinusoidal signal of around 20 kHz in frequency is fed to a power amplifier. The amplified electrical signal is converted into mechanical vibrations using two piezoelectric crystals (converter unit). The converter has very high resonance frequency (about 20 kHz). Contrary to conventional fatigue testing, there are not yet testing standards to perform ultrasonic fatigue tests. For this reason we propose a method which combines stress levels and testing frequencies.

During the fatigue test, the strain has been monitored using strain gauges on the top of the edge of turbine blades. Strain gauges allow to find out the correlation between tension and deformation values. These strain gauges are very small (5–8 mm) and have almost zero mass so they do not affect the vibrational characteristics of the mechanical system. The temperature distribution has been monitored by 10 thermocouples on the wing contour and on the edge of the turbine blades. During HCF test it is important to check the temperature of furnace. A thermocouple on the bearing allows us to find the reference temperature of the furnace.

The preliminary LCF part performs a low cycle on every turbine blade to find out the fundamental frequency, while the HCF part remains off. During the HCF part, the instrument elongates and contracts the sample as in the case of a cyclic loading device. The rate of elongation and contraction of the sample is at the frequency of the turbine blade.

The stress measurements are acquired with regard to a tension level $J_{ref} = 100$ MPa. A particular furnace provokes the distribution of temperature (Figure 3). Hauled profile is warmed up by irradiation process, while the bottom of the profile is warmed by a conduction process. Referring to the Figure 3 we illustrate the distribution of temperature. After the transient phase real temperature distributions have been acquired in two different equilibrium conditions:

- from the ambient temperature to 900° with 50 A and 140 V
- from 900° to 813°C with 48 A and 105 V.

Figure 2 Measurement chain

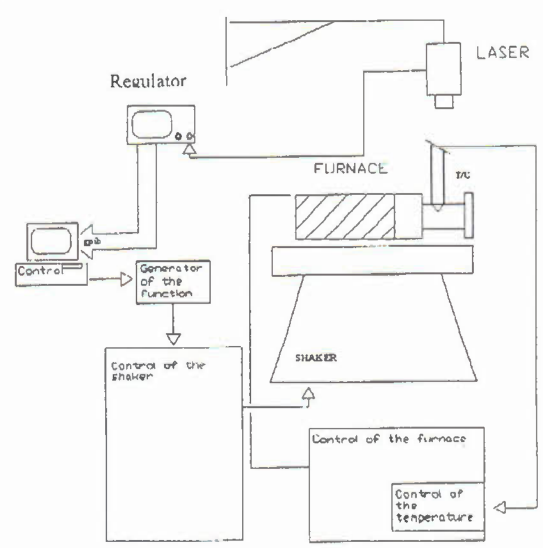
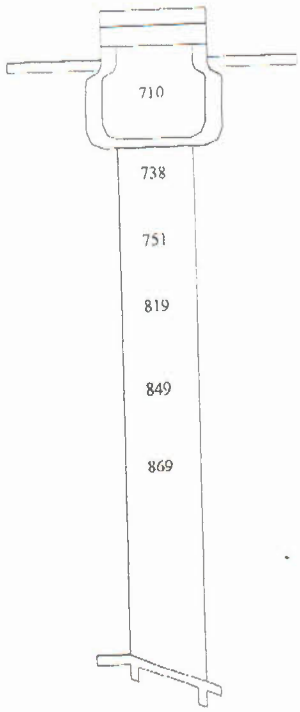


Figure 3 Distribution of temperature



The procedure may be summarised in the following way:

- Determination of the fundamental frequency at the reference temperature.
- Definition of the correlation tension – amplitude with regard to the reference tension level J_{ref} = 100 MPa and to the reference temperature.
- Warming up of turbine blades at the required temperature checked by thermocouples.
- Automatic temperature check by thermocouples on turbine blades.
- Investigation on lateral vibration frequency by a digital counter at high temperature.
- Turbine blades are forced by centrifugal load at lateral vibration frequency and with a required peak – peak displacement of the top of turbine blade. This displacement has been measured by a microscope with the axis ranged into the perpendicular direction to the motion of the point of the edge exit, characterised by an alignment tolerance of ±5° (Figure 2) (Artusio, 1994).

4 Experimental results

The Young's modulus of turbine blades is 207,000 MPa at 20°C. From relations (1) and (2), where is f = 158.7 Hz, S_0 = 297 MPa and K_1 = 2.57, the peak-to-peak vibration amplitude A (Figures 4 and 5) becomes:

$$A = \frac{297\times10^6}{2.57\times158.7\times2.7\times10^8} = 0.0027\text{m}.$$

The increment or decrement of the stress value is, generally, equal to the 10% of the previous stress level. In this case the decrement load is d = 30 MPa. As the first turbine blade does not overcome 10^7 cycles of the fatigue test, the second blade will be tested by the amplitude A, decreased by a value 'd' compared with the first one. From equation (3) we obtain:

$$S_1 = S_0 - d = 297 - d = 297 - 30 = 267\ \text{MPa}.$$

Figure 4 Fundamental frequency of the blades

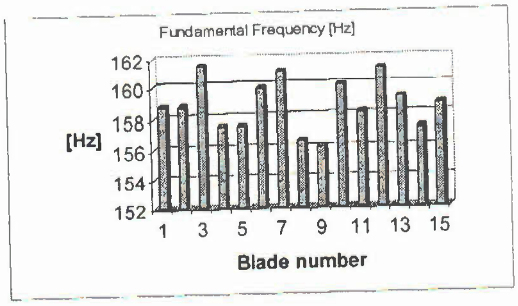
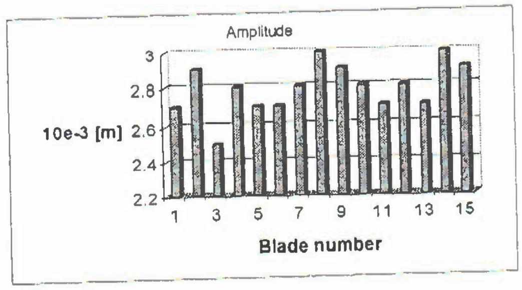


Figure 5 Vibration amplitude of the blades



By relation (5) we obtain the second stress value:

S_2 = 267 – 30 = 237 MPa.

From equations (6) we obtain the constants A = 6 and B = 6 (Tables 1 and 2). By the relation (7) the value of medium fatigue resistance is:

$$\overline{X} = 237 + 30\left(\frac{6}{7} + \frac{1}{2}\right) = 277.7\text{MPa}.$$

Table 1 Failure of blades (X: Broken blade; O: Unbroken blade)

σ(MPa)	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
297	X		X		X		X		X						
267		O		O		O		O		X		O		O	
237											O				

Table 2 Index values A and B

σ(MPa)	i	n_i	in_i	i^2n_i
297	2	0	0	0
267	1	6	6	6
237	0	1	0	0
Total	–	$\Sigma n_i = 7$	$A = \Sigma n_i = 6$	$B = \Sigma i^2 n_i = 6$

By the expression (8) the value of deviation becomes:

$$S_x = 1.62 \times 30 \times \left[\frac{6 \times 7 - 36}{49} + 0.029\right] = 4.54 \text{ MPa},$$

with the reference term

$$\frac{B\sum n_i - A^2}{\left(\sum n_i\right)^2} = 0.122.$$

By the equation (9) we have the stress S_{min}:

$$S_{min} = 277.7 - 4 \cdot 4.54 = 259.54\text{MPa} \cong 260\text{MPa}.$$

5 Finite element model

The experimental tests have been useful to set-up a numerical model of the blades that could be a valid instrument to predict the mean behaviour of the blades during operating conditions and give the designer suggestions to improve performances.

The model has been developed using finite element techniques, and an example of mesh is shown in Figure 6. The geometry came directly from CAD after the CFD study and solid tetrahedral element have been used. The material has the same properties as described in Section 4, so its Young's modulus is variable with temperature and this affects the behaviour of the blades. The model has been loaded and constrained to obtain the same operating condition of the experimental test. No fluidodynamics coupling has been considered, so there is not fluid around the blade. A centrifugal and temperature field has been introduced, together with a vibrational displacement imposed at the low extremity as described in Section 3. The first frequency has been computed, and the results show a good accordance with those of experimental tests as can be seen in Table 3.

Figure 6 Mesh of the blade and displacement field contour

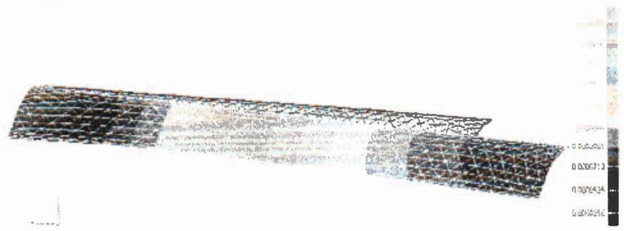


Table 3 Comparison between experimental and numerical results

Measured parameter	*Experimental (mean)*	*Numerical (FEM)*
Vibrational frequency (Hot)	159.0 Hz	160.2 Hz
Vibrational frequency (Cold)	144.0 Hz	147.5 Hz
Amplitude of vibration	0.0027 m	0.0025 m

6 Conclusions

Before the realisation of the HCF test we have to carry out a preliminary test in order to identify some physical measures, important for the following test conditions. From the tests, we are able to calculate minimum, medium, and maximum values of hot or cold natural frequencies (Table 4).

Table 4 Frequency values

Blades number: 15	*Minimum value* (Hz)	*Mean value* (Hz)	*Maximum value* (Hz)
Hot natural frequency	157.3	159.0	161.1
Cold natural frequency	139.5	144.0	148.7
Correlation for J_{ref} = 100 MPa	2.7	2.8	3.0

In order to realise fatigue tests, the HCF method is based on nominal tension level; its tension level, lower or higher, depends on possible breaks of the test piece. During this application, HCF begins with a nominal tension level equal to 297 MPa causing the break of the test piece. During the second phase, the tension is reduced of the 30%, reaching the value 267 MPa. According to row of tests (hot, cold, HCF) we record natural frequencies, stress values, vibration amplitudes, cycles numbers and event of the break. During fatigue tests, we check the temperature distribution in order to respect builder blades specifications in two equilibrium conditions according to the stabilisation time of 5 or 20 minutes (Figures 7 and 8, Table 5). Comparing the experimental results with specified results, concerning the acceptance limits, we are able to test the fatigue resistance of the examined blades group (Table 6).

So, a low fatigue resistance indicates the presence of the following causes (Nicholas, 1999).

- low resistance of the material
- low surface finish
- improper dimensions
- surface galling
- little breaks
- stock covering layer removal
- each combination that is the cause of a lower resistance than calculated for a nominal blade.

Figure 7 Temperature distribution for the blade 3rd level of an aircraft engine

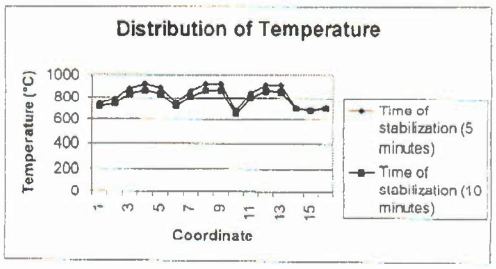


Figure 8 Furnace temperature values

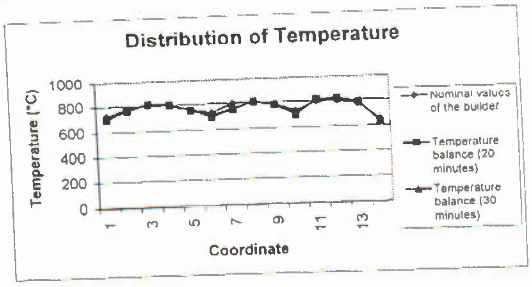


Table 5 Temperatures of the bases

T/C	*Temperature balance (20 minutes)*	*Temperature balance (30 minutes)*
Base no. 1	603	613
Base no. 2	578	590

Table 6 HCF test results

Blade number	*Hot natural frequency*	*Cold natural frequency*	σ (MPa) *Rif. 100*	*Vibration amplitude* (mm)	*Number of cycles* ($x10^7$)	*Blade break (Yes/No)*
1	158.7	145	297	2.7	1.43	Yes
2	158.7	146.2	267	2.9	10.0	No
3	161.5	147.5	297	2.5	2.97	Yes
4	157.5	147.5	267	2.8	10.0	No
5	157.5	148.2	297	2.7	2.29	Yes
6	159.9	146.25	267	2.7	10.0	No
7	161.1	148.7	297	2.8	0.89	Yes
8	156.5	142.5	267	3.0	10.0	No
9	156.2	141.2	297	2.9	1.23	Yes
10	160	143.5	267	2.8	9.95	Yes
11	158.2	142.2	267	2.7	10.0	No
12	161.2	143.5	237	2.8	10.0	No
13	159.2	140.4	297	2.7	0.72	Yes
14	157.3	139.5	267	2.7	10.0	No
15	158.7	140.6	297	3.0	2.81	Yes

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