TIP-TIMING DIAGNOSTICS OF MIDDLE BEARING OF AIRCRAFT ENGINE

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Abstract
The reported problem is the failure of the middle bearing in an aircraft rotor engine. Tip-timing and tip-clearance, variance analyses and RMS are carried out on a compressor rotor blade in the seventh stage above the middle bearing. The experimental analyses concern both an aircraft engine with a middle bearing in good working order and an engine with a damaged middle bearing. A numerical analysis of the seventh stage blade free vibration are conducted to explain the experimental results. This appears to be an effective method of predicting middle bearing failure. The blade vibration variance and RMS increases when there is bearing failure.

Introduction
Rolling element bearings are one of the most essential parts in rotating machinery. During operation, bearings are often subjected to high loading and difficult working conditions, which in turn often lead to the development of defects on the bearing components [1]. One way to increase operational reliability is to monitor incipient faults in these bearings [2-4]. Analytical models for predicting the vibration frequencies of rolling bearings and the amplitudes of significant frequencies with localized defects in bearings have been proposed in [5], [6, 7].

FFT is one of the widely used fault detection techniques [8], [9]. The only drawback of FFT based methods is that it is not suitable for non-stationary signals. In recent years, a new time frequency analysis technique, called Wavelet Analysis, was developed. The advantage of Wavelet Analysis is that the non-stationary characteristic of a signal can be easily highlighted in its spectrum [8], [9], [10], [11], [12], [13].

The tip-timing technique is generally used for diagnosing rotor blade displacements during rotation with a wide range of speeds [14, 15]. In this paper, the tip-timing technique is used for diagnosing displacements of the seventh compressor rotor blades and the middle bearing close to this stage. Such an analysis is important, because the failure of the middle bearing of an aircraft engine was reported in 1993 [14], [15] (Figs. 1, 2).

Measurements of seventh stage compressor rotor blade vibrations were
made using the tip-timing method at the Air Force Institute of Technology in Warsaw. In order to better understand the experimental results, numerical calculations were also conducted for the seventh stage compressor rotor blades using the FE method.

![Fig. 1. The inner running track of the aircraft engine middle bearing](image)

![Fig. 2. Damaged elements of the aircraft engine middle bearing](image)

**Numerical results**

An aircraft engine seventh stage rotor blade solid model was used to generate a mathematical finite element method mesh and perform modal analyses to compute the blade’s natural frequencies. The periodic sector was used for the solid model. Table 1 summarizes the measured and calculated natural frequencies of the 7th stage blade. The mode number is presented in the first column. The second and fourth columns show the calculated natural frequencies for a non-rotating blade at 0 rpm and a rotating blade at 15000 rpm. The third column presents the experimentally obtained natural frequencies. A modal hammer and MsScope software were used to identify the natural frequencies of the cantilever blades. In a real compressor, every rotor blade is different, so the natural frequencies for each rotor blade also differ. For example, in our experiment the first natural frequencies of the seventh-stage rotor blades ranged from 1620 Hz to 1932 Hz. A comparison between the numerical and experimental results (made for stationary blades) was satisfactory. For example, the first calculated frequency was 1765.2 Hz, whereas in the experiment it was <1620, 1930> Hz. Measurements were made only up to the fourth blade mode, because the failure of this blade was reported on the first mode. Fig. 3a presents a Campbell diagram for the rotor blade, which shows that 7EO, 8EO and 9EO (9 x n, n rotor rotation speed) can cause higher rotor blade responses. Aerodynamic cross coupling is not seen here.

**Experimental results**

The tip-timing of the seventh stage compressor rotor blade is presented in Fig. 4. This stage consisted of 48 rotor blades. The graph shows displacements of each of the 48 rotor blades at rotation speeds ranging from 7000
rpm to 15500 rpm. As can be seen, 7EO and 8EO create greater rotor blade responses. Each blade responded at a different time (see circled regions in Fig. 4). Tip-timing and Tip-clearance is a non-contact measurement technique which uses probes mounted in the casing to determine the vibration of all the blades.

\[ \text{Var}(t) = \int \left( \frac{1}{N} \sum_{i=1}^{N} x_i^2 - \frac{1}{N} (\sum_{i=1}^{N} x_i)^2 \right) dt \] (1)

where \( x_i(t) \) is the measured signal in period \( T \).

The results for individual blades could not explain the bearing failure. The entire test run time was divided into short periods \( T \). A variance of all the seventh stage blades was calculated for each \( T \) period:

Fig. 3a. Campbell diagram of the seventh stage compressor rotor blade of an aircraft engine

Fig. 3b. Campbell diagram of the seventh stage compressor rotor blade of an aircraft engine

Fig. 4. The tip-timing measurement of the seventh stage compressor

Fig. 5a presents the variance displacements value of seventh stage rotor blades (Eq. (1)) for an assumed test run (plane x-y, rpm versus time) with the middle bearing working properly. The rotor blade amplitude for 7EO was close to nominal speed 15000 rpm (see Campbell diagram Fig. 3). The resonances appeared in the region of 15000 rpm and were excited by 7EO (see Campbell diagram). There were some other resonances whose origin was difficult to explain using a Campbell diagram of a seventh stage compressor rotor blade, especially at 7000 rpm.

Fig. 5b presents variance versus time results derived from Fig. 5a. The maximal value of the variance amplitude of rotor blades was equal to 1.04 (see Fig. 5b).
Experiments were carried out at the Air Force Institute of Technology in Warsaw to reconstruct a real engine failure that was caused by a damaged middle bearing. In order to model the middle bearing failure, metal filings were successively added to the bearing’s oil filter while the engine was working. Tip-timing and tip-clearance were used to measure only the 7th stage blade vibration amplitudes because this stage was closest to the middle bearing.

Fig. 6a presents the variance displacements values of seventh stage rotor blades (Eq.(1)) for an assumed test run (plane x-y, rpm versus time) with a small amount of metal fillings added to the oil filter of the working middle bearing. The rotor blade amplitude for 7EO was close to that of nominal speed (Fig. 3a). The highest resonances, variance 61, appeared after the engine had been working for a longer period at 14000-15000 rpm (Fig. 6b).

Tip-clearance was measured above the seventh stage rotor blades. Eccentricity of rotor motion is one of the symptoms of a problem with the bearing; hence, the minimum-to-maximum tip clearance value of tip-clearance was shown in Fig. 6c. This clearly shows that the variation value is very small, in the region of 0.95-0.97.
Fig. 6b. The variance of vibration amplitude in seventh stage compressor rotor blade with slightly damaged middle bearing.

Fig. 6c. The measured tip-clearance of the seventh stage compressor rotor blade with slightly damaged middle bearing.

Fig. 7a presents the variance displacements value of seventh stage rotor blades (Eq.(1)) for an assumed test run (the plane x-y, RPM versus time) with more metal filings added to the working bearing oil filter. The maximal rotor blade amplitude for 7EO was close to the nominal speed (see Campbell diagram Fig. 3a). The maximal variance value of rotor blade amplitudes decreased from 61 (Fig. 6b) to 46 (Fig. 7b). The minimum-to-maximum tip clearance value of tip clearance is very small, in the region of 0.94-0.97 (Fig. 7c).

Fig. 7b. The variance of vibration amplitude in seventh stage compressor rotor blade with slightly more damaged middle bearing.
In the next test, more metal filings were added to the bearing oil filter (Fig. 8).

Fig. 8a presents the variance displacements value of seventh stage rotor blades (Eq. (1)) for an assumed test run (plane x–y, RPM versus time) with the working middle bearing on the brink of failure. The maximal rotor blade amplitude for 7EO was close to the nominal speed (see Campbell diagram Fig. 3). In Fig. 8b we see more peaks with amplitudes gradually increased to 47 (from 45 in Fig. 7b). Maximum-to-minimum tip-clearance increased from 0.93 to 0.97. Here the sampling for tip-clearance was 500 kHz. The sampling for tip-timing was 80 MHz.

In the final test (Fig. 9), 7th stage rotor blade rubbing was so severe that, after 300 s, the engine had to be stopped and the bearing was found to be in a plastic state.

Fig. 9a presents the variance value of the seventh stage rotor blades (Eq. (1)) for an assumed test run (plane x–y, RPM versus time) with a fully damaged middle bearing. In Fig. 9b, we see more peaks with amplitude variance increasing to 10.3, at which point the bearing failed and the 7th stage blades started rubbing against the casing.
In comparison with a normally working engine, the variance values increased from 1.04 (see Fig. 5b) to 10.3 (see Fig. 9b). However, this increase only occurred in the region of 190 - 200 s. Next, when the state of the bearing became plastic, it rapidly decreased, remaining reduced to the end of the test. The tip-clearance minimum-to-maximum value was low, 0.91 to 0.96, for a longer period of time see time, up to 294 s (Fig. 9d). In the 294 to 310 range (Fig. 9e), the tip-clearance value approached zero. The trajectory of the rotor blades with near zero tip-clearance is shown in Fig. 10.

The peak detector method was also applied to analyze the bearing failure. The entire test run time was divided into short periods: T. An average amplitude of all the seventh stage blades (RMS) was calculated for each T period:

$$\text{RMS}(t) = \left( \frac{1}{T} \int_{t_0}^{t} r^2(t) \, dt \right)^{\frac{1}{2}}$$  \hspace{1cm} (2)
where \( r_w(t) \) is the measured signal in period \( T \).

A DIL1 filter was used to measure the vibration signal of the rotor blades from 0.1 Hz to 1000 Hz.

The maximal RMS displacements value of seventh stage rotor blades (Eq. (2)) for an assumed test run (the plane x-y, RPM versus time) with the middle bearing working properly was equal to \( 7 \times 10^{-5} \) m.

A peak detector analysis provided an alternative method of studying the experimental results. This analysis is based on calculating new averaged blade amplitudes RMSn in periods \( T \) according to the equation [15]:

\[
\text{RMSn}(t + \Delta t) = \text{RMSn}(t) \left( \frac{t-\tau}{\tau} \right) + \frac{\text{RMS}(t + \Delta t)}{\tau}
\]

(3)

if \( \text{RMS}(t) > \text{RMSn}(t) \) then \( \tau = 400 \),

\( \text{RMS}(t) < \text{RMSn}(t) \) then \( \tau = 2000 \),

with \( \text{RMS}(t) \) taken from Eq (2).

A peak detector analysis Eq. (3) was also conducted in the case of bearing failure (see Fig. 12). The blue line in Fig. 12 is the run test (RPMn), while the red dashed line represents the RMSn (Eq. (2)). The damaged bearing caused increasing vibration amplitude fluctuations. The level of RMSn amplitude increased to 1.62 (Fig. 12), whereas in the engine with a normally working middle bearing it only increased to 0.9. The calculated signal variations reached 70% and were a symptom of increasing bearing failure, though the engine was still working. As mentioned above, in the case of the undamaged bearing, the RMSn varied by no more than 20%.

Fig. 13 presents the RMS displacements value of seventh stage rotor blades (Eq. (2)) for an assumed test run (the plane x-y, RPM versus time) in the case of bearing failure. The maximal rotor blade amplitude for 7EO (7x) was close to the nominal speed (see Campbell diagram Fig. 3). There were some other resonances whose origin was difficult to explain using the Campbell diagram of a rotor blade of the seventh stage compressor. The maximal values of RMS amplitude of rotor blades was equal to \( 7 \times 10^{-4} \) m.
Fig. 12 The peak detector amplitude (RMSn) of seventh stage compressor rotor blade of SO-3 engine – experiment with the damaged bearing

Fig. 13. The measured vibration amplitude RMS of the seventh stage compressor rotor blade in the case of bearing failure

Next a new test run was started (Fig. 14), in which, after 2000 s, bearing failure caused the seventh stage rotor blades to rub against the casing and the engine, had to be stopped. In this case, the RMSn value was almost 1.84.

Fig. 15 presents the RMS displacements value of seventh stage rotor blades (Eq. (2)) for an assumed test run (the plane x-y, RPM versus time) with fully damaged middle bearing. The maximal rotor blade amplitude for 7EO was close to the nominal speed (see Campbell diagram Fig. 3). There were some other resonances whose origin was difficult to explain using the Campbell diagram of a rotor blade of the seventh stage compressor. The maximal values of RMS amplitude of rotor blades was equal to $2e^{-3} \text{ m}$.

Fig. 14. Peak detector amplitude (RMSn) of seventh stage compressor rotor blade vibration with fully damaged middle bearing

 RMS blade amplitude increases with bearing failure increase.

**Conclusions**

Failure of the middle bearing in an aircraft engine was reported. In order to reconstruct this failure, metal filings were added to the bearing’s oil filter in a working engine. Tip-timing and tip-clearance analyses of seventh stage compressor rotor blades situated directly above the middle bearing were carried out. A comparative experimental analysis was also
carried out on an aircraft engine with an undamaged middle bearing. Next, a numerical analysis of the free vibration of a seventh stage blade was conducted to verify the experimental results. The method presented in this paper enables the prediction of middle bearing failure in an aircraft engine before it happens. The results show that variance first increases with bearing failure increase, but still before complete failure it next decreases, stabilizes and decreases when the bearing takes on a plastic state. Tip-clearance provides information about blade eccentricity. In our experiment, however, zero tip clearance only appears towards the very end.

A peak detector analysis was also conducted. RMS blade amplitude increases with bearing failure increase.

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**References**


