EXPERT SYSTEM TO SUPPORT OPERATIONAL SAFETY OF THE TS-11 "ISKRA" AIRCRAFT AND OVERHAULS OF THE SO-3 ENGINES

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Abstract. The paper has been intended to introduce the Tip Timing Method (TTM) to monitor rotating blades in an aircraft compressor, that was implemented in Poland: in maintenance of the TS-11 "Iskra" aircraft in 1993 and in the SO-3 engine overhaul technology in 1997. The scope of research performed before the implementation of TTM and its advisory expert-software have been outlined. On the base of conducted experimental research, it has been confirmed that computer-based system to support operational safety of the TS-11 "Iskra" aircraft and overhauls of engines SO-3 has enabled to actively control a process of material fatigue. As a result of reason identification of fatigue problems, flight safety of the TS-11 "Iskra" aircraft was improved and the following problems were eliminated from operational use: a fatigue problem with I stage rotor blades in compressors and instable operation of the SO-3 engines. Unconscious human errors were also eliminated and technical culture of maintenance and overhauls of the SO-3 engines was improved. Reasons for other technical problems with the SO-3 engines were reliably determined. Presented topic has been illustrated by means of many practical examples.

Introduction

There are many different low cycle fatigue (LCF) and high cycle fatigue (HCF) failures we can observe throughout the turbine engine's life. Fatigue cracks propagating in rotor blades, incorrect control of the engine's fuel system and lack of knowledge on the loads affecting the bearing system generally cause a formidable hazard to flight safety, as well as to engine life and reliability. Detailed analysis of each of the engine faults that have occurred lately in the Polish Air Force shows that about 70% of all faults observed during the turbine engine operation resulted from the poor quality of operation of the fuel system (combustion chamber and exhaust system, fuel system, turbine section) [1,2]. Therefore, the AFIT keeps looking for methods of recognising stochastic loads during the engine's running, and the effects thereof upon the engine's structural reliability [3-5]. Moreover, the Institute seeks methods that will enable the user to both detect the structure degradation (e.g. cracks, overheating) and actively control the material fatigue process [4].

The paper presents a non-contact blade-vibration measuring technique – the tip timing method (TTM), which is one of the most interesting methods of complex diagnosing of jet engines/power turbines/compressors and a powerful tool to investigate dynamic phenomena during operation of the machine [6-10]. This method is used and developed mainly by producers of aircraft engines, despite the fact that it has origins in power engineering [6]. The method has been used in the Polish Air Force since 1993 with the SNDŁ-1b/SPŁ-2b as a diagnosing system developed for the SO-3 engines. Since 1997 this method has been also used in the post-repair/post-overhaul acceptance tests with the CTM-PER/SPŁ-2b diagnosing system. The theoretical bases of the TTM and selected experiences from its use have been presented in the paper. The attention has been drawn to the non-contact monitoring of rotating blades' vibration and detection of their cracks.

1. Motivation

In the years 1975-91 as many as 25 first-stage compressor blades of ten SO-3 engines suffered fatigue-attributable break-offs, which caused two accidents. The metallographic examination of damaged blades made out of the 18H2N4WA steel has proved that the crack initiation centres were located either on the leading edges (55%) or on the blade-back surfaces (45%), in the areas of nodal lines of the first mode vibration. Crack propagation occurred at low-level stresses (HCF problem). Fatigue fracture covering as much as 95% of the blade’s cross section was found in one of the blades. Furthermore, it has also been found that erosion and corrosion, both occurring on the blade’s face surface, as well as fine mechanical damages on the leading edge are stress concentrators [1]. Only on few blades the low cycle fatigue has been observed - the initiation and very fast crack propagation occurred at high-level stresses of the blade - Figure 1. The HCF problem was also observed in titanium blades (Ti5.8Al-3.7Mo) in the TW3-117 engines in the years 2005-2007 [4]. The gigacycle fatigue of compressor blade (VHCF problem) with “fish eye” symptoms under the blade surface has been observed at foreign users [11]. The VHCF crack is initiated inside the material in the area of soft inclusions at a normal level of the material stress and the number of load cycles $N > 10^8$ [12].

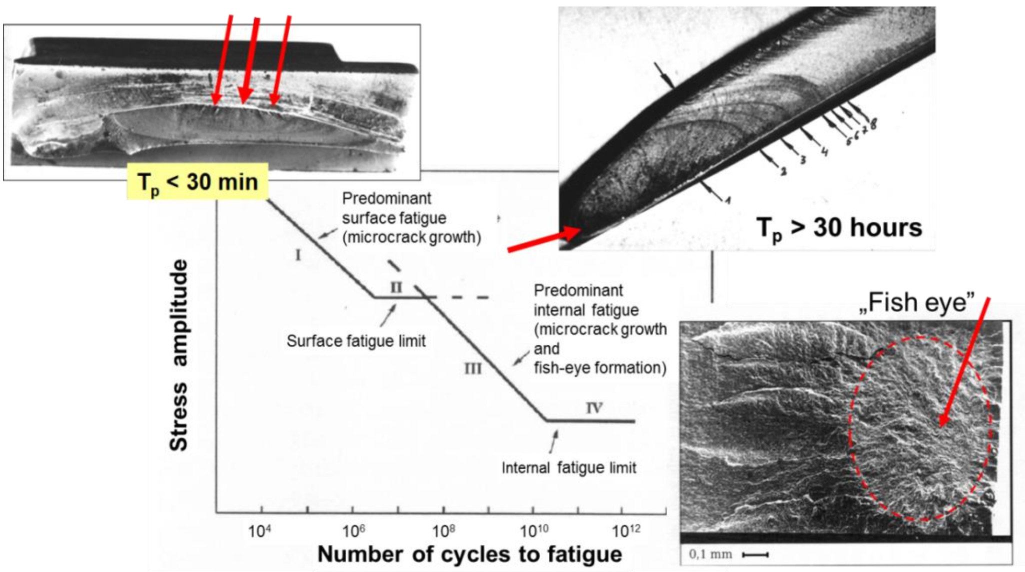


Fig. 1. Fatigue problem of compressor blades [4,12]

Uncontrolled blade fatigue:

- is a threat to service safety,
- limits of the aircraft engine life time,
- increases maintenance costs.

It is also a great challenge for a diagnostics engineer.

If blade fatigue is found, who/what is responsible for the problem?

Classical NDT methods (eddy current, ultrasound, magnetic and fluorescent) proves very low effectiveness of diagnosing blade crack before damage. This is because:

- crack gap closing during engine standstill (about 50% of crack area after 12 hours);
- lack of reliable information about real operating conditions;
- lack of knowledge about early cracking symptoms and mechanisms;
- difficult access to tested blades (because of inlet stator vane).

Other disadvantage of the NDT methods in use (during overhaul and service) is no possibility of fatigue prognosis. In the case of the SO-3 engine it is very important because of errors found in the design of the 1st compressor blades – too low first-mode mistuning form the 2nd rotational harmonic excitation. Due to this, too high stress and fast fatigue crack initiation can occur during operation. These conditions take place during the take-off phase when there is a foreign object lying in the inlet or the inlet icing occurs. Under such conditions the time between crack initiation and blade damage can be shorter than the time of a single flight.

Using both statistical records of operation of the SO-3 engine and the results of bench tests (non-destructive tests) it has been concluded that it is necessary to monitor real operating conditions of the first stage compressor blades to increase safety of the engine operation, without costly correction of faults the design of the blades. An intuitive diagnostic symptom of a blade crack is a change in its modes frequency. The cracking propagation and blade break off occur at a limited decrease in frequency, the value of which depends on the crack centre's position and the loading history. Blades' frequency check during ground handling offers too short prognosis horizon. It is sufficient in the system monitoring only; for example, in the tip-timing method.

2. Tip Timing Method

The tip timing idea consists in observing displacement of a loaded component part. In our case, it will be rotating and vibrating blades – *rotor's rotation phase markers* The sensor (observer) is built on a fixed part of machinery. A palisade of rotating and vibrating N_B blades and a stationary sensor create a specific encoder, in which the time of arrival of blades $TOA_B(k)$ depends on [4]: ζ_B – jitter of blades group components, ζ_ω – jitter of rotor group components, $TOA_T(k)$ - theoretical time of arrival of a blade from an ideal rotor (without errors of scale, vibration and seating of the rotor in supports resulting from the momentary angular speed $\omega(k)$.

$$TOA_B(k) = \frac{[1+\zeta_B(k)]}{[1+\zeta_\omega(k)]} TOA_T(k) = [1+\zeta(k)] TOA_T(k) \tag{1}$$

$TOA_T(k) = (2\pi/N_B)/\omega(k)$ is an aperiodic component resulting from the momentary average rotational speed of the ideal compressor rotor n.

The jitter $\zeta_B(k)$, described by the equation (2), is generated by:

- scale errors ζ_P (N_B variables);
- vibration of the blades taking part in the measurement cycle with the frequency dependent on the rotational speed $\zeta_{L,k}$ ($k \cdot N_B$ variables, where k – number of the analysed blade modes);
- frame vibration ζ_K;
- effects and phenomena used in the sensor, e.g. magneto-mechanical effects for the inductive sensor ζ_{ZD};

$$\zeta_B(k) = \zeta_P(k) + \zeta_{Ł,k}(k) + \zeta_K(k) + \zeta_{ZD}(k) \tag{2}$$

The jitter $\zeta_\omega(k)$, described by the equation (3), is generated by:

- fluctuations of the rotational speed (low frequency interferences of the fuel system by stationary power/engine thrust lever) ζ_{UP};
- transverse and torsional vibration of the rotor: ζ_{WP} and ζ_{WS}, respectively;
- alignment errors of the real rotor - parallel shift and axis inclination ζ_O and ζ_S.

$$\zeta_\omega(k) = \zeta_{UP}(k) + \zeta_{WP}(k) + \zeta_{WS}(k) + \zeta_O(k) + \zeta_S(k) \tag{3}$$

2.1 Measurement

In the tip timing method, the value that is precisely measured is the time of arrival $TOA_S(k)$ of the characteristic signal point $U(t)$ of the encoder, which indirectly reflects the blades' time of arrival under the sensor $TOA_B(k)$. The shape of the signal $U(t)$ and characteristic signal points depend on the sensor type and geometric and physical features of the blades, that is, among others: blade magnetisation, tip clearance, blade incidence angle, chord thickness and health. In the case of variable reluctance sensors, the characteristic point is most often the passing of the signal over the zero level on the trailing edge. The encoder's signal is most frequently delayed in relation to the real blade position. The resultant delay t_{delay} of the measurement chain (system error) changes with the rotational speed n, which is described by the following relation:

$$TOA_S(k) \approx TOA_B(k) + t_{delay}(n) \tag{4}$$

The interval between the blades is measured by the frequency method with the resolution $\Delta t \in \langle 5\,ns; 1000\,ns \rangle$, resulting from the criterion of the required blades' vibration amplitude resolution – the shorter the blade, the higher resolution of the time measurement is required.

2.2 Signal Analysis

The continuous signal $U(t)$ contains aperiodic part $A(t)$, oscillating part $P(t)$, noise/weak oscillating components $I(t)$

$$S(t) = A(t) + P(t) + I(t) \tag{5}$$

Similarly to the discrete signal of $U(t)$, the time $TOA_S(k)$ contains aperiodic part $A(k)$, oscillating part $P(k)$ and noise/weak oscillating components $I(k)$, it is possible, thus, to design a general-purpose observer for <u>real operating conditions</u> of rotating parts and have a complex view on:

- ✓ Disadvantageous dynamic phenomena (flutter, stall, surge, resonance, load coupling);
- ✓ Influence of production, overhaul and maintenance real conditions on the level of malfunctioning and fatigue prognosis.

The jitter ζ_B and ζ_ω of (1) are the source of the AM-FM modulation. The resultant jitter ζ of (1), after taking (2) and (3) into account is described by the relation (6), which reflects the analytical complexity of the TTM. The spectrum ζ contains characteristic stripes of the diagnosed process and stripes of the AM-FM modulation of the jitter components [4,10].

$$\zeta(k)=\frac{1+\zeta_P(k)+\zeta_{Ł,k}(k)+\zeta_K(k)+\zeta_{ZD}(k)}{1+\zeta_{UP}(k)+\zeta_{WP}(k)+\zeta_{WS}(k)+\zeta_O(k)+\zeta_S(k)}-1 \tag{6}$$

For the analysis of the engine elements' health on the basis of the TOA_S, decomposition of the measured signal $TOA_S(k)$ on the A, P, I components from the equation (5) and determination of the blade jitter components ζ_B and rotor jitter ζ_ω are required. For this purpose specialised algorithms are used, which have to control also the measurement signal quality and correct "grave" errors. Signal components of TOA_S are obtained with the numerical processing, narrow-band filtering and AM/FM demodulation – Figure 2. Blade vibrations are shown in the form of phase distributions as points of phase trajectory crossing the phase plane (Poincare map). Such imaging allows detection of the LCF and HCF crack initiation and propagation symptoms in the blade spectrum during the engine operation.

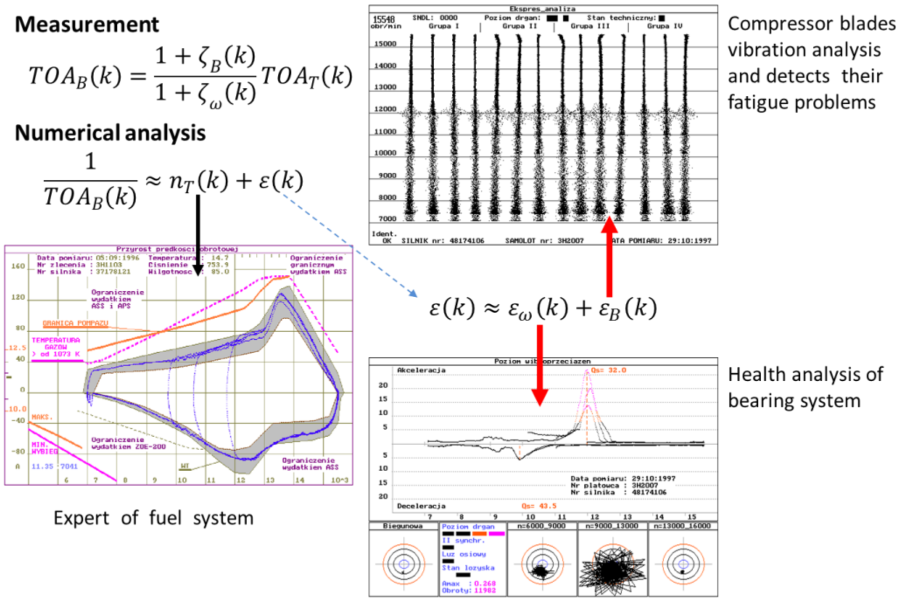


Fig. 2. Complex engine health analysis on the basis of TOA_S signal in the SPŁ-2b software [1,2]

3 AFIT Experience

To solve the expert analysis of compressor blade and engine health, the qualitative evaluation of applicability of the tip timing method was done in 1987-1993. Particular attention was paid to the possibility of estimating the blade health (crack initiation and propagation) on a running engine [1]. Validation of the non-contact blade-vibration measuring method with the strain - gauges was followed with the bench tests of the SO-3 engines, in the course of which the following activities were carried out:

- initial identification of vibration of the first stage compressor blades, and
- verification of algorithms of the ‘reason - effect’ identification of sources of the engine’s accelerated LCF and HCF.

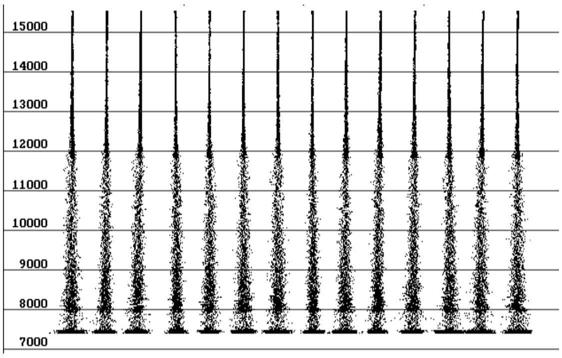
In 1993, a diagnostic system was developed and introduced into the service on the TS-11”Iskra” trainer. It consists of [13]:

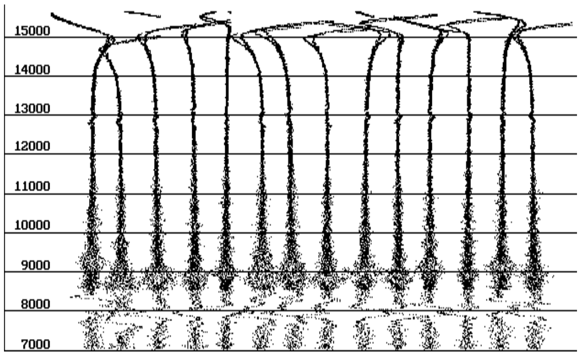
- the blade excessive vibration warning device SNDŁ-1b,
- the ground-based inspection instrument SPŁ-2b,
- the SPŁ-2b software.

In 1997 a real time module on the basis of the Keithley's CTM-PER counter card [14] with the clock rate of 10 MHz has been added to the already existing diagnostic system and is now used in the repair works.

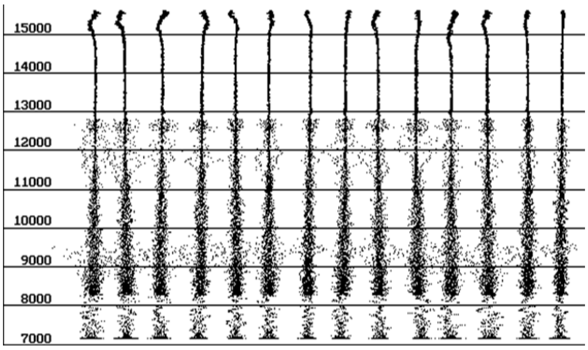
3.1 Monitoring of Blade Vibration

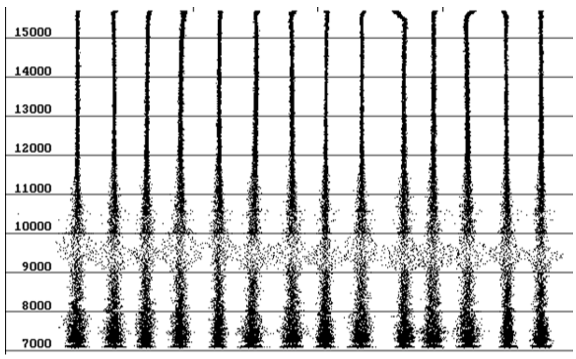
During the examination with a strain gauge no evident symptoms of interrelationships between the disk and blade vibrations were observed. However, it was observed that within the take-off range of the SO-3 engine (n=15600 rpm) operation, resonance with forces from the 2^{nd} harmonic of the rotational speed (f_{1mode} = 520 Hz) may occur – Figure 3.b). The blade vibration spectrum is a very sensitive indicator of the quality of the adjustment of the fuel system, in particular of the engine's work on the border of the surge (compressor's static work) - Figure 3.c). It is also possible to identify the range of the coupling of rotor vibration with blade vibration - Figure 3.d).





a) b)





c) d)

Fig. 3. Phase mapping of blade vibration and ratio of the SO-3 engine compressor [1]: a) normal stress level; b) synchronous resonance - influence of a foreign object in the compressor inlet; c) asynchronous resonance - influence of developed rotating braking-off areas; d) asynchronous resonance – influence of rotor resonant vibration

3.2 Bench Tests of Blade Cracking

After the analysis of destructive testing results (controlled propagation of blade cracking under normal conditions of operating the SO-3 engine) it was found that [1]:

- during the initiation of blade cracking (no open crack visible on the blade surface) – Figure 4.a), only change in the B factor of dynamic increment of blade vibration frequency is seen – frequency of the blade’s free vibration is constant. The observed changes in modal properties reflects the end phase of cyclic material weakening;

- the occurrence of a blade crack – Figure 4.b), decrease of the range of excitations from the rotational-speed II harmonic by 1000 rpm (Δf_n = 16.6 Hz). At the moment, the frequency (the 1st mode) of the blade's free vibration changed by less than 3 Hz;
- when the crack reaches about 30% of the blade profile – Figure 4.c), evident reduction in the frequency of free vibration and decrease in the range of excitations from the rotational-speed III harmonic was observed;
- just before the blade breakaway – Figure 4.d), (65% profile – the crack from the edge of attack, 95% of profile – the crack from the back of the blade), a clear effect of stiffening due to centrifugal forces was observed. Changes in dynamic scale inflicted by the broken blade are comparable with those in other dynamic scales (the influence of the engine's rotational speed).

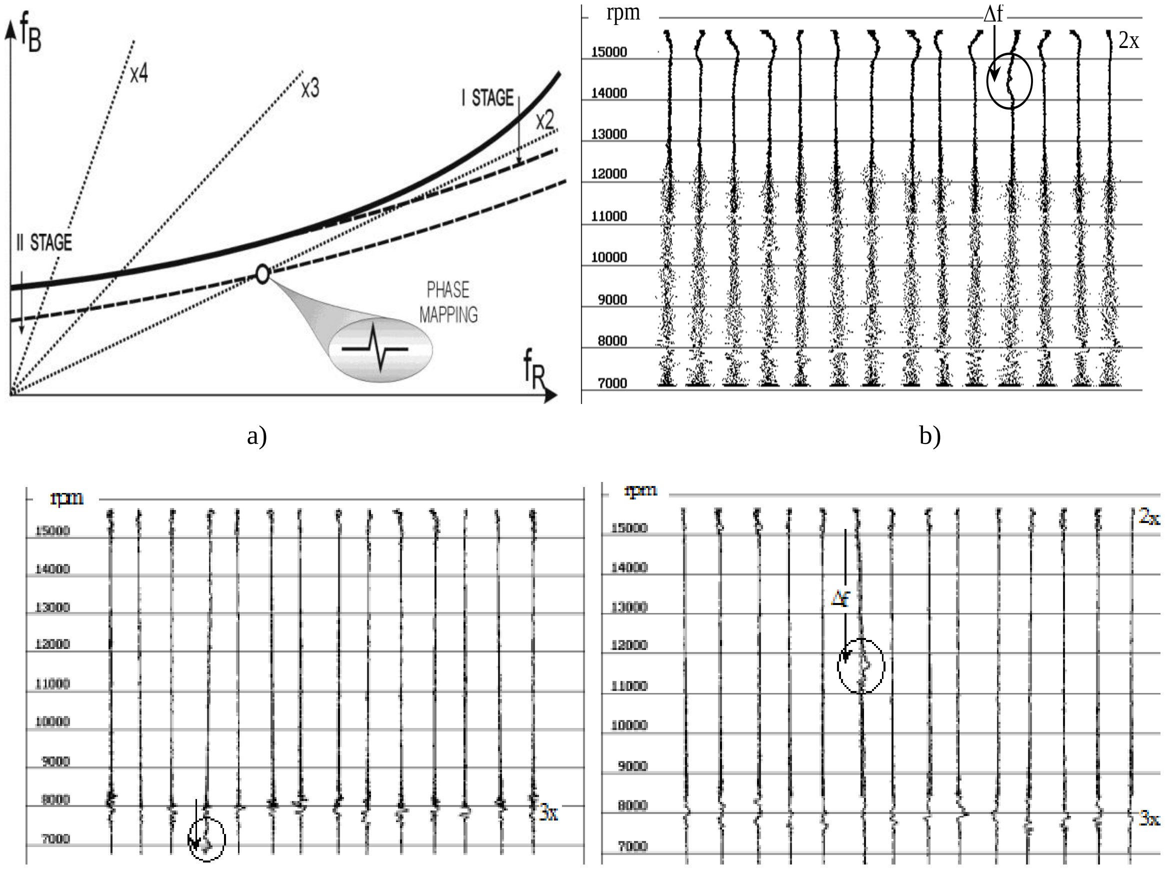


Fig. 4. The effect of blade cracking as phase representation of blade vibration [1]: a) blade frequency plotted in the Campbell diagram; b) first stage crack of blade – changes only B; c) second stage crack of blade – changes f_B(n=0) and *B*; d) finish stage crack of blade – 5 minute before break

Repeated destructive of the SO-3 engine indicated the credibility of the above mentioned diagnostic symptoms of the cracking blade. Some safe diagnosing time-horizon has been accepted for both the diagnostic system under design and the user:

- **50±10 hrs** – if the level of blade vibration for the cruising and take-off ranges of speed is low, and all blades of the compressor's 1st stage show the II harmonic of the I-mode vibration above 15200 rpm;
- **25±5 hrs** – if the level of blade vibration for the cruising and take-off ranges of speed is raised, and at least one blade shows the II harmonic of the I-mode vibration within the range 14700 ÷ 15200 rpm;
- **12±3 hrs** – if the level of blade vibration for the cruising and take-off ranges of speed is raised, and at least one blade shows the II harmonic of the I-mode vibration within

the range 14300 ÷ 14700 rpm. The aircraft can be operated during the day time only, and under favourite weather conditions.

3.3 Safety results

Thanks to the active control of the user over the material fatigue process the following have been eliminated: cracking of compressor blades in operation despite the existence of uncorrected design flaws and high LCF risk in the take-off range. The statistical time between blade cracks has been prolonged for over **1500%**. Other fatigue problems occurring in the SO-3 engines have also been minimised.

4 Conclusion

Over the 20-year operation of the SNDŁ-1b/SPŁ-2b diagnostic system proved that the Tip Timing Method is very effective as:

- contactless method of non-destructive testing and health monitoring of rotating blades,
- objective method for monitoring the operation quality and repairs of the SO-3 engines.

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