Structural Health Monitoring of Compressor and Turbine Blades with the Use of Variable Reluctance Sensor and Tip Timing Method

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Abstract. The comprehensive approach for rotating fan, compressor and turbine blades diagnostic has been introduced in the paper. Critical slow- and high-speed rotating fluid-flow machinery (fans, steam turbines and aero jet engines) running at risk of mechanical damage (by foreign objects and erosion), corrosion and other forms of material fatigue (LCF, HCF, VHCF, TMF). Objects endangered with variations of modal characteristics by blade mass change (influence of deposit) and rate of material anisotropy.

For blades' real operating condition and technical state monitoring purposes a rotating blade observer method (Tip-Timing method) has been used. Combination of supervised rotating blade row and magnetic reluctance sensor create sort of an encoder which output signal contain simultaneous information about:

- blade vibration induced by aerodynamic and mass force input;
- instantaneous rotor rotation speed;
- rotor unbalance and vibration;
- coupling conditions of reluctance sensor with vibration and rotating blades.

Measured values are blades' times of arrival (TOA) at the stationary observer - reluctance sensor mounted on assembly casing. TOA is modulated by aperiodic (instantaneous ideal rotor rotation speed) and periodic components (blades and rotor vibrations). Measurements of TOA were implemented by frequency method available for typical counter card and AD/DA converters. Numerical processing for recorded (non-uniformly sampled) data was utilized in order to separate TOA signal components and extract diagnostic symptoms. Phase mapping method was employed for data analysis as well.

Discussed problem is supported with experimental data from destructive running engine testing (active research) and perennial method operation in Polish Air Force as well as implementation works for aviation and power industry (passive research).

Introduction

There are many different low cycle fatigue (LCF), high cycle fatigue (HCF) and thermo-mechanical fatigue (TMF) failures we can observe throughout the turbomachine's life [1,2], among others cracks and destruction of the blades (fan, axial compressor and turbine), Figure 1. To detect the growing threat against the state of emergency and safe operation are required: condition monitoring (CM), non-destructive testing (NDT) and structural health monitoring (SHM).

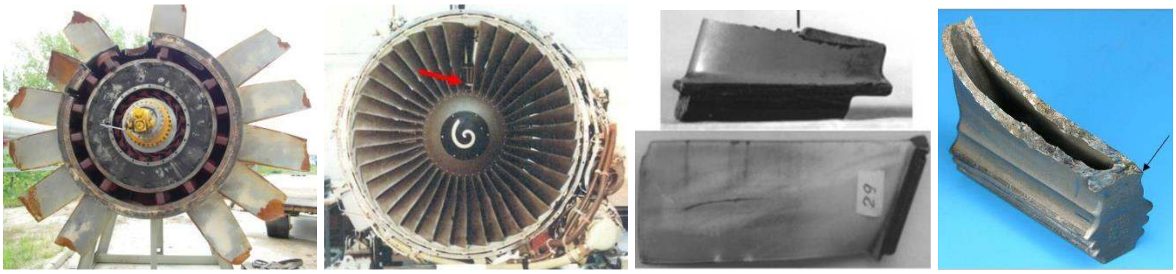


Fig. 1. Fatigue problems of fan, compressor and turbine blades [3, 4]

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 turbomachines (e.g. jet engines, power turbines, compressors) and a powerful tool to investigate dynamic phenomena during operation of the machine [4-13]. This method is used and developed mainly by producers of aircraft engines, despite the fact that it has origins in power engineering [14]. The TTM 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 [15]. Since 1997 this method has been also used in the post-repair/post-overhaul acceptance tests. Currently, the Institute has knowledge, a multi-channel test instrumentation and software for monitoring the rotating blades of various turbomachinery (fans, compressors, turbine of aero-engines, steam and gas turbines).

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. Tip Timing Method

The tip timing idea consists in observing and analysing displacement (quasi-static and vibration) of a loaded component part which rotate. In our case, it will be rotating and vibrating blades of fan, compressor and turbine. The sensor (or some sensors) is built on a fixed part of machinery, Figure 2. A palisade of rotating and vibrating N_B blades and a sensor create a specific rotary encoder, in which both changing the angular distance between the phase marks $\varphi_{i,i+1}(t)$ and the instantaneous angular velocity $\omega_{i,i+1}(t)$.

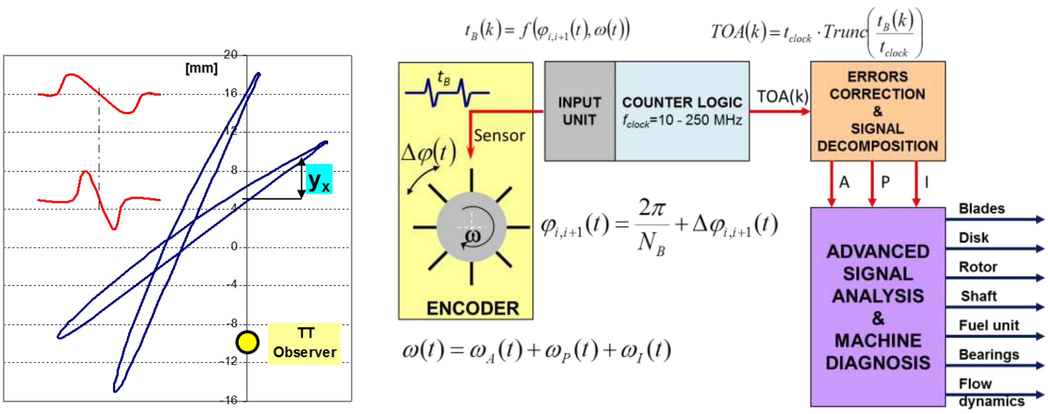


Fig. 2. Idea of the tip timing method (input unit includes low-band signal filter, preamp and Schmidt trigger; counter logic includes the standard frequency generator and counter; A – aperiodic component, P – periodic component, I – interference and noise, y_x – the impact of blade vibration to change the trigger point)

The discrete time of blade arrival $TOA_B(k)$ depends on: TOA_T - theoretical time of arrival of a blade from an ideal rotor without errors of scale, vibration and seating of the rotor in supports, ζ_B – jitter of blades group components, ζ_ω – jitter of rotor group components

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

$TOA_T(k) = (2\pi/N_B)/\omega(k)$ is an aperiodic component resulting from the number of the phase marks (blades) N_B and the momentary average angular velocity of the rotor ω.

The jitter $\zeta_B(k)$ 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};

The jitter $\zeta_\omega(k)$ 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.

The spectrum of the total jitter ζ contains both characteristic striations of the diagnosed process and striations of the AM-FM-PM modulation of the jitter components [3].

Equation (1) can be presented in the additive form (2) wherein A(k) is aperiodic part, P(t) is periodic part, I(t) is interference and noise/weak oscillating components.

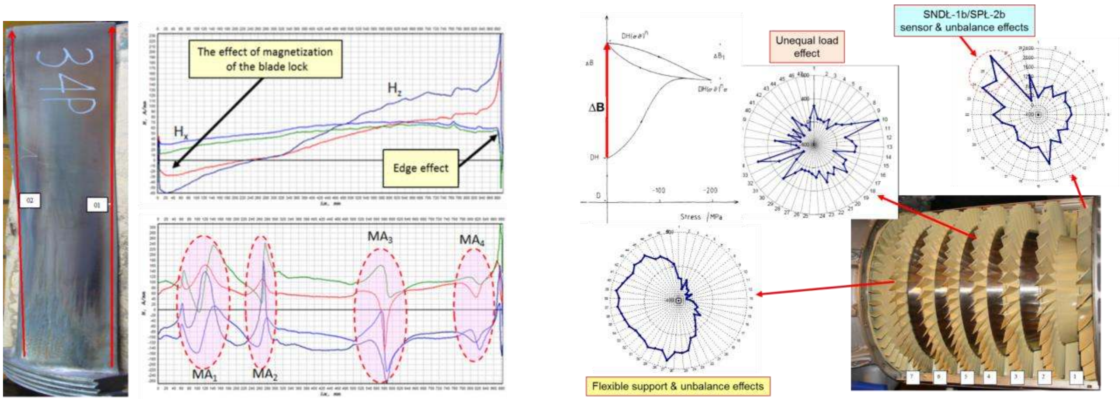
$$TOA_B(k) \cong (1+ f(\zeta_\omega)) \cdot (1+\zeta_B) \cdot TOA_T(k) = A(k) + P(k) + I(k) \quad (2)$$

1.1. 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 S(t) of the encoder (voltage or current), which indirectly reflects the blades' time of arrival $TOA_B(k)$ under the sensor. The shape of S(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 [3,9]. In the case of variable reluctance sensors (VR, used by authors in the described applications) and thin blades, the characteristic point is most often the passing of the signal over the zero level on the trailing edge [16]. For thick blade profile or a large turning angle of the blade chord characteristic points of signal S(t) are local extremes (max and min value) that map the edges of the blades. The optical sensor accurately detects the blade edges [9,17], but does not provide information about:

- the state of blades magnetisation - a symptom of load history and early phase of ferromagnetic material degradation, Figure 3 and Figure 8;
- tip clearance, modulated by the vibrations of the rotor, machine housing and monitored the blades, Figure 8.

The interval between the blades is measured by the frequency method with the resolution $t_{clock} \in \langle 5\,ns; 1000\,ns \rangle$, resulting from the criterion of the required blades' vibration amplitude resolution – the shorter the blade and/or higher rotational speed of the rotor, the higher resolution of the time measurement is required [3, 9]. For the very precise TOA measurements, when the resolution equal to $t_{clock} \in \langle 10\,ps; 5\,ns \rangle$ is required, the **T**ime to **D**igital **C**onverter (TDC) with the delay line method is used [18].



a) b)

Fig. 3. The influence of an unknown load history on the magnetization of: a) LP blades of steam turbine (new blade and after overload, MA_i – the magnetic anomaly correlated with nodal line of vibration, the state before crack) [19]; b) compressor blades of jet engine SO-3 [3]

1.2. Signal Analysis

The measured time $TOA_S(k)$ includes aperiodic part A(k), periodic part P(k), interference and noise/weak oscillating components I(k), it is possible, thus, to design a general-purpose observer for real operating conditions 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.

For the analysis of the machine elements' health, decomposition of the measured signal $TOA_S(k)$ on the A, P, I components and determination of the blade jitter ζ_B and rotor jitter ζ_ω are required. For this purpose specialised algorithms are used [3,9,20-22], which have to control also the measurement signal quality and correct "grave" errors. The impact of resolution time measurement for spectrum of $TOA_S(k)$ components is also taken into account, Table 1.

Table 1. The impact of measurement resolution on $TOA_S(k)$ components (for one-tone ζ_ω and 1st stage compressor blades of engine SO-3) [3].

$f_{clock} \in (1\,MHz, 10\,MHz)$	$f_{clock} \in \langle 10\,MHz, 800\,MHz \rangle$
$A(k) = TOA_T(k)$	$A(k) = TOA_T(k)$
$P(k) = (L + W_1) \cdot TOA_T(k)$	$P(k) = (L + W_1 + W_2) \cdot TOA_T(k)$
$I(k) = (W_2 + L \cdot W_1) \cdot TOA_T(k) + noise(k)$	$I(k) = \left(W_3 + L \cdot (W_1 + W_2)\right) \cdot TOA_T(k) + noise(k)$
where: $L = \zeta_B(k)$ and $W_i = \zeta_{\omega,i}(k)$ is the i-harmonic of ζ_ω from $f(\zeta_\omega) = \sum_{i=1}^{N} \zeta_{\omega,i}$ with relationships (2).	

The major concept of expert analysis of the machine health is the projection of the actual characteristics of some machine systems (e.g. compressor or turbine blades, a fuel system or a bearing system) in the phase space [3,9,15], Figure 4.

2. Some Results

2.1. Monitoring of Blade Vibration Frequency

The advantage of the tip timing method is the possibility of fast identification of modal properties of tested blades, which for a tester most often are "black-box" objects, including the assessment of own vibration frequency of a given f_i mode and dynamic increase coefficient of this frequency B_i under the influence of centrifugal forces – Figure 5. The obtained measurement result enables to define new diagnostic criteria, which decrease the risk of work of blades in adverse conditions, e.g. in asynchronous resonance during transition states.

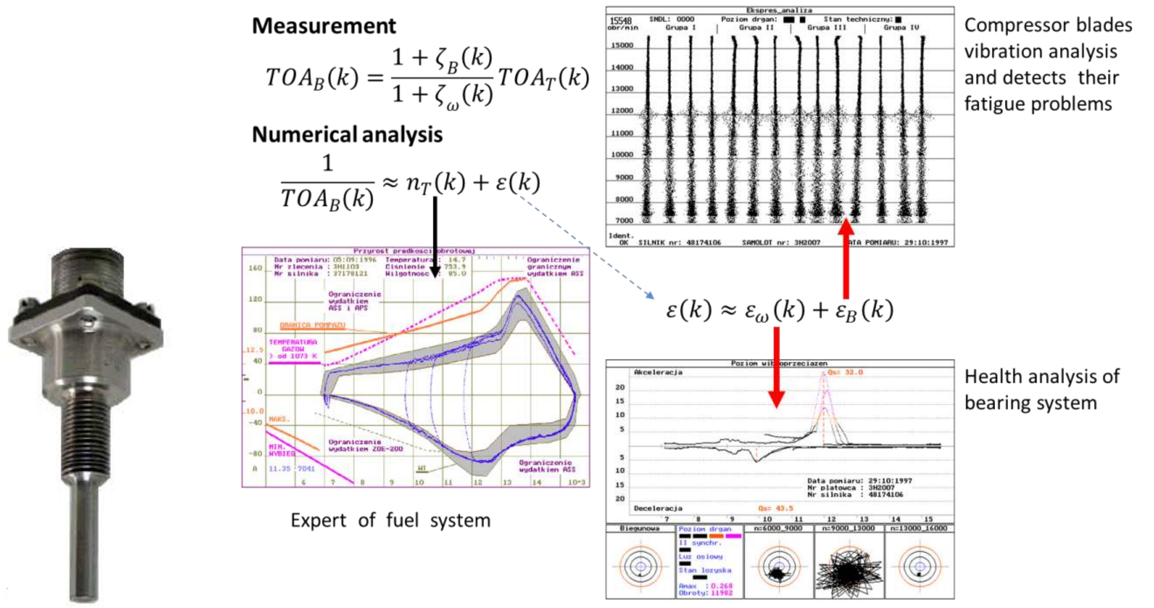
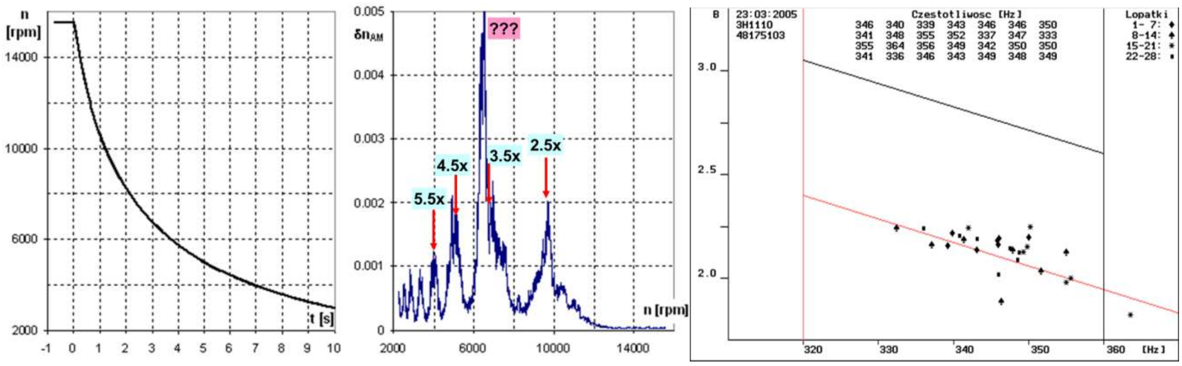


Fig. 4. Examples of VR sensor and imaging results on the phase plane (SO-3 engine, f_{clock} = 10 MHz) [15]



a) b)

Fig. 5. Visualised [3,9]: a) identification of reaction order of blade vibration during dynamic transition states; b) aggregated result of the 1st mode's own vibration frequency $f_{L,1}(0)$ and $f_{L,1}(n) = f_{L,1}(0) + B \cdot f_n^2$

2.2. Health Monitoring of Compressor Blades

The TTM allows the identification of the adverse effects of blade dynamic, Figure 6. Symptoms of cracking blade (increase of cyclical weakness and level of non-linearity, decrease of 1st mode frequency) illustrated in Figure 7.

2.3. Monitoring of Fan Blades

The fan blades in a desulphurisation plant are exposed to deposition of dust and reduce the frequency of the different vibration modes. The surface of the blades is also exposed to chemical corrosion. Ability to monitor the fan blades with variable pitch (load) illustrated in Figure 8. Application of TTM was dictated by two cases of the break blades in one year and the search for the cause accelerated fatigue.

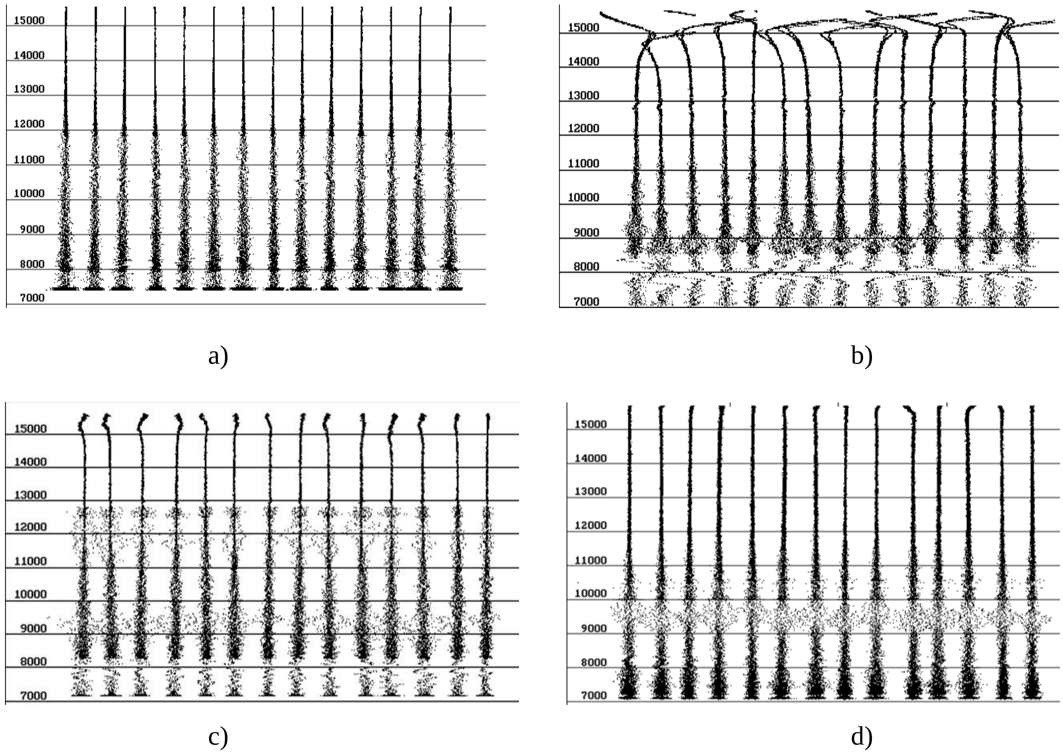


Fig. 6. Phase mapping of blade vibration and ratio of the SO-3 engine compressor [6]: a) normal stress level; b) synchronous resonance - influence of a foreign object in the compressor inlet; c) asynchronous resonance - influence of stall during acceleration; d) asynchronous resonance – influence of rotor resonant vibration

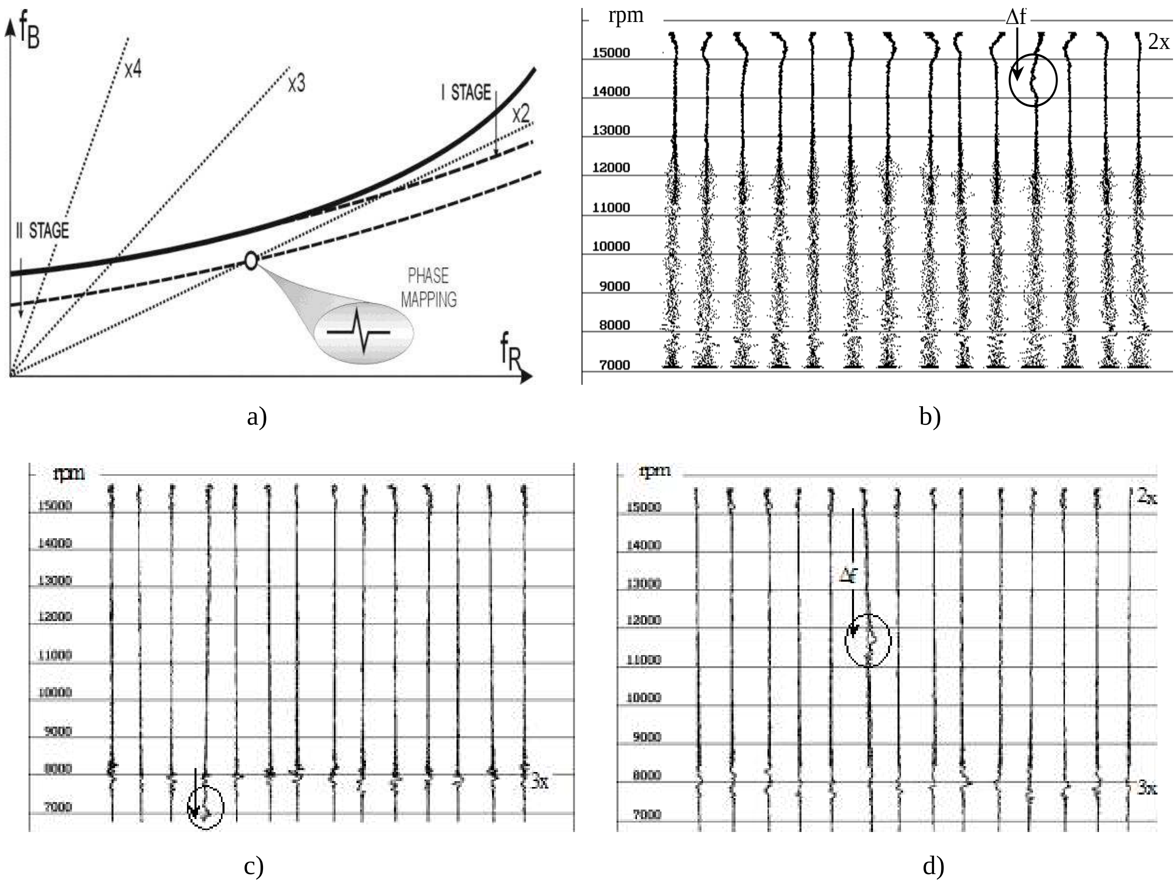


Fig. 7. The effect of blade cracking as phase representation of blade vibration [6]: a) blade frequency plotted in the Campbell diagram; b) cyclical weakness - first stage crack of blade fatigue, only decrease in B ratio growth dynamic frequency; c) second stage crack of blade – changes $f_B(0)$ and B, signal after low band filtration; d) finish stage crack of blade – 5 minute before break, signal after low band filtration

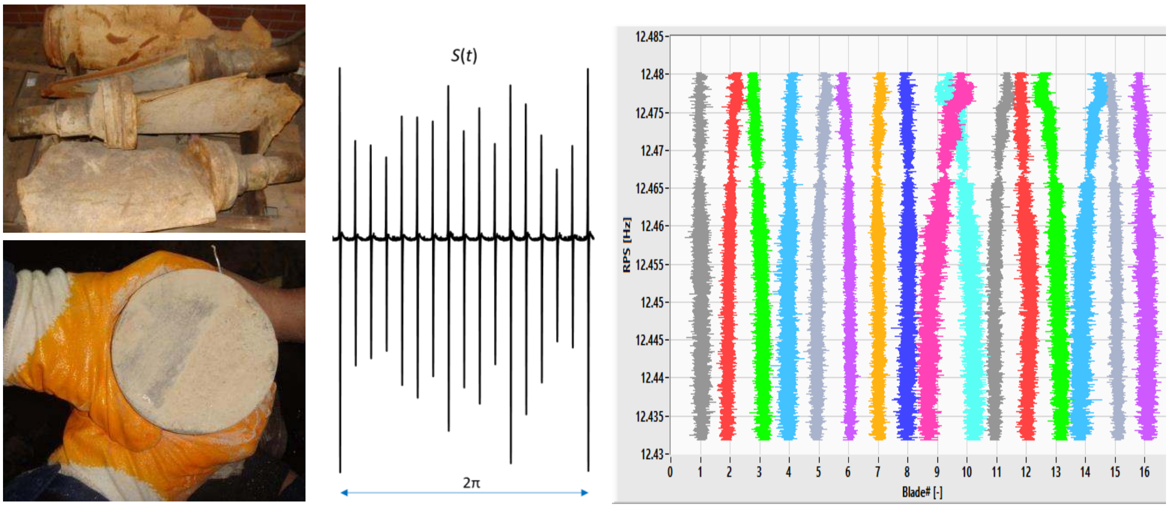
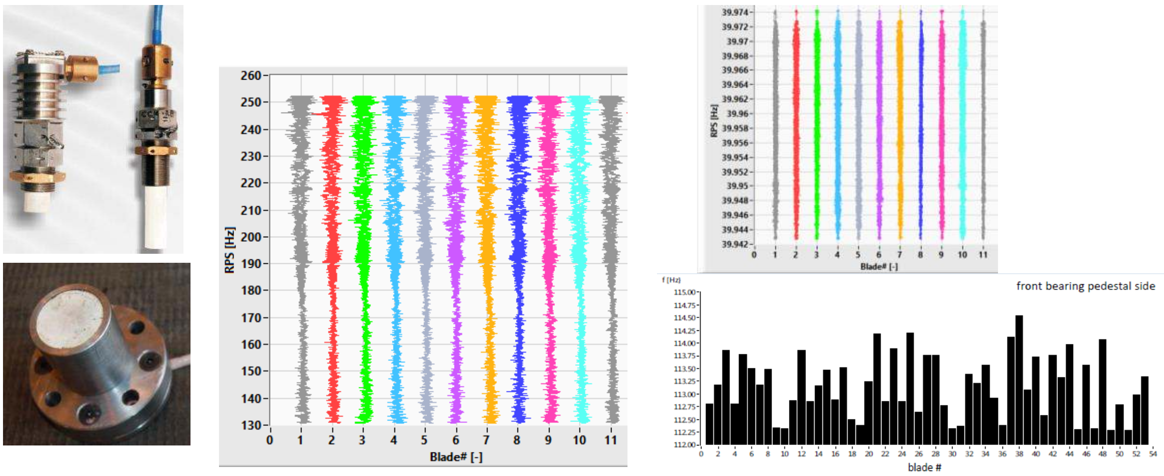


Fig. 8. Non-contact monitoring of fan blade flue gas desulphurisation plant (n_{max} = 750 rpm, f_{clock} = 80 MHz)

2.4. Monitoring of Turbine Blades

Fatigue problems occur also in turbine blades (aircraft engines, steam and gas turbines). In the objects working at the temperatures from 400 to 1500 K. The turbine blade vibration spectrum depends on modal properties of the blade and spectrum of the main source of temperature/pressure oscillation [9].

Non-contact measuring of rotating turbine blade vibration requires to use special sensors adjusted to suit a long work in high temperatures [9,23], Figure 9.a). The maximum amplitude of turbine blade vibration is at least one order of magnitude smaller than compressor blade vibration with similar size of blades, which requires the use of instruments with higher resolution of time measurement. The order of blade vibration (the frequency of individual modes related to the rotational frequency) and level of interferences (e.g. frame vibration excited from the outlet nozzle) are also higher. As a result, signal to noise ratio (SNR) and the level of diagnostic symptoms is lower. For their identification algorithms and methods of discrete signal analysis more sensitive than algorithms used for diagnostic of compressor blades are required. The Institute takes the above mentioned diagnostic needs reported by various users of turbomachinery (from aviation and power industry) into account. Examples of the TTM measurements of the turbine blades illustrated in Figure 9.b) and Figure 9.c).



a) b) c)

Fig. 9. Illustrated: a) high temperature metallic-ceramic VR sensors contain one and four coils (the top and bottom photo, respectively); b) vibration of turbine blades of turbofan engine RD-33 (f_{clock} = 200 MHz); c) vibration of ND-37 blades of steam turbine 18K370 (f_{clock} = 100 MHz)

3. Conclusion

Over the 20-year operation of the SNDŁ-1b/SPŁ-2b diagnostic system proved that the tip timing method is very effective as non-contact method of NDT and SHM of rotating blades. Since 1991 no compressor blade crack in the SO-3 engine operation has occurred - the statistical time between blade cracks has been prolonged for over **1500%**.

The possibility of active controlling of the material fatigue by the aircraft engine user through interference in the fuel system adjustment quality has also been confirmed.

The tip timing method and variable reluctance sensors are also effective in diagnosing industrial turbomachinery. Sensors with 4 coils have been operating reliably in steam turbine 18K370 since 2013.

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