CONDITION – BASED MAINTENANCE OF TURBOJET ENGINES BASED ON COMPRESSOR BLADE VIBRATION MEASURING METHOD

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ABSTRACT

The structure of a condition monitoring system to diagnose some turbojet engines has been discussed in the paper. System approach to the actual technical problem has been exemplified with fatigue crack growth within the first-stage compressor blades. Results of statistical analysis of the process of blades fatigue cracking have issued and followed with a concept of solving the problem. An optimum measuring method has been suggested as well. The method of measuring precisely the engine rotational speed and the amplitude of vibration of the first-stage compressor blades has been also discussed. Results of numerical analysis of measurements follow. The application of only one measuring line to complex, dynamic diagnostic of the engine technical condition of the engine fuel and bearing systems. The operational experience gained in the course of implementing the diagnostic system into the Air Force of the Republic of Poland is also presented.

Nomenclature

t	- time variable;
n	- engine rotational speed;
m	- the number of analysis signals;
CV	- condition vector;
TS	- technical specifications;
m_{pal}	- mass fuel consumption;
m_{pow}	- mass air consumption;
Q_{pal}	- volumetric fuel consumption;
M_T	- turbine rotational moment;
M_S	- compressor rotational moment;
M_F	- others assemblies rotational moment;
Par_i	- simple signal;
T_H	- ambient temperature;
p_H	- ambient pressure;
w	- humidity;
MDS	- mechanize of compressor and exhaust nozzle;
PG	- thermodynamic function of state;
ω	- rotational speed;
φ	- initial phase shift;
f	- frequency.

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1. INTRODUCTION

There are many different technical engine failures we can face in turbine engine life and exploitation process. The rotor blade fatigue cracks propagation and as a consequence the blade breakaway, the excessive main engine fuel supply and bearing system getting out of order always as a rule cause a formidable danger for flight safety, engine life and reliability.

A detailed analysis each of engine fault occurred during last years in Polish Air force units showed that:

- there were some cases of fatigue blade breakaway in spite of an earlier, check using blade ultrasonic and eddy current inspection and technology,
- there were some cases of bearing system damages in spite of complying with engine maintenance technical requirements,
- the engine fuel system getting out of order was observed as a result of fuel aggregates dynamic characteristics failure of their components as well as exploitation adjustment activities faults caused by deficient adjustment technology or methodology.

About 70% of all faults observed during turbine engine maintenance appeared to be located with fuel systems faults or strictly maintenance faults – fig.1 (combustion chamber and exhaust system, fuel system, turbine section). Turbine engines maintenance monitoring process resulted with conclusion that some of them were caused by external reasons like poor condition of fuel and power supply airfield devices. Some of engine faults originated in pilots operating engine faults relying on not obeying engine warming up and conditioning procedures. Such deficiencies cause as a rule mechanical or thermal low cycle fatigue (LCF). In spite of dynamic measurement equipment development as well as providing aircrafts with flight parameters recorders the expert diagnostic system were not properly so far elaborated. There are many reasons of that. First there is a little knowledge about the engine construction data specially with Soviet Union origin. Second: there is a lot of discrepancies between the maintenance technical conditions and technical specifications.

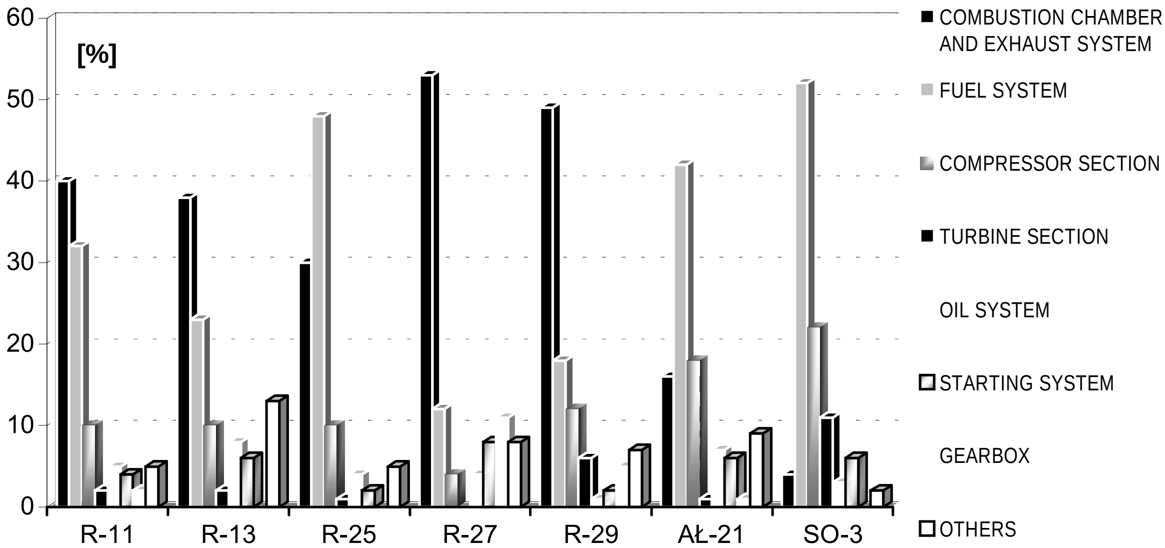

Fig.1. Maintenance problems in Polish Air Force (1991-1997)

Base on above there is of utmost interest the need of looking for new method to recognise stochastic loads during engine operation, their influence on structural engine reliability and running engine technical condition. These methods should comply with present aviation trends i.e. should increase sensitive identification of engine technical condition and reduce the number and minimise the weight of control devices as well as reduce the quantity of measured parameters.

The paper presents non-interference technique of turbo-machine blade vibration phase method, one of the most interesting such complex jet engine diagnostic method as well as the tool for dynamic phenomena investigation of a running engine. The method is based on synchronous blade vibration amplitude measurement and its numerical response analysis referred to the jet engine technical condition analysis [9, 11÷14]. The described method has been used in some units of Polish Air Force since 1993 as SNDŁ-1b/SPŁ-2b SO-3 jet engine diagnostic system. This engine powers polish TS-11 „ISKRA" training aircraft.

2. THEORETICAL BASIS

2.1 Analysis of a turbine engine maintenance

The most popular, non-destructive testing methods and diagnostic systems of turbojet engine e.g.: visual inspection, eddy current or ultrasound have been applied to structures in an effort to locate structural faults. There are based on MIN–MAX formulae identification (usually used in stationer machines) with idea:

„If all observed values of all observed parameters are in <MIN_i, MAX_i> range complying with technical maintenance condition requirement it means that engine is fit to fly".

Assumption to this formula is all observed parameters have no any links and interference between them and controlled object is in constant state of operation. In case of turbine engine with observe signals are depends so this assumption has to be changed to the following shape:

„If all observed values of all observed parameters are in <MIN_i, MAX_i> range complying with technical maintenance condition requirement it means the engine is with high probability fit to fly".

The deficiency of this method is not zero probability of hidden machine fault. These faults are cause aircraft crash.

2.2 Idea of the expert system SNDŁ-1b/SPŁ-2b

The consciousness of low plausibility of exiting diagnostic systems based on multi engine parameters MIN-MAX control formulae as well as repeated first stage compressor blades fatigue breakaway causes (damages) created the urgent need to elaborate more effective, sensitive system based on minimum quality of measured and analysed parameters.

The major concept of expert analysis of engine technical condition is projection of momentary working object characteristics point (i.e. blade, fuel system or bearing system) on phase plane, which equals an 3m – vector of technical state (m – the number of analysed signals).

$$\forall i \in (1,2,...,m)\ \ CV = \left(\mathrm{Par}, \frac{d\mathrm{Par}_i}{dt}, \frac{d^2\mathrm{Par}_i}{dt^2} \right) \in TS \Rightarrow \text{ENGINE OPERATIONAL} \tag{1}$$

It can happen that signal Par_i can comply with technical requirement of the engine but first or second derivative will not comply with new widen technical requirement installed during object dynamic characteristics identification process by inverse operation. This case usually projects (the most frequently) the hidden faults of the engine or disadjustment of the particular systems.

In simplest case, having only one analysed signal, for example, engine rotational speed, the working engine range is described in three-dimensional space.

The momentary point of engine performance is monitored not only on the transient value of rotational speed but basing on first and second derivative of rotational speed which project some trends of parameters changes (and assisting them excitations).

To solve this complicated problem a qualitative evaluation of applicability of non-contact blades vibration measuring method was done. A particular attention was drawn to the possibility of estimating the technical condition of a blade (crack initiation and propagation) on a running engine [8, 9]. The measured blades vibration spectrum is analysed using the phase plane. It is worth noticed that such solution enables measurement of rotational speed of the engine rotor by frequency method and the basic difference rely on using the vibrating and rotating blades as a phase marks (fig.2).

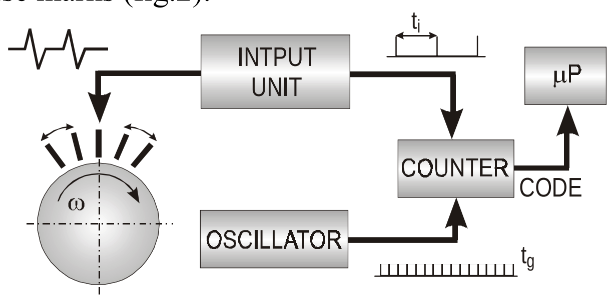

Fig. 2. Measurement method

This enables to consider measured signal as a composition of two different signals:

- one is aperiodic signal of rotational speed (configured by average time value of consecutive rotation of engine rotor),
- second describes oscillation of distance between two adjacent vibrating blades which is stroboscoped by rotational speed.

Basing on narrow band filtering of measured signal we are getting some signal components very useful to expert analysis of engine technical condition following assemblies and structure:

- I-st compressor stage blades,
- engine fuel system,
- bearing system.

2.3 **Vibration analysis of turbine engines**

In regard to vibration analysis turbo jet engines are non-linear, multi-frequency resonance systems (fig.3).

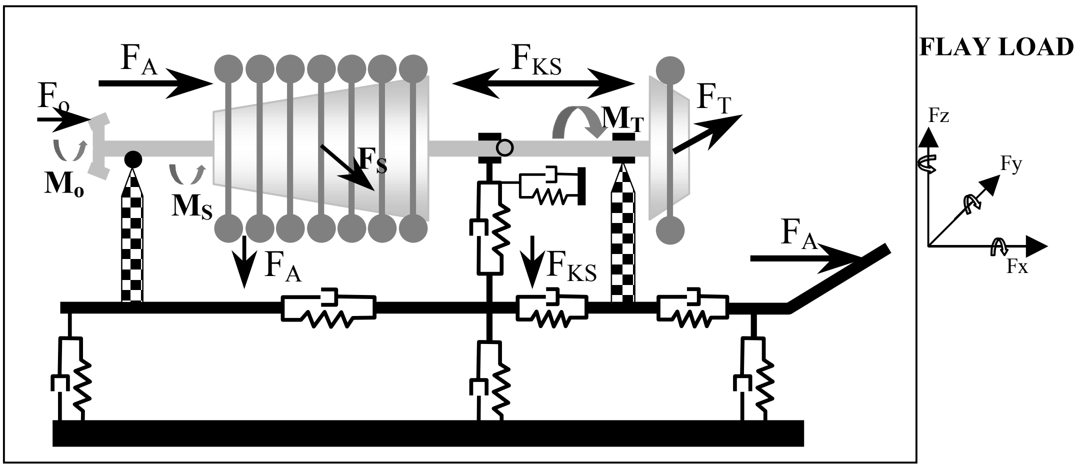

Fig.3. Vibration source of SO-3 engine.

All kind of vibrations of this system is a combination of forced and natural resonant vibration [7]. Forced vibration can due to:

- internally generated forces (e.g. combustion - F_{KS}, power receiver - F_0, M_0);
- unbalances (e.g. compressor - F_S and turbine - F_T);

– external loads (e.g. inlet turbulence - F_A, flight loads)
– ambient excitation (e.g. temperature - t_H, pressure - p_H and humidity – w).

Resonant vibration typically amplifies the vibration response far beyond the acceptable stress and strain level deflection caused by static loading.

Mode (or resonances) are inherent properties of a structure. Resonances are determined by the material properties (mass, stiffness, and damping), and boundary conditions of the structure. For each mode is defined a natural (modal or resonant) frequency, modal damping, and a mode shape. <u>If either the material properties or the boundary conditions of a structure change, its modes will change.</u> This fact is using in machine modal diagnostic.

At or near the natural frequency of a mode, the overall vibration shape (operating deflection shape) of a machine or structure will tend to be dominated by the mode shape of the resonance.

An operating deflection shape (ODS) is defined as any forced motion of two or more points on a structure. Specifying the motion of points defines a shape. Stated differently, a shape is the motion of one point relative to all others. Motion is a vector quantity, which means that it has both a location and a direction associated with it. Motion at a point in a direction is also called a Degree Of Freedom (DOF).

2.4 Blade vibration

Blade's DOF describe as:

$$y(t) = \sum_{i=1}^{k} A_i(t)\sin(\omega_i t + \varphi_i) \tag{2}$$

for example:

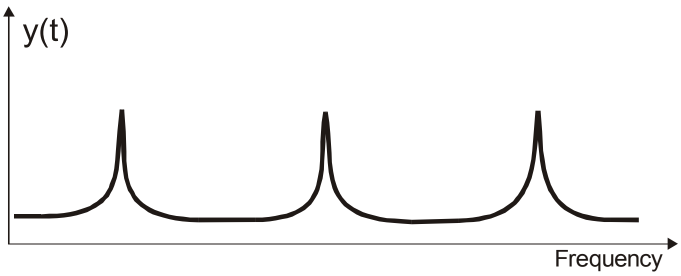

Fig.4. Spectrum of blade vibration.

For steel blade resonance factor Q_s is more then 400 (e.g. first mode frequency f_c=320Hz, bandwidth B=5Hz) so we observe clear and separate modes in frequency domain. It enables solving blade vibration problem as sum up narrow-band signals. Nyquist condition of minimal frequency sampling without aliasing is describe as:

$$f_{smin} > 2B \tag{3}$$

As we can see this sampling frequency is much smaller than minimal sampling frequency low-band signals ($f_s > 2f_{max}$). This fact enables sampling vibration signal e.g. compressor blade with frequency <u>much smaller then first-mode frequency (e.g. only one sampling per shaft revolution).</u> After sampling with frequency ω_s blade's DOF describe as:

$$y(k) = \sum_{i=1}^{k} A_i(k)\sin\left(2\Pi\frac{\omega_i}{\omega_s}k + \varphi_i\right) \tag{4}$$

Taking into account blade resonance characteristic in particular damping coefficient we can recognise and investigate different mode vibrations. Consequence of sampling is spectrum duplicate of every mode of vibration (ω_i/ω_s) with period 2Π (fig.5.) This fact enables observing higher mode of blade vibration in <0÷0,5 ω_s> band frequency (fig.6)

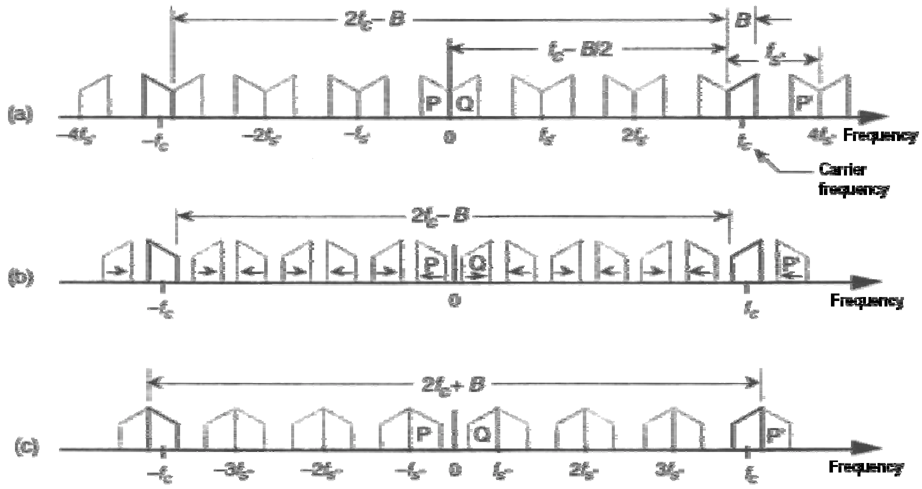

Fig.5. Narrow-band signal sampling and spectrum duplicate
(f_c – mode frequency, B – mode width band).

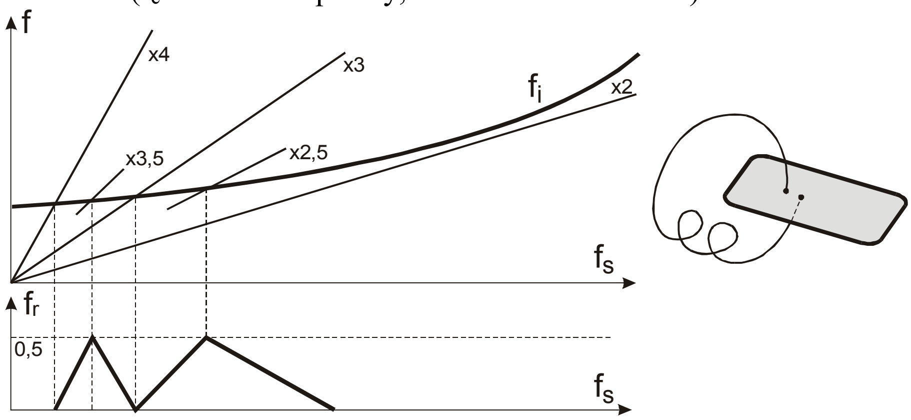

Fig.6. Influence of probe frequency blade vibration on the phase spectra
($f_r = f_i / f_s$, where $f_i = \sqrt{f_{sti}^2 + Bf_{rpm}^2}$).

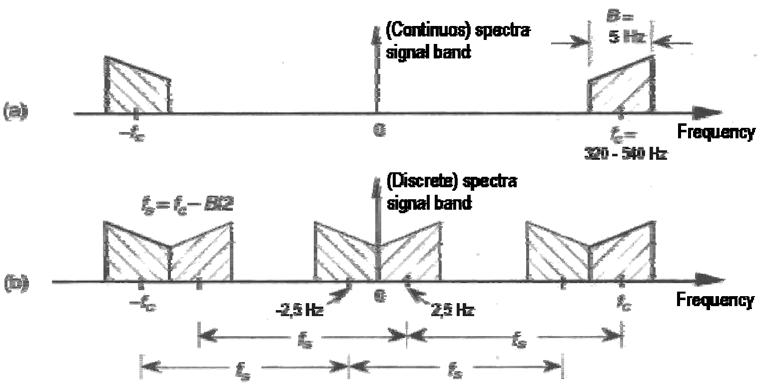

Fig.7. Sampling signal band's: a) original spectra signal (continuous),
b) multiplication spectra sampling's signal.

In discrete signal spectrum doesn't step out aliasing if this is condition [15]:

$$\frac{2f_c - B}{m} \geq f_s \geq \frac{2f_c + B}{m+1} \tag{5}$$

If sampling frequency is equal engine rotation frequency (variable time sampling) then $2\Pi\omega_i/\omega_s$ is order-tracking frequency as in Vold-Kalmman filter.

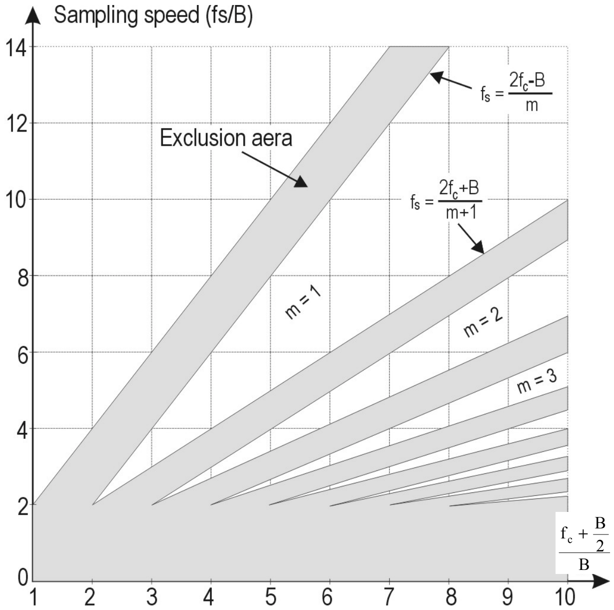

Fig.8. Accepted and exclusion area of (5) equation validation.

During engine operating conditions rotor blades are excited by different sources. Therefore the resonance blades synchronizing phenomena can be used as sensitive detector of:

– flow disturbances in engine channel;
– mass forces (for example created by dynamic rotor unbalance).

Blades vibrations are shown as phase portraits as crossing points of phase trajectory with phase plane. Characteristic feature of the method is information last about total number of the blade modal frequency periods between two subsequent crossing points of phase trajectory with phase plane with simultaneous preservation of basic blades modal parameters. This phenomena allows to detect blades fatigue cracks initiation and propagation during engine performance.

2.5 Diagnostics of engine fuel system

Dynamical system: jet engine turbine – fuel system (fig.9) is strongly non-linear system of inertial. Procedure compares real dynamic characteristic taken during non-stabilized range of engine operation with model dynamic characteristics.

Having in view structure and principles of the engine fuel system work, procedure allows:

– all fuel system aggregates and the whole fuel system adjustment and control;
– early revealing of engine fuel system and fuel aggregates failures.

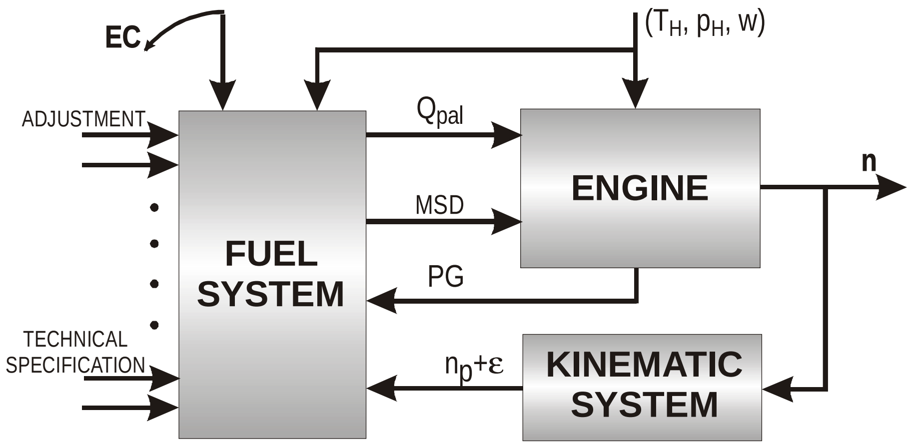

Fig.9. Mathematical model of jet engine turbine – fuel system.

The link between engine thermodynamics, fuel system and kinematics parameters describe equation (6). It should be noticed that engine dynamic characteristics identification is based on simply measured signal.

$$
\begin{aligned}
&\frac{d^2n}{dt^2} + a(n)\frac{dn}{dt} + b(n)n = f(m_{pal}, m_{pow}) = f\left(Q_{pal}, Q_{pow}\right) \approx f(Q_{pal}) \\
&\frac{dn}{dt} = \text{const}\,\frac{M_T - M_S - M_F}{I}
\end{aligned}
\tag{6}
$$

During identification process we were looking for transient function, which was describing the line between technical condition of an engine (characterised by multidimensional technical condition vector CV) and measured rotational speed (residue method). During engine technical state analysis we employ finite set of phase characteristics m points:

$$
\text{TECHNICAL CONDITION} = \left(CV, \frac{dCV}{dt}, \frac{d^2CV}{dt^2}\right) \approx \left(n, \frac{dn}{dt}, \frac{d^2n}{dt^2}\right)_m \tag{7}
$$

2.6 **Diagnostics of engine bearing system**

The idea of bearing technical condition estimation of SO-3 engine is based on correlation phenomena between rotor shift and measured blades vibration spectra changes.

Defined coefficient describing this correlation as:

$$
K = \frac{\Delta T}{\Delta y} \tag{8}
$$

where:

ΔT – apparent amplitude blade vibration change cause by rotor shift at about Δy;

Δx – demurrage shift blades vibration;

Δy – rotor shift (see Fig.10).

Basic principle of this method is that blades of rotor are independent generators. Such phenomenon like blades flutter vibration or rotor shift are projected into blades vibration spectra (so called „apparent” blades vibration).

”K” coefficient of SO-3 engine is:

- axial shift - K=1.17; ΔT=0,20% T (theoretical value – non appear for a ball bearing);
- across shift - K=0.10; ΔT=0,21% T;
- radial shift - K=0.46; ΔT=0,96% T.

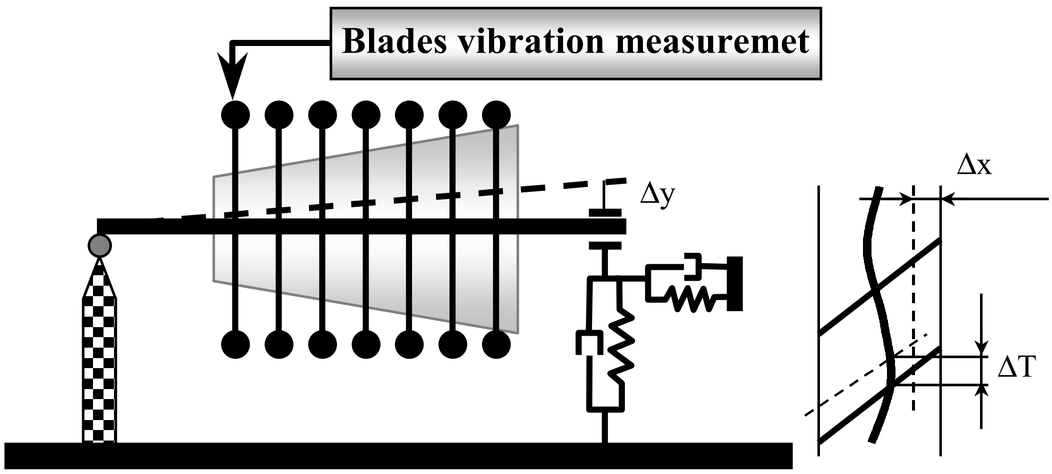

Fig.10. Correlation analysis of blades vibration.

3. DIAGNOSTIC SYSTEM SNDŁ-1b/SPŁ-2b

The system SNDŁ-1b/SPŁ-2b consist of:

- blade excessive vibration signalling device SNDŁ-1b;
- ground control device SPŁ-2b for periodic:
  - recording and computer analysis of first compressor stage blade vibration spectrum,
  - SNDŁ-1b technical condition control performance without its disassembly,
  - The engine rotational speed in I and II cabin indicator faults control without their disassembly,
- SPŁ-2b software.
  - Set of specialised procedures for SPŁ-2b numerical maintenance and numerical data analysis, particularly: blade vibration spectra; fuel and bearing system technical condition analysis.

3.1 Numeral analysis of blades vibration

The effect of blade upon its numerically analysed by considering the blades as appropriately shaped, beam fixed to the rotor disc (self – likeness condition) in centrifugal force field. An actual flow disturbances level running engine is evaluated by employing the linear dependence of disturbances with vibration amplitude (in elastic deformation range). Numerical procedures employ sectional statistical analysis to get average values of blade vibration amplitudes and standard deviation. The analysis results are compared with the engine type model see fig.11 and 12.

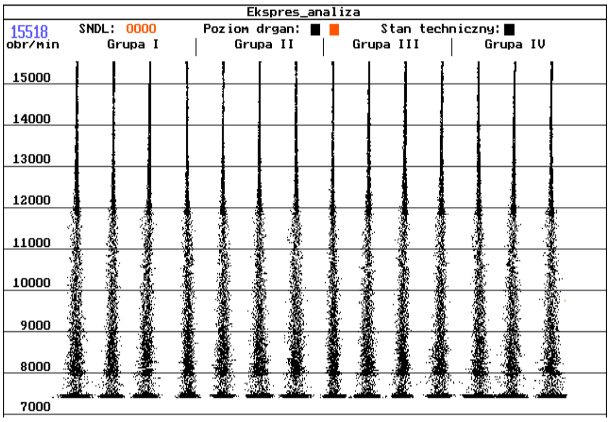

Fig.11. First stage amplitude–phase compressor blade vibration model spectra of SO-3 engine – standard spectrum.

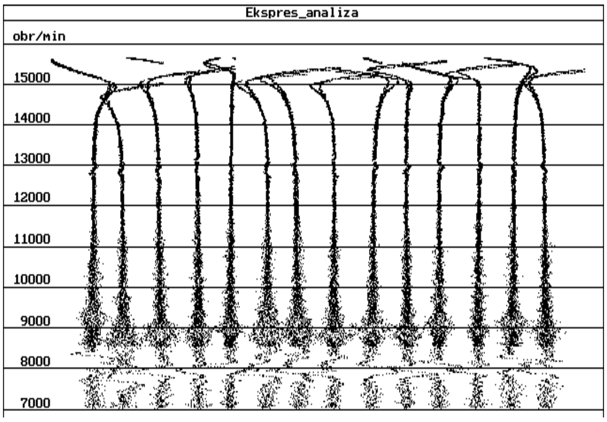

Fig.12. Influence of foreign object dwelling in compressor first stage stator blades on first stage amplitude – phase compressor blade vibration of SO-3 engine.

Effect of blade phase resonance characteristic project by low fluctuation of vibration amplitude (in dynamic pitch) was used in numerical procedure. Narrow band filtering of measured signal was implemented with use if aliasing effect in numerical procedure. The core blades technical condition analysis is based on comparing of the engine dynamic pitch projection to the model projection of an engine.

To get explicit identification of crack suspect or suspect blade an numerical procedure was used to estimate blade free vibration frequencies. Narrow band filtering of measured signal was implemented by low square (LS) method.

After blades free vibration frequencies estimation an identification process with factory or overhaul basic blades frequencies is following. During test bench investigations the lowering of blades vibration amplitude was observed particularly in blade resonance excitations range. This phenomenon is lowering of Q -–factor the mechanical narrow band filter which the blade creatures. During cracks propagation process the crossection of the blade is decreasing which causes decrease of free vibration frequency f_{Bst} and blade dynamic frequency factor B. During engine run these changes project in decreasing of blades forced vibration frequencies f_B – fig.13 (x-axis – dynamic pitch phase, y-axis rotational speed). For first compressor stage blades of SO-3 engine the dynamic changes were the most distinct near maximum of rotational speed of the engine in which the second synchronous excitation of resonance vibration occurs.

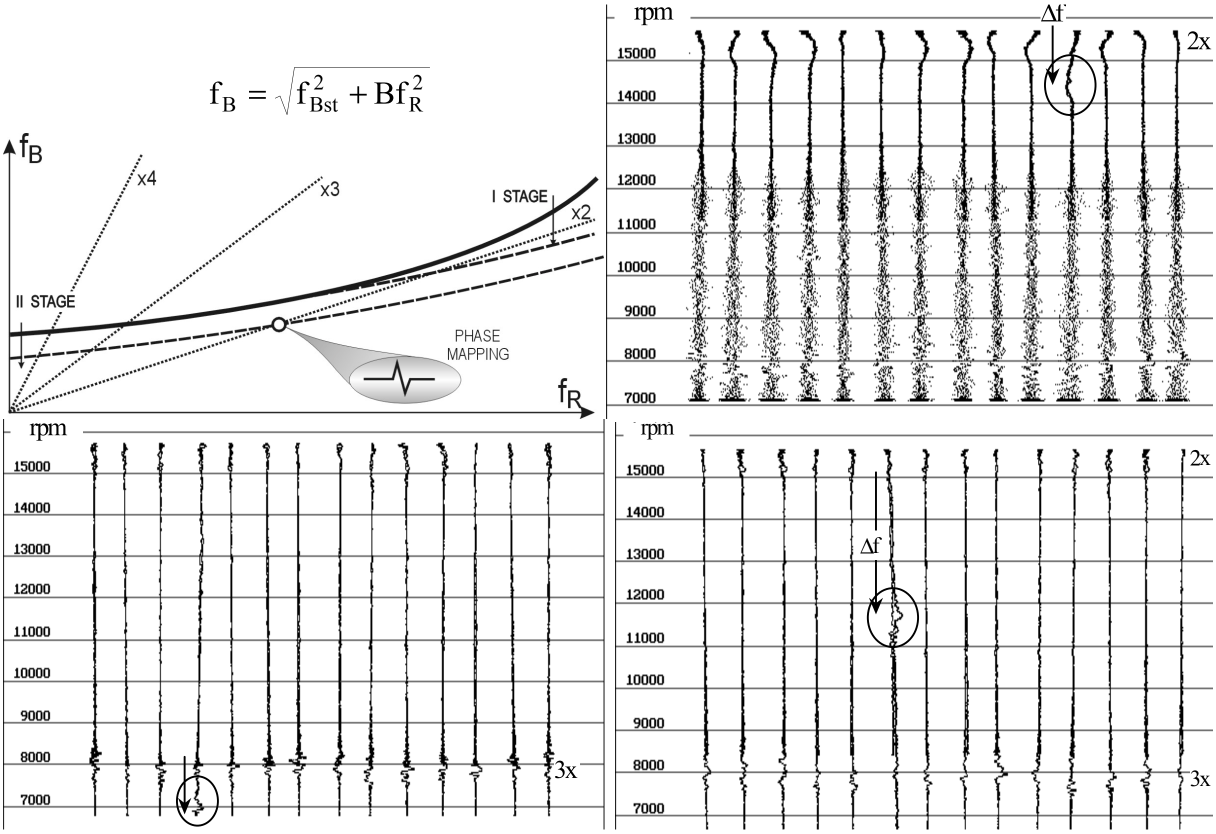

Fig.13. Projection of blade crack propagation process in blade phase vibration spectra:

a) blade frequency on the Campbell diagram;
b) first stage crack of blade – change only B;
c) second stage crack of blade – change f_{Bst} and B;
d) finish stage crack of blade – five minute after break.

3.2 Technical estimation of fuel system condition

Big frequency of sampling of the rotational speed of the engine (once for each rotor turn) allows in very simple way to possess information concerning transient value of rotational speed. Having in view relations between engine fuel consumption and rotational speed (related to real atmospheric weather conditions) an numerical procedure for fuel system technical conditioned estimation was elaborated. Procedure compares real dynamic characteristic taken during non-stabilized range of engine operation (acceleration and deceleration) with model dynamic characteristics – fig.14 (Y-axis – engine rotational increase of rotational speed). Having in view structure and principles of the engine fuel system work, procedure allows expert identification:

- all fuel system aggregates and the whole fuel system adjustment and control, fig.15,16;
- early revealing of engine fuel system and fuel aggregates failures, fig.17.

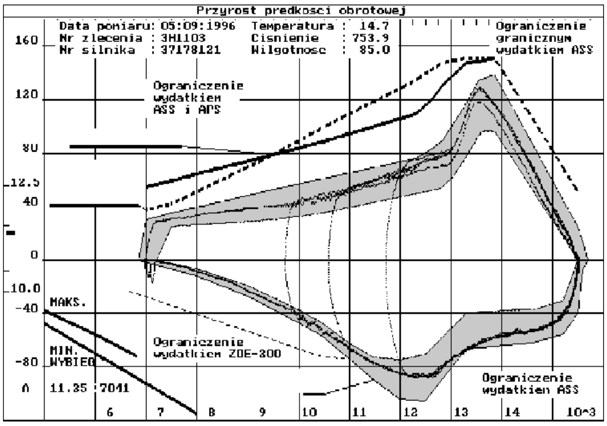

Fig.14. Model characteristic of engine fuel system during transient condition of SO-3.

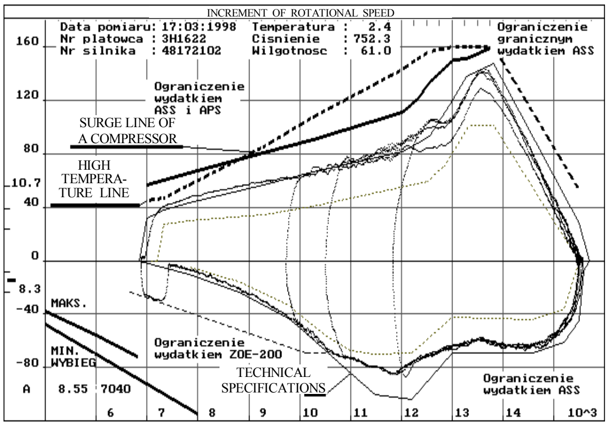

Fig.15. The presentation of improper engine acceleration time adjustment – reduction of the margins of compressor stability.

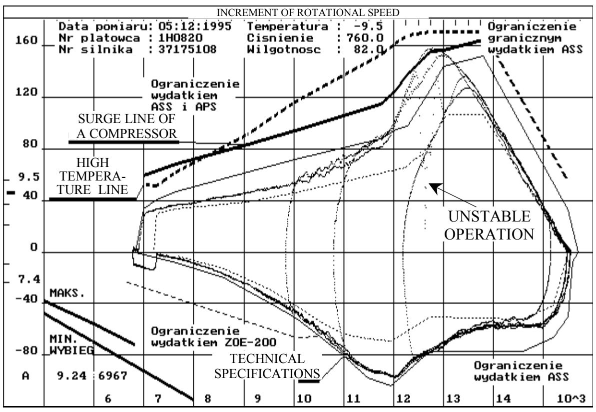

Fig.16. Control error of fuel system and surge threat.

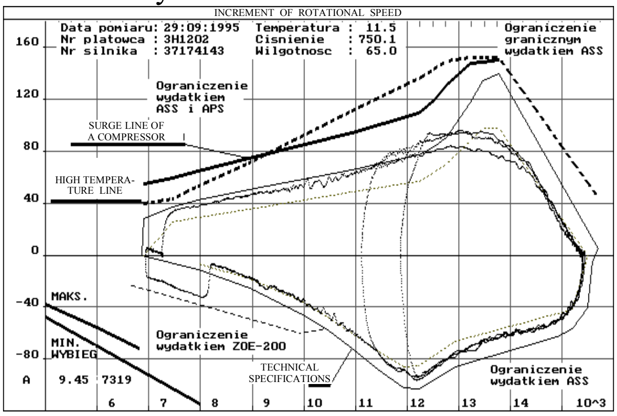

Fig.17. Typical presentation of engine acceleration aggregate fault.

3.3 Numeral analysis of dynamic load of engine rotor bearings

Blades amplitude and vibration frequency data having got during numerical analysis of all rotor blades were used for identification of some disadvantageous dynamic phenomenons from time to time occurred during engine exploitation process.

For this numerical procedure gives relation between:

- bearing system technical state and blade load effect,
- blade characteristic vibration and bearings dynamic load level.

Procedure has sufficient sensitivity for revealing such phenomenons like synchronous and asynchronous rotor blades resonance vibration – fig.18.

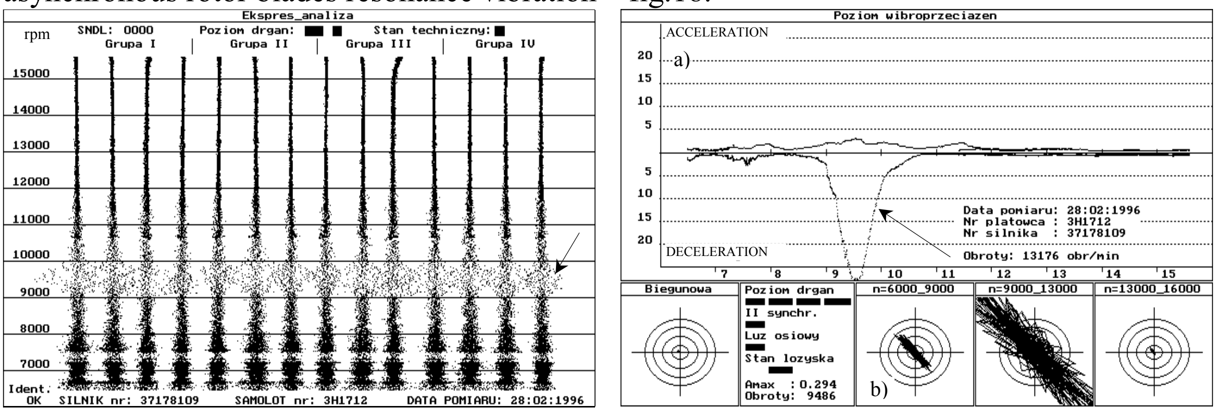

Fig.18. Display of blades vibration and the rotor dynamic conventional unbalance level.

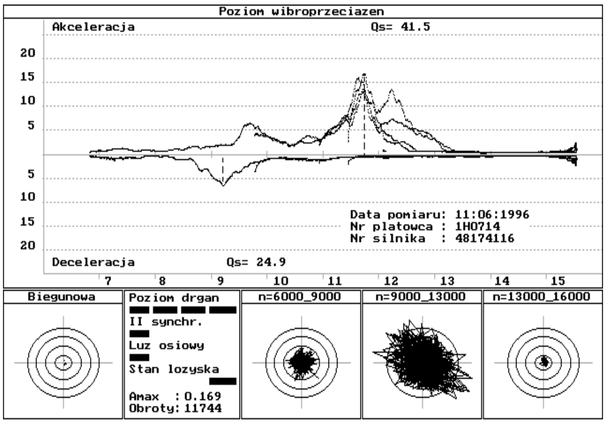

Fig.19. Influence of bearing fault of median support on the engine unbalance.

The results of the analysis are projected conventionally in the shape of normalized radius and dynamic rotor unbalance phase (on fig.18a: y-axis - engine rotational speed, x-axis – conventional, normalized dynamic unbalance radius; on fig.18b: Nyquist diagram).

Different, very interesting example of the engine unbalance is shown on the fig.19-21.

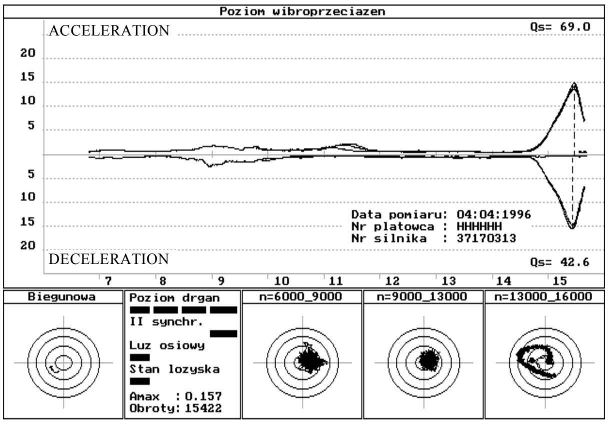

Fig.20. Influence of foreign object dwelling in compressor first stage stator blades on the engine unbalance.

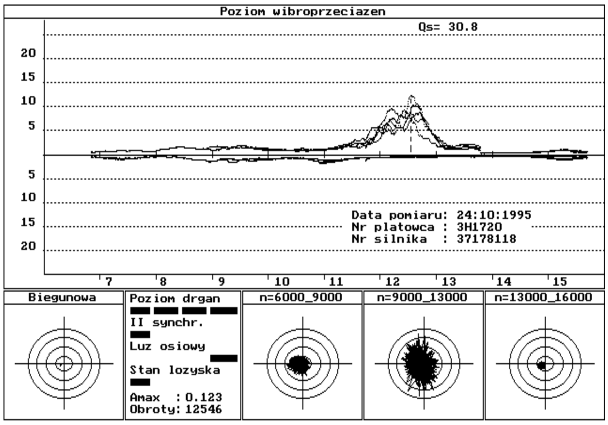

Fig.21. Influence of undo a nut drive shaft on the engine unbalance.

3.4 Experience having got during SNDŁ-1b/SPŁ-2b exploitation process

The functional abilities of described measuring method were verified during some experimental tests on the Institute test bench and during system SNDŁ-1b/SPŁ-2b maintenance in aviation units. In period 1993÷2000 system SNDŁ-1b/SPŁ-2b has embraced above 200 of SO-3 engines.

Having in view running aircraft maintenance (above 2500 registrations), very interesting results were collected specially concerning early detection of cracks in compressor blades of turbojet engine and to crack initiation propagation and process during engine normal condition of operation. In effect we gained:

- technical blades life prolongation;
- engine fuel system technical state improvement;
- simple bearing systems technical state diagnostic system creation.

Besides, capability to significantly increase reliability and safety of aircraft operation was succeeded with rather low funds to be spent on preventive activities.

4. CONCLUSIONS

1. Described method is of great importance for flight safety and may be recommended as a completion of the existing systems of diagnostic of turbine engines.

2. By using the discrete, non-contact method for measuring blades vibration and numerical analysis of blade vibration spectrum it is possible to conduct complex monitoring of technical state of major engine components like:
   - compressor blades
   - fuel system
   - bearing system.

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