ICAS-96-6.3.3

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TURBOJET ENGINE DIAGNOSTIC SYSTEM BASED on COMPRESSOR BLADE VIBRATION and VIBROACOUSTIC ANALYSIS

Introduction

Many technical engine failures occur in turbine engine life and exploitation process i.e:

- the rotor blade fatigue cracks propagation and as a consequence the blade breakaway
- the fatigue damages of the rotor bearings
- the excessive disadjustment of fuel system negatively influence on engine life and reliability.

Existing flight safety impendence creates the need to look for new engine diagnostic systems. This paper presents some diagnostic methods based on compressor blade vibration spectrum measurement and numerical response analysis of vibroacustic process referred to technical condition of the blades,[4] synchronous rotor blades vibration phenomena identification as well as engine fuel system condition evaluation. Described methods were implemented in polish SO-3 jet engine diagnostic system. This engine powers polish training aircraft TS-11 "Iskra".

Measuring and Analysis Methods Decsription

For engine technical condition evaluation two methods of measuring and analysis were implemented:

- the blade natural frequency measurement (engine shut down)
- the forced blade vibration measurement and numerical analysis of a running engine

Measurement of the Blade Free Vibration Frequency

There was an assumption taken in advance, that for constant condition of its fixing in the rotor lock the blade free vibration changes can be only as a result of blade cross section changes. This condition refers to initiation and propagation of a blade cracks. For measuring of blades free frequencies typical Bruel - Kjar set was used consisted of measuring microphone, preamplifier and narrow band vibration spectrum analyser. The blades were under transient mechanical excitation.
The blade free vibration measurement was done to get:

- data described expected blades free vibration changes assisting typical blades crack propagation process (laboratory investigations precluding the blade forced vibration measurement and analysis method).

- precise identification of craked blades (based on earlier blade free vibration measurement or similar data contained in manufacturing factory or overhaul documentation.

Simulation of typical cases of blades fatigue cracks enables to estimate expected blades free vibration changes. Quick and simltaneously quite enough exact simulation method is blade cutting using electro-erosion machine giving "cracks" about 0.1mm width.
Some simulated investigations results referred to compresor first stage steel blade of the SO-3 engine were introduced on fig.1 (where y-axis-relative cracked cross section of the blade, x-axis-relative free vibration change). The shape of suceeding cuts was refering to phases of typical blades cracking. To estimate simulation failures coming mainly because of negligence of energy scattering in the crack as well as taking into consideration two - dimension blade craking model, the results were verified during some blades cracks develop investigations performed on test bench. Fig.2 reflects blade real vibration changes due to crack going from blade edge (crack similar to "B" simulation see. Fig.1). Blade cracks initiation (flat crack) observed as small blade free vibration drop (measurement after 1,5 hours of engine run). During following 12 h of engine run an stability period of cracked blade free vibration was obserwed as a result of three - dimension cracking. At the final phase of blade cracking the intensity of cracking was increased. Since crack initiation to blade breakaway 15% free vibration change range was observed.
The advantage of described method is speed in getting the results of simulation (compare with computer simulation) as well as using classic methods for vibroacoustic signal analysis (FFT analysis). The functional limitation of the method is ability to use it for blades stiffly fixed in rotors locks (blades with "fir-tree" or "trapezium" root fixing).
Above exploitation limitations are because strong influence of stiffness of fixing blade lock on its free vibration frequency and logarithmic decrement. The possibility to use it to real object is up to easy access to the blades. In reality it limits it to the first compresor or fan stage or to the last stage of turbine.

Measuring of Blades Forced Vibration

The measurement of blades forced vibrations during engine run is performed for definition of:

- blades energetic (projected by flow disturbances) and technical conditions,
- mutual blades ring vibration correlation (revealing synchronous vibration phenomena).

Additionally exact information about real rotational speed of the rotor is got during measurement which allows for objective engine fuel system condition evaluation. The blades ring forced vibration and transient value of rotational speed was measured using the descrete, precise measurement of time intervals between following pulses generated in a sensor by rotating blades [11]

The method is based on the relation between blades free vibration f_s and forced vibration f_d:

$$f_d = \sqrt{f_s^2} + B\omega^2 \qquad (1)$$

where:
B - blade dynamic frequency factor
ω - frequency of rotor rotation

and measured and analysed physical values described with formulae:

a) rotor rotation time:

$$T_{rot} \approx kxt_i \qquad (2)$$

where:
k - rotor stage blades number
t_i - time intervals between following pulses

b) engine rotational speed:

$$n_{rot} = \frac{60}{T_{rot}} \qquad (3)$$

c) relative parameter - the Amplitude (the sensor is located in the nlane of the mass centre of blade tip section rotational speed as reference quantity:

$$Amplitude(t) = \frac{\Delta L_j + \Delta c^j_{j-1}(t)}{L_t} \quad (4)$$

t_{sr} - average time between following pulses
$\Delta c^j_j(t)$ - i, j blade deflections substraction
ΔL_j – dynamic pitch error referred to theoretical value
L_t - theoretical value

Characteristic feature of the described method of measurement, are;
- measuring sensor is placed in stator of the engine
- there is a recording of the all blade vibration spectra of monitored compressor stage (nearly simultaneously)
- consciously advantage of using aliasing phenomena in amplitude blade vibrartion spectra analysis is taken - i.e. theorem Nyquista-Shanona-Kotielnikowa (frequency of measured pulses is two to ten time lower than first flexural blade vibration mode)
- measured value (time) proofs high accuracy of measurement.

Numerical Analysis of Blades Vibration

The effect of blade upon its generalized displacement is numerically analysed by considering the blades as as appropriately shaped, beam fixed to the rotor disc (self - likeness condition) in centrifugal force field. An actual flow disturbances level of a running engine is evaluated by employing the linear dependance of disturbences with vibration amplitude (in elastic deformation range).
Numerical procedures employ sectional statystical analysis to get average values of blade vibration amplitudes and standard deviation. The analysis results are compared with the engine type model - see fig. 3 and 4. Fig.3 introduces projection of first compressor stage amplitude vibration spectrum model of SO-3 engine, fig.4 - amplitude vibration spectrum during foreign object dwelling on the first stage compressor stator blades. (Y-axis-amplitude of vibration x-axis-rotational speed of the engine).
Effect of blade phase resonance characterictic projected by low fluctuation of vibration amplitude (in dynamic pitch) was used in numerical procedure. Narrow band filtering of measured signal was implemented with use of aliasing effect in numerical procedure. The core of blades technical condition analysis is based on comparying 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 with equation (1) approximation by low square (LS) method. After blades free vibration frequencies estimation an identification process with factory or overhall basic frequencies is following.
During test bench investigations the lowering of blade vibration amplitude was observed particularly in blade resonance excitations range. This phenomena 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_s" and blade dynamic frequency factor "B" (1). During engine run these changes project in decreasing of blades forced vibration frequencies "f_d" - fig.5 (y-axis - dynamic pitch phase, x - axis - rotational speed). For first compressor stage blades of SO-3 engine the dynamic frequency changes were the most distinct near maximum of rotational speed of the engine in which the second synchronous excitation of resonans vibration occurs - fig.6 (y - axis - engine rotational speed, x - axis - synchronous component of vibration amplitude phase change).

Numerical Analysis of Dynamic Load of Engine Rotor Bearings

Blades amplitude and vibration frequences data having got during numerical analysis of all rotor blades were used for identification of some disadventageous dynamic phenomenas time to time occured 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 phenomenas like synchronous and asynchronous rotor blades resonance vibration - fig.6. The results of the analysis are projected conventionally in the shape of normalized radius and dynamic rotor unbalance phase (y - axis - engine rotational speed, x - axis - conventional, normalized dynamic unbalance radius).

Technical Estimation of Fuel System Condition

Big frequency of sampling of the rotational speed of the engine (once for each turn of the rotor) allows in very simple way to possess information concerning transient value of rotational speed. Having in view relations between engine fuel consumption and engine rotational speed (related to real atmospheric weather conditions) an numerical procedure for fuel system technical conditiond estimation was elaborated. Procedure compares real dynamic characteristic taken during non stabilized range of engine operation (acclereration and deceleration) with model dynamic characteristics - fig.7 (Y - axis - engine rotational increase of rotational speed).
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 - fig.8
- early revealing of engine fuel system and fuel agregates failures - fig.9.

Structure of Diagnostic System

The system consists of:

- blade excessive vibration signalling device SNDŁ-1b.

The board device for crew or ground staff warning about dangerous level of blade vibration above 12000 rpm,

- blade crack signalling device SPŁ-2b.

Ground control device for periodic:

a) recording and compiuter analysis of first compressor stage blade vibration spectrum,
b) SNDŁ-1b technical condition control performance without its disassembly,
c) 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 specrtra; fuel and bearing system technical condition analysis,

- blade vibration recorder RDŁ-1. The board device destinated for blade vibration recording during flight.

Features of the System of the Exploitation

Functional features of diagnostic system were estimated during active, direct experimental investigation on test bench [7] and indirectly during exploitation of 120 polish aircrafts TS-11 "Iskra" in some Polish Air Force units [9].
Having in view running aircraft maintenance, very interesting results were collected specially concerning early detection of cracks in turbine aero-engine compressor blades and to monitor crack propagation process during engine normal condition of operation. In the course of the destroying test bench investigations the following characteristics have been determined:

a) model characteristics of blade vibration spectrum allowing self identification of technical condition of the blades.

It was proofed that sensitivity of the method is much higher compare with traditional non-destructive diagnostic method (like visual inspection, eddy-current, ultrasonic and colour defectoscopy);

b) the link between flow disturbances level and crack initiation and propagation time

- influance of flow disturbance level on blade vibration level, and additional load on bearings was also manifested,

- objective evaluation of the engine fuel supply system technical condition and adjustment influence on blade vibration level.

The implementation of described diagnostics system decreased significally the cost of engine SO-3 maintenance (derease of labour work durning maintenance and service).
Simulataneously the flight safety and reliability coefficients were increased.
The discrete, non-contact method of measuring blades vibration is used in Poland during factory, certification tests of new designed engines.

Conclusions

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

2. By using the discrete, non-contact method od measuring blades vibration and numerical analysis of blade vibration spectrum it is possible to conduct complex, monitored and analysed parameters.

References

1. Kudelski R., Szczepanik R., "Urządzenie od bezdotykowego wyznaczania amplirudy drgań w łopatkach sprężarki silników lotniczych", V KNT "Metody pomiarowe w technice lotniczej", ITWL Warszawa 1987.

2. Szczepanik R., "Analiza i badanie przyczyn urywania sie łopatek I stopnia sprężarki silników SO-3/SO-3W", Sprawozdanie ITWL Warszawa 1987 (nie publikowana).

3. Szczepanik R. "Badanie stanu drgań i naprężeń w łopatkach I stopnia sprężarki silników typu SO-3 w eksploatacji", Sprawozdanie ITWL Warszawa 1988 (nie publikowana).

4. Szczepanik R, Witoś M., "Quality Assessment of Diagnostic od Non-Interference Discrete-Phase Method of Measuring Compressor Blades Vibration", XIX Konferencja ICAS Anheim 1994.

5. Szczepanik R, Witoś M., "Wpływ procesu eksploatacji turbinowego silnika odrzutowego na żywotność łopatek wirnikowych", VII KNT "Przepływowe Maszyny Wirnikowe", Rzeszów 1993.

6. Szczepanik R., Witoś M., "Computer Aided Engine Condition System based on non-interference discrete phase blade vibration measuring method, SAE/SAS Symposium/Workshop Stockholm 1995.

7. Szczepankowski A., Witoś M., "Obiektywna kontrola stanu technicznego układu paliwowego turbinowego silnika odrzutowego typu SO-3 (SO-3, SO-3W, SO-3W22), Konferencja DIAG 95 Szczyrk 1995.

8. Szczepankowski A., Witoś M., "Wpływ stanu technicznego układu paliwowego turbinowego silnika odrzutowego na drgania łopatek wirnikowych sprężarki", Konferencja DIAG 95 Szczyrk 1995.

9.Witoś M., "Bezdotykowa metoda kontroli stanu technicznego łopatek wirnikowych sprężarki osiowej", Konferencja DIAG 95 Szczyrk 1995.

10. Witoś M., "Diagnozowanie stanu technicznego łopatek sprężarki turbinowego silnika spalinowego metodą bezdotykowego pomiaru drgań", Rozprawa doktorska, ITWL Warszawa 1994.

11. Zabłockij I.E., Korostielew J.A., Szipow R.A., "Bieskontaktnyje izmierienia kolebanij łopatok turbomaszin", Maszinostrojenije Moskwa 1997.

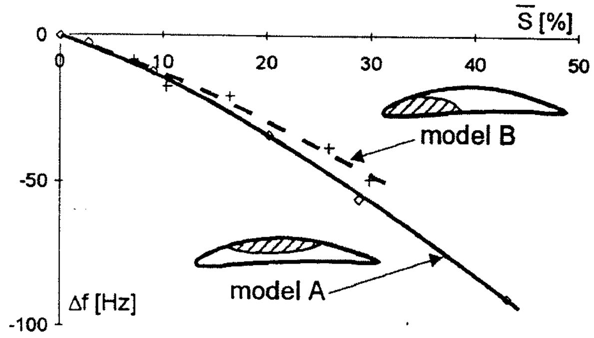


Fig.1. The change of basic flexural free vibration frequency of the blades (after making cut A or B) % = Su/So
where: Su, So - cross section of blade respectively: damaged and undamaged.

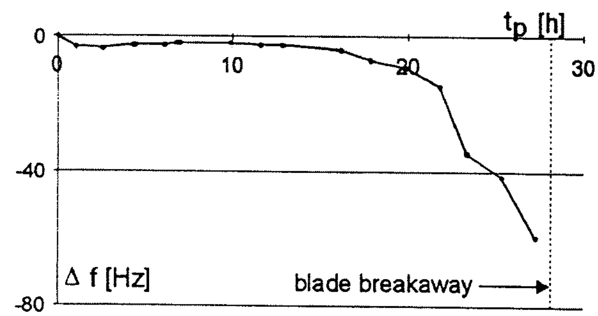


Fig.2. The change of basic flexural free vibration frequency of cracked blade during engine run on the test bench. The time of engine run up to blade breakaway was drawn by interrupted line.

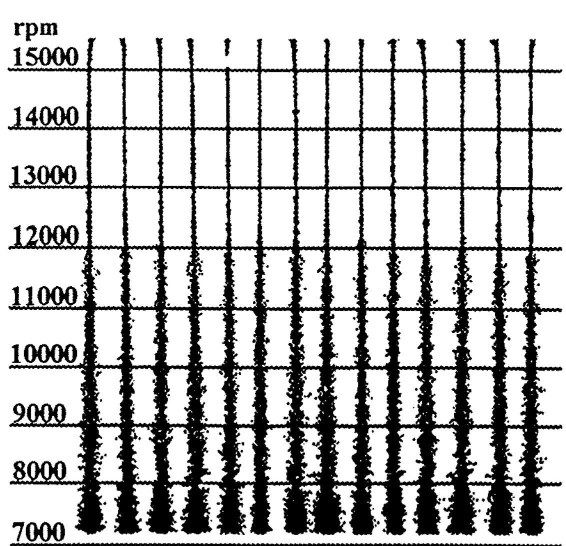


Fig.3. First stage amplitude - phase compressor blade vibration model spectra of SO-3 engine.

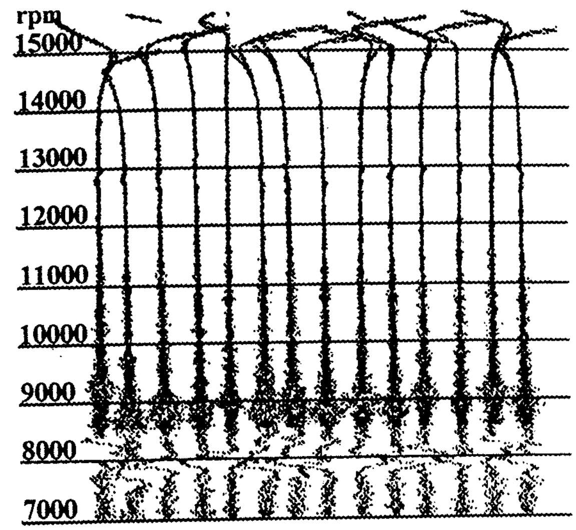


Fig.4. Influence of foreign object dwelling in compressor first stage stator blades on first stage amplitude - phase compressor blade vibration of SO-3 engine.

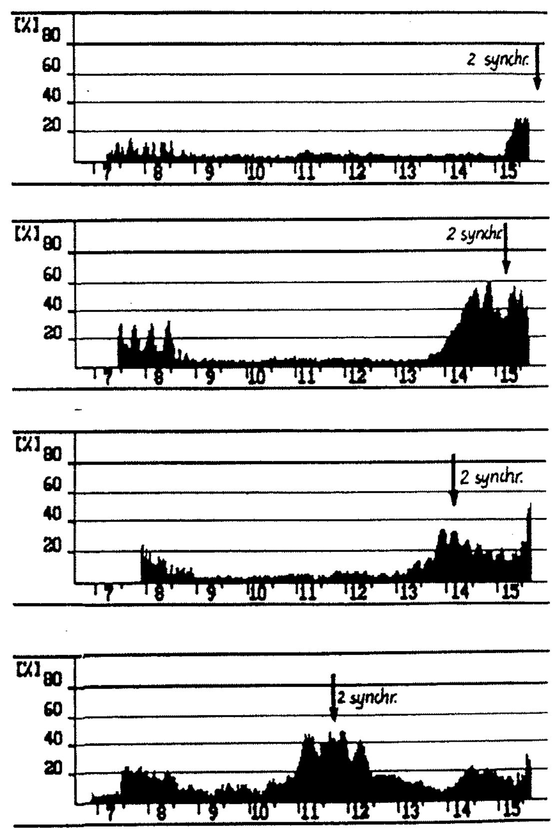
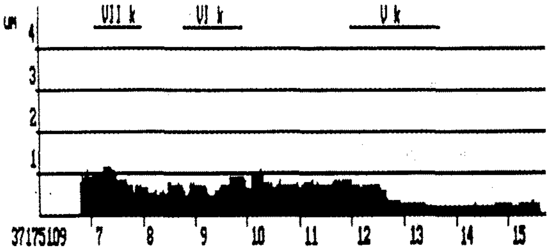
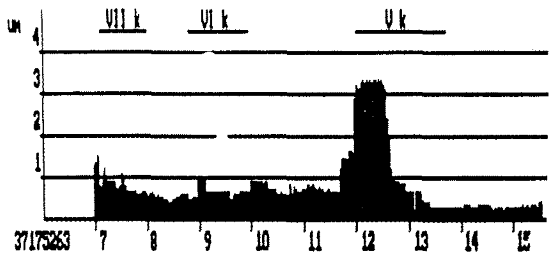


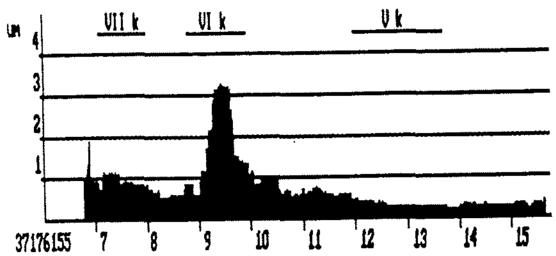
Fig.5. Projection of blade crack propagation process in blade phase vibration spectra.



a) normal level of blades vibration



b) excessive blades vibration level during engine acceleration (close to compressor surge)



c) excessive blades vibration level during engine decele:ation

Fig.6. Influence of blade vibration on the rotor dynamic conventional disbalance level:

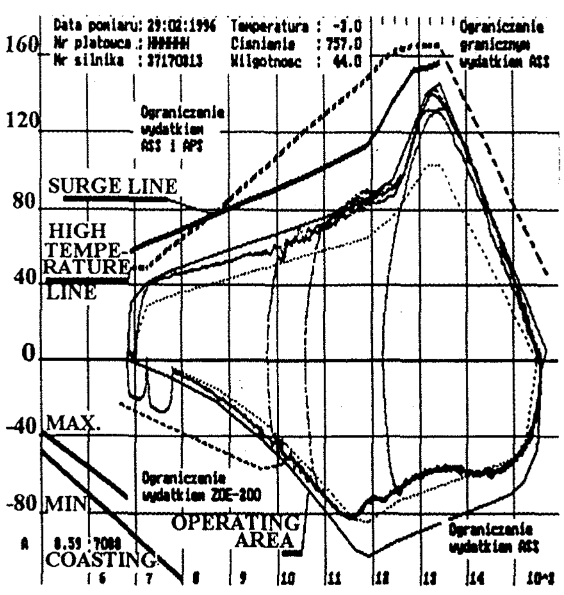


Fig.7. Model characteristic of engine fuel system during transient condition of SO-3 engine operation.

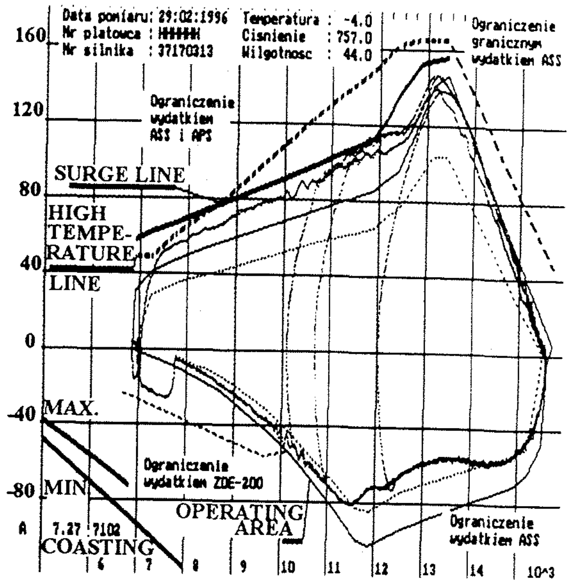


Fig.8. The presentation of improper engine acceleration time adjustment - reduction of the margins of compressor stability (surge threat)

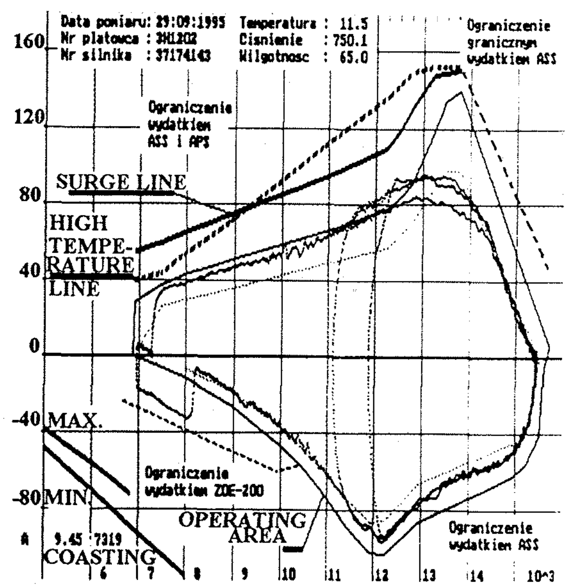


Fig.9. Typical presentation of engine acceleration aggregate fault.