EARLY IN-FLIGHT DETECTION OF FATIGUE CRACKS IN AERO-ENGINE COMPRESSOR AND TURBINE BLADES WITH VIBROACOUSTIC AND DISCRETE-PHASE METHODS

J. Lewitowicz*, R.Szczepanik, H. Dąbrowski, R. Kudelski
Polish Society of Mechanical Engineers and Technicians
00-050 Warsaw, Świętokrzyska 14A, Poland

Abstract

A phenomenon of dynamic change of an aero-engine compressor blade vibration spectra in course of fatigue crack propagation in blade roots /Polish turbojet engine being an example/ is discussed. On the grounds of this phenomenon a diagnostic electronic microprocessor device to measure vibrations of turbine engine rotor blades with the noninterferent discrete-phase method /DPM/ /used for early detecting compressor and turbine blade fatigue cracks/ is described. The device can estimate and signal the fatigue crack sizes to the crew and advise the crew how to operate the engine to slow down the crack propagation process to get safely to the homebase. Examples of vibroacoustic method applications to the ground evaluation of technical condition of some aero-engine compressor and turbine blades and bearings are discussed.

1. Introduction

Noninterferent technique of turbo-machine blade vibration measuring with the discrete-phase method has been known and described in technical literature for over ten years /ref. 1/. It consists in registration of vibrating and rotating blades' deflections from the neutral position, once per a full rotor revolution, with two sensors located at blade roots and tops, respectively. In case blade vibrations are asynchronous with rotation speed, 1000 points of blade deflections are recorded by the sensors in a few seconds, thus reflecting the amplitude of vibrations. If blade vibrations are synchronous with the rotation speed, transition through resonant scope of rotation speed is practised or a few couples of sensors, i.e. 3-4, are installed around the perimeter of analysed compressor stage /ref. 2/. So far, analyses of vibrations of blades in every compressor stage have required us to install two sensors i.e. a measuring sensor located in a hole above the rotating blade and a reference one installed inside the engine, cooperating with the special profile supplied with the number of "teeth" equivalent to the number of blades in the compressor stage. Thus, preparations for testing works included the engine dismantling and sensor adapting and were inconvenient and expensive. The necessity to install a separate reference sensor for every compressor stage inside the engine has been eliminated with the application of a computer-aided discrete-phase method of blade vibration measuring. In this method the blade vibrations of the consecutive stages /the second, the third one etc./ can be related to reference pulses coming from the reference sensor of the first compressor stage.

2. Principle of blade vibration measuring

Investigations of blade vibration dynamics are conducted, as a rule, with strain gauging technique which demands previous engine preparation i.e. channel

*Professor, Aerospace Eng., Member AIAA

drilling for installation of gauges. Such method is useless for diagnostics of compressor blade vibration state. In this case the noninterference discrete-phase method /DPM/ is very useful /ref. 1,2,3/. Considerable development of devices functioning according to DPM principle is noted down in professional literature. These devices are suitable for investigations of blade vibrations, both synchronous and asynchronous ones, with engine rotational speed /e.g. compressor blade vibrations caused by rotating stalls/. Their essential advantage is that vibrations of all blades of a stage may be examined simultaneously. Typical vibration picture of compressor first stage blades taken by DPM devices during engine acceleration from idling to $n=n_{max}$ is presented in Fig. 1. At the beginning of the engine acceleration procedure vibrations asynchronous with rotational speed are visible /stall blade vibrations/.

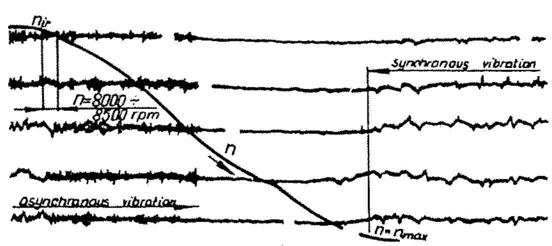

Fig. 1. Typical representation of the 1st stage compressor blade vibrations during engine acceleration.

Within the speed range from 8000 to 8500 rpm the blade resonance caused by the third harmonic of rotational speed is visible. At the end of the acceleration phase synchronous blade vibrations /of the first flexural mode/ caused by the second harmonic of rotational speed can be seen; at speeds ranging from 11000 to 14500 rpm blade vibrations are minimal. It has been examined that a picture of gentle blade vibrations /and stresses/, observed for the whole speed range, is distributed by ingested foreign objects found on the stator blades /e.g. bird remains/. In this case a portion of blades that previously had only weak resonance are now subjected to heavy fluctuations of forces /synchronous with rotational speed/. The blade resonance vibrations are strongly amplified /see Fig. 2/. Stresses inside blades are increased along with the amplitude of vibrations which during a long flight may lead to initiation and propagation of fatigue cracks in the blade roots.

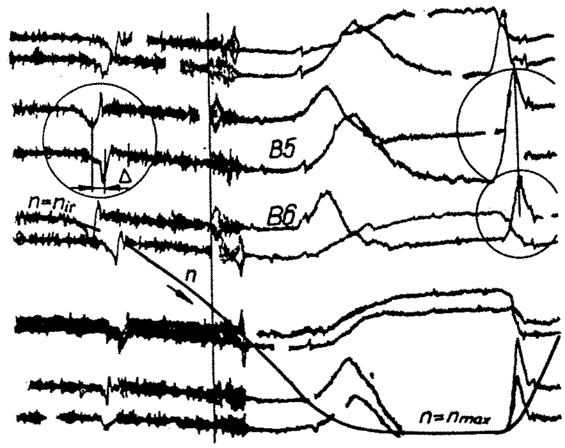

Fig. 2. Representation of the first stage blade vibration during engine acceleration and deceleration /with steady-state stator distortions/.

3. Principle of DPM measuring procedure

The arrangement of sensors for blade vibration analysis practised so far and the modified arrangement /typical of the computer-aided measuring procedure/ are presented in figs 3 and 4, respectively.

Fig. 5 illustrates the measurement principle. A series of reference pulses reflecting the rotation of roots of the first stage compressor blades is seen at the top. Below, the series of measured pulses reflecting the rotation of the 1st, 2nd and 3rd stage blades are seen.

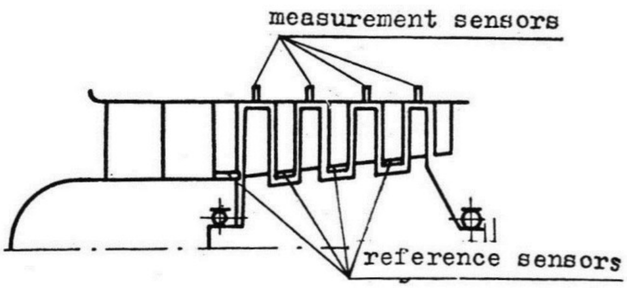

Fig. 3. Arrangement of sensors typical of the traditional DPM measurement.

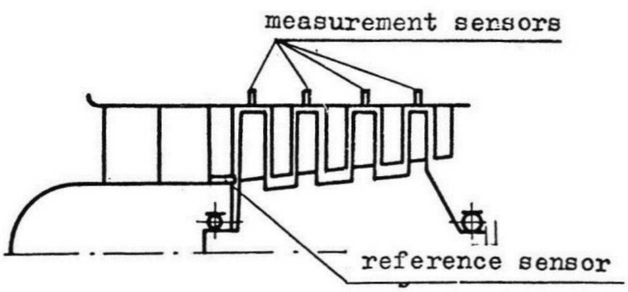

Fig. 4. Arrangement of sensors typical of the computer-aided DPM measurement

As the number of blades in the 2nd, 3rd and further stages is larger than of ones in the 1st stage, the pulses representing the momentary positions of the 2nd /3rd/ stage blades are frequently related to the same reference pulse. The series of pulses are processed, configured to digital form and then analysed by the computer according to a developed algorithm. Fig. 6 shows the vibrations of a selected blade on a display screen.

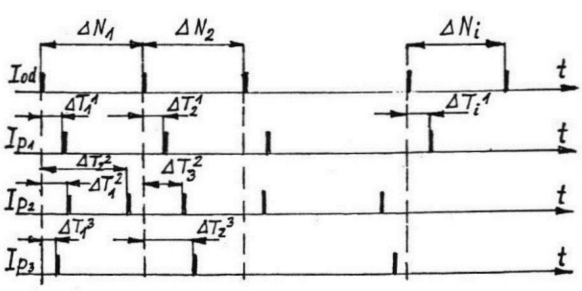

Fig. 5. Principle of blade vibration measurement, of the 2nd and further stages, with computer-aided DPM.

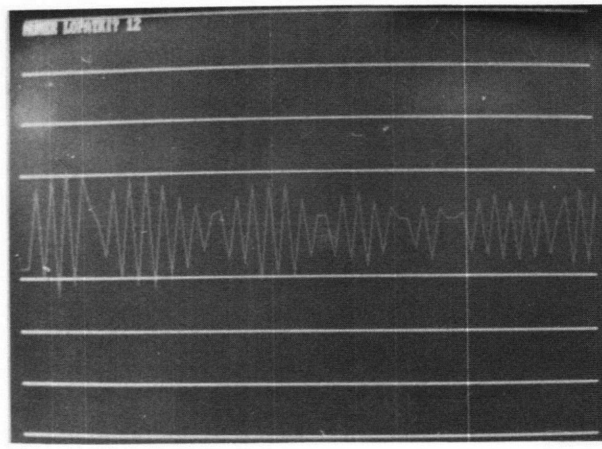

Fig. 6. Vibrations of the selected compressor blade, visualised on a computer display.

4. Advantages and capabilities of the developed measuring system

The developed system /see Fig. 4/ makes the measurement and analysis of blade vibration of any compressor stage possible, with limited interference into the engine structure under test. The vibrations of particular blades can be recorded on floppy discs and used to monitor the changes in blade vibrations, e.g. to monitor engine condition in the course of engine installation life /the wear and tear of bearings and seals, etc./. The analysis of vibration spectra of the blades under test is also possible and may include:

- evaluation of changes in frequency of natural vibrations of tested blades in function of rotation speed,
- finding-out the correlations between the amplitudes of blade vibrations and the frequencies of vibrations for given stage arrangements within the given resonant range of rotational speed,
- printing-out the maximum amplitudes of vibrations for all the blades of the stage under testing, for defined rotation speed, and printing-out the maximum vibration amplitudes for particular blades within the full range of rotation

speed,
- signalling the occurrence of cracks in blades, which is important for long--time testing of engine prototypes.

At present development efforts are in progress, aimed at extending the scope of system functioning.

5. Vibroacoustic method

The discrete-phase method enables us to precisely determine the compressor blade failure /i.e. cracking/ initiation and development. The technique requires the measuring equipment to be installed inside the engine /ref. 3/.

Numerous aero-engines are operated with the DPM equipment neglected. The damaged blades generate the engine vibroacoustic signal /spectrum/ fluctuations due to their free vibration frequency variations. The vibroacoustic spectrum fluctuations can be generated by the damaged bearings as well. Variations of this kind can easily be recorded and used for diagnostic purposes, see /refs 4,5,6/.

Aero-engine with simulated failures in four blades was put to tests at an engine test bench to evaluate the influence of the first compressor stage rotor blade cracking upon the vibroacoustic signal level. A notch of 0.5 mm depth was cut on the convex surface of the blade, 5 mm from the blade root. The simulated failure corresponded to the cracks frequently occurring in the aero-engines. The measurements followed by the analysis of a signal were carried out using Brüel-Kjaer equipment. The tests were conducted at five different engine rotor speeds. Figs 7, 8, 9 illustrate the results of the analysis of the vibroacoustic spectra, the engine rotor speed n_s = 9000 rpm being the exemplification. At this speed the differences between the vibroacoustic signal spectra of a defected engine and a healthy one were most significant.

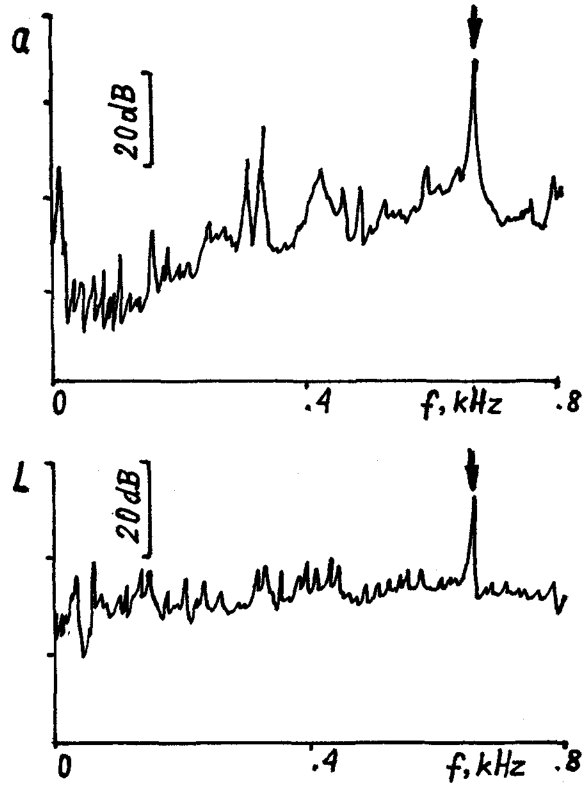

Fig. 7. Acceleration spectrum - vibration amplitude /a/ and noise acoustic pressure level /L/ of a turbine engine with damaged rotor blades of the 1st compressor stage, n_s = 9000 rpm.

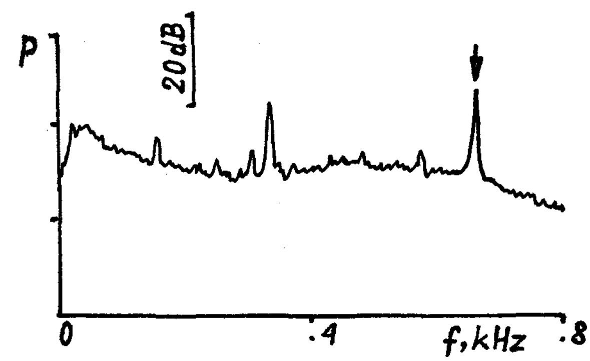

Fig. 8. Spectrum of air pressure fluctuations in a compressor duct of a turbine engine with damaged rotor blades of the 1st compressor stage p - fluctuation amplitude, n_s = 9000 rpm.

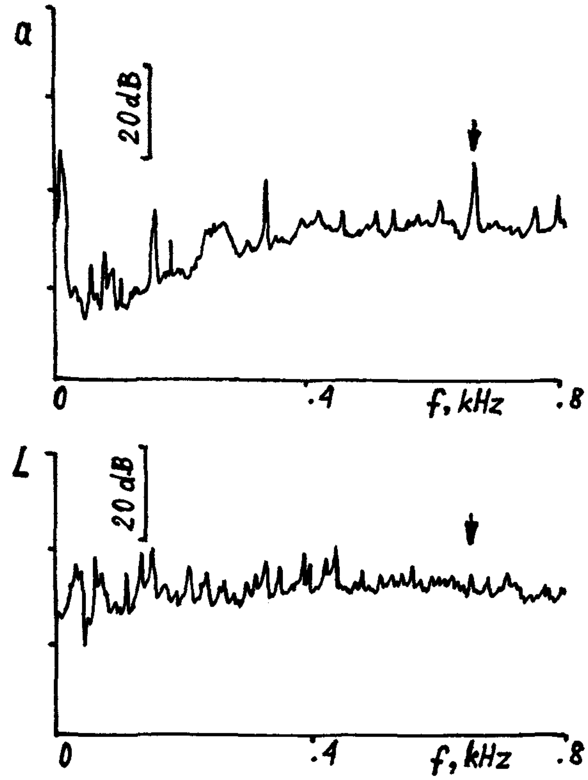

Fig. 9. Acceleration spectrum - vibration amplitude /a/ and noise acoustic pressure level /L/ of a healthy turbine engine, n_s = 9000 rpm.

In case of an engine with damaged blades high levels of a discrete component /corresponding to frequency f=660 Hz/ feature the spectra of vibration accelerations /a/ and noise /L/ /fig. 7/ as well as pressure fluctuations in the compressor duct /fig. 8/. The vibration acceleration level for the discrete component of the above-mentioned frequency is more than ten times higher in case of an engine with damaged blades than in case of a healthy engine /see /a/ plots in figs 7, 9/. The noise spectrum /L/ of a healthy engine /fig. 9/ is - in a wide range of frequencies f - of the noise-like nature whereas in the corresponding noise spectrum of an engine with damaged blades /see /L/ in fig. 7/ a component of frequency f = 660 Hz predominates. Diagnostic frequency f = 660 Hz coincidence for all spectra is a fact of real importance.

In case of defected roll-bearing of a SO-3 engine /engine test bench examinations, n = 7000 rpm/ the high level amplitudes in a band below 1 kHz can be found, in the vibration accelerations spectra /ref. 8/.

Interesting results have been obtained from vibration analysis of the SO-3 engine with a thermocouple of a central bearing incorrectly fitted. Due to the false fitting the thermocouple exerts excessive axial pressure on an outer ring of the bearing, attached to the engine body. As fig. 10 shows, a considerable increase of the amplitude levels, especially in bands above 3 kHz, occurred. Table 1 shows the amplitude ratios /α/ of vibration accelerations of harmonic components of rotor primary /rotational/ frequency.

Vibration spectra shown in fig. 10 have been obtained in the course of measurements in the axial direction equivalent to the direction of the thermocouple pressure on the bearing. The spectra seen in fig. 11 are equivalent to results of measurements in the horizontal direction perpendicular to the main engine axis. They do not show any substantial qualitative differences.

Tab. 1. Amplitude ratios /α/ of vibration accelerations of harmonic components of a rotor primary /rotational/ frequency.

Spectral line number	Frequency, f kHz	$k = \frac{f}{f_o}$	$\alpha = \frac{a_2}{a_1}$
1	0.47	4	2.9
2	3.27	28	3.7
3	3.50	30	8.2
4	4.55	39	16.4
5	4.78	41	11.0
6	5.02	43	17.5
7	5.60	48	6.0
8	5.83	50	5.8
9	7.00	60	20.0

$f_o = \frac{n}{60}$, where: n - rpm,

a_1, a_2 - amplitudes of vibration accelerations of the engine with a thermocouple correctly and incorrectly assembled, respectively.

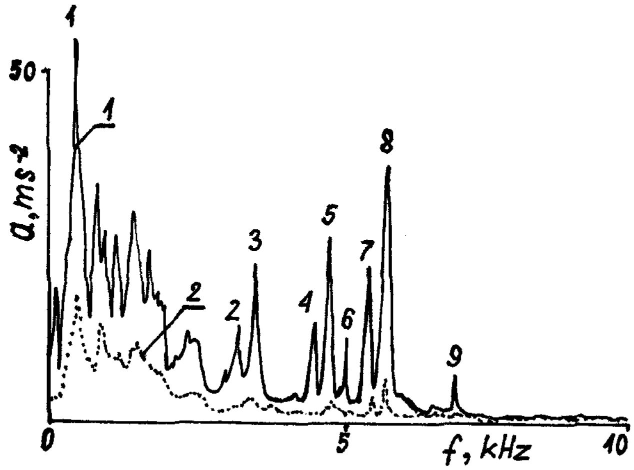

Fig. 10. Vibration acceleration spectra of the SO-3 engine with a thermocouple assembled /1/ incorrectly and /2/ correctly. The measurement made along the axial direction /longitudinal/.

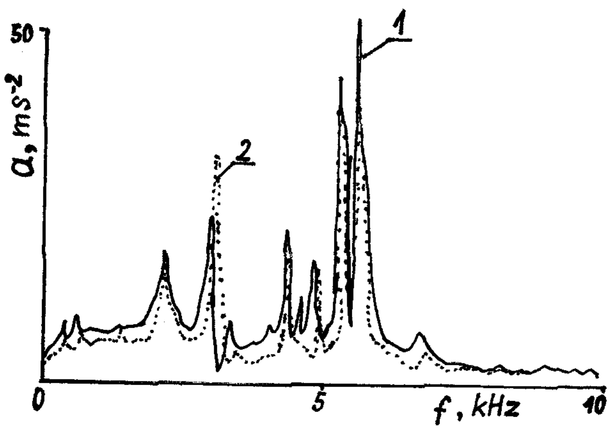

Fig. 11. Vibration acceleration spectra of the SO-3 engine with a thermocouple assembled /1/ incorrectly and /2/ correctly. The measurement made along the horizontal axis of the engine.

References

1. I.E. Zablockij, J.A. Korostieliew, R.A. Szapirow: Noninterference Measurement of Turbo-Machine Blades Vibration. Maszinostrojenije, Moscow 1977 /in Russian/,
2. H.T. Jones: Development of Noninterference Technique for Measuring Turbine Engine Rotor Blade Stresses, AIAA-85-1472, 1985, July, pp. 1-5,
3. R. Kudelski, R. Szczepanik: In-flight Early Detection of Cracks in Turbine Aero-Engine Compressor Blades, NDT 12th World Conference, Amsterdam 1989, Vol.1 pp. 616-623,
4. R. Kudelski, R. Szczepanik: Computer--Aided Noninterference Discrete-Phase Method of Vibration Measurement of Turbine Aero-Engine Compressor Blades, AIRDIAG'89, Poland 1989, Ref. 10, pp. 1-8,
5. E.W. Horne, L.R. Price, H.S. Edwards: Programmed Engine Maintenance C-5A Malfunction Detection Analysis and Recording System, MADRAS, SAE prepr.No 700820,
6. V.A. Karasev, V.P. Maksimov, M.K. Sidorenko: Vibracjonnaja Diagnostika Gazoturbinnych Dwigatielej, Maszynostrojenije, Moscow, 1978, /in Russian/,
7. S.M. Dorosko: Kontrol i Diagnostirowanije Techniczeskovo Sostojanija Gazoturbinnych Dwigatieliej po Wibracjonnym Parametram, Transport, Moscow, 1984, /in Russian/,
8. H. Dabrowski, J. Lewitowicz: Vibroacoustic Diagnostics of Aircraft Mechanical System, Conf. AIRDIAG'89, Poland, 1989, Ref. no 12, pp. 1-12.