EXAMINATION OF THE DYNAMIC PROPERTIES OF 1ST STAGE ROTOR BLADES IN ONE-PASS ENGINE COMPRESSORS UNDER OPERATING CONDITIONS

Ryszard Szczepanik

Air Force Institute of Technology,
Ksiecia Boleslawa Street 6, 01-494 Warsaw, Poland
tel.: +48 22 6851101, fax: +48 22 8364471
e-mail: ryszard.szczepanik@itwl.pl

Abstract

This article includes studies of vibration and stress amplitudes in the 1st stage rotor blades of jet trainer one-pass engine compressors before and after refurbishment in operating conditions.

The presented results were obtained using SAD-2 blade vibration amplitude registering and measuring apparatus. The same tests were carried on the same one-pass engine after modernisation. Example oscillograms from the vibration tests of the 16 blades are shown in this article, where show the vibrations of three randomly selected one-pass engines after refurbishment (marked 1, 2 and 3) out of a total of 50 engines. The engine vibration spectra cover the full rotation speed range. The difference between the vibration amplitudes of 1st stage rotor blades, in one-pass engine compressors before and after the refurbishment, is results from the difference in how the blades were attached to the disc. Before modernisation the hammer-type root was used, whereas after refurbishment dovetail-type fittings were applied. Furthermore, it was confirmed that there is nocoupling via the blade disc occurred when the blades were arranged on the disc according to the sinusoidal order of their free vibration frequencies. In such cases recorded vibration amplitudes remain within the average range (from 100 to 120 MPa in terms of stress).

Keywords: *diagnostics of aircraft engines, compressor blade dynamics*

1. Introduction

This article includes studies of vibration and stress amplitudes in the 1st stage rotor blades of jet trainer one-pass engine compressors before and after refurbishment in operating conditions. Before the refurbishment the 1st stage compressor blades were attached to the rim using the hammer root design (Fig. 10), whereas after the refurbishment the dovetail design was applied (Fig. 20).

Fig. 1. Hammer root of 1st stage rotor blade in one-pass engine compressor before refurbishment

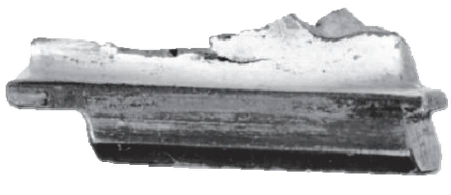

Fig. 2. Dovetail root of 1st stage rotor blade in one-pass engine compressor after refurbishment

The experimental studies into the dynamic properties of 1st stage compressor blades taking into account operational practice included [4]:

- strain gauge measurements of vibration and stress amplitudes in compressor blades on a test bench,

- measurements of blade vibration and stress during regular and irregular flights, i.e. during various manoeuvres such as high angles of attack and sideslip,
- research into how the selection of blade free vibration frequencies affects their vibration and stress and helps to explain the coupling of blades via the disc,
- measurements of 1st stage blade vibration and stress in aeroplanes with various degrees of throttling and misalignment, foreign objects in the inlet guide vanes and in unsteady (surge) conditions.

2. Analysis of 1st stage rotor blade vibrations in aeroplane engine compressors

Tests were carried on several decommissioned engines. Fig. 30 presents vibration amplitudes for 10 rotor blades with rotation speed fluctuating from n_{mg} to n_{max} and then back down to n_{mg}. The rotating stalls occurred in 6,500 to 10,300 rpm range. Very stress amplitudes were recorded when rotation speed was reduced from n_{max} to $n_{mg.}$ They were estimated to be ca. 350-400 MPa (1 mm of doubled amplitude corresponds to a stress' amplitude of 61 MPa). Such a level of vibration could lead to crack initiation, especially when we bear in mind that it is possible for it to be sustained when the DSS is very gradually reduced from n_{max} to n_{mg} (Fig. 40). In such situations there are differences between individual blade vibration amplitudes arising from slightly different rotation speeds. The behaviour of such individual blade vibrations suggests flutter.

The presented results were obtained using SAD-2 blade vibration amplitude registering and measuring apparatus [4]. The same tests were carried on the same one-pass engine after modernisation. The vibration measurements were recorded on oscillographic paper tape using two K-20 oscillographs with a special 32-channel adaptor.

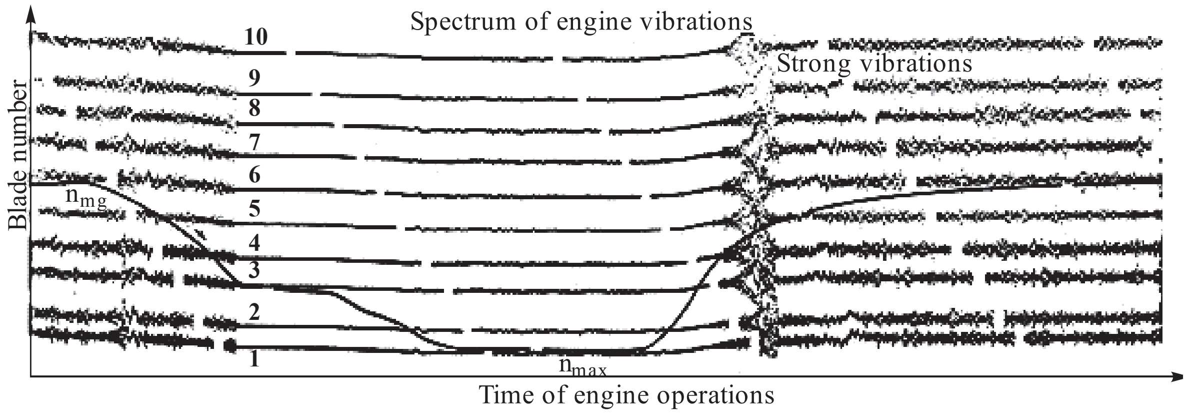

Fig. 3. Vibration spectrum of 1st stage rotor blades in one-pass engine compressor before refurbishment

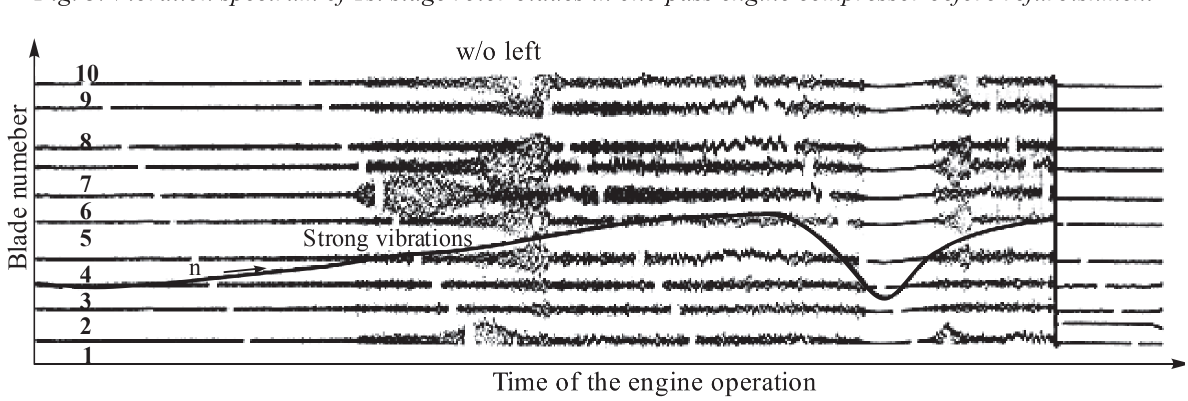

Fig. 4. Vibration spectrum of 1st stage rotor blades in one-pass engine compressor before refurbishment, without felt

Example oscillograms from the vibration tests of the 16 blades are shown in Fig. 5-7. These show the vibrations of three randomly selected one-pass engines after refurbishment (marked 1, 2 and 3) out of a total of 50 engines. The engine vibration spectra cover the full rotation speed range.

Fig. 50 shows forced blade vibrations exited by the third engine order (3EO = 3n) during acceleration from n_{mg}. This does not occur simultaneously in all the blades, which indicates free vibration frequency differences in the blades on the rim. Fig. 60 shows blade vibration forced by the 2nd EO (2n) at a rotation speed close to n_{max}, where the vibrations of disc and blades appear during acceleration. In Fig. 70 disc and blade vibrations appear during reduction of rotation speed to n_{max}, and vibrations forced by the second and third excitation frequencies are also visible.

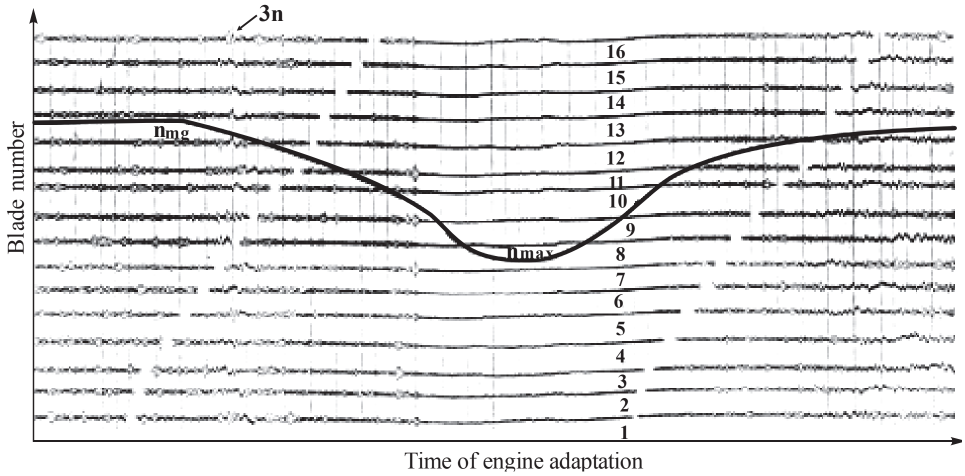

Fig. 5. Vibration spectrum of one-pass engine rotor blades after refurbishment (no. 1)

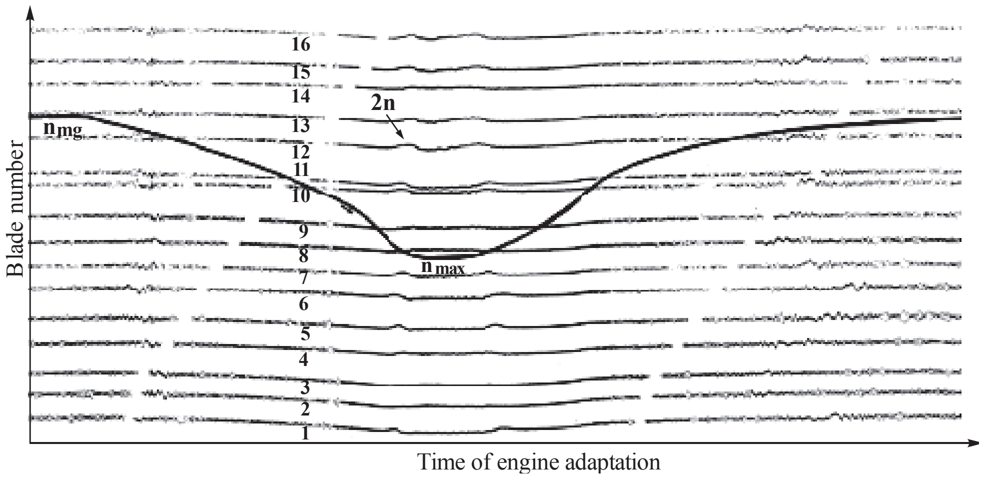

Fig. 6. Vibration spectrum of one-pass engine rotor blades after refurbishment (no. 2)

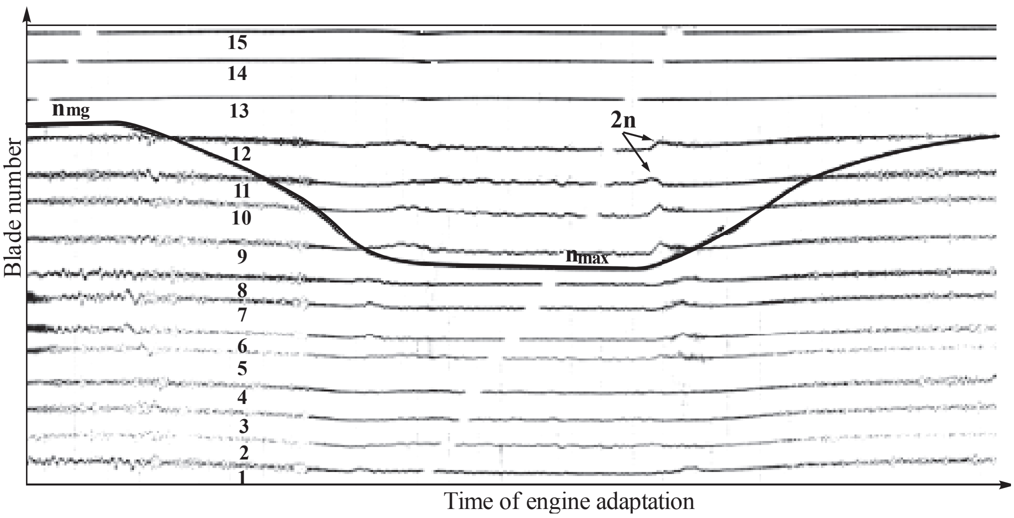

Fig. 7. Vibration spectrum of one-pass engine rotor blades after refurbishment (no. 3)

3. The influence of the selection of rotor blades with given free vibration frequencies on the overall vibration level

In 1998 research at the ITWL test bench was carried out into how the selection of 1st stage compressor rotor blades (with regard to their free vibration frequencies) affects stress amplitudes. Another area of investigation was to measure the time of crack initiation and propagation in 1st stage guide vanes were partly clogged by, e.g. by a piece of ingested cloth.

Research was also carried out into the possible influence of blades with given free vibration frequencies on the occurrence of blade vibration coupling via the blade disc. For this purpose a 32-channel adapter to the SAD device was designed enabling simultaneous recording on the readouts of two oscillographs, K-20 and K-12, the vibration amplitudes of all 28 1st stage compressor blades. Thus it possible to explain the eventuality of blade coupling via the blade disc.

This equipment considerably reduced the expenditure of time and money by eliminating the painstaking and time consuming process of applying strain gauges.

When the tensometric method was applied in the engine test bench, first blades with the broadest natural frequency range were selected for a compressor's 1st stage, i.e. f = 318-385 Hz, whereas the normal natural frequency range is within f = 320-350 Hz [5]. In subsequent tensometric tests the blades were selected in a random manner, and these revealed no excessive stress. This suggested that if the selection was not random (e.g. established according to a sinusoidal function) there may be the transfer of energy between blades (e.g. between those of the same natural vibration frequencies).

Air flow through the engine inlet was disrupted by placing a piece of paper in the guide vanes of the compressor's 1st stage. The engine was started and its rotation speed was smoothly increased from $n = n_{mg}$ to $n = n_{max}$. Blade vibrations were registered on a K12 oscilloscope (0). It turned out that at n = 8,000 rpm vibrations are forced by 3EO with a considerable amplitude. Subsequently, strong resonance vibrations (from the 3EO) occurred at n = 14,800 rpm.

Analysing the first resonance vibrations (from 3EO) one can clearly see that with the increase of rpm, the blades' resonance amplitudes form a sinusoid (0). This effect is used in a tip-timing analysis of blade vibration amplitudes to determine, on the basis of engine rotation speed, the vibration frequencies of synchronous blades. Knowing the exact rotation speed (excitation frequency), one can determine the free vibration frequency of blades subjected to mass force and working fluid flow.

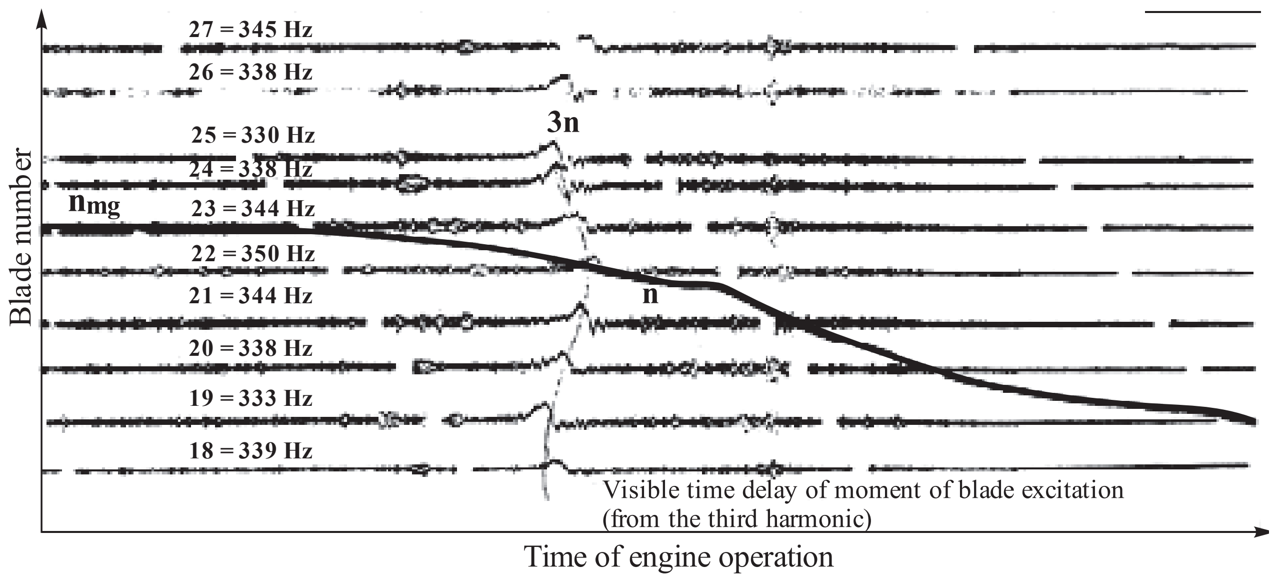

Fig. 8. Rotor blade vibration amplitudes. Visible excitations of blade vibration due to flow disruption by a sheet of paper lying flat on the guide vanes

The results show that in aforementioned engine no coupling via the blade disc occurred when the blades were arranged on the disc according to the sinusoidal order of their free vibration frequencies. In such cases recorded vibration amplitudes remain within the average range (from 100 to 120 MPa in terms of stress).

4. Computer analysis of measured rotor blade vibrations in the 1st stage of a compressor

This section presents the measured results of 1st stage rotor blade vibration amplitudes in a one-pass engine compressor before and after refurbishment using the SAD (system for the analysis of 1st stage of compressor blade vibrations) systems well as processed results using dedicated computer software [6].

Figure 90 presents the vibration amplitude standard spectrum of 14 1st stage compressor rotor blades in an aeroplane inlet. The rotation speed, from 8,000 rpm to 15,000 rpm, is on the ordinate axis, and the blade numbers are on the abscissa axis. Asynchronous vibrations prevail up to$n$=12,000 rpm. Above 12,000 rpm synchronous vibrations dominate and blade vibration amplitudes substantially diminish. Fig. 100 shows the vibration amplitude spectrum of compressor 1st stage rotor blades where one blade (nr 10) stands out on account of its deformed shape or faulty attachment to the disc. When the rotation speed is changed, this rotor blade bends differently in relation to the other blades (see incline between blades 10 and 11 in Fig. 100).

The blade amplitude curves in time (vertical lines in Fig. 110) indicate a tendency for blades to excite 2EO (2n) vibrations, visible at n = 14,800-15,200 rpm. This is due to the selection of rotor blades on the compressor disc whose vibration frequencies are from 490 to 500 Hz at 14,800 to 15,200 rpm.

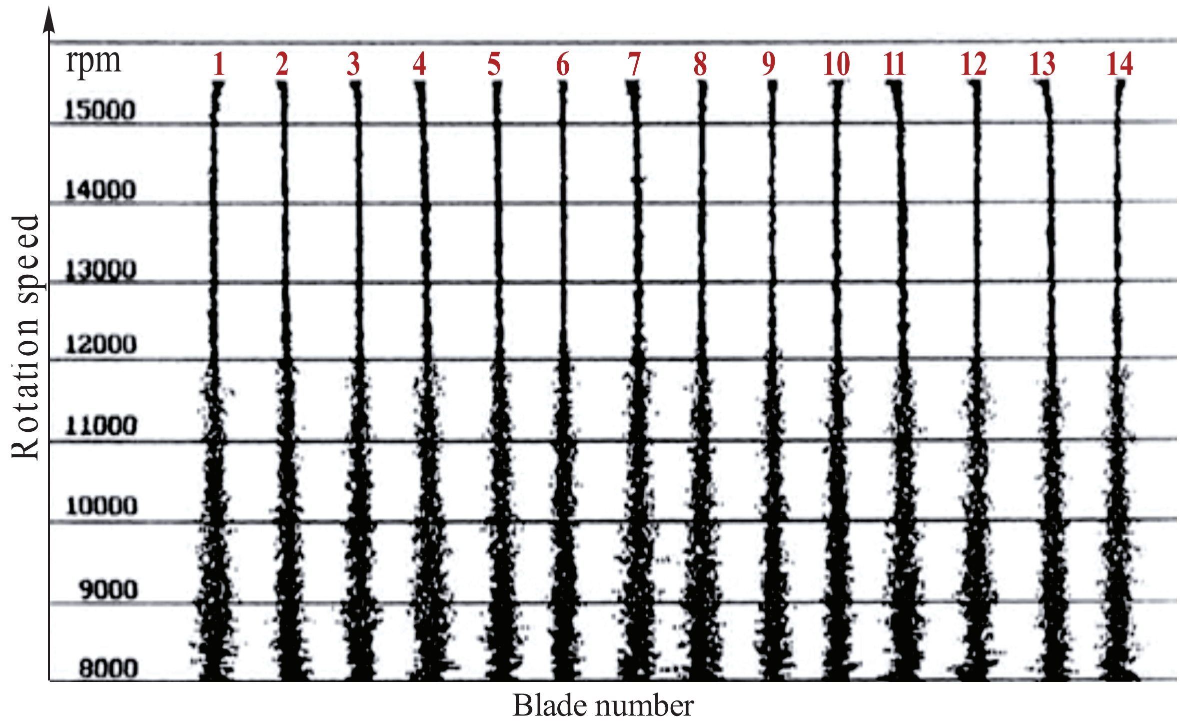

Fig. 9. Vibration amplitudes of rotor blades in the 1st compressor stage. Standard spectrum of one-pass engine compressor 1st stage rotor blades in aeroplane inlet

In the case of a foreign object, other flow asymmetry or rotor disruption the blades will have greater vibration amplitudes (Fig. 12), made apparent in vibrations excited by the second synchronous rotation speed (2n) at n = 14,800 to 15,500 rpm. This effect is not observed for other rotor blades depicted in Fig. 9 at n = 14,800-15,200 rpm.

Figure 12 also shows increased blade vibration amplitudes at 11,000 to 12,000 rpm, indicating susceptibility to surges.

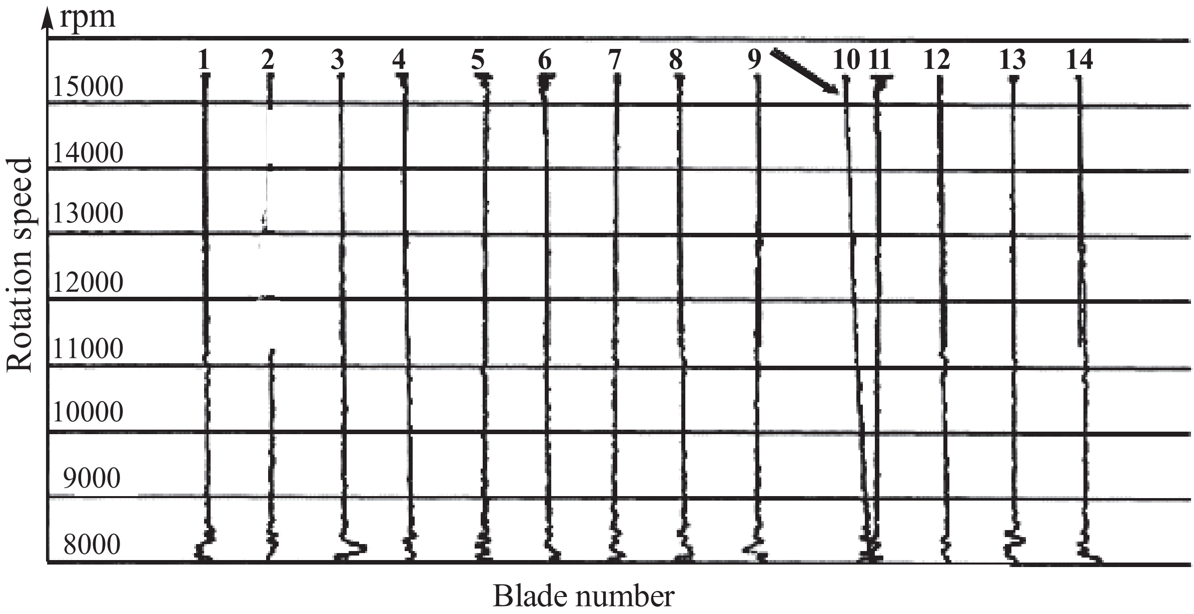

Fig. 10. Vibration amplitudes from 1st compressor stage rotor blades, with one rotor blade excessively deformed

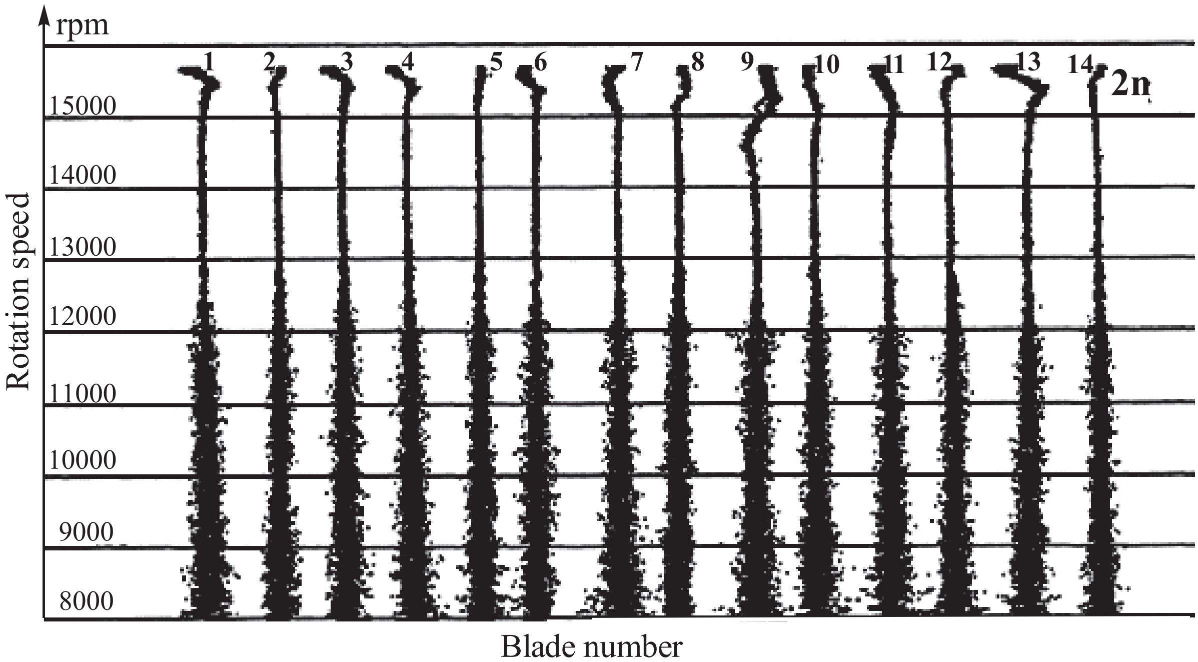

Fig. 11. Vibration amplitudes of one-pass engine compressor 1st stage rotor blades, with tendency to be excited by the second harmonic

Figure 13, in turn, presents blade vibration amplitudes during a surge caused by a large foreign object on the rotor blades. It occurs at above 14,900 rpm and the blade vibration amplitudes are much higher than in the case of 2EO (2n) as shown in Fig. 12. Fig. 13 also shows large blade vibration amplitudes excited by the 3EO at 8,000 to 9,000 rpm.

Figure 14 shows the course of rotor blade displacement with one blade cracked (marked with red circle). The initial cracking phase may be indicated by a reduction in the blade's vibration frequency or the appearance of resonance generated by 2EO at 14,300-15,000 rpm.

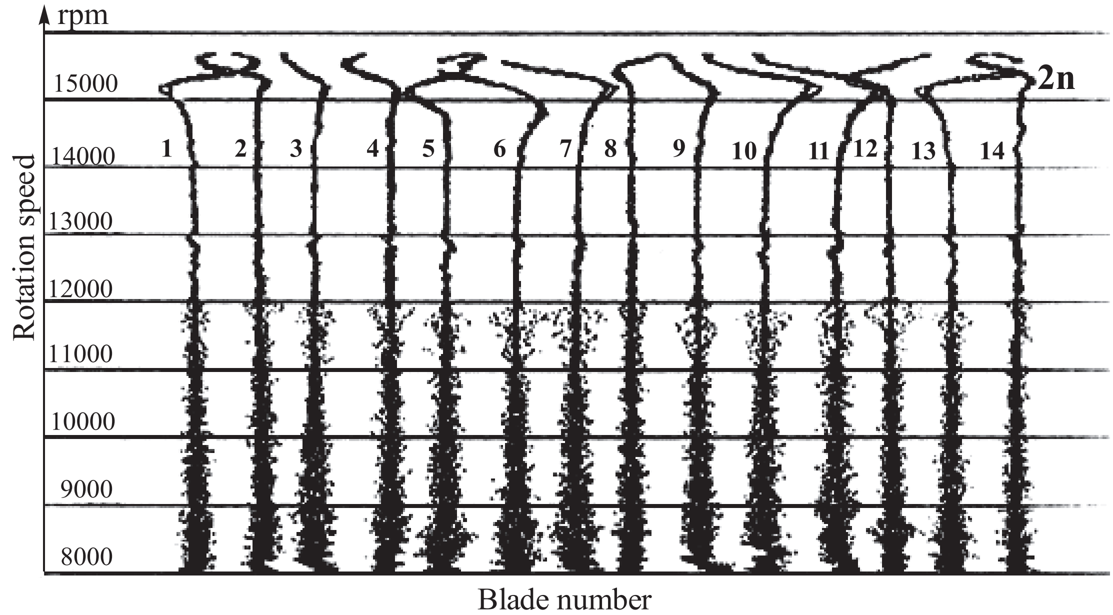

Fig. 12. Vibration amplitudes of compressor 1st stage rotor blades, with small foreign object on guide vanes

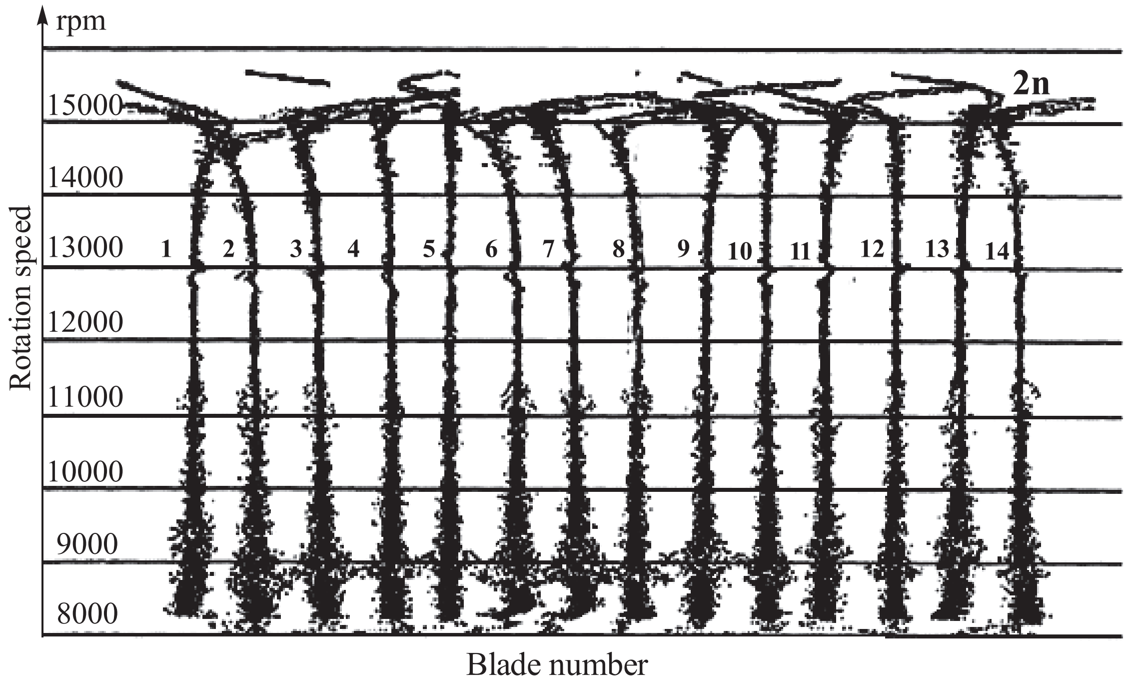

Fig. 13. Vibration amplitudes of compressor 1st stage rotor blades, with surge caused by a large foreign object on the guide vanes

The SNDŁ-1b and SPŁ-2b devices [2] make it possible to record increased blade stress amplitudes in working engines during aircraft flights. For instance, during one-pass engine trials [6] the compressor 1st stage rotor blade vibration amplitudes were additionally monitored by the SNDŁ-1b excessive vibration warning device. This test revealed (Fig. 15 and 16) that blade vibration amplitudes were larger at n = 7,000-8,000 rpm than at n = 11,000-13,000 rpm. It also showed that rotating stall zones disappear when rotation speed is increased to$n$ = 13,200 rpm. During repeated accelerations greater stress amplitudes appear at n = 12,000-12,500 rpm than at 11,000 rpm. Two blades, marked in Fig. 16 with arrows, demonstrate different rigidity or mass distribution from the rest.

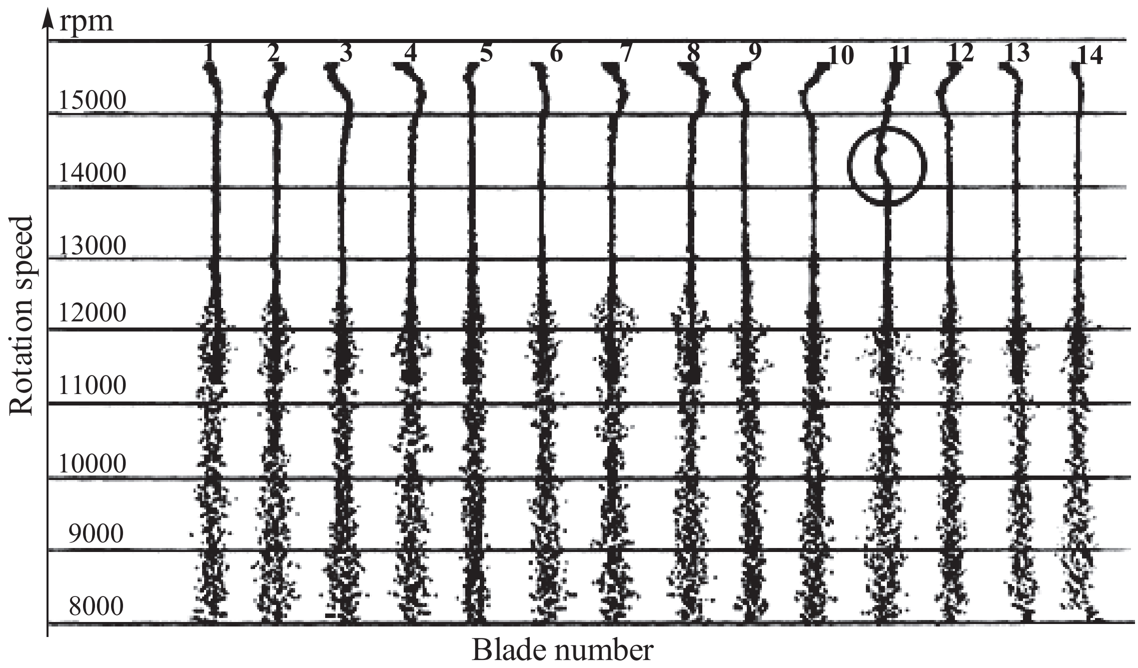

Fig. 14. Vibration amplitudes of compressor 1st stage rotor blades, with crack initiation in one of the rotor blades

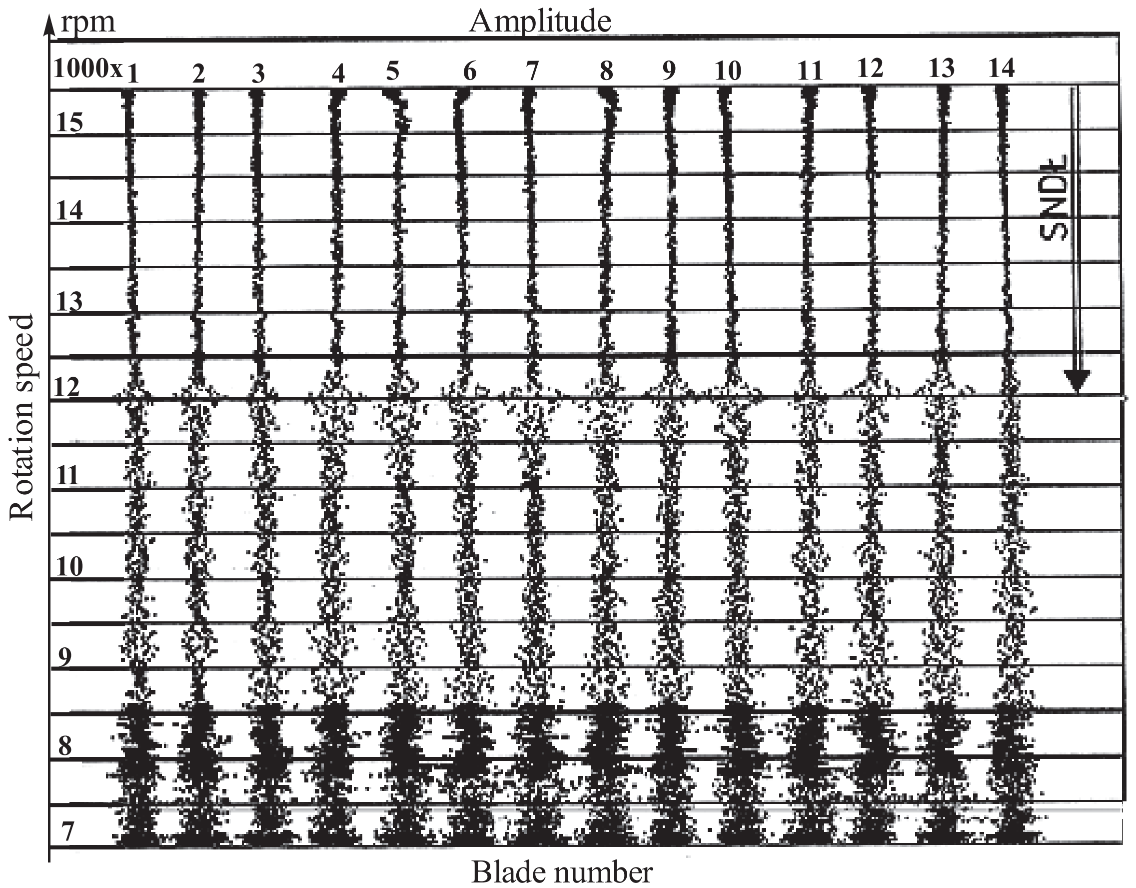

Fig. 15. Vibration amplitudes of compressor 1st stage rotor blades 1 and 14

The engine showed a permanent tendency of reducing the compressor safety margin and premature metal fatigue in its 1st stage blades caused by:

- impact of 3EO at about 8,000 rpm,

– impact of asynchronous rotating stall zones at 10,000 to 12,000 rpm,
– extensive vibrations that periodically appear at 12,000-12,500 rpm.

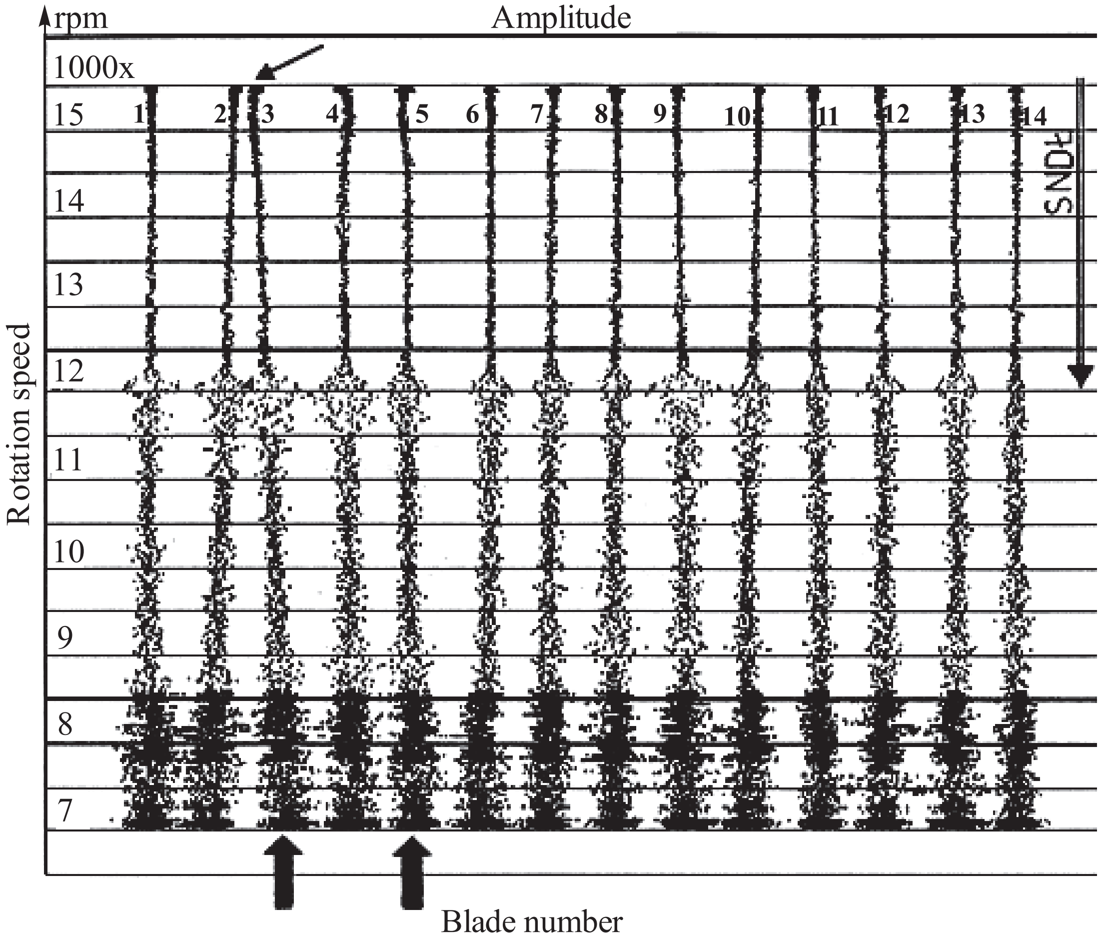

Fig. 16. Vibration amplitudes of compressor 1st stage rotor blades 15 to 28

5. Conclusions

Analysis of oscillograms showing the vibration spectra of all 1st stage rotor blades in 50 one-pass after refurbishment allows for the following conclusions:

– there is considerable repeatability in the measured vibration spectra of all the examined engines, both in the case of rotating stall zones as well as vibrations synchronous to rotation speed,
– none of the examined engines had registered vibration amplitudes exceeding 180-200 MPa (and these included an engine that had experienced a surge as well as one that had ingested a bird),
– the maximum vibration amplitudes of most of the examined were within the 60-100 MPa range (i.e. quite safe).

Metallographic studies showed that the blades in the refurbished engines had no initiated cracks in the region of 25% of their cross-section [3].

The difference between the vibration amplitudes of 1st stage rotor blades in one-pass engine compressors before and after the refurbishment results from the difference in how the blades were attached to the disc. Before modernisation the hammer-type root was used (Fig. 1), whereas after refurbishment dovetail-type fittings were applied (Fig. 2).

The above test results confirm the considerable potential of the presented devices in measuring compressor rotor blade displacements. The vibration amplitude measurements allow for the following diagnostic conclusions:

- a tendency for vibrations to be excited by 2EO (which make the presence of a foreign object very dangerous),
- the presence of a small foreign object in the guide vanes,
- occurrence of compressor surges,
- resonance caused by rotating stall zones,
- blade resonance vibration caused by synchronous and asynchronous excitations during acceleration and deceleration,
- rotor blade cracks,
- misshaped blades.

References

[1] Rządkowski, R., Drewczyński, M., Szczepanik, R., *Dynamic Analysis of Rotor Blades in One-Pass Engines,* ITWL Publishing House, Warsaw 2012.

[2] Szczepanik, R., Witoś, M., Szuty, S., Szczepankowski, A., Szewczyk, J., Krypa, M., *The SNDŁ-1b/SPŁ-2b Diagnostic System, Training Manual,* Studies of ITWL, No. 11650, Warsaw 1996.

[3] Szczepanik, R., Witos, M., Kowalski, M., Szczepankowski, A., *The Method for Alteration of the Vibration Frequency for a Rotating Blade of a Fluid Flow Machine, in Particular a Rotating Blade of a Compressor in a Turbine Aircraft Engine,* Patent No. 189360, Application date: 29th July 1998, Patent Assignee: The Air Force Institute of Technology, Warsaw 1998.

[4] Szczepanik, R., Kudelski, R., *Report from the Project 'Engineering and Construction of SAD Instruments for Studies on Vibrations of Blades Rotors in Turbine Engines',* Study of ITWL, No. 9121/I, 1988.

[5] Szczepanik, R., *Experimental studies on Dynamic Properties Demonstrated by Rotor Blades under Various Operating Conditions,* Scientific Journal No. 26, ITWL, Warsaw 2010.

[6] Witoś, M., *Diagnostics of the Technical Condition Demonstrated by Compressor Blades of Internal Combustion Turbojet Engines with Use of the Non-contact Method for Vibration Measurements* PhD Thesis, ITWL, Warsaw 1994.