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ROTORS FAULT DETECTION USING VIBRATION METHODS

Summary. Ships' propulsion plant usually works in a hard environment caused by static forces and permanent dynamic loads. Basic elements of propulsion systems are rotation machines like gas turbine engines, gear boxes, propulsion shafts etc. Another loads coming from technological faults of rotation machines like misalignment, unbalancing or resonance. Exciding of tolerated values of shaft alignments or unbalancing can cause a damage of radial and thrust bearings in relative short time. Similar situation is occurred when the mode or modes of rotors natural resonances are in the range of operational speed. The paper compares three methods of calculating and recognizing modes of rotors' natural frequencies using laboratory model of rotational machine. Results of FEM modeling, modal hammers measurements and synchronous vibration measurement show that free stop-down process is an interesting area for the vibration diagnosing of rotational machines.

IDENTYFIKACJA USZKODZEŃ MASZYN WIRNIKOWYCH Z ZASTOSOWANIEM METODY DRGANIOWEJ

Streszczenie. Okrętowe układy napędowe pracują w trudnych warunkach powodowanych obciążeniami zarówno statycznymi jak i dynamicznymi. Podstawowymi elementami układów napędowych są maszyny wirujące takie jak turbinowe silniki spalinowe, przekładnie redukcyjne, wały napędowe itd. Inne obciążenia pochodzą od błędów technologicznych występujących w maszynach wirujących takich jak odchyłki położenia geometrycznego osi wałów, niewyrównoważenie lub praca w zakresie rezonansu. Przekroczenie wartości tolerowanych parametrów współosiowości lub wyrównoważenia może relatywnie krótkim czasie spowodować uszkodzenie łożysk nośnych i oporowych. Podobna sytuacja wystąpić może w przypadku pracy maszyn wirnikowych w zakresie występowania rezonansów. W artykule dokonano porównania trzech metod identyfikacji postaci drgań własnych modelu laboratoryjnego. Rezultaty analiz wykonanych w środowisku MES, przy zastosowaniu młotka modalnego oraz synchronicznych pomiarów drgań wskazują, że podczas wybiegu maszyn wirnikowych możliwym jest identyfikacja zmian stanu technicznego maszyn wirnikowych z zastosowaniem pomiaru drgań.

1. INTRODUCTION

Monitoring of gas turbine rotors using vibration method allows recognizing changes of their technical state. Exceeding tolerated values of unbalancing or value of axes misalignment cause the increasing of the dissipation energy – vibration. Arise of the resonances vibration and bearings loads is the consequence of this situation.

Common methods of assessing technical state of the rotors systems often used off-line vibration measurements in steady states, usually without signals synchronisms. Alternative are measurements in non-steady states with the signals synchronism or with application the *Order Tracking* procedure. Another interesting method of rotors unbalancing identification is *Auto Tracking* procedure, very useful in the case of the lack of accessibility of synchronous signal.

Usefulness of mentioned methods should be identified in the laboratory stand because of the possibility of execution of effective experiments and cutting costs. Results could be adopted for identification numerical models of rotors system as well.

2. ROTORS UNBALANCING

Rotated object, caused oscillated, dynamic loads of bearing system is defined as unbalanced. Bering vibrations are effects of mutual interactions unbalanced mass and radial acceleration of the rotor system. Rotated unbalanced mass changed the direction of centripetal force. It tries to move the rotor in the bearing system along line of operation of the force. The rotor system with rotational speed ω is loaded by centrifugal force represented by the following equation (1) [4]:

$$\overline{F} = m_n \cdot \overline{r} \cdot \omega^2 \tag{1}$$

where: $\overline{r}$ - the leading vector defining site centre of gravity mass m_n

The static moment of the unbalanced mass respect the rotor axis $\overline{N} = m_n \cdot r$ is called unbalancing. Modulus of the unbalanced vector $|N| = m_n \cdot r$ is called value of unbalancing and the angle α is called angle of unbalancing.

The centrifugal force consists on preliminary force acting on bearing, from part of first unbalanced mass and the secondary force from second unbalanced masses in other surfaces of unbalancing.

Because of available methods of measurements the unbalancing can be dividing on:

- Statics – when the center of gravity of rotor are placed beyond axis of rotation and the geometrical and mean axis of inertia are parallel;
- Moments – when the rotation and geometrical axes are cut across in the center of gravity of rotor;
- Dynamics – when the center of gravity of rotor are placed beyond axis or rotation and the rotation and geometrical axes are cut across beyond the center of gravity.

All rotated objects are considered as the dynamics unbalanced systems because of the technological procedure of balancing.

3. BENDING VIBRATION AND CRITICAL SPEED OF ROTOR

Growths of value of bending vibration of gas turbine rotors are observed in define ranges of rotary speed. Mainly it is an effect of unbalancing or axes misalignment. Value of amplitude of first harmonic is depending of unbalanced mass and rotational speed. The axes slope causes increasing of value second harmonic of velocity of vibration. Additional, the reaction coming from inequality field of flow of air and gases forced on the rotor bearing system on both directions - axial and radial. That kind of tensions can makes, in relative short time, fatigues damage not only bearing system but all construction of rotor. Most exposed, on this type of loads is a gas generator rotor, which works in the range of temperature from 15^0 C up to 450^0 C. The rotated machines usually have also a forbidden range of rotational speed connected to the critical speed – n_{KR}. It is range of 0,7 – 1,4 n_{KR} where the operation of machine is allow during start – up, acceleration, deceleration and stop – down process.

Task of the determination critical speeds, which make resonance, bring to determination of the frequency of free bending vibration. It can be calculate value of period T of the rotor with 10 – 20% approximation using equation (2).

$$T = 2\pi\sqrt{\frac{f}{g}} \tag{2}$$

where: T - period of the free vibration [s], f – bending from own weight of the shaft (rotor) between supports [m], g – 9,81 [m/s^2].

4. SIMPLIFIED MODEL OF GAS GENERATOR ROTOR

Presented model simulates a real rotor of gas generator for analyzing usefulness three different methods of assessing natural frequencies and modal analyses (shapes) of resonance – fig. 1.

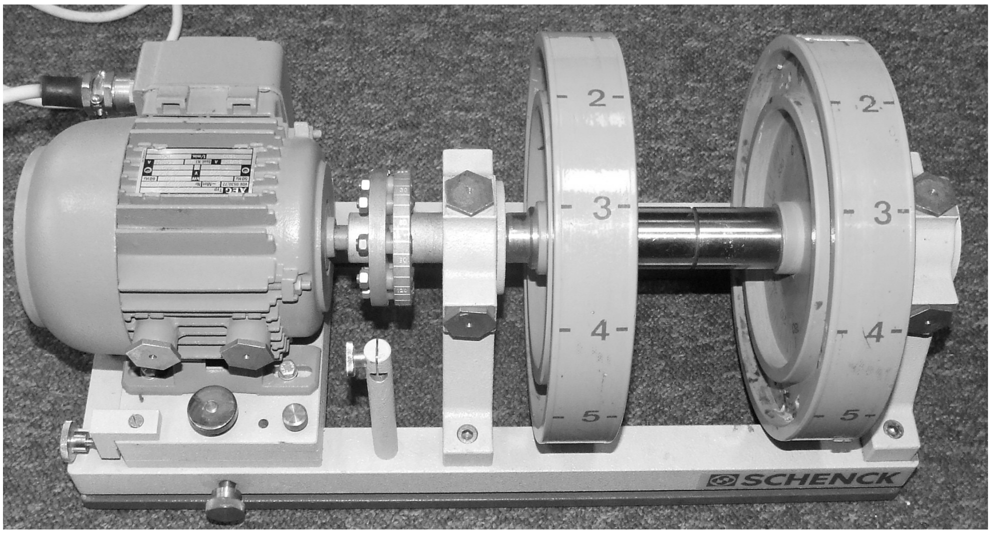

Fig. 1. Simplified laboratory model of gas turbine rotor
Rys. 1. Uproszczony model laboratoryjny wirnika silnika turbinowego

First test had verification of usefulness of modal hammer technique on purpose of modes of natural frequency of the rotor. The tests were made with use of analyzer type B&K 3560-B-120, gauges type B&K 4398 and modal hammer type B&K 8206-003. The rotor was taken out from the support and next six measurements, in two main directions – axial and radial, were accomplished – figure 2.

It gets in result of research as frequency characteristics and diagram of coherence – figure 3 and 4.

Table 1

Natural frequencies of rotor (shapes)

	1	2	4	5	6	7	8	9	10
freq [Hz]	16	410	414	710	804	1438	1494	1804	2798
freq [rad/s]	100,53	2576,1	2601,2	4461	5051,6	9035,2	9387	11334	17580

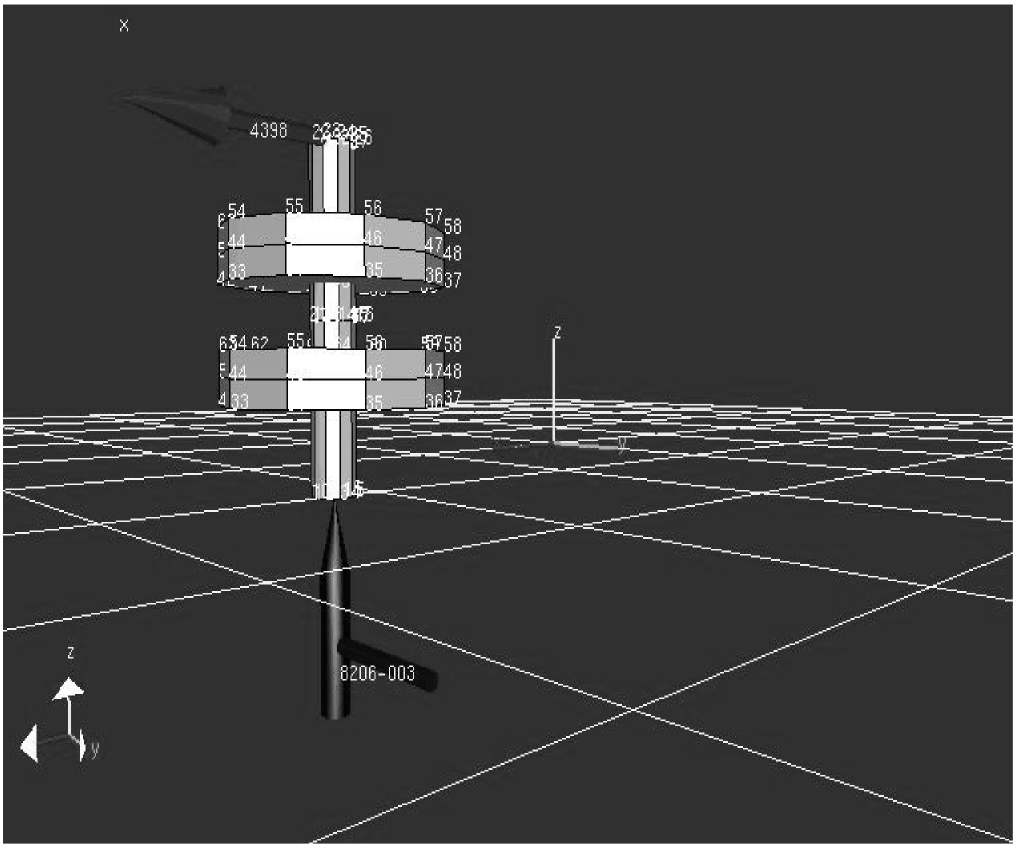

Fig. 2. Directions of analysing axes using the modal hammer
Rys. 2. Kierunki pomiarów z wykorzystaniem młotka modalnego

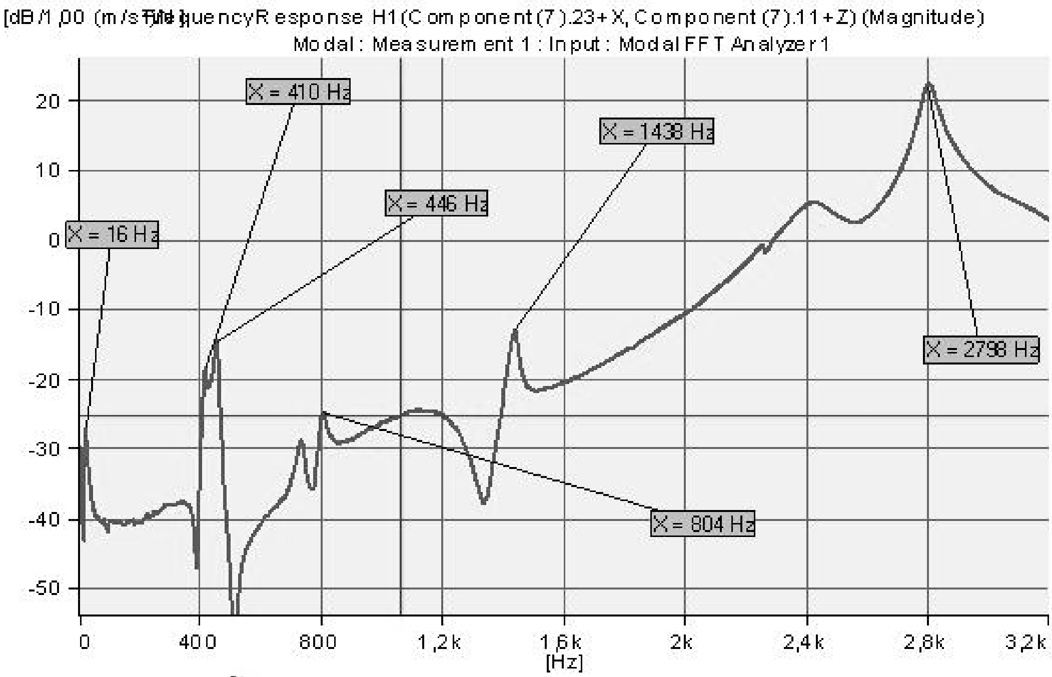

Fig. 3. The frequency response of modal hammer impact
Rys. 3. Odpowiedzi w dziedzinie częstotliwości z wykorzystaniem młotka modalnego

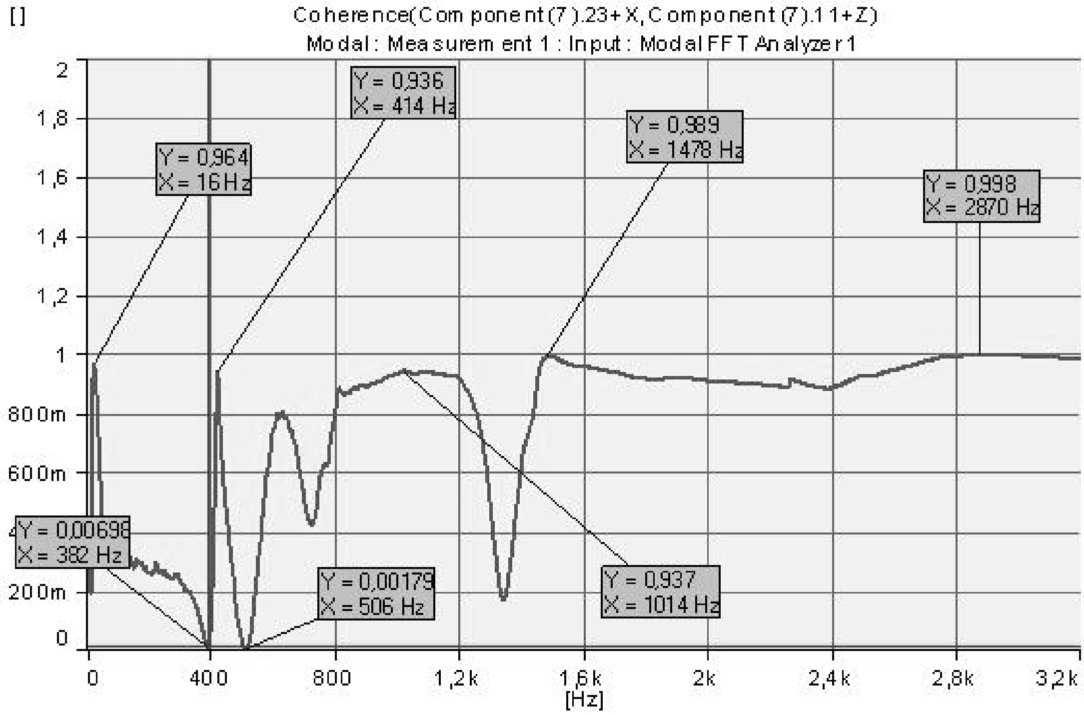

Fig. 4. The diagram of coherence using modal FFT analyser
Rys. 4. Charakterystyka koherencji z użyciem analizy modalnej FFT

Analysis of both characteristics allows presenting following modes of natural frequencies – table 1.

5. RESULTS OF „ANSYS WORKBENCH" ANALYSIES

Analyze of the exact virtual model of the rotor of gas generator is complicated and it is require the high advanced work station. Preparing an adequate model of rotor is a not a technical problem – figure 5.

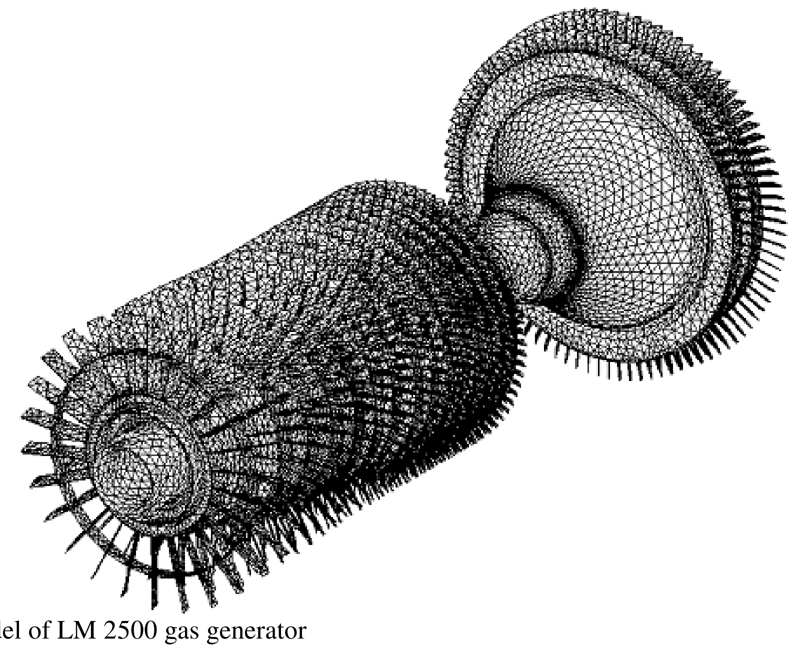

Fig. 5. Virtual model of LM 2500 gas generator
Rys. 5. Wirtualny model wirnika wytwornicy spalin silnika LM 2500

Much more complicated task is the calculation modes of shapes of the model witch consists of over 100 000 elements. It was a reason to try verified proposed method for much more simplified object like rotor presented on the figure 4. Simplified laboratory model of rotor was prepared as a virtual model in the Solid Works software – figure 6. The model has been subjected analyses in the CAE „Ansys Workbench 11.0" for calculating natural resonances and modes of shapes.

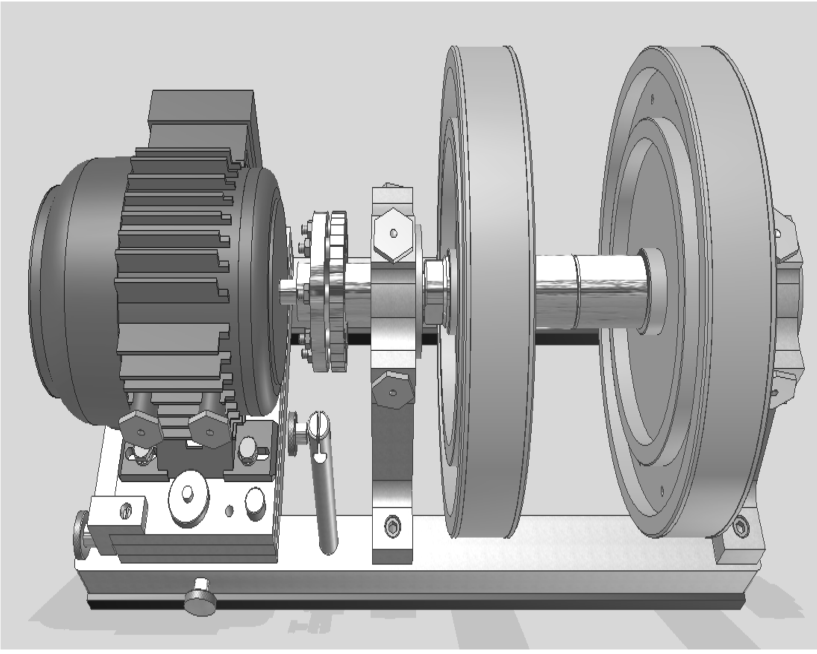

Fig. 6. Virtual model of laboratory's rotor
Rys. 6. Wirtualny model wirnika laboratoryjnego

It carries research for two constrains (red points in the fig. 7 and 8):

- like in the bearing system of support – fig. 7;
- along axis of rotation – fig. 8.

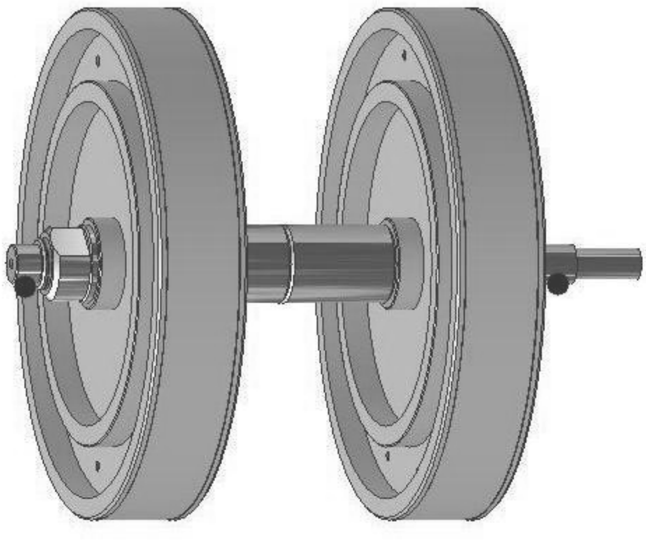

Fig. 7. First stickseed support of rotor
Rys. 7. Pierwszy sposób utwierdzenia wirnika

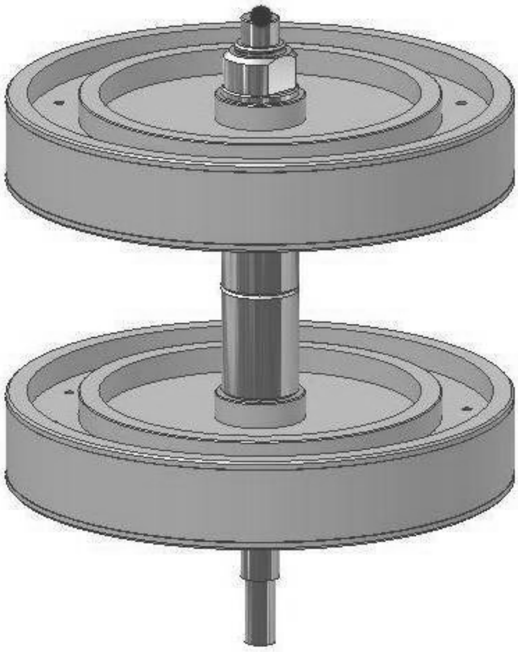

Fig. 8. Second stickseed support of rotor
Rys. 8. Drugi sposób utwierdzenia wirnika

The researches carried out two results for both methods of rotors handle – table 2.

Table 2
Results of „ANSYS WORKBENCH" analyses

Freq.	Handle nr 1		Handle nr 2	
	Natural frequen	Natural frequen	Natural frequen	Natural frequen
nr	ω [rad/s]	f [Hz]	ω [rad/s]	f [Hz]
1	67,8051	10.797	67,80516	10,797
2	2603,1856	414,52	116,88964	18,613
3	3204,37	510,25	158,78352	25,284
4	5197,8304	827,68	3204,37	510,25
5	7604,452	1210,9	4271,7816	680,22
6	–	–	7591,892	1208,9

6. RESULTS OF ORDER TRACKING ANALYSIES

Researches were carrying out on the rotor presented in the figure 1. The main task of the test was identification of natural frequencies of the rotor during the start-up and stop-down process. The acceleration and velocity of vibration were adopted for synchronous measurements with the use of optical tachometer.

First tests analyzed vibration signals in the typical synchronous spectrum FFT – figure 9.

It was appeared that typical, synchronous Autospectrum of vibration do not bring enough information for recognizing natural frequencies. Next step of researches was analyzing vibration signals with the use of the Order Tracking procedure, without forced braking – free stop-down deceleration up to stoppage – figure 10 and 11.

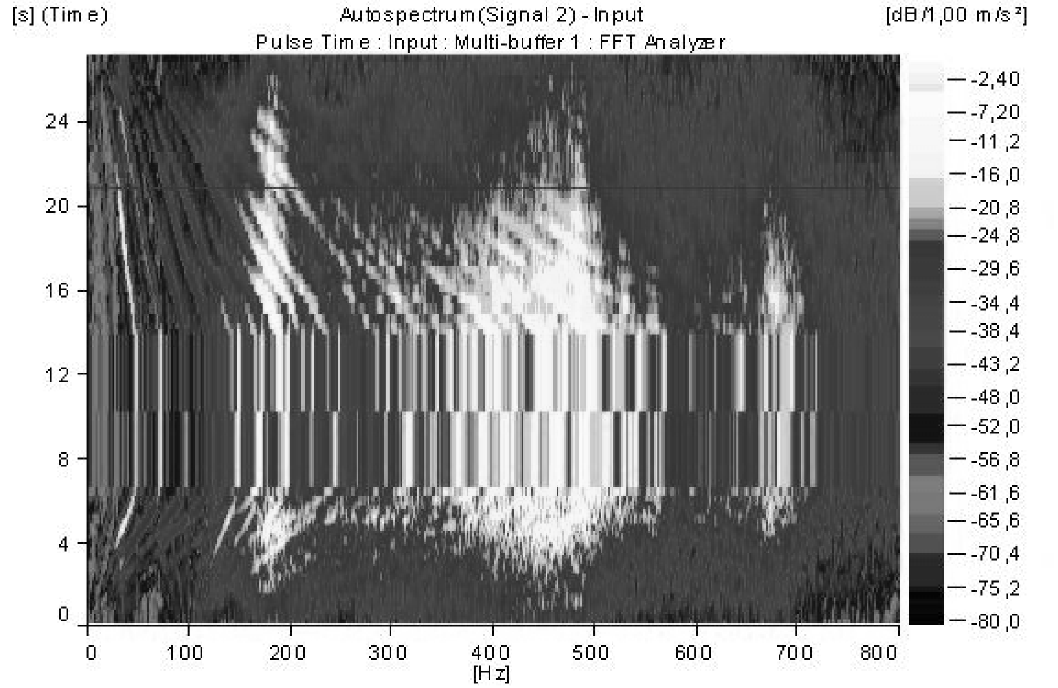

Fig. 9. Autospectrum FFT of acceleration with forced breaking of rotor
Rys. 9. Widmo FFT z przyspieszenia wirnika z wymuszonym oddziaływaniem

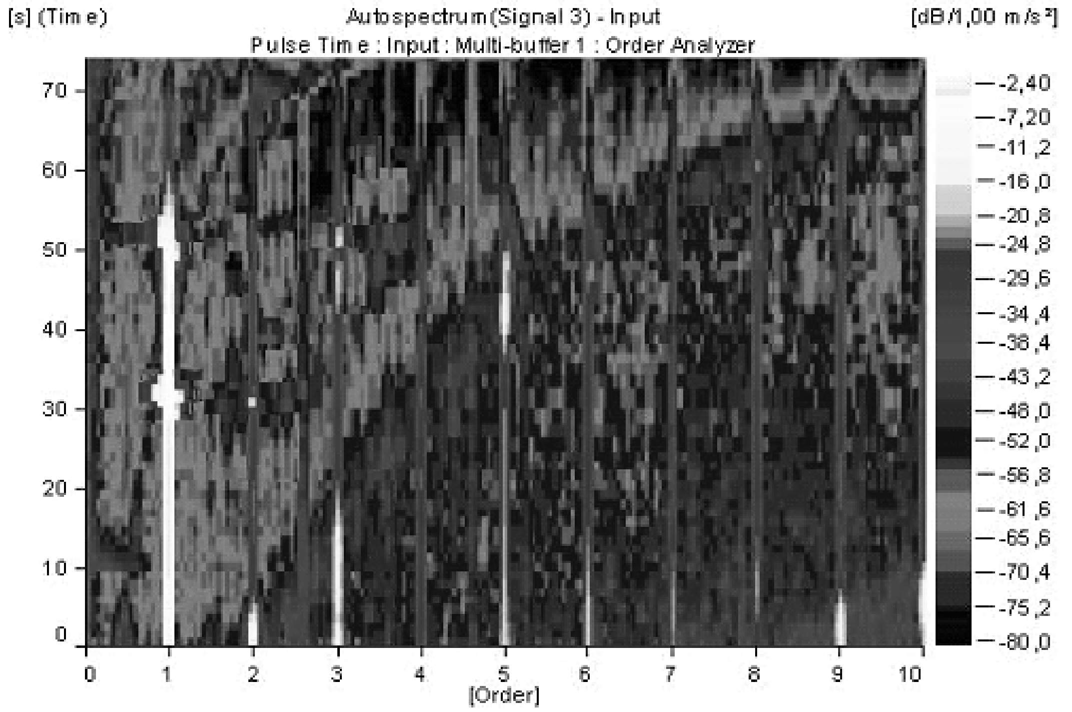

Fig. 10. Order tracking procedure for free stop-down process of the rotor
Rys. 10. Wyniki analizy rzędów podczas swobodnego wybiegu wirnika

It is well visible on the figure 11, those amplitudes of acceleration of 1-st harmonic in time points 29 second and 51 second rapidly grown up and for next 2 – 3 second drop down.

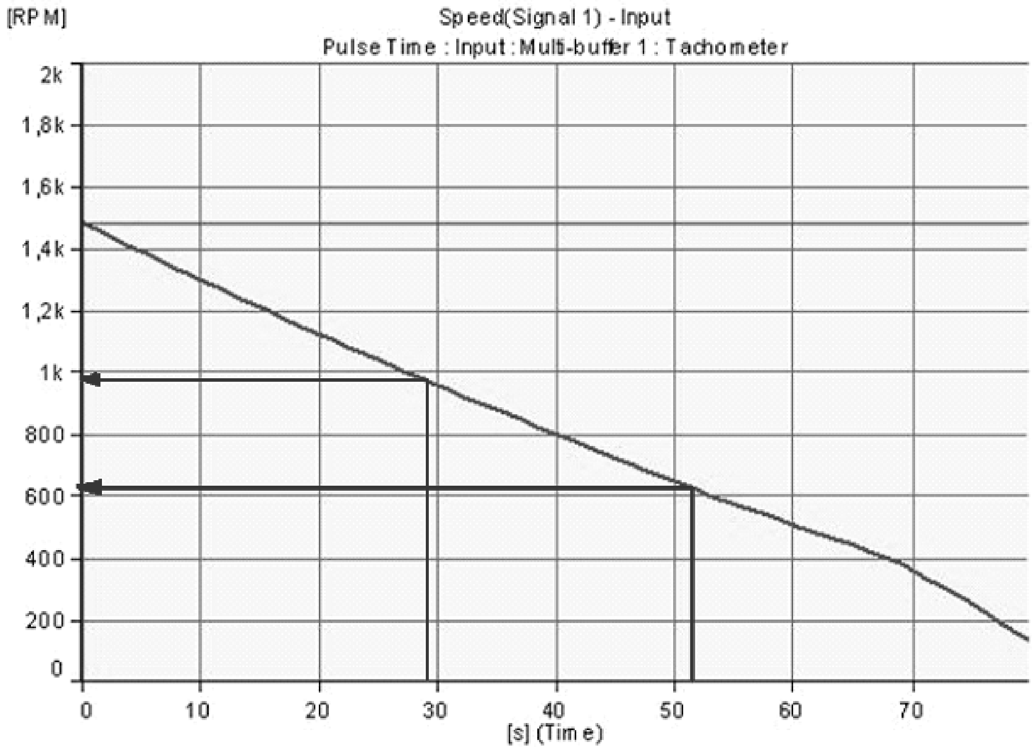

Fig. 11. Characteristic of rotors rotational speed during free stop-down deceleration
Rys. 11. Charakterystyka zmian prędkości obrotowej wirnika podczas swobodnego wybiegu

Analyze of characteristic of rotors rotational speed (fig. 11) shows that arising amplitudes of acceleration had place near 610 rpm (10,2 Hz) and 980 rpm (16,3 Hz). It means that during free stop-down process of deceleration it is possible to recognize natural frequencies - compare tables 1 and 2.

7. FINAL CONCLUSIONS

Table 3 includes all results of identification natural resonances. The presented results of modelling and measurements related to the performed experimental tests confirm it possible to recognize natural frequency in the free stop-down deceleration process. It is important knowledge because all rotated machines can be described by vibration spectra like a fingerprint. Any changes of mass of inertia, unbalanced masses or changes of supports stiffness provoke changes the natural frequencies. It means that process of the comparing with the previous and present acceleration spectra allows identified changes of technical state of the rotor system.

Table 3

Results of natural frequencies of the laboratory rotors' system

	Natural frequencies [Hz]							
Modal hammer	–	16	–	414	–	710	804	1438
Simulation nr 1	10,7	–	–	414	510	–	827	1210
Simulation nr 2	10,7	18	25	–	510	680	–	1208
Order tracking	11	17,5	63	408	–	–	–	–

The presented conclusion is important because the most of vibration monitoring systems of gas turbine engines analyse signals from start-up point up to pressing button STOP. Successive experimental tests will make it possible to verify features of the signals assumed for the analysis, to be able to build reliable models of the stop-down spectra for the monitoring system of-line type.

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