Diagnostics of Fatigue Cracks in Rotor Systems of Machines

A. Zakhezin, Y. Pryadko, O. Kolosova, and S. Aliukov

***Abstract*— High-speed loaded rotor systems are widely distributed in machines and equipment of various branches of technology, including transport. Maintaining rotor systems at the appropriate high technical level is an important task, the solution of which is necessary to reduce current costs, to decrease the number of emergency situations. In this paper, the process of appearance of fatigue cracks in rotor systems is considered and the degree of their development is determined. Based on the carried out studies, the method for predicting the development of the fatigue cracks has been developed. The destruction of structural elements under variable loads usually occurs gradually, as a result of the accumulation of microdamages that pass into the developing fatigue microcracks. The appearance of cracks does not mean that the structural elements are completely out of order and that immediate repairs are necessary. In practice, structural elements with cracks can continue to function reliably for a considerable time, and a slight decrease in operating loads can increase their durability and even completely halt the growth of cracks. Therefore, when vibrating the rotor machines, it is necessary to determine not only the presence and depth of the crack, but also its location, since the rate of crack growth depends on the level of stresses arising and on the length of the present crack. Early diagnosis of the appearance of defects in these systems solves many problems. The most important element of the analysis of the state of the structure is the determination of the degree of development of the crack, the determination of the time to reach the critical size. This allows us to remove the costly risks of disturbing critical technological processes. As shown in this paper, if detection of defects is possible with the help of modal and Fourier analysis, wavelet analysis makes it possible to determine the degree of crack development over time. Using the sample of 10 experiments and calculations involving modal, Fourier and wavelet analysis of one of the rotor systems, we obtained in the paper the results allowing extending them to rotor systems of different designs.**

***Index Terms*— Vehicles, diagnostics, rotor systems, fatigue cracks.**

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A.M. Zakhezin is with the South Ural State University, 76 Prospekt Lenina, Chelyabinsk, 454080, Russian Federation (e-mail: a.zakhezin@gmail.com).

O.P. Kolosova is with the South Ural State University, 76 Prospekt Lenina, Chelyabinsk, 454080, Russian Federation (e-mail: kolosovaop@susu.ru).

Y.G. Pryadko is with the South Ural State University, 76 Prospekt Lenina, Chelyabinsk, 454080, Russian Federation (e-mail: yuri_pryad@mail.ru).

S.V. Aliukov is with the South Ural State University, 76 Prospekt Lenina, Chelyabinsk, 454080, Russian Federation (corresponding author), home phone: +7-351-267-97-81; sell phone: 8-922-6350-198; e-mail: alysergey@gmail.com.

I. INTRODUCTION

NOWADAYS many theoretical and experimental researches in the field of vibration analysis devoted to the investigation of the rotating rotor and shaft with crack, particularly in the field of vehicles. The purpose of these studies is to develop a method for diagnosing the location of a crack and determining its depth. Some researchers used the simple mechanism of the non-linearity «breathing» mechanism of the crack [1-3]. There is a complete closing and opening the crack according to cross moving of this rotor under action of own weight during the each revolution of this system. This model takes into account the change in the stiffness of the system, and the coefficients of the stiffness matrix depend on time. Other researchers determine the process of opening and closing of the crack is by the bending moment in the crack location [4]. The main axis and rigidity changes are represented by the function of the crack instant area. The experimental investigations show that the process of the crack opening and closing can be described by sine wave dependence such as $1+cos\,\psi$, where ψ - angle between a main inertia axis of the weakened section and weight force direction during one revolution of the rotor [5]. In these models the local coefficients of the compliance to the relations of loads are considered as dependence for density deformation energy of the core. The model describing in this article with help of the modal analysis of the nonlinear dynamic system and modal contributions of the eigenvectors and eigenvalues changes allows to determine the 2-5% crack from diameter.

The detection of a growing fatigue crack of a shaft is based on the diagnostical model of the system consisting of the rotors, flexible shaft supported by the ball bearings. The purpose of these investigations is to development the diagnostic method of the crack location and determination of its depth.

Experimental results have been obtained for different depth of the crack - 5%, 10%, 15% and 30% from shaft diameter. Experimental results shown that when a cracked occurs in the rotor system, it can be detected by the statistical characteristics of the vibrosignals.

The modal analysis of the three-dimensional finite-elemental model has been executing using computer program STARDYNE. This system has got distributed weight, rigidity and damped parameters. 600 eigenvectors and eigenvalues have been computed for 1100 Hz frequency range for the various models: without crack and crack in the rotor (5% from diameter); crack in the shaft (5%, 10%, 15%, 30% from diameter). The executing of the modal contributions appropriated to different frequency ranges allows to determine the frequency range where the change of the vibrosignal should be most appreciable from the crack.

II. CRACK PARAMETERS

The destruction of the construction elements on variable loads happens usually gradually in consequence of microdamages accumulation which are developed into fatigue cracks. In practice the construction elements with crack can function reliably during considerably length of time. Sometimes the decrease of the operation loads can increase their longevity. Therefore it is necessary to determine not only crack depth but also its location because growth rate of crack depends on stress level and from crack length. Rotation of the rotor machine parts leads to cyclical loads alternating inelastic deformation connected to crack gradual development. The expansion stresses reaching yield stress levels are replaced compression stresses during unloading stage.

Ten samples of the shaft were made for vibrodiagnostic of the fatigue cracks in rotor systems. The cracks various depth from shaft diameter have been grown in stress concentration (trough near ball bearing) with help of the vibrostand simulating a cyclical load. The installation scheme is shown in Figure 1.

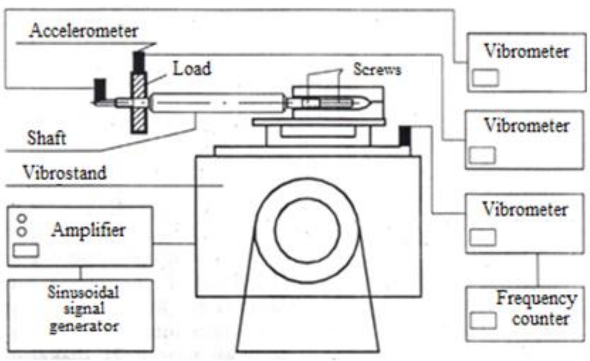

Fig. 1. Installation for crack growth

The shaft was fixed by screws in special devise and was install on vibrostand. The thick end of the shaft was hung up the load and was fixed the nut lock (Figure 2).

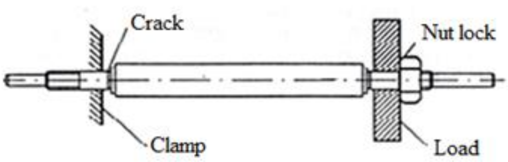

Fig. 2. Crack location

The frequency of the sinusoidal signal generator was set near third resonance frequency of the system during first part of the test. The crack emerging was defined by increase of the vibro-acceleration amplitude. After crack detecting frequency of the sinusoidal signal was set in neighborhoods of the third resonance frequency. The amount load cycles before destruction was essentially increased during second part of the test. The use of the two part of the loading could appropriate the process of the crack development and growth to real condition of the crack fatigue growth in shaft on non-stationary loading [4,5].

III. EXPERIMENTAL SETUP OF THE ROTOR SYSTEM

The experimental installation is shown in Figure 3.

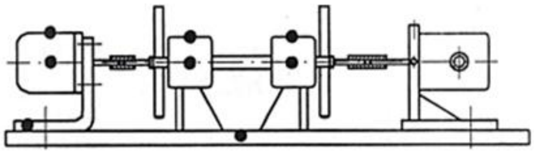

Fig. 3. Experimental installation

The shaft without crack was rotated and vibrations were measured using accelerometers 4370 (B&K), dual channel analyzer 2034 (B&K), tape recorder 7005 (B&K), analog digital converter (ADC) and analyzed by the computer.

The shaft was replaced by shaft with different depth from diameter and vibrations were measured again. The vibration probes were located on two bearing pedestals in horizontal and vertical directions. The vibroaccelerations of the rotor system were recorded and converted in digital forms and used in finite-elemental model as input excitation [8].

IV. FINITE-ELEMENTAL MODEL

The spatial model of the shaft and flexible rotor was constructed using finite elemental analysis and consists of 200 three-dimensional solid elements connected by 440 nodal points (Figure 4). The 5% depth crack has been simulated in the shaft. The model of the crack has been created by special finite elements with nonlinear damping and rigidity characteristics simulating the loss energy because of "breathing" mechanism of the real crack [6]. The ball bearings and bearing pedestals have been simulated by rigidity coefficients enclosed on the nodes of the finite-elemental model. The vibroaccelerations of the rotor system were recorded and converted in digital forms and used in finite-elemental model as input excitation.

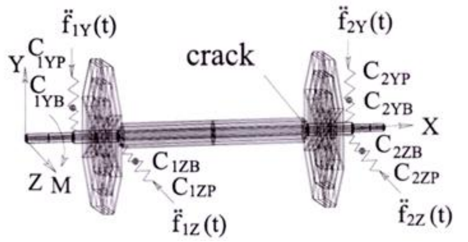

Fig. 4. Finite-elemental model

V. MODAL ANALYSIS OF THE SYSTEM

The modal analysis of the three-dimensional finite-elemental model has been using computer program STARDYNE [8,9]. The system has distributed weight, rigidity and damped parameters. 600 eigenvectors and eigenvalues have been computed for 2 kHz frequency range for the various models: without crack and crack in the rotor (5% from diameter). In model analysis a complex deflection pattern of the vibration structure is resolved into a set of simple mode shapes with individual frequency and damping parameters.

The appearance and increase of the crack will be change forced dynamic deflection of a structure represented as a weighted sum of its mode shapes and therefore change eigenvectors and eigenvalues of the system. The elements of the mode shape vector are the relative displacements of the each DOF hence the crack location influence on the specific eigenvectors connected to different frequency range. Thus the increase of vibrosignal amplitude in the certain frequency range will be connected to change of the modal contribution for eigenvalues appropriated to this frequency range. The firsts 600 eigenvalues are located in frequency range from 0 up to 2000 Hz, and their full contribution in rotor elements stresses near the crack are considered as 100%. The model contribution of each eigenvalues has been computed for stress in element of this model and has been obtained the most energetic contribution for various models (Tables 1 and 2).

Table 1 - Model contributions of the system (without crack)

№	№ mode	Frequency, Hz	Contribution, %
1	23	61	16.08
2	40	88	6.35
3	80	214	0.421
4	104	285	0.286
5	131	358	0.391
6	157	424	0.934
7	188	482	1.056
8	283	589	0.764
9	296	596	0.145
10	308	686	0.386
11	319	710	2.084
12	348	748	3.334
13	357	754	29.73
14	365	765	11.948
15	387	801	0.139
16	440	899	11.809
17	472	958	0.834
18	536	1031	0.122

Table 2 - Model contributions of the system (crack 5%)

№	№ mode	Frequency, Hz	Contribution, %
1	19	56	10.97
2	81	228	0.223
3	156	431	0.743
4	194	505	0.204
5	228	596	0.109
6	299	685	0.265
7	312	705	1.837
8	336	745	0.68
9	362	771	0.367
10	410	848	2.388
11	417	862	7.346
12	425	870	11.019
13	432	883	11.754
14	443	908	5.51
15	453	918	5.877
16	466	946	5.51
17	472	958	5.51
18	535	1154	2.204

Analyzing a frequent structure of the model contributions in the rotor elements stresses have been observed that about 15% of energy of all 600 modes passes into 650-1150 Hz range with crack arise. Thus system energy is redistributed on this frequencies with crack occurs because of "breathing" mechanism of the real crack excite the fluctuation in this frequency range. If there is system without crack that 18 most energetically modes possess about 86.8% of all energy of the system. But if there is the system with crack that some part of all energy (14.3%) of the system is redistributed on 650-1150 Hz frequency modes (Figure 5). Therefore for system with crack the summarized contribution from 12 most energetically modes becomes 72.516%.

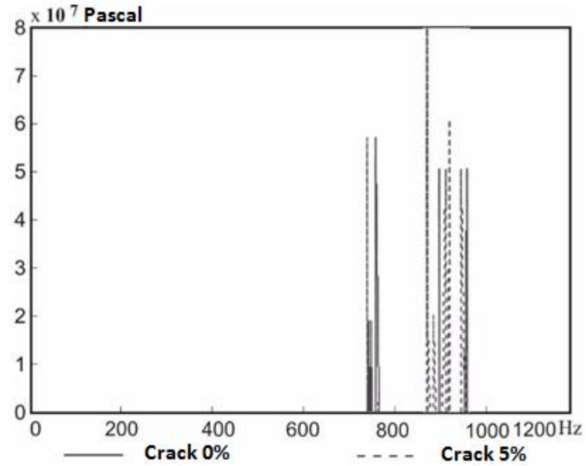

Fig. 5. Change of modal contributions

VI. Statistical Characteristics of the Vibrosignal

The vibrosignal of the rotor system is deterministic data that reflects the dynamic characteristics of the system. It can be captured on-line with help of analog digital converter and digital information can be saved into disk of computer. Statistical characteristics of the vibrosignal consist of peak-factor PF, kurtosis ε and dimensionless amplitude discriminants A_6 and A_8.

The peak-factor PF is the probability characteristics of the vibroacseleration: PF=MAX/RMS, where MAX is the arithmetic maximum of the vibrosignal amplitude; RMS – root mean square value of the vibrosignal amplitude: $RMS = \sqrt{\frac{1}{T}\int_0^T [z(t)]^2\, dt}$, where T is period of data points, z(t) - amplitude of the vibrosignal. The kurtosis ε is the fourth central moment divided by fourth power of standard deviation: $\varepsilon = \frac{\mu^4}{SD^4}$,where ε is the arithmetic mean of the amplitude variation of the signal profile. Mathematically it can be expressed as: $\mu = \frac{1}{T}\int_0^T z(t)dt$, SD is standard deviation of the vibrosignal amplitudes is the square root of the second moment ψ and arithmetic mean μ; $SD = \sqrt{\psi^2 - \mu^2}$. The dimensionless amplitude discriminants A_6 is the sixth central moment divided by sixth power of the standard deviation: $A_6 = \frac{\mu^6}{SD^6}$. The dimensionless amplitude discriminants A_8 is the eighth central moment by eighth power of the standard deviation: $A_8 = \frac{\mu^8}{SD^8}$.

VII. Statistical Researching of the Vibrosignal

The peak-factor PF, kurtosis ε and dimensionless amplitude discriminants A_6 and A_8 can be obtain for all frequency range 2-6535 Hz and only for 6501150 Hz frequency range where the change of the vibrosignal should be most appreciable from the crack. The statistical characteristics have been executed for rotor system with different crack depth (Table 3).

Table 3 - Statistical parameters of the vibrosignal (vertical direction)

Crack depth,% from diameter	Right bearing vertical direction	
	2-6535 Hz	650-1150 Hz
Peak-factor		
0 %	2.34	1.65
2-3 %	1.801	5.81
5-8 %	2.58	8.53
25 %	15.34	38.70
Kurtosis		
0 %	5.16	1.67
2-3 %	3.87	7.07
5-8 %	2.09	14.73
25 %	46.83	134.48
A_6		
0 %	0.0354	1.2753
2-3 %	0.1583	3.333
5-8 %	0.682	11.1
25 %	759	888
A_8		
0 %	0.0447	0.1886
2-3 %	0.188	4.261
5-8 %	0.785	14.03
25 %	874	908

From the Table 3 it can be shown that peak-factor PF, kurtosis ε and dimensionless amplitude discriminants A_6 and A_8 are considerably increase both 2-6535 Hz and 650-1150 Hz diapason of the vibrosignals for 5-8% and especially 25% crack. However for 2-3% crack depth all statistical characteristics for wide-band signal don't vary on comparison with system without crack, but all characteristics are increase for narrow-band signals. Thus peak-factor PF, kurtosis ε and dimensionless amplitude discriminants aren't reflect emerging a small crack in the rotor system for wide-band vibrosignal. But these statistical characteristics can be chosen as diagnostical characteristics for narrow-band signal (for 650-1150 Hz frequency range where the change of the vibrosignal should be most appreciable from the crack). This frequency range can be obtain only from theoretical model analysis finite-elemental model of the rotor system where vibroaccelerations of the rotor system are used in finite-elemental model as input excitations.

All sorts of local damage, reduction of stiffness, cracking, increasing damping, changing the mass-geometric parameters of the system, both in time and in space, leads to non-stationarity of the vibration signal. Wavelet analysis works well with the analysis of such signals.

To establish the informative characteristics of anomalous vibration measurements in time in the high-frequency region, we apply the wavelet basis in the form of a piecewise continuous function or a wavelet with a higher center frequency. As a first approximation, we will use the Morlet wavelet for the basis function. This function gives the minimal discrepancy with the Fourier analysis of the vibrational signal. The Morlet wavelet has a frequency localization, the best among other bases, therefore it is recommended for solving the problem of signal representation in a wide frequency range [10,11].

Experimental measurements of vibration parameters were carried out with the help of Bruel and Kjepr equipment, an accelerometer 4370, and preamplifiers 2635. Measurement signals were digitized and processed using the Atlas multi-channel synchronous recorder and spectrum analyzer [9] included in the State Register of Measuring Instruments under No. 19989-00. A preliminary analysis of the results of the Fourier transform allows us to reasonably choose the implementation of the vibration signal, the width of the frequency band, the wavelet type, and the center frequency of the wavelet transform.

Figure 6 shows the results of the wavelet analysis of these signals, where it can be seen that the main harmonics remain stable throughout the entire time interval. At the same time, the detected high-frequency harmonics, which determine the incipient crack, attenuate with time. Thus, Fourier and Wavelet analysis complement each other. The first one detects harmonic components in the analyzed vibration signal, the second one allows to localize the harmonics in time.

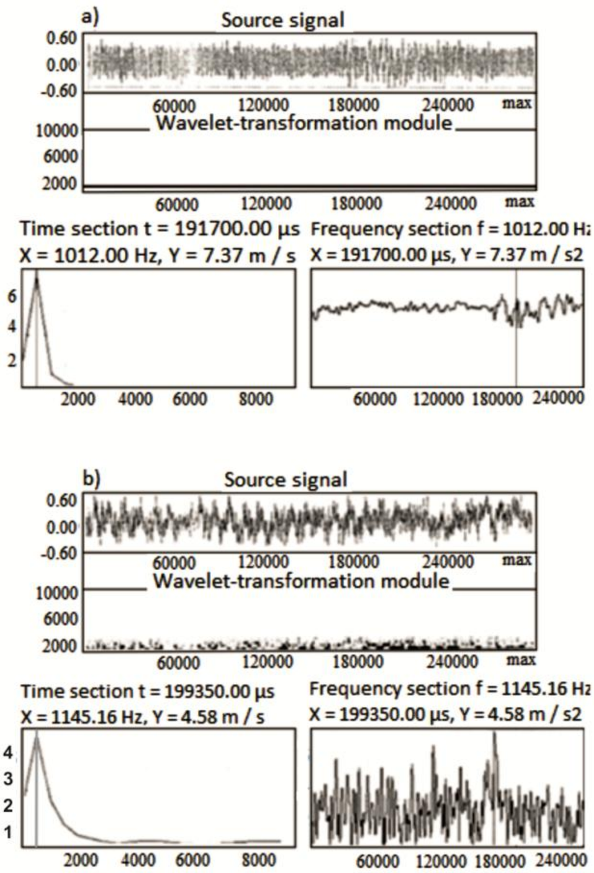

Fig. 6. Results of wavelet analysis of the defective rotor with the crack: a) 5%, b) 25%

In the analysis of technogenic vibrational signals or the response of physical systems, the numerical values of the signal parameters are almost always associated with the mass-geometric characteristics and kinematic parameters of the object. This is possible because the functional properties of the object are described by an exact physical model.

When analyzing the vibrational activity of an object under real operating conditions, especially when defects occur, such a model almost never happens. Therefore, when analyzing a vibrational signal, one has to rely on the previously encountered analogies, and if possible go to the physics of the vibrational signal. The use of one or another method of processing a vibration signal suggests its nature. So, if we have a stationary deterministic signal, then we can restrict ourselves to processing in the time domain with the determination of the fundamental harmonics. The influence of a large number of disturbing factors on the vibration signal requires the use of spectral Fourier analysis for its processing. Spectral analysis based on the Fourier transform is effective in processing stationary random signals. However, in the analysis of interconnected signals, it is necessary to determine the phase relations between several spectra.

Various types of local damage, reduced stiffness, cracking, increased damping, and changes in the mass-geometric parameters of the system both in time and in space will lead to the non-stationarity of the vibration signal. A common property of most practical non-stationary signals is that they can be divided into sections in which they have the character of quasistationary signals. Wavelet analysis does a good job of analyzing these kinds of signals.

Using the expansion in oscillating functions localized both in the time and frequency domains, the wavelet transform reflects a one-dimensional signal on the time-frequency plane, describing the spectral composition of the signal at each moment in time. It is necessary to take into account the strong influence of the type of applied wavelet basis to the analysis of a vibrational signal of complex shape. For a qualitative analysis of the vibrational signal in a fairly wide frequency band, several different bases should be used. To establish the informative characteristics of the anomalous measurements of the vibration signal in time in the low-frequency region, we apply the wavelet basis in the form of a continuous function, in the higher-frequency region in the form of a piecewise-continuous function, or a wavelet with a higher central frequency.

As the first approximation, one can use the Morlet wavelet for the basis function; this function gives a minimum discrepancy with the Fourier analysis of the vibration signal. Wavelet Morlet has a frequency localization, the best among other bases, and in this regard is the most recommended for solving the problem of representing a vibrational signal in a wide frequency range. If we are interested not only in high-frequency components, but also want to reflect the lower frequencies, then the scale factor should be taken quite large. The range of variation of the scale factor depends on the choice of the wavelet and on what frequencies we want to display as a result of decomposition. For a quantitative description of the experimental records and identification of possible defects, spectral analysis methods based on the Fourier transform and wavelet transform were used. The determination of the parameters of the frequency spectrum was carried out on the basis of a bivariate analysis. The measurement signals were digitized and processed using a multi-channel synchronous recorder and Atlant recorder software. A preliminary analysis of the results of the Fourier transform allows you to reasonably choose the length of the vibration signal, the bandwidth, the type of wavelet and the center frequency of the wavelet transform. An excessive increase in the sampling frequency is undesirable, since it will lead to a large sample size of the experimental data and will significantly increase the complexity of calculations using the wavelet transform.

Thus, the Fourier and wavelet spectrograms complement each other.

The first one detects harmonic and quasi-harmonic components in the analyzed vibrational signals, and the second one allows localizing harmonics in time (Figures 7 and 8).

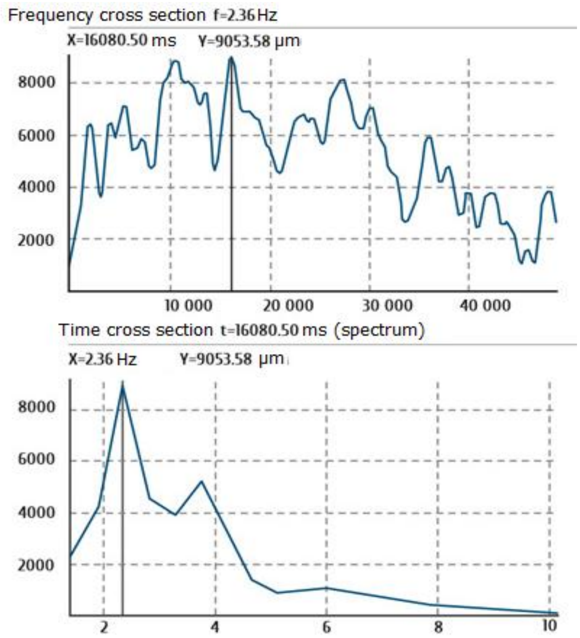

Fig. 7. The results of the wavelet analysis of vibration displacements of the object with a defect

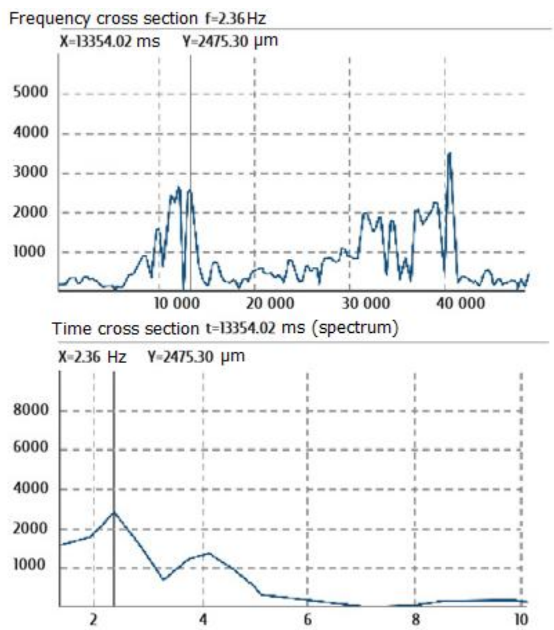

Fig. 8. The results of the wavelet analysis of vibration displacements of the object without any defect

On the example of the analysis of the influence of a defect of the rotor system on the spectral characteristics of the vibrodiagnostic signal, it is concluded that in order to determine the state and performance of any mechanical system, it is necessary to have sufficiently wide and accurate characteristics of the object. These are physical parameters, the main frequencies that determine the behavior of each element of the object, the possible causes of loss of performance. For qualitative research, the use of several methods is necessary. Modal analysis of the model, Fourier and wavelet analysis of the signals are applied sequentially. Complete research and the creation of automatic diagnostic systems is possible for critical systems related to technological disasters. At the same time, the results of the work are also applicable for a quick and approximate analysis of the state of objects using the vibration analysis methods shown.

The advantage of the wavelet transform over the Fourier transform is that it allows you to track the change in the spectral properties of the signal over time and indicate which frequencies (scales) dominate the signal. The Morlet wavelet was chosen as the base, which, unlike the others, makes it possible, after choosing the wavelet frequency (the number of oscillations), to compress or stretch the function as a whole without violating the similarity of the family of functions.

In the paper the detailed comparison of the methods of spectral analysis for diagnosing the appearance and development of cracks in rotor systems is carried out.

VIII. Conclusions

The method of the diagnostic of the grown crack described in this article is based on the model analysis of the system consisting from the two rotor, shaft supported by the ball bearings. Experimental results have been obtained for different depth of the crack, namely: 5%,10%,15% and 30% from shaft diameter.

1.The executing of the model contributions appropriated to different frequency ranges allows to determine the frequency range where the change of the vibrosignal should be most. appreciable from the crack.

2. Experimental results shown that when a cracked occurs in the rotor system, it can be detected by statistical characteristics with help of the data from experimental and theoretical model analysis at its early stage.

3. Another damages of the system such as unbalancing rotor, worn surface of the ball bearing or the geometrical features of the ball bearing some increase power of the vibroacceleration in Y and Z direction for all bearings in different frequency range (rotation speed =50Hz), damage of the ball bearing in high frequency range (5000-15000 Hz). The further increase of the crack can be observed that statistical characteristics are increase in wide-band signal that was investigated in other researches.

4. The peak factor of the kurtosis coefficient, the values of dimensionless amplitude discriminants, poorly reflect the appearance of a small crack in the rotor system for a broadband vibration acceleration signal, but is suitable as an informative characteristic for a narrowband signal allocated in the range of 650–1150 Hz that is the system with the appearance of cracks.

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